Verification of Simulated Wheel-Rail Forces with Measured Data

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Verification of simulated wheel-rail forces with measured data

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Preface and acknowledgements

This thesis represents the final part of my pursuit toward a degree in Master of Science. It was written during the spring and summer of 2015 for the Department of Aeronautical and Vehicle Engineering, Division of Rail Vehicles at The Royal Institute of Technology (KTH) in Stockholm, and in collaboration with Bombardier Transportation AB in Västerås.

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Marcus Jans Bertilsson
Abstract

The purpose of this thesis was to verify the simulated wheel-rail forces from the simulation software Simpack by comparing the results with on-track measurements. The report constitutes a literature study on wheel-rail contact modelling and techniques for measurements of track forces and track irregularities. Furthermore, it describes the verification procedure itself and how the model was set up, and subsequently presents the results from the actual comparisons. Three versions of Simpack have been compared that utilize different types of contact models. Investigations on alternative ways of filtering track irregularities were conducted, and how the simulated forces were influenced by using worn wheel and rail profiles was also studied.

The method was to realistically model the track sections in nine test cases with varying curve radii, considering the nominal track geometry and track irregularities. A vehicle model for the Regina 250 train was used for the simulations, which were then compared with the corresponding on-track measurements. Time histories and frequency contents were compared.

It was found that the simulated wheel-rail forces correspond well to the measurements, at least for frequencies up to 5-10 Hz, and that no considerable deviations could be found when comparing the contact models. However, the simulation times differed and the discrete contact model introduced in Simpack 9.8 was particularly time demanding.

The results shows that it is beneficial to filter the track irregularities before conducting simulations, since it saves simulation time while retaining an adequate accuracy of the track forces.

The use of worn wheel and rail profiles could in this report not be shown to have any major impact on the wheel-rail forces, which is in contradiction with the hypothesis. However, it should be noted that the scope for this part was narrow and it merits a more thorough investigation.
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Chapter 1

Introduction

1.1 Background

The testing and approval of new railway vehicles has over the years been developed to an extensive and time consuming procedure. Many aspects within the design process are largely depending on multi body simulations, for instance in order to assess and evaluate the dynamic running behaviour of the vehicle in accordance with the requirement specifications. However, the final decision for approval is based on results from on-track test runs, conducted with a vehicle that corresponds to a normal production type. In order to minimize the amount of time and cost of such procedures, it is essential to make accurate predictions in the design process. Consequently, it is very important that the multi body simulation results are realistic.

During on-track tests plenty of measurements are conducted, for instance the accelerations in the carbody and the bogie frame, but also the wheel-rail forces are measured. Standards are used which state different allowed limit values, for the wheel-rail forces there are requirements on for instance lateral and vertical forces Y and Q, as well as the ratio Y/Q. These quantities basically determine the vehicle running behaviour and they affect important issues such as wear, fatigue, safety, comfort and much more. By using multi body simulations it is possible to within reasonable accuracy simulate the wheel-rail forces. The vehicle dynamic department at Bombardier Transportation in Västerås employs the software Simpack for their vehicle dynamics simulations, where the wheel-rail forces are one of the most important result outputs. The latest version of Simpack included among many other updates a new contact model, and before it can be fully utilized by Bombardier it would be valuable to verify its accuracy. Also, in some cases, it has been observed that the wheel-rail forces contains large peaks when using the new contact model. These peak forces is a phenomenon that has not been seen in previous versions, thus an investigation on why they occur would be in order.

1.2 Objectives

The objective for this project is to verify the wheel-rail forces from the Simpack simulations, and if possible determine why the peak forces emerge. The knowledge of the reliability and accuracy of these forces is important since they are a fundamental part of the vehicle acceptance process.

1.3 Methodology

The verification is executed by comparing the simulated wheel-rail forces with measurement data from on-track tests. Such data is available from the Green Train Project, where measurements were conducted using the Regina 250 as the test vehicle. The method here is to use the corresponding vehicle model in Simpack, choose a number of test cases from different track
sections and implement the track geometry and quality as accurately as possible, and then compare the simulation results with the experimental measurements. It is also of interest to compare the difference in results (if there are any) between the previous versions of Simpack and the newest.
Chapter 2

Wheel-rail contact modelling

2.1 Background and history

There are early examples of railway lines, dating back to the 16th century, most of which were specially designed for the transport of different mining products. But the utilisation of railways for public and cargo transport grew rapidly during the 19th century, mainly in Europe and North America. The engineers of the time came across problems with the railway operation, mainly regarding damage to the surface of the wheel and rail, a phenomena that today is referred to as wear. Many of the questions that arose during the early days of railway operations are yet to remain unsolved [11, p. 24], but great attempts were made to solve these issues. Scientists Heinrich Hertz, Frederick W. Carter, Hans Fromm and Joost Kalker et. al. have contributed substantially to our better understanding of the contact mechanics and parts of their theories are still used in the modern engineering of today.

Heinrich Hertz was born in Hamburg in 1857 and began his educational pursuit in Berlin where he studied natural sciences, along with the renowned Helmholtz and Kirchhoff [11, p. 9]. The normal contact problem was one of the first subjects he encountered in his career, and in 1881 he proposed a solution. Hertz is perhaps most known for his contributions within the field of electromagnetic waves, but his paper on the normal contact is what he is mostly known for within Contact Mechanics, as it is the basis of the term Hertz theory.

The tangential contact problem was first discussed by Carter and Fromm in the mid 1920s. Noteworthy, it was not a work of collaboration but they individually obtained similar results in characterising the relation between the creep and creep forces [11, p. 20]. Their solutions differ in the sense that Carter proposed an analytical one, whereas Fromms’ was of numerical nature, and therefore much more complex. A more recent development in the subject has been made by Kalker, whose numerical methods for evaluating the rolling contact are widely used in modern solvers, especially his simplified theory and the implementation FASTSIM.

2.2 Short introduction to the wheel-rail contact situation

The contact between the wheels and rails gives rise to many complex problems related to the vehicle running stability, safety, wear, etc., all of which are somehow dependent on the forces that emerge within the wheel-rail contact area. This is a study that will focus on verifying the simulated wheel-rail forces, therefore a review of the theories that describes them is necessary. But a prerequisite to understanding these theories is to first describe the geometry and the general contact situation, for simplicity, the case of a free wheelset rolling on two rails will be introduced.

The common set up of what normally is referred to as a wheelset, is when the two wheels are stiffly connected to an axle. Historically, wheelsets have been designed in different ways and also today there are small deviances in their design, but the general features of modern
wheelsets are very similar. One of the main tasks for the wheelset is to provide a guiding mechanism for the train as it travels along the railway track. Due to the wheels being stiffly connected to the axle, one could claim that it is designed to only travel on straight tracks. For geometrical reasons, curve negotiation implies that the angular velocity of the outer wheel must be greater than that of the inner wheel, due to the difference in rolling distance, as described in Figure 2.1. The solution is to introduce a conicity on the wheels, which allows a different rolling radius for the left and right wheel depending on the lateral position of the wheelset, see Figure 2.1. When the wheelset approaches a curve, the lateral acceleration acting on it causes a lateral displacement towards the outside of the curve. In turn, the rolling radius of the inner wheel becomes smaller, but larger for the outer wheel. This allows the wheelset to have self-steering capabilities, even though the angular velocity is the same for both wheels. For certain combinations of curve radius and lateral acceleration, the wheelset performs radial steering, i.e. when the rolling radius of both wheels perfectly matches the corresponding rolling distance.

In reality, the ability for a train to conduct radial steering is depending on more parameters than only the curve radius and lateral acceleration. A modern train is typically designed with two bogies per carbody, each bogie comprising two wheelsets. The bogie is designed to be able to rotate in the horizontal plane in relation to the carbody. Each wheelset within the bogie has, by regarding the track as completely stiff, two main degrees of freedom: the lateral displacement \( y \) and the yaw angle \( \alpha \) [8, p. 87], also illustrated in Figure 2.1. The maximum yaw angle of the bogie in combination with the allowed yaw angle of the wheelsets gives a limit on the smallest curve radius that the train can negotiate through and still have radial steering abilities.

![Figure 2.1: The two degrees of freedom for a wheelset](image)

The wheelset yaw angle is limited by using longitudinal springs, and there is a metal stop that prevents too large yaw angles which can occur in extreme running conditions, for instance when running with very high speed or in narrow curves. There are also longitudinal dampers to prevent the hunting motion of the wheelsets, a phenomenon that is inevitable when running on straight track using standard wheelsets with conical wheels. Similarly, lateral springs and dampers are used to limit the displacement and motion of the wheelset in lateral direction.

Today, rail vehicles are running with such high speeds that the lateral acceleration becomes large even for quite large curve radii. Therefore, most curves on the tracks have a designed cant in order to reduce the track plane acceleration, i.e. increasing the normal force contribution on the track while reducing the lateral force. When the speed is too high or the curve too narrow, the lateral acceleration becomes so large that the wheel flange comes into contact with the rail, preventing the train to derail; one of the main purposes of the wheel flange. Equivalently, if the train is running slowly through a curve with a designed cant, the wheel flange prevents the train from falling of the rails. Figure 2.2 shows a typical track cant.
For a certain vehicle speed and curve radius, the cant will completely compensate the lateral acceleration acting on the train, this is called equilibrium cant or equilibrium speed. The equilibrium cant $h_{eq}$ can be expressed as

$$h_{eq} = \frac{2b_0 v^2}{g R}$$

where $g$ is the gravitational acceleration, $v$ is the vehicle speed, $R$ is the curve radius and $2b_0$ is the width between the left and right nominal wheel-rail contact point. Obviously, equation (2.1) is not always fulfilled, resulting in either cant deficiency or cant excess, see equation (2.2), $h_t$ is the track cant.

$$h_d = h_{eq} - h_t$$
$$h_e = h_t - h_{eq}$$

### 2.3 Wheel-rail contact modelling

The previous section briefly described why the wheels and the tracks have been designed as they have, and also shortly touched upon the general wheel-rail contact situation. As implied, all forces that are acting on the vehicle, both vertical and lateral, as well as the tangential forces due to traction and braking, must be transferred through the wheel-rail contact interface [8, p. 88]. This interface is a very small contact patch, typically the size of a thumbnail. Rail vehicles in general are very heavy, particularly locomotives which can have weights of over 80 tonnes [2], thus the pressure within this small contact patch is often very high. The contact surface (size and shape), the pressure as well as the tangential forces are parameters that must be identified in order to describe the forces within the contact patch, and subsequently the dynamic behaviour of a vehicle. By regarding the rail and the wheel as two quasi-identical bodies, i.e. that the relationship between the shear modulus $G$ and Poisson’s ratio $\nu$ of the two bodies can be expressed

$$\frac{G_1}{1 - 2\nu_1} = \frac{G_2}{1 - 2\nu_2}$$

then according to [17], it can be assumed that the total normal deformation of the contact interface will not be affected by the tangential forces. Subsequently, this yields the possibility to first identify the contact patch and pressure separately, followed by the tangential forces.
CHAPTER 2. WHEEL-RAIL CONTACT MODELLING

1. The normal contact (Hertz’s theory)
2. The tangential contact (Kalker’s theory)

The following sections will describe these theories, as they are required in order to gain sufficient understanding of how wheel-rail contact models are used in simulations, and also to highlight their possible limitations and approximations.

