Running Dynamics for a Maintenance Railway Vehicle

Master of Science Thesis

by

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Preface

This thesis is a part of my MSc in Vehicle Engineering at KTH Royal Institute of Technology in Stockholm and was carried out in collaboration with TD Rail & Industry in Sundbyberg.

I would like to thank my supervisor and examiner at KTH, Mats Berg, and my supervisor at TD Rail & Industry, Pontus Karlsson, for their valuable support and assistance during the work.

I would also like to thank the people at BS Verkstäder in Falköping for their help with the information about the studied vehicle and the testing of the vehicle. Finally I also want to thank Kristian Karlsson at TD Rail & Industry for his help during the vehicle test.

Michael Gustafsson

Stockholm, August 2014
Abstract

BS Verkstäder has developed a new railway maintenance vehicle, MTR2000, which is in the process of authority approval. A part of the approval is the running dynamics of the vehicle which for maintenance vehicles following SS-EN 14033-1 can be evaluated using a simulation model of the vehicle, which is verified by comparing simulation results to measurements from a vehicle test. The testing, assessment values and limit values for running dynamics follow EN 14363.

The goal of this work was to create a simulation model of the vehicle in the software Gensys and to verify the model with vehicle tests. Using the verified model, simulations were then performed corresponding to the test conditions defined in the standard for approval of running behaviour. The model was also used to investigate how the vehicle’s running behaviour could be improved.

The results of the verification showed that the model gave a good representation of the vehicle up to the frame, while attempts to have similar good representation for the cabin were not successful. There were also other aspects of the verification process which could question the validity of the model.

With the verification only being partly successful all further simulations were done with focus kept on results up to the frame were the verification had given the best results.

The results of these latter simulations showed that the vehicle would most likely pass an authority approval process in its current configuration.

Simulations performed to try to improve the vehicle running behaviour showed that a decrease of 50% in damper constant for the vehicle’s hydraulic dampers would lead to decreased acceleration levels and lower wheel-rail guiding forces.
Sammanfattning

BS Verkstäder har tagit fram ett nytt arbetsfordon, MTR2000, som är i en pågående process för myndighetsgodkännande. En del av detta godkännande avser fordonets gångdynamik vilken för arbetsfordon, som följer standard SS-EN 14033-1, kan utvärderas med en simuleringsmodell av fordonet, som är verifierad med en jämförelse av simuleringsresultat och mätningar från ett fordonstest. De tester, storheter och gränsvärden som sedan används följer standard EN 14363.


Resultaten av verifieringen visade att modellen gav en god representation av fordonet upp till ramen, medan försök att få en lika bra representation upp till hytten inte var framgångsrika. Det fanns även andra aspekter av verifikationsprocessen som kan ifrågasätta modellens validitet.

Då verifikationen endast var delvis framgångsrik, lades fokus på resultat upp till ramen i alla vidare simuleringar, då det var där verifikationen gav bäst resultat.

Resultaten av dessa vidare simuleringar visade att fordonet med stor sannolikhet skulle klara av ett myndighetsgodkännande i sin nuvarande konfiguration.

Simuleringar utförda med syfte att försöka förbättra fordonets gångegenskaper visade att en minskning på 50% av dämparkonstanten för fordonets hydrauliska dämpare skulle leda till minskade accelerationsnivåer samt lägre hjul-räl krafter.
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1 Introduction

In this chapter the background and definition of the problem handled in the thesis are explained. The purpose and delimitations of the work as well as the methodology are also explained.

1.1 Background

The company BS Verkstäder has developed a new rail vehicle, called MTR2000 [1], which is a maintenance vehicle where the customers have the option to add for example a crane or a sky lift on its platform. See Figure 1.

![MTR2000](image)

Figure 1. MTR2000

For a new rail vehicle to be approved by the Swedish Transport Agency a number of requirements needs to be fulfilled [2], a part of these are requirements for the vehicle’s dynamic interaction with the track. The quantities looked for are mainly vertical and lateral accelerations in the vehicle and wheel-rail forces and these can be obtained with simulations using a mathematical model of the vehicle. The model has to be verified with a on-track test of the vehicle and then the simulations can be used for a virtual approval of the vehicle.

1.2 Problem definition

The problem handled in the thesis can be summarized as: Create and verify a simulation model of the vehicle MTR2000 and use that to see if the vehicle meets the running dynamics requirements for authority approval, and to see if any vehicle design improvements can be made.

1.3 Purpose

The purpose of the present work was thus to examine the running dynamics of the vehicle MTR2000 using a simulation model and see if it meets the requirement for authority approval or if any design changes would be necessary. If successful the simulation model could then be used for the approval.
1.4 Delimitations

To keep the work to a reasonable level, a few delimitations were done. The following topics have not been included in this thesis:

- Stationary and quasi-static vehicle testing, which is a part of the approval.
- Ride comfort, no comfort requirements have been covered.
- Flexible bodies, all bodies were assumed to be rigid in the vehicle model.

1.5 Methodology

The vehicle model was created in the simulation software Gensys using data from documentation and a CAD model of the vehicle provided by BS Verkstäder. The CAD model provided was missing some parts, so as a first step the model was completed by creating representative models of the missing parts and adding them to the complete CAD model. The updated CAD model was then verified by weighing of the vehicle and comparing the weight of the model with the actual weight.

To verify the Gensys model, a low-speed test of the vehicle was performed, in which accelerations were measured at four different positions in the vehicle. The data were analysed in the frequency domain and then band-pass filtered and analysed in the time domain. The measurement data was then compared to the simulation model’s response mainly by looking at the amplitudes of the accelerations.

Then simulations were performed with vehicle speeds, track geometries and data processing methods according to the standard EN 14363. The results were then analysed to see if they were below their respective limit values. After these simulations the model was used to see if any changes could be done to the suspension properties which would lead to improvements to the vehicle’s overall running dynamic performance.
2 Frame of reference

2.1 Requirements

The requirements for running safety for the vehicle type handled in this thesis are found in SS-EN 14033-1 [3], chapter 8, which gives reference and some deviations to the EN 14363 [2] standard.

2.1.1 SS-EN 14033-1

This standard contains the technical requirements for running of maintenance rail vehicles. In the chapter for running safety three different ways of determining the running characteristics of the machine are given. It can be done by either running tests, reference to a similar type approved machine or by simulation, the latter method have been used in this thesis.

Proving the running behaviour using simulation is permitted when there is a proven model of the machine. To prove a model it needs to be compared against results from running tests when the same input of track characteristics is used. The model is validated when there is a close correlation between the simulation and running test results in which the suspension is excited sufficiently.

2.1.2 EN 14363

This standard contains information on the vehicle running tests needed to be performed and the quantities and their limit values which are tested. The testing is divided in two parts: stationary tests and on-track tests. In the thesis the focus is on the tests explained in the on-track section of the standard and the quantities there.

2.1.2.1 Assessment values

As a first step it had to be decided if a complete or just a partial on-track vehicle test was needed. Since the test was being performed for an initial acceptance of the vehicle, a complete test was needed. Following this, a measuring method shall be determined with the choices being normal and simplified measuring method, the simplified differs from normal in that no wheel-rail forces are measured. The measuring method is determined using a flow chart and for the studied vehicle it came down to if it was deemed a new-technology vehicle or not. But since simulation was used, and since the software calculates all quantities, the determination of which measuring method that was needed was not done and the normal measuring method was adopted. The quantities that needed to be measured with this method can be seen in Table 1.

<table>
<thead>
<tr>
<th>Quantity</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\Sigma Y_{\text{max}}$</td>
<td>Max sum of guiding forces</td>
</tr>
<tr>
<td>$(Y/Q)_{\text{max}}$</td>
<td>Maximum quotient of guiding force over wheel force</td>
</tr>
<tr>
<td>$\Sigma Y_{\text{rms}}$</td>
<td>Moving RMS of guiding forces</td>
</tr>
<tr>
<td>$\hat{y}_{\text{max}}$</td>
<td>Maximum lateral acceleration in vehicle body</td>
</tr>
<tr>
<td>$\hat{z}_{\text{max}}$</td>
<td>Maximum vertical acceleration in vehicle body</td>
</tr>
<tr>
<td>$\hat{y}_{\text{rms}}$</td>
<td>Moving RMS of lateral accelerations in vehicle body</td>
</tr>
<tr>
<td>$\hat{z}_{\text{rms}}$</td>
<td>Moving RMS of vertical accelerations in vehicle body</td>
</tr>
</tbody>
</table>

Table 1. Quantities to measure, normal measuring method.
Here \( Y \) and \( Q \) are the track forces meaning they are resultant forces in the contact between wheel and rail. \( Y \) is the lateral guiding force and \( Q \) is the vertical force, see Figure 2.

![Figure 2. Track forces.](image)

The accelerations in the vehicle body are normally measured above each bogie, or axle for non-bogie vehicles, as well as in the longitudinal centre of the body. Since the studied vehicle’s cabin is small and there is a platform in the rear the spots where to measure accelerations weren’t clear. The standard doesn’t cover this, so in the thesis accelerations are looked at in a) the cabin above the front axle and b) in the frame above both the front and c) in the frame above the rear axle. These positions were chosen because this is where the response generally would be the highest and also because if the frame and cabin are seen as the vehicle body and the vehicle as only having a primary suspension, these positions would be measured following the standard.

### 2.1.2.2 Test conditions

Testing of the vehicle needed to be carried out in four different test zones. The first test zone is straight track and curves with very large radii. In this zone the testing of the vehicle is carried out in the area of the vehicle’s permissible speed. The second test zone is in large-radius curves. Here the testing of the vehicle is carried out in the area of the vehicle’s permissible speed and with high cant deficiency. The third zone is in small-radius curves with radii between 400 m and 600 m. The fourth and last zone is in very small-radius curves with radii between 250 m and 400 m. The testing of the vehicle for zone 3 and 4 is carried out in the area of the vehicle’s permissible cant deficiency. In each test zone there are different requirements for test conditions for the track sections; these can be seen in Table 2.