2.3.1 The normal contact

The solution to the normal problem provided by Hertz in his paper [7] is almost exclusively used today for determining the normal contact forces between two elastic bodies, but his work was actually based on the research by Winkler. Winkler proposed a solution where the contact between two bodies was modelled by having one rigid body resting on an elastic foundation, the so-called Winker bed, seen to the left in Figure 2.3. Hertz was at the time investigating how the optical properties of lenses were affected if the lenses were stacked, but he was not pleased with Winklers’ solution since it was depending on having accurately defined spring stiffness; in fact, it has later been proved that there is no specific stiffness that gives sufficiently correct results [17, p. 9].

![Figure 2.3: The winkler bed, the elastic foundation is represented by springs](image)

Hertz theory is based on the assumptions that the two bodies in contact:

- are linearly elastic, isotropic, homogeneous and have smooth surfaces.
- can be considered half-spaces within the contact area, which requires that the typical dimension of the contact patch is much smaller than that of each body (e.g. radius).
- can be described by quadratic functions, at least the areas of the bodies that are in contact.
- have a constant curvature within the contact patch.

According to Hertz, then if these assumptions hold, it must follow that the contact patch between the bodies has an elliptic shape, that it is flat and that the contact pressure follows a semi-elliptic distribution. As we have seen, the wheelset has two degrees of freedom, therefore the notation for the movement of the wheelset can not be same as for the global reference system of the track ($x$ and $y$). Instead, the notation for the wheelsets’ local reference system is $\xi$ for the direction of travel and $\eta$ for the lateral direction, see the bottom illustration in Figure 2.4. Lets give the first body an index $R$ (the rail) and the second body index $W$ (the wheel), and each body has two principle radii, one in the $\xi - z$ plane and one in the $\eta - z$ plane, see the top illustrations in Figure 2.4, where $r_0$ is the rolling radius. Then the penetration distance $z(\xi, \eta)$ of the bodies can be expressed as

$$z(\xi, \eta) = A\xi^2 + B\eta^2,$$

(2.4)
the constants $A$ and $B$ are given by

$$A = \frac{1}{2} \left( \frac{1}{r_{\eta R}} - \frac{1}{r_{\eta W}} \right), \quad (2.5)$$

$$B = \frac{1}{2} \left( \frac{1}{r_{\xi R}} + \frac{1}{r_{\xi W}} \right) \approx \frac{1}{2r_{\xi}} \approx \frac{1}{2r_0} \quad (2.6)$$

For this specific case of a wheel and rail in contact, the expression for $B$ becomes simplified since the longitudinal radius of the rail is $r_{\xi} \approx \infty$. The pressure distribution is given by

$$\sigma_{\zeta}(\xi, \eta) = \frac{3N}{2A_e} \sqrt{1 - \left( \frac{\xi}{a} \right)^2 - \left( \frac{\eta}{b} \right)^2} \quad (2.7)$$

where $A_e$ denotes the contact ellipse area, $N$ the normal contact force and $a$ and $b$ are the semi-axes. In Figure 2.5 below, a representation of an arbitrary contact patch and the corresponding pressure distribution is shown. Note that the pressure shown is acting in the local vertical direction $\zeta$, and not in the global reference system $z$. Due to geometry, these two directions do not generally align.

Figure 2.4: (Top) The principle radii of the wheel and rail, (Bottom) notation for the reference system of the contact patch [3, p. 8-11]

Figure 2.5: The shape of the contact patch and pressure distribution [2]
CHAPTER 2. WHEEL-RAIL CONTACT MODELLING

The size of the contact patch and the semi-axes $a$ and $b$ are influenced by the geometry of the two bodies, and also the applied force. The semi-axes can be calculated by using the following equations,

\[
a = m \sqrt[3]{\frac{3}{4} \frac{N}{E^*} \frac{1}{A + B}},
\]

(2.8)

\[
b = n \sqrt[3]{\frac{3}{4} \frac{N}{E^*} \frac{1}{A + B}},
\]

(2.9)

where

\[
E^* = \left( \frac{1 - \nu_1^2}{E_1} + \frac{1 - \nu_2^2}{E_2} \right)^{-1}.
\]

(2.10)

The so-called Hertzian coefficients $m$ and $n$ can be found in tables by firstly calculating the parameter $\Theta$, which gives a relation between geometrical properties of the two bodies. It is defined as

\[
\cos \theta = \frac{|A - B|}{A + B},
\]

(2.11)

With the normal contact pressure and the size of the contact patch known, the next step is to treat the tangential forces, which has a major influence on the behaviour of the vehicle.

2.3.2 The tangential contact

In order for any forces to be transmitted between two bodies in rolling contact, there must be slip present between them. This means that in order to transmit the tractive or braking force from the wheels of a vehicle onto the rails, the peripheral velocity at the contact point on the wheels must be greater or smaller than the translational velocity of the vehicle. This slip or sliding velocity is called the creep or creepage, and it consists of three components; longitudinal creep, lateral creep and spin creep [3, p. 8-2], which are defined as

\[
\xi = \frac{v_\xi}{v},
\]

(2.12)

\[
\eta = \frac{v_\eta}{v},
\]

(2.13)

\[
\phi = \frac{\omega}{v}.
\]

(2.14)

Where $v_\xi$ and $v_\eta$ are the velocities in the $\xi$- and $\eta$-direction of the contact patch respectively, see Figure 2.4 for a description of the directions, $\omega$ is the angular velocity around the normal axis of the contact patch and $v$ is the translational velocity of the vehicle. These creepages can also be summed up into the resulting translational creep using

\[
v = \sqrt{v_\xi^2 + v_\eta^2}.
\]

(2.15)

A train is driven by applying a torque to one or more of the wheelsets, this torque will be balanced by the train resistance (load, drag, etc.) and will subsequently lead to longitudinal creep and creep forces. These forces cause stresses in wheel, tensile stresses behind the contact and compressive stresses in front of the contact. The creepage was firstly described in the theories by Carter [4] and Fromm [6], who investigated a case where two cylinders were rolling on each other, a two-dimensional problem, where the assumed creepage was greater than zero. This creepage causes tangential stresses $\tau_x$ within the contact area which start to increase from the leading edge of the contact patch and continues until the maximum limit $\mu \sigma_z$ is reached.
When reaching this limit the sliding zone emerges, and the particles in contact begin to slide relative to each other, from this point the tangential stresses follow the curve $\mu \sigma_z(x)$ towards the trailing edge of the contact point, illustrated in red in Figure 2.6. The left illustration shows $P_{S\tau}$ and $P_{A\tau}$ which are the slip traction and the adhesion traction respectively, and $P_{\tau}$ is the actual traction (or stress).

**Figure 2.6:** Tangential stresses within the contact patch [9, p. 63]

When calculating the tangential forces, some additional assumptions must be regarded besides the ones already mentioned for the solution to the normal problem. These are:

- Mass forces are not considered
- The input data are constant during the time a certain particle is located within the contact area (stationary contact)
- The friction coefficient $\mu$ is constant
- One of the axes in the contact point reference system aligns with the dominating direction of speed

If all of these assumptions hold, it is possible to express the tangential forces in the longitudinal direction $\xi$ and lateral direction $\eta$ respectively as

$$F_{\xi} = -f_{\xi}\mu N,$$
$$F_{\eta} = -f_{\eta}\mu N,$$

(2.16)
(2.17)

Here, the forces are defined as acting on the wheels and positive in the $\xi$- and $\eta$-direction respectively. $f_{\xi}$ and $f_{\eta}$ are coefficients that can take values between -1 and 1, $\mu$ is the friction coefficient and $N$ is the normal load.

J. J. Kalker has made substantial contributions on the topic of solving the tangential creep force problems, and he proposed both linear and non-linear solutions that are still in use today. His theories, along with other commonly used ones, can be seen in Figure 2.7, which shows the relationship between the creep forces and creepage. It can be seen that the relationship is linear only for a small area in the vicinity of the origin, i.e. for small creepages. It is said that Kalker’s linear theory [10, 247-248] is valid under the condition that

$$\text{abs}(v) + \text{abs}(\frac{\phi}{1000}) \lesssim 0.002.$$

(2.18)

In Kalker’s linear theory the creep forces are again divided in two components, $\xi$ and $\eta$, and the spin moment $M_{\phi}$ is treated separately. They are in [3, p. 8-20] expressed as

$$F_{\xi} = -\kappa_{11} v_{\xi},$$

(2.19)
CHAPTER 2. WHEEL-RAIL CONTACT MODELLING

Figure 2.7: The relation between the creep force and creepage for different theories [8, p. 96]

\[ F_\eta = -\kappa_{22}v_\eta - \kappa_{23}\phi, \quad \text{(2.20)} \]

\[ M_\phi = -\kappa_{23}v_\eta - \kappa_{33}\phi \quad \text{(2.21)} \]

where the creep coefficients \( \kappa_{ij} \) are depending on the semi-axes \( a \) and \( b \), the shear modulus \( G \) and the Kalker coefficients \( c_{ij} \) which can all be found in tables. According to Kalker, the linear theory can be used for friction coefficients of around \( \mu = 0.6 \). But in the general case, equation (2.18) is not valid, this can be due to different external factors such as oil or dirt that affects the friction coefficient, but also when the vehicle is running on tracks with small curve radii. For such cases, non-linear theories must be used in order to accurately describe the emerging creep forces.