<table>
<thead>
<tr>
<th>Test zone</th>
<th>Length of track section</th>
<th>Minimal number of track sections</th>
<th>Minimal length of the sum of all track sections</th>
<th>Mean value of curve radius of all track sections</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>250 m</td>
<td>25</td>
<td>10 km</td>
<td>-</td>
</tr>
<tr>
<td>2</td>
<td>100 m</td>
<td>25</td>
<td>10 km</td>
<td>-</td>
</tr>
<tr>
<td>3</td>
<td>100 m</td>
<td>50</td>
<td>-</td>
<td>500 m ± 50 m</td>
</tr>
<tr>
<td>4</td>
<td>70 m</td>
<td>25</td>
<td>-</td>
<td>300 m, + 50m, - 20 m</td>
</tr>
</tbody>
</table>

Table 2. Test conditions for track sections.

The speed and cant deficiency for the different test zones can be seen in Table 3. Cant deficiency is the difference between equilibrium cant and the real track cant; equilibrium cant is the cant
necessary to have zero track plane acceleration [4]. For test zone 1 & 2 the vehicle speed \( V \) shall be specified as a test parameter. The relevant speed is the desired permissible maximum speed of the vehicle, \( V_{adm} \). The vehicles desired maximum cant deficiency is denoted \( cd_{adm} \).

<table>
<thead>
<tr>
<th>Test zone</th>
<th>Speed, ( V )</th>
<th>Cant deficiency, ( cd )</th>
<th>Tolerance</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>( V = \max(1.1 \cdot V_{adm}, V_{adm} + 10 \text{ km/h}) )</td>
<td>( cd \leq 40 \text{ mm} )</td>
<td>( \pm 5 \text{ km/h} )</td>
</tr>
<tr>
<td>2</td>
<td>( V_{adm} \leq V \leq 1.1 \cdot V_{adm} )</td>
<td>( 0.75 \cdot cd_{adm} \leq cd \leq 1.1 \cdot cd_{adm} )</td>
<td>( \pm 5 \text{ km/h, } \pm 0.05 \cdot cd_{adm} )</td>
</tr>
<tr>
<td>3</td>
<td>( V \leq 1.1 \cdot V_{adm} )</td>
<td>( 0.75 \cdot cd_{adm} \leq cd \leq 1.1 \cdot cd_{adm} )</td>
<td>( \pm 0.05 \cdot cd_{adm} )</td>
</tr>
<tr>
<td>4</td>
<td>( V \leq 1.1 \cdot V_{adm} )</td>
<td>( 0.75 \cdot cd_{adm} \leq cd \leq 1.1 \cdot cd_{adm} )</td>
<td>( \pm 0.05 \cdot cd_{adm} )</td>
</tr>
</tbody>
</table>

Table 3. Test conditions for speed and cant deficiency.

Track irregularities also constitute an important aspect of the test conditions, see [2].

2.1.2.3 Processing

The processing of the measurement signals differs between the different quantities and also if looking at running safety or ride characteristics. The unit and filtering for the different quantities can be seen in Table 4 for running safety and in Table 5 for ride characteristics. Here \( \Sigma Y \) is the resultant lateral track force of a wheelset, \( \ddot{y}^* \) and \( \ddot{z}^* \) are lateral and vertical accelerations respectively of the vehicle body.

<table>
<thead>
<tr>
<th>( \Sigma Y_{max} )</th>
<th>kN</th>
<th>Low-pass filter 20 Hz</th>
<th>Sliding mean method with:</th>
</tr>
</thead>
<tbody>
<tr>
<td>( (Y/Q)_{max} )</td>
<td>-</td>
<td>Low-pass filter 20 Hz</td>
<td>- Window length 2.0 m</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>( \Sigma Y_{rms} )</th>
<th>kN</th>
<th>Low-pass filter 20 Hz</th>
<th>Sliding RMS method with:</th>
</tr>
</thead>
</table>

\[ \ddot{y}^*_{max} \text{ m/s}^2 | \text{Low-pass filter 6 Hz} | - \]

\[ \ddot{z}^*_{max} \text{ m/s}^2 | \text{Band-pass filter 0.4 – 4 Hz} | - \]

Table 4. Unit and filtering for measurement signals for running safety.

\[ \ddot{y}^*_{max} \text{ m/s}^2 | \text{Band-pass filter 0.4 – 10 Hz} \]

\[ \ddot{z}^*_{max} \text{ m/s}^2 | \text{Band-pass filter 0.4 – 10 Hz} \]

\[ \ddot{y}^*_{rms} \text{ m/s}^2 | \text{Band-pass filter 0.4 – 10 Hz} \]

\[ \ddot{z}^*_{rms} \text{ m/s}^2 | \text{Band-pass filter 0.4 – 10 Hz} \]

Table 5. Unit and filtering for measurement signals for ride characteristics.

2.1.2.4 Limit values

There are different limit values with different significance with regard to safety issues; there are limit values for running safety and for ride characteristics. The limit values for running safety are relevant limits which shall be used restrictively. These values can only be changed nationally if the track and operating conditions differ from the conditions used by the International Union of Railways (UIC) for definition of the limit values. The limit values for ride characteristics are no running safety-relevant limits but need to be considered with regard to vibrational load on passengers, freight and vehicle.
components. The limit values can be seen in Table 6. Here $k_1$ is a constant which is set to 1.0 and $2Q_0$ is the static axle load.

\[
\begin{array}{|c|c|c|}
\hline
\text{Quantity} & \text{Running safety} & \text{Ride characteristics} \\
\hline
\Sigma Y_{\text{max}} \ [\text{kN}] & k_1 (10 + 2Q_0/3) & - \\
\Sigma Y_{\text{rms}} \ [\text{kN}] & \Sigma Y_{\text{max}} / 2 & - \\
(Y/Q)_{\text{max}} \ [-] & 0.8 & - \\
\dot{y}_{\text{max}} \ [\text{m/s}^2] & 3.0 & 4.0 \\
\dot{z}_{+\text{max}} \ [\text{m/s}^2] & 5.0 & 5.0 \\
\dot{y}_{\text{rms}} \ [\text{m/s}^2] & - & 1.5 \\
\dot{z}_{+\text{rms}} \ [\text{m/s}^2] & - & 2.0 \\
\hline
\end{array}
\]

Table 6. Limit values for running safety and ride characteristics.

In the work there was no consideration taken to ride comfort, for which the limit values for acceleration levels are lower than the limit values presented here. The requirements for ride comfort can be found in SS-ISO 2631-1 [5].

### 2.2 Vehicle description

The vehicle, which is illustrated from the side in Figure 3, is a two-axle vehicle with a platform in the rear to carry load or tools.

![Figure 3. Side view of the vehicle, produced from CAD-model.](image)

The vehicle is driven by a diesel engine which propels both axles via a hydrostatic transmission. The basic vehicle data are summarized in Table 7.

\[
\begin{array}{|c|c|}
\hline
\text{Length over buffers} & 9.180 \text{ m} \\
\text{Width} & 2.400 \text{ m} \\
\text{Height} & 3.600 \text{ m} \\
\text{Platform height} & 0.900 \text{ m} \\
\text{Weight (with crane)} & 15000 \text{ kg} \\
\text{Axle load (with crane)} & 73.6 \text{ kN} \\
\text{Axle distance} & 5.700 \text{ m} \\
\text{Wheel diameter} & 0.860 \text{ m} \\
\text{Maximum speed} & 80 \text{ km/h} \\
\hline
\end{array}
\]

Table 7. Basic vehicle data.
2.2.1 Suspension

The primary suspension, between axles and frame, used in the vehicle is Metalastic chevron springs [6] combined with vertical hydraulic dampers. Other than the primary suspension the cabin and some components are suspended on the frame with rubber springs to either lower the effect of the component’s vibrations on the frame or to protect the component from vibrations in the frame.

2.2.1.1 Metalastic chevron

Metalastic chevron springs are metal plates with rubber elements in between fitted in a “vee” configuration, see Figure 4.

![Figure 4. Metalastic chevron springs [6].](image)

This configuration enables shear and compression compliance within the rubber elements giving the spring three modes of flexibility; these can be seen in Figure 5.

![Figure 5. Modes of flexibility for axle box suspension [6].](image)

For a spring, the force is ideally given by equation (1).

\[ F = k \cdot \Delta x \]  

(1)

Here \( k \) is the spring coefficient and \( \Delta x \) is the deflection. Since these springs use rubber to create their flexibility it can’t be assumed that they are linear springs but have nonlinear properties. More on rubber properties is explained in the 2.2.1.3 Rubber springs section of the report.
2.2.1.2 Hydraulic dampers

The hydraulic dampers, see Figure 6, have linear damper characteristics with a blow-off function which limits the maximum damper force.

![Hydraulic damper](image)

The damper force for linear damping is given by equation (2).

\[ F = c \cdot \Delta \dot{x} \]  

(2)

Here \( c \) is the damper coefficient and \( \Delta \dot{x} \) is the piston velocity. This expression gives the damper force for velocities up to the blow-off force. When the blow-off force is reached the damper force will remain constant, this gives the force as a function of velocity graph that is illustrated in Figure 7.

![Force as a function of velocity](image)

Figure 7. Force as a function of velocity for a linear hydraulic damper with a blow-off function at 10 kN.

2.2.1.3 Rubber springs

Rubber has different characteristics for static and dynamic conditions meaning that when the spring has a spring coefficient for static load, then it has a higher one for dynamic load [7]. A simple model of a rubber spring and how it differs from a steel coil spring can be seen in Figure 8.

![Difference between steel and rubber spring](image)

Figure 8. Difference between a steel spring model and a simple rubber spring model.
Here a parallel viscous damper is added to account for the rubber’s internal damping. The damping coefficient for this simple model can be calculated using equation (3).

\[ c = 2 \cdot \zeta \cdot \sqrt{k \cdot m} \]  

(3)

Here \( k \) is the spring coefficient, \( m \) is the mass that the spring holds and \( \zeta \) is the ratio of critical damping which for rubber is around 0.05.

This simple rubber model can be expanded by putting a spring in series with the damper which together with the parallel spring gives a coupling that has different static and dynamic stiffness. This model can be seen in Figure 9.

![Figure 9. Rubber spring model with spring parallel with spring and damper in series.](image)

The next step from this is to add friction. Including friction force gives increased stiffness at small displacement amplitudes [8]. This model can be seen in Figure 10.

![Figure 10. Rubber spring model with spring parallel with friction block and a spring and damper in series.](image)

In the vehicle there are several parts that are supported with rubber elements to reduce the vibrations on the part. In this work only the rubber elements supporting heavier parts have been looked at since these can have a noticeable effect on the overall running dynamics. In Figure 11 the rubber springs for the cabin and engine can be seen; these are the ones that are considered in the model.
2.2.2 Wheel profile

The vehicle’s wheel profile follows the UIC 510-2 standard [9] and since the vehicle is new and have barely been driven the wheels are assumed to be unworn. The wheel profile can be seen in Figure 12.

![Figure 12. Wheel profile.](image)

2.3 Gensys

Gensys is a multi-body simulation software with focus on rail vehicles. The development of the software began in 1992 and at the same time the company AB DEsolver started which is the company behind developing and supporting the software. The development of the software is carried out in cooperation with KTH Royal Institute of Technology and Chalmers University of Technology [10].

In Gensys models are created by defining masses and their properties and then connect them to each other using any of the predefined couplings, for instance a spring or damper. With the vehicle model a track model is needed together with track design geometry and track irregularities. More on modelling in Gensys is explained in the Simulation model chapter.

Using the model some different calculations can be made including frequency analysis of the vehicle and time simulations over a defined track. When performing time simulations there are some different methods that solve ordinary differential equations to choose from, in this work Heun’s method is used [11].

In simulations the software also uses Hertz theory to calculate the contact patch and contact pressure between wheel and rail. This theory says that an elliptic contact area arises if two bodies are pressed together with a normal force [4].
The software also uses Kalker’s simplified theory of rolling contact to calculate tangential forces in the contact. This is a frequently used model of wheel-rail contact since it often gives a sufficiently exact description while being faster to use than the so-called complete theory [4].

From a simulation there is a lot of different output data available including displacements and accelerations at any position in all bodies, spring and damper displacements and forces, wheel-rail forces and wheel-rail wear.
3 Simulation model

3.1 Vehicle model

The model was built with five masses (rigid bodies): the cabin, frame, engine and the two wheel sets. To keep the model as simple as possible all masses were assumed to be rigid, see Figure 13.

![Vehicle model](image)

Figure 13. Vehicle model.

3.1.1 Vehicle data

To relate different masses to each other and to the program’s fixed coordinate system, each mass is given its own moving coordinate system. To get the coordinates systems to move a separate moving coordinate system is created which moves with the vehicle’s speed and follows the track design geometry. Then the coordinate system for the first mass is defined in relation to this and the following masses’ coordinate systems are defined either in relation to the first mass system or to another already defined mass system. The different systems can have a longitudinal distance between themselves but no lateral or vertical. In the lateral direction the zero point is in the middle of the vehicle and in the vertical direction the zero point is at the nominal top of the rail head.

In the model the cabin mass coordinate system was the first one defined, then the frame and engine mass systems were defined in relation to that, and the axle mass systems were defined in relation to the frame. In Figure 14 the longitudinal distances between the different coordinate systems can be seen.
The value for these distances can be seen in Table 8.

<table>
<thead>
<tr>
<th></th>
<th>ace</th>
<th>acf</th>
<th>afa1</th>
<th>afa2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Longitudinal distance to CoG [m]</td>
<td>1.130 m</td>
<td>2.885 m</td>
<td>3.265 m</td>
<td>2.435 m</td>
</tr>
</tbody>
</table>

Table 8. Longitudinal distance between coordinate systems.

The inertia properties of each mass are defined in their own coordinate system, it’s from those origins the location of the mass centres of gravity are defined. Since the coordinate systems were defined relative to each other, so became the mass centres of gravity (CoG). Other than the location of the centre of gravity, the weight of the mass and its corresponding moments of inertia around axes through its centre of gravity were needed. These quantities were all determined using a CAD model of the vehicle. Since the CAD model wasn’t complete, some approximated models of the missing parts were created.

In Table 9 the properties for all masses can be seen. The frame mass includes the crane as a load and the diesel tank with 50 litres of diesel, which corresponds to the amount of diesel in the tank during the vehicle test.

<table>
<thead>
<tr>
<th>Longitudinal distance to CoG [m]</th>
<th>Cabin</th>
<th>Frame</th>
<th>Engine</th>
<th>Axles</th>
</tr>
</thead>
<tbody>
<tr>
<td>Lateral distance to CoG [m]</td>
<td>−0.10</td>
<td>−0.40</td>
<td>0.00</td>
<td>0.00</td>
</tr>
<tr>
<td>Vertical distance to CoG [m]</td>
<td>2.28</td>
<td>1.00</td>
<td>0.69</td>
<td>0.43</td>
</tr>
<tr>
<td>Mass [kg]</td>
<td>3083</td>
<td>8960</td>
<td>900</td>
<td>1013</td>
</tr>
<tr>
<td>Moment of inertia, roll [kgm²]</td>
<td>3.59 · 10³</td>
<td>6.70 · 10³</td>
<td>9.69 · 10</td>
<td>5.80 · 10²</td>
</tr>
<tr>
<td>Moment of inertia, pitch [kgm²]</td>
<td>3.76 · 10³</td>
<td>5.70 · 10⁴</td>
<td>1.68 · 10²</td>
<td>7.70 · 10</td>
</tr>
<tr>
<td>Moment of inertia, yaw [kgm²]</td>
<td>4.13 · 10³</td>
<td>5.75 · 10⁴</td>
<td>1.66 · 10²</td>
<td>5.81 · 10²</td>
</tr>
</tbody>
</table>

Table 9. Properties for all masses in the simulation model.
3.1.2 Metalastic chevron model

The chevron model is built using the simple rubber model with a spring and a damper element in parallel. To get the flexibility in x, y- and z-directions there is a spring-damper for all three directions at each wheel except for the z-direction where the dampers are ignored since the effect of it would be negligible in comparison to the effect of the hydraulic dampers which work in the same direction. The couplings are introduced with no lengths which eliminates the problem of figuring out were attachment points for each direction would be, which since the chevrons have a rather complex geometry would be tricky. Also if a spring has a length Gensys gives it a pre-load force of the spring’s length times the spring coefficient, so giving the spring no length eliminates the need to compensate for this to avoid initial value problems.

3.1.3 Hydraulic damper model

To model the dampers with the blow-off force, viscous damping elements with asymmetric non-linear coupling properties were used, in which the properties can be set using damper force and corresponding velocity for as many points as needed. In between two points the damping is seen as linear and after the end points the program extrapolates using the previous two points. So to get the desired damper element using zero force and zero velocity as a starting point, the velocity for the blow-off force were calculated using the damper coefficient giving linear characteristics up to the blow-off point. Then an extra point was added for a higher velocity than the blow-off but still with the blow-off-force as the corresponding force so the force would be kept constant at the blow-off force for all higher velocities. Since the asymmetric coupling properties were used the properties are mirrored for negative force and velocity, the entered data points give the graph seen in Figure 7.

Since the dampers are mounted with an angle the elements’ working directions was given the direction ‘cu’. This is a built in function in the program where the program itself calculates the direction using the attachment points of the elements and the directions are then updated automatically when the bodies move.

3.1.4 Rubber spring model for cabin and engine suspension

These rubber springs were modelled in the same way as the chevron springs with a spring and damper in parallel for each direction. But for these rubber springs the damping was included in the vertical direction since there are no other dampers and its effect couldn’t be neglected.

3.2 Track model

The track model, which can be seen in Figure 15, was built with individual track pieces for each wheelset which move along with the vehicle. Each individual rail piece has lateral and vertical springs and dampers in parallel allowing for movement in these directions and the track similarly allows for movement in the lateral direction. For the contact between rail and wheel the model permits two-point contact with stiffness set by \( k_{fr} \) and \( k_{nr} \).
In the model the track has the mass $m_t$ whereas the rails are considered massless for numerical reasons. In Table 10 the data for the track model are listed.

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>$m_t$</td>
<td>1000 kg</td>
</tr>
<tr>
<td>$k_{ytg}$</td>
<td>30 MN/m</td>
</tr>
<tr>
<td>$c_{ytg}$</td>
<td>173 kNs/m</td>
</tr>
<tr>
<td>$k_{zrt}$</td>
<td>180 MN/m</td>
</tr>
<tr>
<td>$c_{zrt}$</td>
<td>250 kNs/m</td>
</tr>
<tr>
<td>$k_{yt}$</td>
<td>42 MN/m</td>
</tr>
<tr>
<td>$c_{yt}$</td>
<td>250 kNs/m</td>
</tr>
<tr>
<td>$k_{fr} = k_{nwr}$</td>
<td>600 MN/m</td>
</tr>
</tbody>
</table>

Table 10. Track model data.

The rail profile used is nominal BV50 with inclination 1:30. This is used since it’s the rail profile of the rails on the test track used for the model verification. The track irregularities used, which are denoted S22, are regarded as normal for Swedish main line tracks. Since the irregularities for the test track were not known S22 irregularities were used for the verification too, but then with amplitudes modified to a level where the acceleration amplitudes at one of the axle boxes in the simulations matched the results from the test.
4 Model verification

To verify the vehicle model a test of the vehicle was performed. In this chapter the measurement setup, processing of data, results from the test and the comparison to simulation results are presented.