In the non-linear theory proposed by Kalker, the creep forces depend on a number of variables that must be calculated as a first step. These are

- The resulting creep \( v \) (2.15),
- The direction of the resulting creep \( v \),
- The spin creep \( \phi \),
- Poissons’ ratio \( \nu \) for the wheel and the rail,
- The semi axes and the corresponding ratio \( a/b \),

and when they are found, the general expression for the creep forces in equation (2.16) and (2.17) become

\[ F_\xi = -f_\xi(v_\xi, \phi, a/b, \nu)\mu N, \quad \text{(2.22)} \]

\[ F_\eta = -f_\eta(v_\eta, \phi, a/b, \nu)\mu N. \quad \text{(2.23)} \]

One computational implementation that applies Kalkers’ non-linear theory is called \textit{CONTACT}, but since most of the variables that the coefficients \( f_\xi \) and \( f_\eta \) are depending upon must be recalculated in each integration step, the calculations become highly time consuming. Therefore, the variables are calculated and preprocessed outside the actual simulations, still this theory requires too much time to be used in industry applications where the time factor is of great
interest. Instead Kalker proposed the so called *Simplified theory* [9], where the relationship between the surface deformation and traction is described linearly [17, p. 36] as

\[ \bar{u} = L \bar{\tau}, \] (2.24)

the deformation vector is described as \( \bar{u} = (u_x, u_y, u_z) \) and \( \bar{\tau} = (q_x, q_y, q_z) \) is the traction vector, \( L \) is the flexibility parameter. Kalkers’ simplified theory is similar to Winklers’ theory that was mentioned in section 2.3.1. Like the Winkler bed, the points within the contact surface can in the simplified theory be deformed in any way regardless of how any other near points are deformed, see Figure 2.8. This is because the deformation in one point is only depending on the load in that exact point. It is not useful to apply the simplified theory to the normal contact, since the Hertz theory already provides a quick and proper solution, but when applying it to the tangential contact it yields much faster calculations [17, 36].

**Figure 2.8:** (Left) Hertzian normal solution, (Right) Simplified tangential theory[8, p. 96]

Despite being an approximate solution, it provides solutions with a maximal error or 15%, and its’ corresponding computational algorithm *FASTSIM* is roughly 1000 times faster compared to *CONTACT*.
Chapter 3

Measurement techniques

The goal for this study is to assess the accuracy of simulated wheel-rail forces by comparing the results with on-track measurements, hence, the outcome will depend upon external factors that can not be altered. For instance, the measurement data is used as a reference that should represent the reality, but there is of course a limitation on how accurate such measurements can be. Additionally, the track irregularities that are implemented in the simulation software are also measured using a system with a certain precision. These are factors that will affect the results, and even though it is not possible to alter the data at hand, it is important to be aware of its quality and accuracy.

3.1 Measurement of Track Forces Using Measuring Wheelsets

There is no device available that can directly measure the forces occurring in the wheel-rail interface [3][15-4], partly because of the very finite space in the vicinity of the contact area. Still, the need to evaluate these forces is highly present, especially for the acceptance testing of new or modified rail vehicles, as prescribed in the accepted standards EN14363 CITE and UIC518 CITE. Techniques for estimating the track forces have been developed, Iwnicki et. al. [8][434] state three types.

The H-force Method

The most basic technique is to apply strain gauges between the wheelset axle extension and the axle box. Their relative displacement yields a strain that through the relationship between strain and force gives the H-force, see Figure 3.1. This technique does not regard the wheelset mass force contribution and it is not possible to separate the left and right Y-forces. However, compensation for the wheelset mass force can be made by using accelerometers mounted on the wheelset, which gives a rough estimation of the lateral track shift force $S$. The technique is mainly used for testing low speed vehicles ($< 160$ km/h) or freight wagons ($< 120$ km/h) where it is not obligatory to measure Y- or Q-forces.

Figure 3.1: Schematic view of the H-force [8][435]
CHAPTER 3. MEASUREMENT TECHNIQUES

The Axle Method

This method utilises the relationship between bending moment and force to estimate the wheel forces. The lateral, vertical and longitudinal wheel forces for both left and right wheel can be determined by measuring the moments on six cross sections of the axle. The method suffers from two disadvantages, one being that the wheel-rail contact point moves along the wheel tread depending on the load case, which causes the force application point to change and thus also changing the measured moment. Another being that the moments are affected by the vertical mass forces from the unsprung wheelset. A principal sketch over the measuring points can be seen in Figure 3.2.

![Figure 3.2: Points A-F where the moments are measured using the Axle Method](image)

The Wheel Web Method

The measurements used in the validation and comparison part of this report have been carried out with this technique. Therefore, this method will be described more thoroughly. There are variations of the "wheel methods", one historically common method is the spoked wheel method in which the strains were measured in specially manufactured spoked wheels. Today it has been replaced with the wheel web method, which is very similar but the strains are measured in radial direction on the wheels of a "standard" wheelset. The strain gauges are carefully distributed on the inner and outer surface of the web, and since the gauges are placed only a few centimetres vertically from the wheel-rail interface, the measurements contain most of the dynamic forces from the unsprung mass. High frequent forces of up to at least 100 Hz can be measured, and the accuracy is during normal working conditions around 5-10 % [8][438], provided that the system is well designed and calibrated.

![Figure 3.3: Position of the strain gauges, (L) Y bridge, (M) 1st Q bridge (R) 2nd Q bridge](image)

The output signals must be proportional to the applied loads, which is ensured by connecting the gauges for the Q- and Y-forces to separate Wheatstone bridges. In the measurements conducted
on the Regina 250 train in 2007, three bridges were used per wheel; one for the Y-forces and two for the Q-forces. Figure 3.3 shows the position of the corresponding strain gauges.

There are 20 strain gauges connected to each wheel, 10 on the outer side and 10 on the inner side. The signals are combined into one Q- or Y-signal by connecting the wiring from the gauges to wheatstone bridges. For such a case as for the tests in 2007, where two bridges were used for the Q-forces, the two signals from the Q-bridges would need to be combined as well. Figure 3.4 below shows a picture of a typical strain gauge set up and a connecting scheme for one of the wheatstone bridges.

Additional signal processing is also required to handle parasitic effects due to that the gauges are transmitting data continuously. For instance, the signal from each Q-gauge will be a sinusoidal signal with frequency $\omega$ (the angular velocity of the wheel), but the signal will most likely not wear off until the subsequent signal starts to rise. Therefore, there might be some data overlap that needs to be handled, and the signals must also be calibrated in such a way that the Y- and Q-gauges only transmits Y- respectively Q-forces. The parasitic effects of signal overlapping is illustrated schematically in Figure 3.5 below, for one of the Q-bridges.

**Figure 3.4:** (L) Wheatstone bridge connecting scheme [19], (R) Typical gauge positioning [Courtesy of Lars Andersson, *Interfleet Technology]*

**Figure 3.5:** Parasitic effects
3.2 Measurement of track irregularities

Irregularities are measured with regular intervals in order to monitor the track condition. This is mainly for safety reasons, but also to ensure that a good ride quality is obtained. These are two of the most important factors for the railway as a passenger transport mode: high safety and a fast and comfortable ride. The condition of the track is classified as either $QN_1$, $QN_2$ or $QN_3$ [3], meaning tracks that can be maintained within standard schedule, tracks that need short-term maintenance and tracks that need acute maintenance, respectively. There are also different accepted values for each of the three classes depending on the vehicle speed; the higher speed, the more strict this value is. In this study, track irregularities are needed in order to reproduce the track condition that was present when the on-track measurements were conducted, i.e. to achieve simulations that agree well with reality. This chapter will cover the basic principles that are applied when measuring track irregularities.

![Figure 3.6: The four track irregularity quantities. [3]](image)

3.2.1 Standard for measuring track irregularities, EN13848-1

The Swedish measuring vehicles used by Trafikverket are designed with respect to the guidelines provided in standard EN13848 [13]. It should be noted that the track irregularity data available were measured in 2007, and since then a new version of this standard has been released. In some cases, the format of the available data is therefore not exactly the same as they are today according to EN13848, but the general features of how the measurements are conducted are still the same. Four quantities are usually measured, illustrated in Figure 3.6.

Longitudinal level

The longitudinal level measures the vertical errors of each rail, from a reference line situated a certain distance over the rail head. In Figure 3.7, point 2 refers to the reference point and point 1 to the rail head, and $Z_{pi}$ is the measured value. There are three wavelength ranges defined in which measurements should be done, their specifications can be seen in Table 3.1. The wavelength lower limit of 3 m can be changed to 1 m if short wavelengths should be considered. NB: This is where the data available from 2007 differs from the standards, as we will see later, these are not the ranges that have been used before.
Table 3.1: Specifications for measurement of longitudinal level.

<table>
<thead>
<tr>
<th>Type</th>
<th>Range [m]</th>
<th>Resolution [mm]</th>
<th>Uncertainty [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>D1</td>
<td>$3 &lt; \lambda &lt; 25$</td>
<td>$\leq 0.5$</td>
<td>$\pm 1$</td>
</tr>
<tr>
<td>D2</td>
<td>$25 &lt; \lambda &lt; 70$</td>
<td>$\leq 0.5$</td>
<td>$\pm 3$</td>
</tr>
<tr>
<td>D3</td>
<td>$70 &lt; \lambda &lt; 150$</td>
<td>$\leq 0.5$</td>
<td>$\pm 5$</td>
</tr>
</tbody>
</table>

Figure 3.7: Longitudinal level [13, 12]

Alignment (lateral position)

The alignment of the rails or their lateral position is measured as the deviation $Y_p$ from a reference line located in the center of the track, measured at a distance of $Z_p$ below the running surface, as described in Figure 3.8. Similarly as for the longitudinal level, three wavelength ranges are defined with specifications according to Table 3.2 below.

Table 3.2: Specifications for measurement of the rail alignment.

<table>
<thead>
<tr>
<th>Type</th>
<th>Range [m]</th>
<th>Resolution [mm]</th>
<th>Uncertainty [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>D1</td>
<td>$3 &lt; \lambda &lt; 25$</td>
<td>$\leq 0.5$</td>
<td>$\pm 1.5$</td>
</tr>
<tr>
<td>D2</td>
<td>$25 &lt; \lambda &lt; 70$</td>
<td>$\leq 0.5$</td>
<td>$\pm 4$</td>
</tr>
<tr>
<td>D3</td>
<td>$70 &lt; \lambda &lt; 200$</td>
<td>$\leq 0.5$</td>
<td>$\pm 10$</td>
</tr>
</tbody>
</table>

Figure 3.8: Alignment, lateral position of left and right rail [13, 16]
Gauge

The gauge $G$ is the smallest distance between the left and right rail, and is measured on a horizontal line between the two rails, at a distance of $Z_p = 14$ mm below the running surface. An illustration is shown in Figure 3.9. The gauge is measured as the actual value and not a deviation. No wavelength ranges are stated, since there are no limitations on how small or large errors that can be measured. The resolution is still $\leq 0.5$ mm and the uncertainty $\pm 1$ mm.

![Figure 3.9: Gauge irregularity measurement [13, p. 10]](image)

Cross level (cant)

The cross level or cant can be calculated by measuring the angle between the track plane and a horizontal reference line. This is typically done using a gyroscope. Knowing this angle, it is possible to calculate the cant by using the nominal track gauge plus the width of the rail head as the hypotenuse. The data resolution is $\leq 0.5$ mm and the uncertainty $\pm 5$ mm.

![Figure 3.10: Cross level, cant [13, 14]](image)

### 3.2.2 Frequency content in measured track irregularity signals

The data resolutions and uncertainties for the four measured quantities are typically within limits of what can be regarded as sufficient. However, the resolution differs between the wavelength ranges in for instance the longitudinal level. But, as previously mentioned, the available data has not the exact format as defined in EN13848, but 1-25 m, 1-70 m and 1-140 for the longitudinal level and lateral alignment. However, no information regarding the resolution or uncertainty could be found.

The question arose if there were different resolutions for the three wavelength ranges. Since all of them have the lower wavelength limit of 1 m, but different upper limits, then it could be...
possible that the resolution for especially the short wavelength could be worse for the signals that cover the entire wavelength span 1-140 m. Some samples of the track data available were evaluated by looking at the Power Spectral Density (PSD) of each of the four quantities. As an example, Figure 3.11 shows the original signals from each of the three wavelength ranges. It can be clearly seen that none of the longer wavelengths are picked up in the first wavelength range.

![Longitudinal level (original signals)](image)

Figure 3.11: Original signals showing the difference between the wavelength ranges

It is however more difficult to directly conclude anything regarding the resolution, for that purpose it is more suitable to look at the PSD of the signals. Figure 3.12 shows the PSD for the corresponding signals, it can be seen that the resolution seems to be almost exactly the same regardless of the wavelength range that was used. Since it is desired to reproduce irregularities of all wavelengths, then if the resolution would differ that would require the signals from the different ranges to be combined in some way, but this shows that that will not be necessary. The signals for the entire wavelength of 1-140 m will be used throughout.
3.3 Measurement of wheel and rail profiles

Due to wear, the shape of the wheels and rails are changing continuously as the vehicle is operating. As for the track irregularities, the shape of the wheels and rails are measured with scheduled intervals and there are limits regarding how much wear that is allowed. The wheel and rail profiles are worn differently, and the amount of wear depends on the operating conditions as well as on the type of track. The rails are most prone to wear in turn-outs and tight curves. There is obviously rail wear on tangent track as well, but in that case the wear is mostly subject to the top of the rail leading to a flattened rail head, which is not as critical as the type of wear that emerges in curves.

Figure 3.13: A typical appearance of worn rails, showing the difference between the inner and outer rail [3, p. 10-10]

In a narrow curve or a curve where the vehicle is passing with a high cant deficiency, the outer wheel flange will come into contact with the outer rail, and only the outermost part of the inner wheel tread will be in contact with the inner rail. This leads to a drastically different contact.
situation between left and right wheel, and correspondingly a great difference in force and creep. A typical appearance of worn rail profiles in a curve can be seen in Figure 3.13. The wheels are worn much faster than the rails, since the rails are only worn during the specific instance when a vehicle passes over it whereas the wheels are constantly worn. When the wheels have reached the limit for allowed wear, they can be re-profiled a number of times before the entire wheelset must be replaced. A desired lifetime before reprofiling is around 300 000 - 600 000 km [2, p. 4:33]. The typical worn shape of a wheel is shown in Figure 3.14 below.

For the use of worn profiles in dynamics simulations, there is a high requirement on the accuracy of the measured data, the general requirement is around 0.02 mm. Stationary measuring systems such as the MiniProf system or SPAK system can provide such accuracies. The MiniProf system measures the profile by using a magnetic measuring wheel that is connected to two arms with optical rotary encoders [8, p. 447], see figure 3.15. This system can be used to measure both wheel and rail profiles. However, it is not practically suitable to measure the rail profiles continuously over an entire track, since it must be handled manually. The SPAK system, which was used when measuring the wheel profiles used in this report, has a high accuracy but must also be handled manually. New measuring vehicles are equipped with laser systems that measure both the position of the rails but also the shape of the rail profile. This provides the possibility to measure the profiles along an entire track, but the drawback is a reduced accuracy of the measurements. According to [8, 445], one should be cautious when profile data obtained from on-track laser measuring systems is used for dynamics simulations.
Chapter 4

Model validation and verification

The process of validation and verification (V&V) is carried out in every research and industry area where computational simulations are utilized. It is a crucial part of the development, regardless of what is simulated, since it is the only way of accurately knowing to what extent the obtained results are true. The following sections will review the process of verification and validation, and also treat some topics regarding how to compare simulations with measurements.

4.1 Definitions

The topic of (V&V) is perhaps best described by L. E. Schwer, who comprehensibly says that

“The processes of verification and validation are how evidence is collected, and documented, that help establish confidence in the results of complex numerical simulations” [16, p. 245].

The American Society of Mechanical Engineers assigned a Standards Committee with the purpose of developing standards for assessing the credibility and correctness of computational solid mechanics simulations. Their guide, [16], was originally published in 2006 and has been approved by the American National Standards Institute (ANSI). Even though it is mainly intended for use within the field of solid mechanics, it comprised valuable guidelines in how to approach the problem of verifying and validating simulation results. What is the difference between validation and verification? The process-tree illustrated in Figure 4.1 provides a good overview.

In everyday speech, verification and validation are frequently used in incoherent ways. A definition is provided by [16], which states the following:

- **Verification**: The determination of how accurate a computational model represents its mathematical basis.
- **Validation**: The determination of how accurate a computational model represents the real world, in terms of its intended use.

According to these definitions, it could be claimed that the title of this thesis is somewhat ambiguous. When ”comparing simulations with measured data”, the process should perhaps be treated as a validation process and not a verification. On the other hand, a major part in this work constitutes a literature study on wheel-rail contact modelling which was needed to understand and interpret the simulation results. This may justify the use of the word verification, even though very little has been done in assessing the accuracy of the mathematical model itself. The following text will focus on the validation process.

The goal of a validation procedure is to evaluate the predictive capability of the computational model. The simulation results are compared with validation experiments (in this case
on-track tests), and if there is a coherent relationship between the two, then the model is validated for its intended use. Noteworthy, in a normal case, for instance when designing a new train, it is not possible to conduct on-track tests since the train has not yet been produced. The validation in this study is carried out on a train that has been produced multiple times and has been operating since 2007, so the goal here is to provide knowledge to Bombardier of how well their general simulation results corresponds to reality, and not to validate this vehicle in particular. The two most important factors within the validation procedure are:

1. Perform validation experiments for the sole purpose of validating the model at hand.
2. Assess the accuracy by quantifying how well the simulations compare to the experiments.

Regarding the first point, the measurement data that is available was acquired in 2007, however not for the sole purpose of verifying the model, but rather for the acceptance process of the vehicle. Still, the data is very comprehensive and has been conducted in a great variety of running conditions according to pertinent standards, why it must be regarded as sufficiently detailed for use within this study. The second point is a bit more complex, assessing the accuracy is obviously one of main tasks here, but exactly how to compare simulations with measurements is not as evident.

### 4.2 Comparing simulations with on-track tests

Several issues must be considered when comparing simulations with on-track measurements. L-O. Jönsson et. al in [12] provide plenty of useful information to be regarded. They state that in order to obtain comparable simulations, it is not only the vehicle model that is of importance, but also to realistically model the track, both the nominal geometry as well as the irregularities. Additionally, the measurement data must also have a certain quality level if the comparisons are to be of any use. A good representation of the difficulties in comparing simulations with measurements can be seen in Figure 4.2. Problematic features that most often are unknown...
is the adhesion or the friction that was present at the time of the measurements. The friction coefficient is very difficult to measure and in the protocols there is often only a note about the general weather conditions, such as 'dry' or 'wet'. Therefore this parameter has to be assumed. Also, other external factors like wind speeds, temperature, dynamic properties of the track and the condition of the vehicle are parameters that may have an influence on the results. The most important factors to consider according to [12, p. 871] are

- to have an accurate vehicle model,
- to obtain a proper representation of the track geometry and irregularities,
- to use track force measurements with an adequate quality.

Often the vehicle models available from manufacturers are only representing a nominal vehicle, to increase the detail level in the models it is suggested that the model is fitted in relation to measured data for the springs, damper or any other component that will influence the vehicle dynamic behaviour. It is also noted that, especially when comparing simulated track forces, simulations using measured wheel and rail profiles should be evaluated as well. Regarding the actual comparison of the measured and simulated signals, it is advised to check that the track geometry corresponds to the measured result. This can be done by positioning the signals to see if for instance the curve lengths fit by looking at the guiding force Y, which will increase within the curve.

Figure 4.2: The many difficulties in comparing simulations with measurements are often due to the amount of influencing parameters [12, p. 870].
Chapter 5

The verification procedure

The verification procedure and how the actual comparison between simulation and measurement was carried out will be described in this chapter. It will also cover the different test cases that were used and how they were decided upon. But firstly an overview of the vehicle that was used in the validation procedure is given; its technical features but also some background of the development within the Green Train project.

5.1 The Green Train project and the vehicle Regina 250

The Green Train project started in May 2005, it was a collaboration between Bombardier Transportation, The Royal Institute of Technology (KTH) and the Swedish national railway agency (Trafikverket). The purpose was to develop a modern train designed for Nordic conditions, and with the possibility to handle speeds of over 250 km/h (hence the name) on standard Swedish lines while still being track friendly, i.e. low wear levels particularly in curves.

![Figure 5.1: The candidate vehicle Regina 250 in its rightful environment [1]](image)

The standard vehicle Regina 9062, which had already been in service for a few years, was decided to be the vehicle platform for the project. But in order to achieve the goals that were set up, the vehicle had to be modified in many ways. The main challenge was to find a bogie configuration that could achieve good stability on straight track in higher speeds, but at the same time being able to perform radial steering in narrow curves. Generally, there is a conflict between such features. Two bogie configurations were tested: the original configuration from the standard Regina train called ”Medium” and another configuration called ”Soft”, which had optimally configured springs and dampers to achieve better passive radial steering. In this study, the simulation model for the ”Soft” configuration has been used. The general specifications for the
Regina 250 can be found in Table 5.1.

Table 5.1: Vehicle specifications for Regina 250 (X52)

<table>
<thead>
<tr>
<th>Vehicle type</th>
<th>Bombardier Regina 250 (X52)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length</td>
<td>2*26.95 m</td>
</tr>
<tr>
<td>Gross weight</td>
<td>2*60 tonnes</td>
</tr>
<tr>
<td>Axle load</td>
<td>15 tonnes</td>
</tr>
<tr>
<td>Max. permitted operational speed (v_{adm})</td>
<td>250 km/h</td>
</tr>
<tr>
<td>Max. operational cant deficiency (c_{d,adm})</td>
<td>165 mm</td>
</tr>
</tbody>
</table>

5.2 Selection of appropriate test cases

In order to achieve a reliable basis for the process of verifying the simulated track forces, it was decided that a number of test cases should be evaluated. The selection of the test cases to be used is not an entirely intuitive procedure; the measurement data available comes from on-track tests conducted on several hundreds of kilometres of railway line, comprising a great number of curvy and straight sections combined with different vehicle speeds, all of which could be possible to use. For the purpose of investigating the observed peak forces, which seem to emerge arbitrarily, one way could be to simulate over the entire track length for which there are track force measurements available, and then choose to compare the instances where peaks occur. But that would result in extremely time consuming simulations and the amount of result data would most likely require even more time to analyse. Another alternative would be to choose the test cases arbitrarily, but that would not be appropriate for a technical investigation since it does not correlate with good scientific code.