4.1 Measurement setup

In the test accelerations were measured in four different positions in the vehicle. These were in the cabin above the front axle, on the frame above the front axle, on the platform above the rear axle and on the front right axle box. The positions are illustrated in Figure 16.

![Figure 16. Measurement positions.](image)

In Table 11 the lateral and vertical distances to the measuring points can be found. The lateral distance is measured from the middle of the vehicle with the positive axis pointing to the right. The vertical distance is measured from the top of the rails.

<table>
<thead>
<tr>
<th>Position</th>
<th>Lateral distance [m]</th>
<th>Vertical distance [m]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cabin</td>
<td>0.83</td>
<td>1.45</td>
</tr>
<tr>
<td>Frame over front axle</td>
<td>-0.16</td>
<td>1.19</td>
</tr>
<tr>
<td>Frame over rear axle</td>
<td>-0.53</td>
<td>0.96</td>
</tr>
<tr>
<td>Axle box</td>
<td>0.94</td>
<td>0.43</td>
</tr>
</tbody>
</table>

Table 11. Lateral and vertical measurement positions.

To measure the lateral and vertical accelerations at these positions two smart phones were used with an app that allowed accessing the phones’ accelerometer via a computer and collecting up to 10 000 samples per measurement with a sampling frequency of 60 Hz. In Figure 17 the measuring setup for the measurements at the frame over the front axle can be seen.
Due to time limitation of the work and the current state of the vehicle, a simpler test was performed on BS Verkstäder’s own railyard with a speed limitation of 30 km/h. During the test the vehicle was pulled or pushed, depending on direction of travel, by another vehicle. This was done to eliminate disturbances from the MTR2000 vehicle’s own motor. The tests were performed with two different speeds: 15 km/h and 30 km/h. The effective test section consisted of 100 m track with a very large curve radius. The complete plan, covering 48 test runs, for the test can be seen in Appendix A.

4.2 Data processing

The data were collected after each test run and the data from the selected test section were extracted out using times from a stop watch, which was started when the measuring was activated on the phones. Intermediate times were clocked at the beginning and end of the test section. The vehicle’s speed differed slightly between nominally identical tests, and this combined with the manual clocking made the test result from the same measuring point somewhat different between test cases. An example of this can be seen in Figure 18 where three test cases are shown in which the vehicle had the same speed and direction, so the graphs should have been more or less identical.
The original plan was to average the three test cases, but with the difference between them it was chosen to select one from each measuring point from each speed and travel direction. This was done by selecting a test case where the biggest cabin acceleration peaks were located in the middle of the data, since a peak in the middle of the data definitely origins from the selected test track. Then test results for the other measuring points were analysed to find the test case which had most similarities with the selected one, especially when looking at the larger peaks. This procedure was then repeated for both speeds and travel directions. To easier separate individual peaks in the data, the selection was done after band-pass filtering at 0.4-10 Hz. The complete set of raw and filtered measuring data can be seen in Appendix B.

The data selected were then analysed as raw data in the frequency domain, to see which frequencies that had the most effect on the results. After studying the data in the frequency domain, they were analysed in the time domain band-pass filtered at 0.4-10 Hz, which as mentioned earlier is the procedure for processing measurement data for ride characteristics in the standard EN 14363.

### 4.3 Measurement results

In this section, only the response from each point with the speed of 30 km/h and travel direction forward are presented. The response for both speeds and directions for all the selected test cases can be seen in Appendix B.

#### 4.3.1 Frequency response

The frequency response of the raw axle box accelerations (lateral and vertical), see Figure 19, shows no specific frequency range where the response is higher. This is to be expected since the axles are unsprung masses.

Figure 19. Frequency response from axle box.

In Figure 20 the response from the frame over the front axle can be seen. If compared to the response from the axle box it shows a decrease in magnitude which since there is a suspension between them is expected. In the lateral direction the decrease is low which suggests that the
suspensions have very little effect on the accelerations in that direction. Besides that, the response shows no significant peaks in the lower frequency area.

The response from the frame over the rear axle, which is seen in Figure 21, shows a larger decrease of magnitude than in the front, but since no measurements were made on a rear axle box there is no information of the effect of the suspension. However, it can be assumed that the front and rear axles should have a similar response and, since the suspensions are identical in the front and rear, the difference in response, particularly in the lateral direction, needs to be investigated. One possible reason for this is that in the test of the vehicle it was pulled or pushed, depending on direction by another vehicle. This vehicle was connected in the rear of the test vehicle and might have prevented the test vehicle’s ability to move laterally. Other than this it can be seen that there is a peak in lateral magnitude between 3-4 Hz, presumably this peak is caused by the vehicle having a natural frequency around there.

Figure 20. Frequency response from the frame over the front axle.

Figure 21. Frequency response from the frame over the rear axle.
The cabin’s response, which can be seen in Figure 22, shows a further decrease in magnitude from the measuring point in the frame over the front axle. But it also shows some clear peaks at lower frequencies, at 2 Hz and 4.5 Hz in the lateral direction and at around 3 Hz in the vertical direction. Once again these are presumed to exist due to natural frequencies in the vehicle.

4.3.2 Band-pass filtered accelerations

The data presented in this section are from the front right wheel axle box, frame over the front axle and cabin. These are presented adjacent to each other to visualize the effect of each suspension step.

In Figure 23 the lateral accelerations for these points can be seen. Here it shows that there is no decrease at all between the axle and frame, meaning that the primary suspension has no effect in this direction. Going up to the cabin, on the other hand, a significant decrease is shown which is to be expected if the springs are dimensioned and working properly.
Looking at the vertical accelerations, which can be seen in Figure 24, a clear decrease by both suspension levels can be seen. So in this direction the suspension seems to be working properly.

4.3.3 RMS values

The RMS values for the data were analysed since these also are a part of the requirements in the standard. Here both directions are included in the same figure for tests in 30 km/h, while the same type of figures for the 15 km/h tests can be found in Appendix B.

The RMS values for lateral acceleration for the different measuring points, see Figure 25, shows the same effect as previously discussed, with the primary suspension in the front not having any effect in the lateral direction. In the rear the primary suspension seems to have a significant effect on the accelerations.
The vertical RMS values which can be seen in Figure 26 show a more expected behaviour from the suspension with the accelerations being reduced for each suspension level.

Worth noticing is that the cabin’s RMS values is well under the limit values, cf. Table 6, but the vehicle speed in the test was well below the vehicle’s maximum speed of 80 km/h. Note also that the limit values are for ride characteristics and not comfort for which lower accelerations are usually needed.

4.4 Comparison simulation-measurement

The comparison was done on band-pass filtered accelerations and using the S22 track irregularities in the simulations, with the amplitudes of the irregularities up-scaled by 2-5 times of their original value. Since the actual track irregularities on the test track were not used in the simulations the
focus was on comparing acceleration amplitudes. Because of numerical reasons, simulations for 15 km/h couldn’t be performed. So only comparisons for 30 km/h are presented with travel direction forward being presented here, results for travel direction backwards can be seen in Appendix B.

In Figure 27, which shows lateral accelerations from the test and simulation at the axle box, it can be seen that the test in some of the bigger peaks has accelerations higher than from the simulation. It can also be seen that in the areas with smaller accelerations the values are higher from the test.

![Figure 27. Lateral accelerations at front right axle box, Simulation vs. Test.](image)

For the vertical axle box accelerations which are shown in Figure 28, the same thing as for the lateral accelerations is seen with the test having larger peaks. In areas with smaller amplitudes the test shows higher accelerations than the simulation.

![Figure 28. Vertical accelerations at front right axle box, Simulation vs. Test.](image)

When looking at the lateral accelerations in the frame over the rear axle and over the front axle, initial simulations did not show the same behaviour as the test did, with no decrease in accelerations
at all in the front and a big decrease in the rear. With the assumption that the rear acceleration was lowered because the vehicle was coupled in that end to another vehicle, a lateral spring connected to a ground point was added at the rear in the model to imitate this. The spring, which is used in the comparison for all four measuring points, was given a relatively large stiffness and the result which can be seen in Figure 29 showed a large decrease in lateral accelerations in the frame over the rear axle.

![Figure 29. Lateral accelerations in the frame over the rear axle, Simulation vs. Test.](image)

Looking at the vertical accelerations in the frame over the rear axle, which can be seen in Figure 30 there are similar amplitudes with no peaks being notably larger in the test or simulation.

![Figure 30. Vertical accelerations in the frame over the rear axle, Simulation vs. Test.](image)
In the lateral accelerations in the frame over the front axle, which are shown in Figure 31, it can be seen that the amplitudes are similar between the test and simulation and that the added lateral spring in the rear of the model hasn’t affected the results in the front.

![Lateral accelerations in frame over front axle, going forward 30 km/h. Simulations vs. Test](image1)

Figure 31. Lateral accelerations in the frame over the front axle, Simulation vs. Test.

From the vertical accelerations in the frame over the front axle which are seen in Figure 32 the same as in the rear can be seen with the simulations having similar amplitude in the largest peak as the test while not containing as many large peaks.

![Vertical accelerations in frame over front axle, going forward 30 km/h. Simulations vs. Test](image2)

Figure 32. Vertical accelerations in the frame over the front axle, Simulation vs. Test.
In Figure 33 which shows the lateral accelerations of the cabin a large difference between the test and simulation can be seen. In the simulation the acceleration has increased between the frame and the cabin while in the test there is a large decrease.

![Lateral accelerations in cabin, Simulation vs. Test](image)

Figure 33. Lateral accelerations in the cabin, Simulation vs. Test.

In Figure 34, which shows the vertical accelerations in the cabin, the same effect as in the lateral direction is shown with the simulation having a much larger response than the test.

![Vertical accelerations in cabin, Simulation vs. Test](image)

Figure 34. Vertical accelerations in the cabin, Simulation vs. Test.
The maximum values for the lateral and vertical accelerations from the measurement and simulation used in the comparison can be seen in Table 12.

<table>
<thead>
<tr>
<th>Position</th>
<th>Lateral</th>
<th>Vertical</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Test</td>
<td>Simulation</td>
</tr>
<tr>
<td>Axlebox</td>
<td>1.58</td>
<td>1.44</td>
</tr>
<tr>
<td>Frame over rear axle</td>
<td>0.46</td>
<td>0.51</td>
</tr>
<tr>
<td>Frame over front axle</td>
<td>1.75</td>
<td>1.82</td>
</tr>
<tr>
<td>Cabin</td>
<td>0.62</td>
<td>1.93</td>
</tr>
</tbody>
</table>

Table 12. Maximum values for accelerations in comparison between simulation and test (in m/s²).
5 Analysis of verification

From the comparison it can be concluded that the model doesn’t give a good representation of the vehicle when looking at the accelerations in the cabin. Looking at the stiffness of the rubber springs on which the cabin stands, there is no reason to think that those alone would give the decrease in accelerations that were seen in the tests. Possible reasons to why they are low anyway are that the cabin structure or the floor is flexible, and with that the response could have been lowered. In this case the response could be different in different positions in the cabin. Another possible reason is that during the tests the smartphones were attached to the surface using duct tape and if this was not done properly the phone might have been given some suspension during the tests and thereby lower the response. One thing that speaks against this theory is that the phones were moved and then reattached between the tests in different speed and the results show similar response which would mean that it had to be attached poorly both times and assumed to only have been that in the cabin. But one thing that differs between the cabin and the other test positions is that the cabin floor is covered with a rubber carpet with may have affected the tapes ability to attach properly.

The comparison also shows that differences in lateral accelerations in the back and front of the frame were most likely caused by the vehicle being coupled to another vehicle in the rear. Since the primary suspension is the same at both axles and the vehicle weight is almost evenly distributed between them there is no reason to believe that they should have different effects on the accelerations. So with this the assumption of the coupling in rear lowering the accelerations there is assumed to be correct and the added lateral stiffness at the rear buffers is removed for the simulations at higher speeds (Chapter 6).

With the comparisons showing similar response in the frame between the test and simulations it is assumed that the model gives a fairly good representation of reality up to this level. So for the simulations at higher speeds the focus is on the accelerations in the frame and since the wheel-rail forces are mostly affected by the primary suspension these are also assumed to be correct.
Simulations according to EN 14363

In this chapter results from simulations with track design geometry and vehicle speeds following the standard EN 14363 are presented with figures and values for the assessment quantities. For the accelerations, figures only with the accelerations in the frame over the front axle are presented here while the figures for the accelerations in the cabin and the frame over the rear axle can be seen in Appendix C. All accelerations are band-pass filtered between 0.4 and 10 Hz which is the procedure for ride characteristics.

The vehicle’s limit values for sum of Y forces, max and RMS, can be seen in Table 13, cf. Table 6.

| $\sum Y_{\text{max,lim}}$ (with crane) [kN] | 34.5 |
| $\sum Y_{\text{rms,lim}}$ (with crane) [kN] | 17.3 |

Table 13. Limit values for sum of Y forces, max and RMS, for the vehicle.

The desired maximum cant deficiency $c_{\text{adm}}$ used was 100 mm.

6.1 Test zone 1, tangent track

For this zone with the desired maximum speed of the vehicle at 80 km/h the test speed becomes 90 km/h according to Table 3.

In Figure 35 the lateral and vertical accelerations are shown.

![Figure 35. Lateral and vertical accelerations in the frame over the front axle, simulation in 90 km/h on tangent track](image)
The maximum values for the accelerations, including accelerations from the frame over the rear axle and the cabin, and their RMS values can be seen in Table 14.

<table>
<thead>
<tr>
<th></th>
<th>Cabin</th>
<th>Frame over front axle</th>
<th>Frame over rear axle</th>
</tr>
</thead>
<tbody>
<tr>
<td>$y_{max}^*$ [m/s²]</td>
<td>0.66</td>
<td>0.66</td>
<td>0.77</td>
</tr>
<tr>
<td>$z_{max}^*$ [m/s²]</td>
<td>1.82</td>
<td>1.31</td>
<td>1.68</td>
</tr>
<tr>
<td>$y_{rms}^*$ [m/s²]</td>
<td>0.19</td>
<td>0.17</td>
<td>0.21</td>
</tr>
<tr>
<td>$z_{rms}^*$ [m/s²]</td>
<td>0.46</td>
<td>0.29</td>
<td>0.34</td>
</tr>
</tbody>
</table>

Table 14. Maximum accelerations and RMS values for the cabin and frame over front and rear axle, simulations in 90 km/h on tangent track.

### 6.2 Test zone 2, large radius curve

For test zone 2 a curve with 700 m radius and cant at 50 mm was used. The vehicle speed for the simulations was 88 km/h which is the upper limit for speed in the test zone. With this speed the cant deficiency becomes 81 mm.

Figure 36 shows the lateral and vertical accelerations.

![Lateral and vertical accelerations in the frame over the front axle, curve radius 700 m, speed 88 km/h](image)

Figure 36. Lateral and vertical accelerations in the frame over the front axle, simulation in 88 km/h on curved track, 700 m radius.

In Table 15 the maximum values and RMS values for the accelerations, including the values from the frame over the rear axle and the cabin can be seen.

<table>
<thead>
<tr>
<th></th>
<th>Cabin</th>
<th>Frame over front axle</th>
<th>Frame over rear axle</th>
</tr>
</thead>
<tbody>
<tr>
<td>$y_{max}^*$ [m/s²]</td>
<td>1.72</td>
<td>1.27</td>
<td>0.95</td>
</tr>
<tr>
<td>$z_{max}^*$ [m/s²]</td>
<td>1.54</td>
<td>0.68</td>
<td>0.96</td>
</tr>
<tr>
<td>$y_{rms}^*$ [m/s²]</td>
<td>0.45</td>
<td>0.37</td>
<td>0.27</td>
</tr>
<tr>
<td>$z_{rms}^*$ [m/s²]</td>
<td>0.46</td>
<td>0.22</td>
<td>0.28</td>
</tr>
</tbody>
</table>

Table 15. Maximum accelerations and RMS values for the cabin and frame over front and rear axle, simulations in 88 km/h on curved track, 700 m radius.
The sum of lateral forces for both axles can be seen in Figure 37.

![Figure 37. Sum of Y forces for both axles, simulation in 88 km/h on curved track, 700 m radius](image)

Figure 38 shows the Y/Q quotient for both front axle wheels.

![Figure 38. Y/Q for both front axle wheels, simulation in 88 km/h on curved track, 700 m radius](image)
Figure 39 shows the $Y/Q$ quotient for both rear axle wheels.

![Quotient Y/Q rear axle, curve radius 700 m, speed 88 km/h](image)

**Figure 39.** $Y/Q$ for both rear axle wheels, simulation in 88 km/h on curved track, 700 m radius.

The maximum sum of $Y$ forces, RMS of sum of $Y$ forces and $Y/Q$ quotient can be seen in Table 16.

<table>
<thead>
<tr>
<th>$\sum Y_{\text{max}}$ [kN]</th>
<th>11.4</th>
<th>Front axle</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\sum Y_{\text{rms}}$ [kN]</td>
<td>4.78</td>
<td>Front axle</td>
</tr>
<tr>
<td>$(Y/Q)_{\text{max}}$ [-]</td>
<td>0.41</td>
<td>Front outer wheel</td>
</tr>
</tbody>
</table>

**Table 16.** Maximum sum of $Y$ forces, RMS of sum of $Y$ forces and quotient $Y/Q$, simulation 88 km/h on curved track, 700 m radius.

### 6.3 Test zone 3, small radius curve

For test zone 3 a curve with 400 m radius and cant at 145 mm was used. The vehicle speed for the simulations was 88 km/h which is the upper limit for speed in the test zone. With this speed the cant deficiency becomes 83 mm.

The lateral and vertical accelerations can be seen in Figure 40.
The maximum values for the accelerations and their RMS values, including values from the cabin and the frame over the rear axle, can be seen in Table 17.

<table>
<thead>
<tr>
<th></th>
<th>Cabin</th>
<th>Frame over front axle</th>
<th>Frame over rear axle</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\ddot{y}_{\text{max}}$ [m/s²]</td>
<td>1.42</td>
<td>1.04</td>
<td>1.81</td>
</tr>
<tr>
<td>$\ddot{z}_{\text{max}}$ [m/s²]</td>
<td>1.55</td>
<td>0.63</td>
<td>0.91</td>
</tr>
<tr>
<td>$\ddot{y}_{\text{rms}}$ [m/s²]</td>
<td>0.44</td>
<td>0.37</td>
<td>0.59</td>
</tr>
<tr>
<td>$\ddot{z}_{\text{rms}}$ [m/s²]</td>
<td>0.47</td>
<td>0.22</td>
<td>0.29</td>
</tr>
</tbody>
</table>

Table 17. Maximum accelerations and RMS values for the cabin and frame over front and rear axle, simulations in 88 km/h on curved track, 400 m radius.

In Figure 41 the sum of the $Y$ forces for both axles can be seen.
The \( Y/Q \) quotient for both front axle wheels can be seen in Figure 42.

![Figure 42. \( Y/Q \) for both front axle wheels, simulation in 88 km/h on curved track, 400 m radius.](image)

The \( Y/Q \) quotient for both rear axle wheels can be seen in Figure 43.

![Figure 43. \( Y/Q \) for both rear axle wheels, simulation in 88 km/h on curved track, 400 m radius.](image)

Table 18 shows the maximum values for the sum of \( Y \) forces, RMS of sum of \( Y \) forces and the \( Y/Q \) quotient.