O. Polach et al state in [14] a number of criteria that should be fulfilled when conducting validations using on-track test measurements. For instance, they claim that the test sections used should represent all 4 test zones defined in standard EN14363 [18]. Therefore, in order to obtain a qualitative selection, it was decided to select the test cases according to the guidelines in EN14363, but also UIC code 518 [5] regarding on-track tests (which provides an additional zone). Note that these two standards are typically very similar. Since the measurements available were used in the acceptance process of the modified vehicle Regina 250, it should not be difficult to find such test cases. An additional requirement was that the nominal track geometry found in databases from Trafikverket must not be dated later than the time when the on-track measurements were conducted. This data is used to build the track geometry in the simulation software, so if the track has been maintained or modified, that will induce errors when comparing the simulations with measurements.

5.2.1 Guidelines according to EN14363 and UIC518

Both of the standards EN14363 and UIC 518 state plenty of requirements that should be taken into account when conducting on-track tests for the purpose of vehicle acceptance testing. The essential quantities used for this study are those regarding the curve radii, cant deficiency, vehicle speed and curve length. By compiling the relevant requirements, it was possible to reduce the number of possible test cases and a proper selection could be done. The compiled information can be seen in Table 5.2. Additionally, the standards also propose that the vehicle should be running with a constant speed through the curves. Note that the columns containing "section length" are in the standards referring to the minimum length of the parts that can be used within a certain curve. They are used to obtain an adequate statistical basis needed for the vehicle acceptance process, so instead of using only one curve,
Table 5.2: Compilation of requirements according to EN14363 and UIC518

<table>
<thead>
<tr>
<th>Radii ($R$)</th>
<th>Speed ($v$)</th>
<th>Cant deficiency ($c_d$)</th>
<th>Section length ($l$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>250-400 m</td>
<td>$v \leq 1.1v_{adm}$</td>
<td>$0.75c_{d,adm} \leq c_d \leq 1.1c_{d,adm}$</td>
<td>70 m</td>
</tr>
<tr>
<td>400-600 m</td>
<td>$v \leq 1.1v_{adm}$</td>
<td>$0.75c_{d,adm} \leq c_d \leq 1.1c_{d,adm}$</td>
<td>100 m</td>
</tr>
<tr>
<td>600-900 m</td>
<td>N/A</td>
<td>$0.75c_{d,adm} \leq c_d \leq 1.1c_{d,adm}$</td>
<td>250 m</td>
</tr>
<tr>
<td>Large / very</td>
<td>$v_{adm} \leq v \leq 1.1v_{adm}$</td>
<td>$c_d \leq 40$ mm</td>
<td>500 m</td>
</tr>
<tr>
<td>straight track</td>
<td>/ \ max(1.1v_{adm}, v_{adm} + 10$km$/h)</td>
<td>$c_d \leq 40$ mm</td>
<td></td>
</tr>
</tbody>
</table>

it can be divided into multiple sections if it is sufficiently long. In this study, no statistical calculations will be made, but the requirement of the section length will be used as the minimum length of the curve to use as a test case.

5.2.2 Selected test cases

By using the guidelines provided by the information tabulated in Table 5.2, a number of test cases could be chosen. The procedure was to investigate the measurement data using SiView, which is a software developed by Interfleet used to analyse measurement data. By looking at the guiding force $Y$, interesting curves could be found, these were then looked up in the Trafikverket database to check that the curve radius and length suits some of the 5 zones and that the geometry had not been altered since the measurement took place. The next step was to check that the vehicle speed was constant during the curve negotiation and that it fits within the prescribed range. That last point was to calculate the cant deficiency using equations (2.1) and (2.2), shown in section 2.2, and if that also was within the allowed range, that curve would be used as a test case.

This resulted in the selection of 9 test cases, ranging from very narrow curves with high cant deficiency up to large radius and also straight track. The test cases are shown in Table 5.3 below. NB., it was not easy to find test cases where all parameters according to the previous Table 5.2 were fulfilled. Therefore, some of the test cases are not in full compliance with the guidelines, but close.

Table 5.3: The selected test cases

<table>
<thead>
<tr>
<th>#</th>
<th>$R$ [m]</th>
<th>$h_t$ [mm]</th>
<th>$v$ [km/h]</th>
<th>$c_d$ [mm]</th>
<th>$l$ [m]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>280</td>
<td>145</td>
<td>86</td>
<td>166</td>
<td>189</td>
</tr>
<tr>
<td>2</td>
<td>411</td>
<td>135</td>
<td>100</td>
<td>152</td>
<td>226</td>
</tr>
<tr>
<td>3</td>
<td>589</td>
<td>130</td>
<td>120</td>
<td>158</td>
<td>216</td>
</tr>
<tr>
<td>4</td>
<td>645</td>
<td>135</td>
<td>119</td>
<td>124</td>
<td>89</td>
</tr>
<tr>
<td>5</td>
<td>1000</td>
<td>150</td>
<td>169</td>
<td>187</td>
<td>1282</td>
</tr>
<tr>
<td>6</td>
<td>3206</td>
<td>60</td>
<td>256</td>
<td>181</td>
<td>709</td>
</tr>
<tr>
<td>7</td>
<td>4994</td>
<td>40</td>
<td>263</td>
<td>123</td>
<td>1915</td>
</tr>
<tr>
<td>8</td>
<td>10000</td>
<td>30</td>
<td>273</td>
<td>58</td>
<td>145</td>
</tr>
<tr>
<td>9</td>
<td>Straight track</td>
<td>0</td>
<td>274</td>
<td>(18)</td>
<td>(18)</td>
</tr>
</tbody>
</table>
that could be exported by SiView while still containing the curve of interest.
Chapter 6

The Simpack vehicle model

The simulations are a main part of both the results and work behind them within this project, thus, it is essential to provide the reader information regarding the simulation software in general, and specifically about the vehicle model that has been used. These areas will be handled in the following sections. Apart from the main objective, which is to evaluate the accuracy of the simulations in general, it is also of interest to assess if and how different changes to the model parameters affect the results. There are plenty of parameters that would be interesting to investigate to see if the simulation results could be improved, but due to the limited amount of time, the scope for testing different parameters had to be restricted. Therefore, the parameters that were investigated and also how they have been altered will be covered in this chapter as well, this will ease the understanding of the results presented in Chapter 7.

6.1 About Simpack

Simpack is a multi-body simulation (MBS) software that can be used for a range of applications; airplanes, wind turbines, road vehicles and rail vehicles to mention a few. At Bombardier in Västerås it is used to simulate the dynamic properties of fully assembled rail vehicles, but also to calculate the eigenfrequencies of specific components or conduct wear calculations. The vehicle models are produced by creating mass elements (bodies) that represent different equipment, for instance the wheelsets or the carbody. The bodies can be chosen as either rigid or flexible. In the graphical user interface, these bodies can be illustrated with different geometric shapes, however, Simpack treats the rigid bodies as point masses. The mass and inertia of the bodies can be automatically calculated or manually inserted. When using flexible bodies, it is possible to import a body from a CAD file. The bodies are then connected to each other by assigning
joints and producing force elements to represent springs and dampers. Constraints are used to limit the movement of the bodies.

6.2 Vehicle model - Bombardier Regina 250

The vehicle model that has been used in this study was provided by Bombardier Transportation, comprising one carbody that is connected to two bogies by an air spring. The secondary suspension level has springs and dampers in the vertical, lateral and longitudinal direction, as well as in yaw and roll, see Table 6.1 for more details.

The primary suspension level consists of two axle boxes per wheelset, connected at the outside of the wheels via the axle journal. The axle boxes are modelled using non-linear springs and dampers, and the axle-boxes themselves are connected to the bogie using linear spring-damper elements. The typical locations of the spring and damper elements in the bogie are illustrated in Figure 6.2, where spring components are represented in red and dampers in yellow.

![Figure 6.2: Graphical representation of one of the bogies in the model](image)

Table 6.1: Secondary suspension level properties in the Simpack model (per bogie)

<table>
<thead>
<tr>
<th>Direction</th>
<th>Properties</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vertical</td>
<td>Two non-linear air springs and two non-linear dampers, connected using linear spring-damper elements in series</td>
</tr>
<tr>
<td>Lateral</td>
<td>Two non-linear dampers, with linear spring-damper elements in series. Also two bumpstops, which are modelled as spring elements with damping</td>
</tr>
<tr>
<td>Longitudinal</td>
<td>Parallely connected spring-damper elements which represent the traction rod</td>
</tr>
<tr>
<td>Yaw</td>
<td>Two linear yaw dampers (one per side), serially connected spring-damper elements. These dampers have a blow-off level, so there is a constant damping up to a specific force and after which the damping decreases.</td>
</tr>
<tr>
<td>Roll</td>
<td>One anti-roll bar, modelled as a linear spring</td>
</tr>
</tbody>
</table>

The model contained an imported CAD file that represented the flexible carbody, this gives
the possibility to calculate the movement of several point of the body during different loading conditions, see Figure 6.1. However, for the study of investigating simulated wheel-rail forces it was decided to only use a rigid carbody.

6.3 Wheel-rail contact models

Three different contact models are evaluated in this study, the background to why and how they differ will be covered here. Bombardier has been using Simpack v.8 for many years, and it has also been previously tested in similar studies where simulations were compared with measurements. However, a few years ago the new v.9 was released but it has so far not been fully implemented as their main simulation tool. This is partly because its accuracy has not yet been tested, but also because an adaptation from one version to another requires time, since many of the projects are spanning over several years. Future projects should be started using Simpack v.9.

The main features regarding the wheel-rail simulations are essentially the same in version 8 and version 9, but the latter involved a major upgrade of the entire software engine and several areas were changed in terms of how models are built up and how different elements relate to each other. For both versions, the calculation steps for each contact patch are done in the following order:

1. evaluation of the profile geometries such as curvatures and profile shapes, and then finding the position of the contact point,
2. evaluation of the creepages,
3. calculating the normal force,
4. calculating the tangential forces and torque.

Test simulations in Simpack v9.7 have shown that peak forces with magnitudes 2-3 times greater than the average peaks would sometimes emerge, which had not been seen in previous versions. This was thought to be a result of calculation errors from when the number of contact points on one wheel changes, for instance from one-point contact to two-point contact. When version 9.8 was released during the spring of 2015, a new contact search algorithm had been implemented that takes into account the actual shape of the contact patch, and not limiting it to have an elliptic shape.

![Shape of the contact patch, load : 37kN](image)

**Figure 6.3:** Representation of the real contact shape compared to an elliptic requirement [20, p. 801]
The algorithm is based on the work presented in [20], but with some modifications. Figure 6.3 shows the conceptual difference when a requirement on having an elliptic shape (Hertz theory) is used compared to using the actual shape (STRIPES). In each integration step, the contact shape is updated, so there should not be any “jumps” between the number of contact points.

To conclude, the three versions of Simpack will be compared: a general assessment by comparing both v8.9 and v.9.7 against the measurements, in order to assess if any particular differences between them can be seen. And also one comparison with v9.8 to assess if the new contact search algorithm has any effect on the sometimes observed peak forces. The contact models that have been used are shown in Table 6.2, which basically use different ways to calculate the normal force. The major difference is in version 9.8, discrete elastic, where the actual contact shape is used and then split into a number of sections. The normal and tangential forces are discretely applied in each section, whereas in the other versions, the forces are applied on the area center of gravity of the entire contact patch.

Table 6.2: Contact models applied in this study

<table>
<thead>
<tr>
<th>Version</th>
<th>Contact model</th>
<th>Properties</th>
</tr>
</thead>
<tbody>
<tr>
<td>8.9</td>
<td>Constraint + quasi elastic</td>
<td>Elliptic contact shape</td>
</tr>
<tr>
<td>9.7</td>
<td>Equivalent elastic</td>
<td>Elliptic contact shape</td>
</tr>
<tr>
<td>9.8</td>
<td>Discrete elastic</td>
<td>Non-elliptic contact shape</td>
</tr>
</tbody>
</table>

6.3.1 General properties

The following Table 6.3 shows the settings that have been used for the contact and the solver in the model. They can be regarded as the standard settings provided by Simpack.

Table 6.3: Contact and solver settings used in the Simpack model

<table>
<thead>
<tr>
<th>Contact settings</th>
</tr>
</thead>
<tbody>
<tr>
<td>Contact search</td>
</tr>
<tr>
<td>Normal force</td>
</tr>
<tr>
<td>Young’s modulus</td>
</tr>
<tr>
<td>Poisson’s number</td>
</tr>
<tr>
<td>Ref. damping</td>
</tr>
<tr>
<td>Friction coeff.</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Solver settings</th>
</tr>
</thead>
<tbody>
<tr>
<td>Integration method</td>
</tr>
<tr>
<td>Sampling rate</td>
</tr>
</tbody>
</table>

6.4 Track irregularities

Track irregularities have a great impact on the track forces, the high frequent dynamic response of the vehicle is tightly connected to the amount and type of track irregularities. And they are of special interest when simulation results are compared with on-track measurements; the
track irregularities must be correctly described in the model in order to achieve a successful representation of the real world.

### 6.4.1 Coordinate systems for track excitations

SIMPACK provides two alternative ways of implementing track irregularities, either rail related or track related. In the rail related type, the vertical and lateral movement of the left and right rail is described by separate input signals. Whereas for the track related type, the vertical and lateral movements are considered the same for both left and right rail, and the gauge and cant irregularities are implemented as separate signals. This difference is schematically described in Figure 6.4.

![Figure 6.4: Alternatives to implement track irregularities in Simpack.](image)

For practical reasons related to the data-format of the track irregularity measurement file from Trafikverket, only the track related type was used in this study. It would have been possible to use the rail related type, which was actually the plan in the beginning, but that required the measurement data to be combined in a clever way. For instance, the measuring vehicle measures vertical errors as a separate signal, and the cant (including nominal value) as another. Therefore, if it is desired to implement both errors, the signals must be combined. Here arises another problem; the cant is realised as a rolling motion of the entire track plane, i.e. it is not equivalent to simply rise the higher rail according to the cant. That would introduce a very strange wheel-rail contact case. In addition, the measured gauge irregularity represents the difference between the nominal gauge and the measure one. Thus again, it is not evident how the lateral rail position signal should be combined with the gauge signal. One way would be to add half of the gauge error to each of the left and right lateral position signal, but that would most likely induce errors. For example, the gauge is usually widened in curves, but it is probable that the higher rail is shifted outwards more than the lower rail is shifted inwards. Summed up, the rail related type seemed to be connected to plenty unnecessary work, hence the track related type was chosen.

### 6.4.2 Filtering

Normally, when carrying out simulations with track irregularities, it is not required to fully reproduce the irregularities corresponding to the track in question. Instead, it is often sufficient to produce artificial track irregularities that statistically corresponds to a certain track condition. There are standards that state the level of frequency content that the irregularity signals need to comprise, and also the corresponding maximum acceleration and force levels that the vehicle must satisfy. For this reason, Bombardier Transportation still utilizes the simulation software GENSYS for most of their handling of track irregularity files. In Gensys there is a module for the sole purpose of creating track irregularity files with different quality criteria, a feature that is not available in Simpack. And since Gensys is a Swedish software, it can directly handle the data-format produced from the Swedish track measuring vehicles. This is obviously very neat as it makes for a simple and quick procedure, but for the purpose of verification where the track
conditions must be reproduced as accurately as possible, it is not desired to tamper the data with unnecessary filtering. It is also likely that some of the peak forces that are arising due to irregularities might be filtered out.

Table 6.4: Simpack excitation signals, standard procedure

<table>
<thead>
<tr>
<th>Input</th>
<th>Measured signal</th>
<th>Wavelengths</th>
<th>Filtering</th>
</tr>
</thead>
<tbody>
<tr>
<td>Lateral $\Delta y$</td>
<td>mean(Lat. L + Lat. R)</td>
<td>1-140 m</td>
<td>No filtering</td>
</tr>
<tr>
<td>Vertical $\Delta z$</td>
<td>mean(Vert. L + Vert. R)</td>
<td>1-140 m</td>
<td>No filtering</td>
</tr>
<tr>
<td>Roll angle $\Delta \phi$</td>
<td>Cant</td>
<td>N/A</td>
<td>High-pass 100 m</td>
</tr>
<tr>
<td>Gauge $\Delta g$</td>
<td>Gauge</td>
<td>N/A</td>
<td>No filtering</td>
</tr>
</tbody>
</table>

Therefore, after analysing the frequency content and resolution of the track irregularity measurements (see 3.2.2), it was decided to use unfiltered signals with the complete wavelength interval (1-140 m). The cant irregularity signal was however high-pass filtered, since it contained the built-in cant as well, but only the cant error was of interest (a decision that will be discussed in the discussion part). It also needed to be converted into roll angle, according to Simpack input function formats, using the equation below.

$$
\Delta \phi = \arcsin \left( \frac{h_t}{2b_o} \right)
$$

(6.1)

To conclude, the lateral and vertical excitation signals in Simpack were the mean value of the measured irregularities for the left and right rail respectively. The roll angle (cant) excitation was calculated using equation (6.1) with the measured cant signal, high-pass filtered at 0.01 [1/m] (corresponding to a wavelength of 100 m). For the gauge excitation, the measured gauge irregularity signal could be used directly. For an overview, see table 6.4.

Table 6.5: Simpack excitation signals, Gensys procedure

<table>
<thead>
<tr>
<th>Input</th>
<th>Measured signal</th>
<th>Wavelengths</th>
<th>Filtering</th>
</tr>
</thead>
<tbody>
<tr>
<td>Lateral $\Delta y$</td>
<td>mean(Lat. L + Lat. R)</td>
<td>1-140 m</td>
<td>Low-pass 2.5 m, high-pass 125 m</td>
</tr>
<tr>
<td>Vertical $\Delta z$</td>
<td>mean(Vert. L + Vert. R)</td>
<td>1-140 m</td>
<td>Low-pass 2.5 m</td>
</tr>
<tr>
<td>Roll angle $\Delta \phi$</td>
<td>Cant</td>
<td>N/A</td>
<td>Low-pass 2.5 m, high-pass 100 m</td>
</tr>
<tr>
<td>Gauge $\Delta g$</td>
<td>Gauge</td>
<td>N/A</td>
<td>Low-pass 2.5 m</td>
</tr>
</tbody>
</table>

For comparative reasons, the test cases were also simulated with track excitations that were produced in Gensys, using its standard filtering settings. The same raw track irregularity data was used, but everything handled via Gensys. A complete description of the calculation steps can be found in the Gensys users manual [15]. A more brief overview can be found in Table 6.5. In the following text, the strategy to not filter track irregularities (Table 6.4) will be referred to as "standard procedure", and the filtering strategy using Gensys (Table 6.5) will be referred to as "Gensys strategy".

6.5 Measured wheel and rail profiles

For most simulations it is sufficient to use nominal unworn profiles, for instance when estimating the quasi-static forces or when conducting comfort evaluations. But the wear on both wheels
and rails affects the shape of the wheel-rail contact patch and also the position of the contact point, which in turn will affect the wheel-rail forces. One hypothesis was that there should be a better correspondence between simulations and measurements by implementing worn profiles.

The wheels of the test vehicle Regina 250 were measured in 2007 using the SPAK system. Figure 6.5 shows the left and right wheel profiles for the first wheelset, which also was the one with the most prominent wear. As can be seen, the wear is not severe, but even small deviations from the nominal profiles is said to possibly have an influence on the wheel-rail forces. For the rail profiles, measured data could only be found for the first three test cases, which luckily were the ones that had the most narrow curves. The rail profile inside the 280 m radius curve (test case 1) is shown in Figure 6.6, which clearly shows a significantly worn profile.

The plan was to conduct simulations with worn wheel and rail profiles on all of the three test cases for which measured rail profiles had been found. However, due to problems when implementing the rail profiles in Simpack, only the first test case could be simulated. The problems occurred when multiple rail profiles were imported in order to simulate over rails with variable profiles, for instance modelling the how the rail profiles vary throughout a curve. The main issue was that the data for the profiles that were used came from on-track laser systems, these systems have a limited accuracy and also provides the profiles as they are rotated in.

![Figure 6.5: The worn shape of the wheels on the first wheelset of the test vehicle Regina 250.](image)

![Figure 6.6: Worn rail profile inside the 280 m radius, test case 1.](image)
space, i.e. not fixed to a certain point in the coordinate system. This becomes a problem since Simpack requires that the origin is located on the top of the rail head, and that the profiles are not rotated. For instance, if a profile is rotated it is possible that the side of the rail aligns with the vertical axis, causing multiple y-values on the same x-value, which is not allowed. There is a built-in function for rotating profiles but it was difficult to make it work.