<table>
<thead>
<tr>
<th>( Y_{\text{max}} ) [kN]</th>
<th>12.2</th>
<th>Rear axle</th>
</tr>
</thead>
<tbody>
<tr>
<td>( Y_{\text{rms}} ) [kN]</td>
<td>5.83</td>
<td>Front axle</td>
</tr>
<tr>
<td>( (Y/Q)_{\text{max}} ) [-]</td>
<td>0.51</td>
<td>Front outer wheel</td>
</tr>
</tbody>
</table>

Table 18. Maximum sum of \( Y \) forces, RMS of sum of \( Y \) forces and quotient \( Y/Q \), simulation 88 km/h on curved track, 400 m radius.
6.4 Test zone 4, very small radius curve

For test zone 4 a curve with 300 m radius and cant at 150 mm was used. The vehicle speed for the simulations was 80 km/h. With this speed the cant deficiency becomes 102 mm.

The lateral and vertical accelerations can be seen in Figure 44.

![Lateral and vertical accelerations in the frame over the front axle, curve radius 300 m, speed 80 km/h](image)

Figure 44. Lateral and vertical accelerations in the frame over the front axle, simulation in 80 km/h on curved track, 300 m radius.

Table 19 shows the maximum values for the accelerations and their RMS values, including accelerations from the frame over the rear axle and the cabin.

<table>
<thead>
<tr>
<th></th>
<th>Cabin</th>
<th>Frame over front axle</th>
<th>Frame over rear axle</th>
</tr>
</thead>
<tbody>
<tr>
<td>( y_{max} ) [m/s²]</td>
<td>1.52</td>
<td>1.05</td>
<td>2.06</td>
</tr>
<tr>
<td>( z_{max} ) [m/s²]</td>
<td>1.68</td>
<td>0.56</td>
<td>0.88</td>
</tr>
<tr>
<td>( y_{rms} ) [m/s²]</td>
<td>0.41</td>
<td>0.36</td>
<td>0.67</td>
</tr>
<tr>
<td>( z_{rms} ) [m/s²]</td>
<td>0.44</td>
<td>0.19</td>
<td>0.27</td>
</tr>
</tbody>
</table>

Table 19. Maximum accelerations and RMS values for the cabin and frame over front and rear axle, simulations in 80 km/h on curved track, 300 m radius.
Figure 45 shows the sum of the $Y$ forces for the front and rear axle.

![Sum of guiding forces both axles, curve radius 300 m, speed 80 km/h](image)

**Figure 45.** Sum of $Y$ forces for both axles, simulation in 80 km/h on curved track, 300 m radius.

Figure 46 shows the $Y/Q$ quotient for both front axle wheels.

![Quotient $Y/Q$ front axle, curve radius 300 m, speed 80 km/h](image)

**Figure 46.** $Y/Q$ for both front axle wheels, simulation in 80 km/h on curved track, 300 m radius.
Figure 47 shows the $Y/Q$ quotient for both rear axle wheels.

![Quotient Y/Q rear axle, curve radius 300 m, speed 80 km/h](image)

Figure 47. $Y/Q$ for both rear axle wheels, simulation in 80 km/h on curved track, 300 m radius.

In Table 20 the maximum sum of $Y$ forces, RMS of sum of $Y$ forces and $Y/Q$ quotient can be seen.

<p>| | | | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>$\sum Y_{\text{max}}$ [kN]</td>
<td>15.6</td>
<td>Rear axle</td>
<td></td>
</tr>
<tr>
<td>$\sum Y_{\text{rms}}$ [kN]</td>
<td>6.94</td>
<td>Front axle</td>
<td></td>
</tr>
<tr>
<td>$(Y/Q)_{\text{max}}$ [-]</td>
<td>0.55</td>
<td>Front outer wheel</td>
<td></td>
</tr>
</tbody>
</table>

Table 20. Maximum sum of $Y$ forces, RMS of sum of $Y$ forces and quotient $Y/Q$, simulation 80 km/h on curved track, 300 m radius.

6.5 Running stability

The running stability of a rail vehicle is affected by the so-called equivalent conicity which is a function of wheel and rail profiles, wheel inside gauge, flange thickness, rail inclination, track gauge and relative lateral displacement [4]. To simulate a higher equivalent conicity the track gauge was tightened and simulations were run on straight track with the same speed used in previous simulations (Section 6.1). The results presented are from simulations where the track gauge was tightened with 8 mm, which is the track gauge where the model started to show running instability. In Figure 48 the lateral and vertical accelerations can be seen.
Figure 48. Lateral and vertical accelerations in the frame over the front axle, simulation in 90 km/h on tangent 8 mm tighter track.

Table 21 shows the maximum values of the accelerations and their RMS values, including values for the cabin and the frame over the rear axle.

<table>
<thead>
<tr>
<th></th>
<th>Cabin</th>
<th>Frame over front axle</th>
<th>Frame over rear axle</th>
</tr>
</thead>
<tbody>
<tr>
<td>$y_{max}$ [m/s²]</td>
<td>2.79</td>
<td>2.32</td>
<td>4.41</td>
</tr>
<tr>
<td>$z_{max}$ [m/s²]</td>
<td>1.92</td>
<td>1.36</td>
<td>1.89</td>
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<tr>
<td>$y_{rms}$ [m/s²]</td>
<td>0.84</td>
<td>0.72</td>
<td>1.35</td>
</tr>
<tr>
<td>$z_{rms}$ [m/s²]</td>
<td>0.49</td>
<td>0.29</td>
<td>0.37</td>
</tr>
</tbody>
</table>

Table 21. Maximum accelerations and RMS values for the cabin and frame over front and rear axle, simulation in 90 km/h on tangent 8 mm tighter track.

With the lateral accelerations showing signs of instability in the vehicle, the wheel-rail forces are of interest to see if the instability can cause a derailment.
Figure 49 shows the sum of the guiding forces for both axles.

![Figure 49. Sum of Y forces for both axles, simulation in 90 km/h on tangent 8 mm tighter track.](image)

In Figure 50 the Y/Q quotient for both front axle wheels can be seen.

![Figure 50. Y/Q for both front axle wheels, simulation in 90 km/h on tangent 8 mm tighter track.](image)
The Y/Q quotient for both rear axle wheels can be seen in Figure 51.

![Figure 51. Y/Q for both rear axle wheels, simulation in 90 km/h on tangent 8 mm tighter track.](image)

The maximum sum of Y forces, RMS of sum of Y forces and Y/Q quotient can be seen in Table 22.

<table>
<thead>
<tr>
<th>Y axis</th>
<th>Value</th>
<th>Location</th>
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<tbody>
<tr>
<td>$\sum Y_{\text{max}}$ [kN]</td>
<td>25.2</td>
<td>Rear axle</td>
</tr>
<tr>
<td>$\sum Y_{\text{rms}}$ [kN]</td>
<td>7.95</td>
<td>Rear axle</td>
</tr>
<tr>
<td>$(Y/Q)_{\text{max}}$ [-]</td>
<td>0.47</td>
<td>Rear right wheel</td>
</tr>
</tbody>
</table>

Table 22. Maximum sum of Y forces, RMS of sum of Y forces and quotient Y/Q, simulation in 90 km/h on tangent 8 mm tighter track.
7 Analysis of EN 14363 simulations

The simulations in Chapter 6, showed that all limit values are fulfilled for all test zones while using track irregularities S22 and with no change to the average track gauge. The results from curve simulations showed that the guiding forces are below the limits, which means that the vehicle handles the curves well which were expected considering the short wheelbase. Acceleration results from the cabin show the same effect as in the verification, namely that the accelerations increase slightly between the frame and the cabin. But given that the model verification showed big differences there, this doesn’t have to be true for the vehicle.

The testing of running stability showed that the vehicle becomes unstable when tightening the track gauge with 8 mm, this would equal to a high equivalent conicity, above what is necessary for the vehicle to handle. Worth noting is that the simulations show that the unstable vehicle wouldn’t exceed the limit values for wheel-rail forces, so it wouldn’t be any risk of derailment.
8 Simulations with changed vehicle data

To test if the accelerations could be lowered some simulations were done where the properties of the suspensions were changed. One change which showed improvements was when the damper constant for the hydraulic dampers was reduced. This change was tested on tangent track and curved track with radius 300 m. The change that was made for the hydraulic dampers was to lower the (high) damper constant with 50%.

8.1 Tangent track

In Figure 52 the lateral and vertical accelerations can be seen, cf. Figure 35.

![Graph of lateral and vertical accelerations](image)

Figure 52. Lateral and vertical accelerations in the frame over the front axle, simulation in 90 km/h on tangent track with the hydraulic damper constant lowered with 50%.

The maximum values and the RMS values and the change in them compared with the unmodified vehicle for the accelerations, including accelerations from the frame over the rear axle and the cabin, can be seen in Table 23.

<table>
<thead>
<tr>
<th></th>
<th>Cabin</th>
<th>Change</th>
<th>Frame over front axle</th>
<th>Change</th>
<th>Frame over rear axle</th>
<th>Change</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\ddot{y}_{max}$ [m/s²]</td>
<td>0.61</td>
<td>-8%</td>
<td>0.46</td>
<td>-30%</td>
<td>0.69</td>
<td>-10%</td>
</tr>
<tr>
<td>$\ddot{z}_{max}$ [m/s²]</td>
<td>1.73</td>
<td>-5%</td>
<td>1.34</td>
<td>+2%</td>
<td>1.54</td>
<td>-8%</td>
</tr>
<tr>
<td>$\ddot{y}_{rms}$ [m/s²]</td>
<td>0.16</td>
<td>-16%</td>
<td>0.14</td>
<td>-18%</td>
<td>0.20</td>
<td>-5%</td>
</tr>
<tr>
<td>$\ddot{z}_{rms}$ [m/s²]</td>
<td>0.38</td>
<td>-17%</td>
<td>0.29</td>
<td>0%</td>
<td>0.30</td>
<td>-12%</td>
</tr>
</tbody>
</table>

Table 23. Maximum accelerations and RMS values for the cabin and frame over front and rear axle, simulation in 90 km/h on tangent track with the hydraulic damper constant lowered with 50%.
8.2 Curved track, radius 300 m

Figure 53 shows the lateral accelerations cf. Figure 44.

![Graph](image)

Figure 53. Lateral and vertical accelerations in the frame over the front axle, simulation in 80 km/h on curved track, radius 300 m and the hydraulic damper constant lowered with 50%.

The maximum values and the change in them compared with the unmodified vehicle for the accelerations, including accelerations from the frame over the rear axle and the cabin, and their RMS values can be seen in Table 24.

<table>
<thead>
<tr>
<th></th>
<th>Cabin</th>
<th>Change</th>
<th>Frame over front axle</th>
<th>Change</th>
<th>Frame over rear axle</th>
<th>Change</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\ddot{y}_{max}$ [m/s²]</td>
<td>1.41</td>
<td>-7%</td>
<td>1.06</td>
<td>0%</td>
<td>1.89</td>
<td>-8%</td>
</tr>
<tr>
<td>$\ddot{z}_{max}$ [m/s²]</td>
<td>1.17</td>
<td>-30%</td>
<td>0.52</td>
<td>-7%</td>
<td>0.77</td>
<td>-12%</td>
</tr>
<tr>
<td>$\ddot{y}_{rms}$ [m/s²]</td>
<td>0.43</td>
<td>+5%</td>
<td>0.39</td>
<td>+8%</td>
<td>0.63</td>
<td>-6%</td>
</tr>
<tr>
<td>$\ddot{z}_{rms}$ [m/s²]</td>
<td>0.39</td>
<td>-11%</td>
<td>0.18</td>
<td>-5%</td>
<td>0.24</td>
<td>-11%</td>
</tr>
</tbody>
</table>

Table 24. Maximum accelerations and RMS values for the cabin and frame over front and rear axle, simulation in 80 km/h on curved track, radius 300 m and the hydraulic damper constant lowered with 50%.
In Figure 54 the sum of guiding forces for both axles are shown.

![Figure 54](image)

Figure 54. Sum of Y forces for both axles, simulation in 80 km/h on curved track, radius 300 m and the hydraulic damper constant lowered with 50%.

The Y/Q quotient for both front axle wheels can be seen in Figure 55.

![Figure 55](image)

Figure 55. Y/Q for both front axle wheels, simulation in 80 km/h on curved track, radius 300 m and the hydraulic damper constant lowered with 50%.
Figure 56 shows the $Y/Q$ quotient for both rear axle wheels.

![Figure 56. $Y/Q$ for both rear axle wheels, simulation in 80 km/h on curved track, radius 300 m and the hydraulic damper constant lowered with 50%.](image)

The maximum sum of $Y$ forces, RMS of sum of $Y$ forces and $Y/Q$ quotient and the change in them compared with the unmodified vehicle can be seen in Table 25.

<table>
<thead>
<tr>
<th></th>
<th>Value</th>
<th>Change</th>
<th>Axle/Wheel</th>
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</thead>
<tbody>
<tr>
<td>$\sum Y_{max}$ [kN]</td>
<td>14.2</td>
<td>-9%</td>
<td>Rear axle</td>
</tr>
<tr>
<td>$\sum Y_{rms}$ [kN]</td>
<td>6.98</td>
<td>+1%</td>
<td>Front axle</td>
</tr>
<tr>
<td>$(Y/Q)_{max}$ [-]</td>
<td>0.55</td>
<td>0%</td>
<td>Rear inner wheel</td>
</tr>
</tbody>
</table>

Table 25. Maximum sum of $Y$ forces, RMS of sum of $Y$ forces and quotient $Y/Q$, simulation in 80 km/h on curved track, radius 300 m and the hydraulic damper constant lowered with 50%.
9 Discussion and conclusions

The first conclusion that has to be drawn from the present work is that the simulation model is only partly verified and that all simulations that are made to test the requirements from the EN14363 standard are made with the assumption that the model gives a close to reality representation of the vehicle. Even if the simulations had shown similar results to the test in the cabin in the verification the model still wouldn’t have been fully verified. A complete verification would require that the same track design geometry and track irregularities are used in the test and simulations. It would also require a test performed at the vehicle maximum speed since it can’t be assumed that a model verified for 30 km/h also is verified for 80 km/h. And also test equipment that can handle a higher sampling frequency would be needed.

When looking at the simulations that are made following the standard they show that the vehicle with its current configuration probably would pass the authority approval. The testing of running stability showed that the vehicle handles high equivalent conicity without becoming unstable, and when instability occurred the wheel-rail forces were still under their respective limit values. The increase in equivalent conicity was achieved by tightening the track gauge, while the wheel and rail had the nominal profiles. If the wheel and rail profiles had been worn, the instability would occur with track gauge less tightened than what the test showed. If running instability would become a problem the easiest solution would be to increase the longitudinal stiffness in the primary suspension. This would however lead to higher wheel-rail forces in curves since it would be harder for the wheels to follow the curves. And even though the wheel-rail forces probably wouldn’t exceed the limit values, so would they lead to higher wear on the wheels.

The simulations where some of the properties of the vehicle suspension were changed showed that there are changes that can be made which can improve the vehicle performance. The change presented with lowering the hydraulic damper constant was one that could relatively easy be done in reality and that showed good results. The reasoning behind that change was that a high damper constant lead to high damper forces even at relatively low piston velocities. Given the vehicle’s relatively low weight this would lead to it being stiff almost always in the damper’s direction. Softening the damper by having a smaller constant could then lead to a better result in the lower frequency range in which the dynamic behaviour is studied. Other changes that were studied but not presented were softer lateral primary stiffness to improve the response for lateral accelerations in the frame but no change that gave a noticeable result was found.

Worth noting is that the S22 track irregularities were used for all simulations in higher speed. And while these irregularities can be seen as realistic in test zone 1 and 2, so would the irregularities likely be larger in the tighter curves used for test zone 3 and 4.
10 Future work

To be able to use the simulation model for an authority approval of the vehicle there is some further work that has to be done.

As a first step, the reason why the verification test showed so low levels in measured accelerations in the cabin should be investigated. The structure of the cabin needs to be looked at to see if there is a possibility that this could have had an effect. To support this, new measurements in the cabin can be performed, where measurements are done in different positions in the cabin to see if the same results are shown everywhere or are just shown in the area where the measurements in this work was done.

The assumption that the difference in accelerations in the measurements between the frame over the front and rear axles was caused by the vehicle being coupled to another vehicle also needs to be confirmed. This could be done by performing new measurements in which the vehicle is running on its own power. These measurements as well as the ones in the cabin could be performed in the same way as the test in this work since they would not be done for complete model verification but only to verify assumptions made in this work.

To verify the model completely, which would be needed if it shall be used for authority approval, a test where the vehicle is running at its maximum speed would be needed. And for the verification the track design geometry and track irregularities on which the test is performed are needed as input in the simulations.

When the model is fully verified, simulations on track geometries and with vehicle speeds following the standard can be performed. The verification of the model has to be audited by an independent third-party reviewer. With the simulations an examination of the ride comfort in the vehicle could also be done.
11 Bibliography


Appendix A, Vehicle test plan

Presented here is the test plan, which was compiled prior to the vehicle test.

A1. Vehicle test plan

- Accelerations \([\text{m/s}^2]\) are measured with smartphones.
- All measuring points are measured 3 times in both travel directions.
- While measuring the vehicle shall hold a constant speed.
- Speeds used in testing are 15 km/h and 30 km/h.
- The vehicle is pulled or pushed by another vehicle.
- One measurement takes about 15 min with preparations, measuring and control of measured data. When switching measuring points longer time might be necessary.
- The measuring app is activated while the vehicle is standing still and at the same time the stopwatch is started. Then intermediate times are taken when the start and end of the test zone is reached.
- The test is estimated to take two days; the table below shows the estimated schedule.

<table>
<thead>
<tr>
<th>Day</th>
<th>Activity</th>
<th>Start</th>
<th>Estimated time</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Estimated arrival to Falköping</td>
<td>09:00</td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>Preparations</td>
<td>10:00</td>
<td>2h</td>
</tr>
<tr>
<td>1</td>
<td>Lunch</td>
<td>12:00</td>
<td>1h</td>
</tr>
<tr>
<td>1</td>
<td>Testing, measurement point 1</td>
<td>13:00</td>
<td>3h</td>
</tr>
<tr>
<td>1</td>
<td>Start testing, measurement point 2</td>
<td>16:00</td>
<td>1h</td>
</tr>
<tr>
<td>2</td>
<td>Preparations</td>
<td>7:30</td>
<td>0.5h</td>
</tr>
<tr>
<td>2</td>
<td>Continue testing, measurement point 2</td>
<td>8:00</td>
<td>2h</td>
</tr>
<tr>
<td>2</td>
<td>Start testing, measurement point 3</td>
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<td>2h</td>
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<td>Lunch</td>
<td>12:00</td>
<td>1h</td>
</tr>
<tr>
<td>2</td>
<td>Continue testing, measurement point 3</td>
<td>13:00</td>
<td>1h</td>
</tr>
<tr>
<td>2</td>
<td>Testing, measurement point 4</td>
<td>14:00</td>
<td>3h</td>
</tr>
<tr>
<td>2</td>
<td>Finish</td>
<td>17:00</td>
<td></td>
</tr>
</tbody>
</table>

Needed of BS Verkstäder

- Access to the track during the testing.
- A vehicle to pull/push with.
- A driver.

Conditions

- The vehicle is tested with the crane on.
- The amount of diesel in the tank is 50 l.
- A 230 V AC outlet is needed in one of the two vehicles to power the computer and a router.
Preparations

- Determine the part of the track which is to be used as the test section.
- Verify that everything works with the transferring of measurement data between the phone and computer.
- Determine the measuring points.