For test case 1, eleven measured rail profiles were managed to be implemented in Simpack. The variable profiles starts with a nominal profile on the straight track and the first measured profile at the point just before the first transition curve begins. The subsequent profiles were placed at the points where the two transition curves begin and end as well as one in between, the rest were evenly distributed within the circular part of the curve.

6.6 Model imperfections

Since the vehicle model provided by Bombardier represents a nominal vehicle, there were some imperfections compared to the vehicle that was used for the on-track measurements. Some of these could be dealt with but others not, which will be highlighted in this section.

Vehicle total mass and center of mass

First of all, the total mass of the vehicle did not correspond to the weighting reports that were available from the on-track tests, therefore the mass of the carbody in the model was adjusted. This was the reason why the simulations were conducted with a rigid carbody; it was not possible to change the mass when using a flexible body. Secondly, not only did the weighting reports show a lower mass compared to the model, it also showed that the center of mass was skew. Attempts were made to adjust for this skewness, but they were not successful. It seemed that it was not only the carbody center of mass that differed, the skewness must have been affected by other factors as well. For instance, the weighting report showed that the left hand side of the vehicle had a higher load, but it was not a symmetrical left-right load distribution. The difference in load between left and right wheel was larger for second and third wheelset, compared to the first and fourth. This suggests that there could perhaps have been a torsion in the bogies of the test vehicle, or some other tensions that affected the load. It would of course have been possible to model this, but it would have required more time and above all more modelling experience.

Speed profiles

Even though the speed of the vehicle in all of the selected test cases was roughly constant, there are small deviations in the speed. These deviations are a result of the drivers’ experience and his ability to retain a constant vehicle speed, but also how rapidly the speed controlling equipment in the vehicle reacts to changes in the load conditions, e.g. incline, etc. It is possible to implement speed profiles in the vehicle model, but it was decided to use the mean speed from the on-track tests as the initial velocity instead. This initial speed will decrease as the simulations run due to resistance (typically in curves), but not more than around 1% since the test cases only cover a few kilometres, and therefore it seemed unnecessary to model the speed profiles. However, the speed deviations in the on-track measurements resulted in a phase difference when comparing the measurements with the simulations, which is apparent in result for some of the test cases.

To reduce the influence of the speed deviation between the measured and simulated signals, it was decided to plot the forces versus distance instead of time, which had a small but improving effect. However, as will be seen in the following chapter, it resulted in Power Spectral Density diagrams with the unit spatial frequency \(1/\text{m}\). This makes it more difficult analyse the PSDs, since the numbers on the x-axis can not directly be interpreted as the frequency of which the
force is acting on the vehicle. However, the spatial frequency at a given point can be converted into time frequency by taking the vehicle speed $v$ into consideration, since

$$\lambda = \left( \frac{1}{m} \right)^{-1}$$  \hspace{1cm} (6.2)

$$f = \frac{v}{\lambda}$$  \hspace{1cm} (6.3)
Chapter 7

Results

In this chapter, the most interesting results will be presented. In order to provide readable information, a selection of the result data had to be done, therefore the focus will lie on only a few of the test cases and certain areas within these will be highlighted. The results from all of the test cases can be seen in Appendix 11.

7.1 Comparison between simulated and measured wheel-rail forces

The main topic for this study was to assess the accuracy of the simulated wheel-rail forces by comparing the simulations with on-track measurements. This section will cover this general accuracy assessment, but also the topic of investigating possible difference between different contact models provided in Simpack. Therefore, all plots will contain the simulated forces from the three contact models, as well as the measured forces. Sometimes this causes difficulties in separating the three simulated signals since they often are located on top of each other, so differences can only be seen when magnifying the plots. When there is need to see differences between the simulated signals, these plots will be magnified.

7.1.1 General assessment of the accuracy of the simulations

Test case 3: Curve radius 589 m, vehicle speed 120 km/h

When comparing the unfiltered forces, it can be clearly observed that the simulated forces have more high frequency content compared to the measured forces. This can be seen in all of the nine test cases. The reason for this may be that the track irregularity signals that were used were unfiltered, and therefore perhaps contained a larger portion of high frequent track irregularities than what was actually present, i.e. errors in the track irregularity measurements. A typical representation of this is illustrated in Figure 7.1, which shows the S-force for the 1st and 2nd wheelset along with the corresponding PSD diagrams. Again, note that the PSD is in the unit [1/m], spatial frequency, since the signals used to produce them are in the distance domain, unit [m].

The PSD diagrams in Figure 7.1 confirm that the simulated forces contains more high frequencies, since they are far above the red curve for the measured forces. The frequency contents only seem to relate up to around 0.1-0.2 [1/m], which by taking the vehicle speed into account corresponds to roughly 5 Hz, from that point the level for the simulated signals are greater than the measured signal. This is especially apparent for the 2nd wheelset, where the simulated forces contain peaks that do not have any correlation with the measured signal and also has much high frequent noise. By applying a filter that filters out frequencies above above 5 Hz, a much better correspondence is obtained, as shown in Figure 7.2. But a good
Figure 7.1: Test case 3. (top) S-force, (bottom) PSD. Unfiltered signals. Correspondence was still seen when filtering out frequencies above 10 Hz, even if the PSD’s suggested otherwise.

Figure 7.2: Test case 3. S-force, low pass filtered at 5 Hz.
Test case 6: Curve radius 3206 m, vehicle speed 256 km/h

In this case, the vehicle is running with very high speed and very high cant deficiency (181 mm). In Figure 7.3, it can be seen that within the 3206 m radius curve the simulations do not pick up the low frequent forces that can be observed in the measurement. At first sight, it might seem like an error in the simulations, but it is actually due to that the vehicle model does not contain the metal stop that restricts the yaw movement of the wheelset. At running conditions like these, the yaw movement of the wheelset becomes so large that it comes in contact with the metal stop, which induces these low frequency forces.

![Image of Figure 7.3: Test case 6. S-force, (top) S-force, unfiltered (bottom) S-force, low-pass filtered at 0.4 Hz.]

Quasi statically the simulations provide good results, as indicated in the bottom diagram in Figure 7.3.

7.1.2 Comparison between the contact models

The results from the comparison between the Simpack versions show that there in general is a very small difference between them. Differences can be seen in some cases, but mostly when no post-processing filters are applied, i.e. the differences that come out clearly are within the high frequency ranges. Some interesting points will be highlighted in the text below.

Test case 7: Curve radius 4994 m, vehicle speed 263 km/h

This is one of the few test case where a peak force can be observed which has been treated differently depending on the wheel-rail contact model. Figure 7.4 shows the S-force from test case 7, where the train passes through a long 4994 m radius curve followed by a few hundred meters straight and then into another similar curve. As highlighted, when the train crosses the transition into the second curve, a significant peak force arises. But when magnified, it can be seen that when using the older version 8.9 the peak is hardly visible, whereas with version 9.7 a more significant peak force has been calculated. And the discrete elastic contact model using 9.8 gives a large negative peak instead. Also notable is that at this level of magnification the
measurement signal is basically flat, which implies that the resolution in the measurement is lower compared to the simulations, and that nothing can be said of whether or not these peaks have any correlation with the measurement. The two identical peaks to the right of the red circle in 7.4 is treated the same with all contact models, but only the green line is visible since it is located "on top" of the others.

![Figure 7.4: Test case 7. S-force, unfiltered, showing a peak force being treated differently depending on contact model.](image)

According to these results, the solution provided in version 9.8 with purpose of removing the peak forces that sometimes has been observed in version 9.7 has some effect. However, for instance the other peaks seen in 7.4 have values that are typically the same regardless of which version that has been used, i.e. there is no consistency observed regarding how the different versions treat peak forces.
By disregarding the peaks and only looking at the general features of the three versions it can be seen that the differences are negligible. In Figure 7.5 below, a section has been magnified which shows that the three versions provide very similar results, both with and without post-processing filtering. It also shows that at this level of detail the simulated forces are not well fitted to the measurement, which is probably due to the problems with the vehicle speed profile that were discussed in section 6.6. It seems that plotting the signals as functions of distance had an effect for larger errors, but looking closely showed that there are still some deviations left between simulations and measurements.

Another point that was very clearly observed was that the simulation time when using the discrete contact model in Simpack 9.8 was extremely long. The simulation times were in some cases about a factor nine longer compared to the equivalent elastic model in version 9.7. And even more if compared to the constraint approach in version 8.9.
7.2 Using Gensys to filter track irregularities

When comparing the filtering strategy according to the Gensys procedure (Table 6.5) with the standard procedure (Table 6.4), two main differences could be found. The first one was that when not conducting any filtering in the post-processor, the Gensys procedure produced results that were more accurate in relation to the measured signal.

Test case 5: Curve radius 1000 m, vehicle speed 169 km/h

The results are presented for test case 5, as shown in the top diagram of Figure 7.6, where the blue curve is the force simulated with the Gensys procedure and the black curve is the force simulated with the standard procedure. The Gensys procedure is not perfectly fitting the red measured signal, but still better than when using the standard procedure.

On the other hand, as shown in the bottom diagram of Figure 7.6, when applying a low-pass filter with a cross over frequency at 10 Hz in the post-processing, both procedures show similarly accurate results. This implies that the differences are mainly seen within the high frequent content, which is to expect.

![Figure 7.6](image)

**Figure 7.6:** Lateral track shift force $S$, difference between the Gensys and the standard procedure.

The corresponding results for the vertical force can be seen in Figure 7.7 below, where instead the standard procedure seems to be closer to the measurement compared to the Gensys procedure when unfiltered. But again, when applying a low-pass filter in the post-processing, both procedures gives very similar results, as can be seen in the bottom diagram of Figure 7.7.
When comparing the PSD for the S-forces and the vertical forces shown in Figure 7.8, it can be seen that the Gensys procedure has best effect for the Q forces, where the frequency content fits the measured signal up to around 1.5 [1/m]. The PSD also confirm that the frequency contents are closer to those of the measured signal when the Gensys procedure is used.

The second main difference that was found is that the simulation time when using the Gensys procedure is significantly lower compared to using the standard procedure. The reason is likely due to that most of the track irregularities with very short wavelengths are filtered out when
using the Gensys procedure, while they are completely retained when using the standard proce-
dure. Track irregularities with very short wavelengths require the simulation solver to use very
short integration steps, which in turn are more time demanding.
7.3 Worn wheel and rail profiles

As mentioned earlier, data for the measured wheel profiles were available but measured rail profiles could only be found for test case 1.