Testing order

<table>
<thead>
<tr>
<th>Measuring point</th>
<th>Measuring case</th>
<th>Direction</th>
<th>Speed [km/h]</th>
<th>Time [min]</th>
<th>Filename</th>
<th>Start/Stop time for the test case</th>
<th>Measuring points</th>
</tr>
</thead>
<tbody>
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<td>1</td>
<td>1</td>
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<td>15</td>
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</tr>
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<td>2</td>
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<table>
<thead>
<tr>
<th>Measuring point</th>
<th>Measuring case</th>
<th>Direction</th>
<th>Speed [km/h]</th>
<th>Time [min]</th>
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<th>Start/Stop time for the test case</th>
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Appendix B, Vehicle test and comparison results

Here, results not presented earlier in the report from the vehicle test and the comparison with simulation are presented.

B1. Raw and band-pass filtered measuring data

Figure B1. Lateral and vertical accelerations in the cabin, raw and filtered, 15 km/h going forward, test 1.

Figure B2. Lateral and vertical accelerations in the cabin, raw and filtered, 15 km/h going forward, test 2.
Figure B3. Lateral and vertical accelerations at the frame over the front axle, raw and filtered, 15 km/h going forward, test 1.

Figure B4. Lateral and vertical accelerations at the frame over the front axle, raw and filtered, 15 km/h going forward, test 2.
Figure B5. Lateral and vertical accelerations at the frame over the rear axle, raw and filtered, 15 km/h going forward, test 1.

Figure B6. Lateral and vertical accelerations at the frame over the rear axle, raw and filtered, 15 km/h going forward, test 2.
Figure B7. Lateral and vertical accelerations at the axle box, raw and filtered, 15 km/h going forward, test 1.

Figure B8. Lateral and vertical accelerations at the axle box, raw and filtered, 15 km/h going forward, test 2.
Figure B9. Lateral and vertical accelerations in the cabin, raw and filtered, 15 km/h going backward, test 1.

Figure B10. Lateral and vertical accelerations in the cabin, raw and filtered, 15 km/h going backward, test 2.
Figure B11. Lateral and vertical accelerations at the frame over the front axle, raw and filtered, 15 km/h going backward, test 1.

Figure B12. Lateral and vertical accelerations at the frame over the front axle, raw and filtered, 15 km/h going backward, test 2.
Figure B13. Lateral and vertical accelerations at the frame over the rear axle, raw and filtered, 15 km/h going backward, test 1.

Figure B14. Lateral and vertical accelerations at the frame over the rear axle, raw and filtered, 15 km/h going backward, test 2.
Figure B15. Lateral and vertical accelerations at the axle box, raw and filtered, 15 km/h going backward, test 1.

Figure B16. Lateral and vertical accelerations at the axle box, raw and filtered, 15 km/h going backward, test 2.
Figure B17. Lateral and vertical accelerations in the cabin, raw and filtered, 30 km/h going forward, test 1.

Figure B18. Lateral and vertical accelerations in the cabin, raw and filtered, 30 km/h going forward, test 2.
Figure B19. Lateral and vertical accelerations in the cabin, raw and filtered, 30 km/h going forward, test 3.

Figure B20. Lateral and vertical accelerations at the frame over the front axle, raw and filtered, 30 km/h going forward, test 1.
Figure B21. Lateral and vertical accelerations at the frame over the front axle, raw and filtered, 30 km/h going forward, test 2.

Figure B22. Lateral and vertical accelerations at the frame over the front axle, raw and filtered, 30 km/h going forward, test 3.
Figure B23. Lateral and vertical accelerations at the frame over the rear axle, raw and filtered, 30 km/h going forward, test 1.

Figure B24. Lateral and vertical accelerations at the frame over the rear axle, raw and filtered, 30 km/h going forward, test 2.
Figure B25. Lateral and vertical accelerations at the frame over the rear axle, raw and filtered, 30 km/h going forward, test 3.

Figure B26. Lateral and vertical accelerations at the axle box, raw and filtered, 30 km/h going forward, test 1.
Figure B27. Lateral and vertical accelerations at the axle box, raw and filtered, 30 km/h going forward, test 2.

Figure B28. Lateral and vertical accelerations at the axle box, raw and filtered, 30 km/h going forward, test 3.
Figure B29. Lateral and vertical accelerations in the cabin, raw and filtered, 30 km/h going backward, test 1.

Figure B30. Lateral and vertical accelerations in the cabin, raw and filtered, 30 km/h going backward, test 2.
Figure B31. Lateral and vertical accelerations in the cabin, raw and filtered, 30 km/h going backward, test 3.

Figure B32. Lateral and vertical accelerations at the frame over the front axle, raw and filtered, 30 km/h going backward, test 1.
Figure B33. Lateral and vertical accelerations at the frame over the front axle, raw and filtered, 30 km/h going backward, test 2.

Figure B34. Lateral and vertical accelerations at the frame over the front axle, raw and filtered, 30 km/h going backward, test 3.
Figure B35. Lateral and vertical accelerations at the frame over the rear axle, raw and filtered, 30 km/h going backward, test 1.

Figure B36. Lateral and vertical accelerations at the frame over the rear axle, raw and filtered, 30 km/h going backward, test 2.
Figure B37. Lateral and vertical accelerations at the frame over the rear axle, raw and filtered, 30 km/h going backward, test 3.

Figure B38. Lateral and vertical accelerations at the axle box, raw and filtered, 30 km/h going backward, test 1.
Figure B39. Lateral and vertical accelerations at the axle box, raw and filtered, 30 km/h going backward, test 2.

Figure B40. Lateral and vertical accelerations at the axle box, raw and filtered, 30 km/h going backward, test 3.
B2. Frequency response of selected raw measuring data

Figure B41. Frequency response from cabin, 15 km/h going forward.

Figure B42. Frequency response from frame over front axle, 15 km/h going forward.
Figure B43. Frequency response from frame over rear axle, 15 km/h going forward.

Figure B44. Frequency response from axle box, 15 km/h going forward.
Figure B45. Frequency response from cabin, 15 km/h going backward.

Figure B46. Frequency response from frame over front axle, 15 km/h going backward.
Figure B47. Frequency response from frame over rear axle, 15 km/h going backward.

Figure B48. Frequency response from axle box, 15 km/h going backward.
Figure B49. Frequency response from cabin, 30 km/h going backward.

Figure B50. Frequency response from frame over front axle, 30 km/h going backward.
Figure B51. Frequency response from frame over rear axle, 30 km/h going backward.

Figure B52. Frequency response from axle box, 30 km/h going backward.
B3. RMS values of band-pass filtered data for 15 km/h

Figure B53. RMS values of band-pass filtered lateral accelerations, 15 km/h.

Figure B54. RMS values of band-pass filtered vertical accelerations, 15 km/h.
B4. Simulation vs. measurement for 30 km/h, travel direction backwards

Figure B55. Lateral accelerations in cabin, going backward, Simulation vs. Test.

Figure B56. Lateral accelerations at frame over front axle, going backward, Simulation vs. Test.
Figure B57. Lateral accelerations at frame over rear axle, going backward, Simulation vs. Test.

Figure B58. Lateral accelerations at axle box, going backward, Simulation vs. Test.
Figure B59. Vertical accelerations in cabin, going backward, Simulation vs. Test.

Figure B60. Vertical accelerations at frame over front axle, going backward, Simulation vs. Test.
Figure B61. Vertical accelerations at frame over rear axle, going backward, Simulation vs. Test.

Figure B62. Vertical accelerations at axle box, going backward, Simulation vs. Test.
Appendix C, Simulation results

Here, additional results from simulations according to EN 14363 and simulations with changed vehicle data, for the cabin and frame over the rear axle, are presented.

C1. Accelerations for the cabin and the frame over the rear axle, test zone 1.

![Figure C1. Lateral and vertical accelerations in the cabin, simulation in 90 km/h on tangent track.](image1)

![Figure C2. Lateral and vertical accelerations at the frame over the rear axle, simulation in 90 km/h on tangent track.](image2)
C2. Accelerations for the cabin and the frame over the rear axle, test zone 2.

Figure C3. Lateral and vertical accelerations in the cabin, simulation in 88 km/h on curved track, 700 m radius.

Figure C4. Lateral and vertical accelerations at the frame over the rear axle, simulation in 88 km/h on curved track, 700 m radius.
C3. Accelerations for the cabin and the frame over the rear axle, test zone 3.

Figure C5. Lateral and vertical accelerations in the cabin, simulation in 88 km/h on curved track, 400 m radius.

Figure C6. Lateral and vertical accelerations at the frame over the rear axle, simulation in 88 km/h on curved track, 700 m radius.
C4. Accelerations for the cabin and the frame over the rear axle, test zone 4.

Figure C7. Lateral and vertical accelerations in the cabin, simulation in 80 km/h on curved track, 300 m radius.

Figure C8. Lateral and vertical accelerations at the frame over the rear axle, simulation in 80 km/h on curved track, 300 m radius.
C5. Accelerations for the cabin and the frame over the rear axle, running stability.

Figure C9. Lateral and vertical accelerations in the cabin, simulation in 90 km/h on tangent 8 mm tighter track.

Figure C10. Lateral and vertical accelerations at the frame over the rear axle, simulation in 90 km/h on tangent 8 mm tighter track.
C6. Accelerations for the cabin and the frame over the rear axle, changed vehicle data.

Figure C11. Lateral and vertical accelerations in the cabin, simulation in 90 km/h on tangent track with the hydraulic damper constant lowered with 50%.

Figure C12. Lateral and vertical accelerations at the frame over the rear axle, simulation in 90 km/h on tangent track with the hydraulic damper constant lowered with 50%.
Figure C13. Lateral and vertical accelerations in the cabin, simulation in 80 km/h on curved track, radius 300 m and the hydraulic damper constant lowered with 50%.

Figure C14. Lateral and vertical accelerations at the frame over the rear axle, simulation in 80 km/h on curved track, radius 300m and the hydraulic damper constant lowered with 50%.