Test case 1: Curve radius 280 m, vehicle speed 86 km/h

The simulations in this evaluation were conducted in Simpack 9.7 using the equivalent elastic contact model. When simulating the wheel forces using worn wheel profiles and nominal rail profiles, the results indicate that the measured wheel profiles had a very small impact. Perhaps this was to be expected, since the shape of the worn wheel profiles did not deviate very much from the nominal profiles. As shown in Figure 6.5, the wheels were mostly worn on the flange, and the depth of the worn area was not more than a few millimetres. This could be an explanation to why the simulated forces shown in Figure 7.9 are so similar. Figure 7.9 also shows that when using worn wheel profiles the simulations has a tendency of improvement since they are closer to the measurement, but this improvement is quite small.

![Figure 7.9: Lateral forces on 1st wheelset, low-pass filtered at 10 Hz](image)

When simulating the same test case again but using worn profiles for both the wheels and the rails, no particular improvement could be seen. In fact, the results indicate a slight impairment, the simulations using worn profiles are generally further away from the measurement compared to using nominal profiles, see Figure 7.10.

It was very unfortunate that only the rail profiles from one test case could be imported to Simpack, since a comparison between different test cases would have been interesting.
Figure 7.10: Lateral forces on 2nd wheelset, low-pass filtered at 10 Hz
Chapter 8

Conclusions

8.1 General assessment of the accuracy of the simulations

According to the results that have been found, it can be concluded that the simulations are in good accordance with the measurements. The simulated lateral quasi static forces can be considered as sufficiently accurate, given that the model itself contains a level of uncertainty such as the vehicle mass and center of mass, as well as the uncertainties of the measured track irregularities and also how the track was modelled. The simulated dynamic forces emerging from track irregularities have also shown a good agreement with the measurements, both regarding lateral and vertical forces. The agreement was found to be best for frequencies below 10 Hz, since the simulations had much more high frequent dynamic content compared to the measurement.

8.2 Comparison between applied contact models Simpack v9.7, 9.8 and 8.9

Apart from a few exceptional occurrences, the results from the simulations are very similar for all of the three contact models in Simpack. Some cases were observed where peak forces occurred using the equivalent elastic model which were presumably a result of calculation errors, but where the magnitude of that same peak was smaller using the discrete elastic model. Due to the rarity of such cases it is however difficult to give any certain claims as to whether or not the simulations with the discrete model provides an improvement. Especially given that the cost for using that model is a tremendously increased simulation time, in many cases a factor of 9 longer compared to using the equivalent elastic model. The possible improvement (or difference) in the results are not significant enough in relation to the increased simulation time.

8.3 Using Gensys to filter track irregularities

The procedure to use Gensys as a tool for filtering track irregularities and creating track excitation files to be imported to Simpack has been proved to be favourable. The simulation results showed that the track irregularities of significance for the dynamic forces were retained, even if the track irregularities had been filtered beforehand. The Gensys procedure provides a smart and efficient way of filtering out high frequent track irregularities while still keeping a realistic representation of the track. The simulated forces were, when using the Gensys strategy compared to the standard strategy (i.e. to not filter the track irregularities), in better agreement with the measured forces before any post-processing filtering had been made. But when applying a low-pass filter at 10 Hz, both strategies provided similar agreement to the measurement. Another great advantage when using the Gensys strategy was that the simulation times were
drastically decreased, which is a consequence of the reduced amount of short wavelength track irregularities, since they require the solver to use shorter integration steps.

8.4 Worn wheel and rail profiles

The hypothesis that the simulation results should agree better with the measurement by using worn wheel and rail profiles could not be verified in this investigation. A slight improvement could be seen when using only worn wheel profiles, but when applying worn rail profiles as well the results rather showed a worse agreement. It should be noted that worn rail profiles could only be applied in one of the nine test cases, and that the procedure of implementing these profiles in Simpack was very laborious, why the results should be viewed upon with a critical approach. NB: In general it was not a problem to import only one profile, the problems arose when trying to have variable rail profiles, i.e when multiple rail profiles where imported.

The wheel profiles could with ease be implemented in Simpack, since they were measured with a device (SPAK) that has a high accuracy and provides the data points for the profile in a correct orientation relative to the coordinate system. On the contrary, the rail profiles were measured using a laser system mounted on board a track measuring vehicle. This system has a modest level of accuracy and the orientation of the data points is depending on the rail inclination as well as on the position of the rail relative to the vehicle. Therefore it is required to first rotate and shift the data points of the profile before they can be implemented in Simpack, which has been proved to be difficult. Thus, it is recommended that the rail and wheel profiles to be used in dynamic simulations should be measured using a device with a higher accuracy than what can be provided using on board laser systems. This is to both ensure reliable simulation results but also to reduce the amount of preparatory work.
Chapter 9

Discussion

The comparison between the simulated and measured wheel-rail forces indicated that the accuracy of the simulations were generally good. However, that statement and the analysis behind it is based upon a subjective engineering judgement, and it is obvious that the conclusions found would benefit greatly from having results that rested on a more rigid foundation. A problematic feature regarding this has been that there are few guidelines on how to compare simulated data with measurements, and limits for when simulations can be regarded as sufficiently accurate have not been found. Perhaps it would have been possible to make a statistical analysis of the forces, according to standards for vehicle acceptance e.g. EN14363, but that never got considered. In retrospect, even though it seems unlikely to have found the time for that, it would have granted a higher credibility to the verification.

On that note, many parts within this study have been very time consuming without yielding commensurable matter to the report. Firstly, the process of selecting appropriate test cases and obtaining the corresponding track irregularities for them became a comprehensive operation, yet absolutely essential to the ensuing verification process. Secondly, the choice to investigate nine test cases resulted in a wide and scattered scope of running conditions, but it may have been overly substantial. Especially with respect to that the investigation over time came to cover not only one version of Simpack but three, plus two methods for filtering track irregularities, which increased the amount of simulations. And last but not least, it was very challenging to implement the measured rail profiles to Simpack, and as the results indicate it maybe never got properly accomplished. It is of course easy to be wise after the event, and some of these slightly additional features to the prime objective have still provided useful and interesting information, but perhaps the time would have come to better use in a more thorough data analysis instead.
Chapter 10

Future work

This section highlights the areas that should be subject for further investigations in a future continuation of the project.

- **Tuning of the vehicle model**

  The vehicle model used in this project, provided by Bombardier, has been used for several investigations in the past. Therefore, the settings for the model elements such as springs and dampers etc. are likely to already have been adjusted to better correspond to the measurements. Even so, there has been no adjustments made in this project and by trying to tune the model it would be possible to further increase the accuracy of the simulations. Furthermore, as mentioned in section 6.6, it was evident that the vehicle used for the on-track test had a skew wheel load distribution. Attempts were made to adjust for this skewness in the vehicle model, but without success. Managing to model this skewness would allow for better possibilities to compare the vertical forces.

  In order to obtain a better correspondence between the simulated and measured forces it is suggested to utilize the speed profiles of the test vehicle when conducting simulations. This would most likely reduce the observed phase shift in the time history comparisons.

- **Tuning of the contact model settings**

  This investigation covers three Simpack versions and contact models, but the settings that have been used are the standard ones that Simpack suggests. It would be interesting to adjust for instance the friction coefficient to see how different values affects the results.

- **Conducting simulations using the discrete elastic model with track excitations from Gensys**

  The simulation time was very long when using the discrete contact model in Simpack 9.8, but on the other hand the simulation times were shown to be decreased when using track irregularities that were filtered in Gensys. It is possible that conducting simulations with the discrete contact model combined with Gensys track excitations might reduce the simulation time to a more reasonable level that may justify the use of the discrete elastic contact model.

- **Measured track geometry**

  At an early stage of the project it was decided to use measured track irregularities but that the nominal track geometry would be modelled manually with respect to the databases
provided by Trafikverket. However, as indicated by the simulated forces in some of the
test cases, there exist small errors in the track geometry. It is likely that these errors
could have been prevented by using measured track geometries as well, which for instance
is possible in one of the features of Gensys.

- **Assessing the accuracy by means of a statistical approach**

The accuracy assessment has in this thesis been based upon the engineering judgement of
the author, which is one of its’ weak points. It would be favourable to also make a more
quantitative evaluation, e.g. by using the required calculations in standards for vehicle
acceptance.
Appendix A - Results from the test cases

The figures in Appendix A show the simulated lateral S-force and vertical Q-force for every test case, and for both the 1st and 2nd wheelset. They are supposed to provide an overview of the results from the nine test cases, and give the reader the possibility to assess them. The figures illustrate the simulation results filtered using a low-pass filter with a cross-over frequency of 10 Hz, since that was the frequency at which the best agreement could be obtained.

In order to maximize the area of the diagrams, it was chosen to remove the legend in the figures and instead show one general legend here, see Figure 11.1 below. It is valid for all of the following figures.

| Simulation Simpack 9.7, equivalent elastic |
| Simulation Simpack 9.8, discrete elastic |
| Simulation Simpack 8.9, constraint elastic |
| Measurement |

**Figure 11.1:** The legend corresponding to the following figures.
CHAPTER 11. APPENDIX A - RESULTS FROM THE TEST CASES

11.1 Test case 1: R = 280 m

Figure 11.2: Lateral track shift force $S$, low-pass filtered at 10 Hz

Figure 11.3: Vertical load $Q$ per wheel, low-pass filtered at 10 Hz
11.2 Test case 2: $R = 411$ m

Figure 11.4: Lateral track shift force $S$, low-pass filtered at 10 Hz

Figure 11.5: Vertical load $Q$ per wheel, low-pass filtered at 10 Hz
11.3 Test case 3: $R = 589$ m

Figure 11.6: Lateral track shift force $S$, low-pass filtered at 10 Hz

Figure 11.7: Vertical load $Q$ per wheel, low-pass filtered at 10 Hz
11.4 Test case 4: $R = 645$ m

Figure 11.8: Lateral track shift force $S$, low-pass filtered at 10 Hz

Figure 11.9: Vertical load $Q$ per wheel, low-pass filtered at 10 Hz
11.5 Test case 5: $R = 1000$ m

**Figure 11.10:** Lateral track shift force $S$, low-pass filtered at 10 Hz

**Figure 11.11:** Vertical load $Q$ per wheel, low-pass filtered at 10 Hz
11.6 Test case 6: $R = 3206$ m

**Figure 11.12:** Lateral track shift force $S$, low-pass filtered at 10 Hz

**Figure 11.13:** Vertical load $Q$ per wheel, low-pass filtered at 10 Hz
11.7 Test case 7: R = 4994 m

Figure 11.14: Lateral track shift force $S$, low-pass filtered at 10 Hz

Figure 11.15: Vertical load $Q$ per wheel, low-pass filtered at 10 Hz
11.8 Test case 8: $R = 10000$ m

Figure 11.16: Lateral track shift force $S$, low-pass filtered at 10 Hz

Figure 11.17: Vertical load $Q$ per wheel, low-pass filtered at 10 Hz
11.9 Test case 9: $R = \infty \text{ m}$ (straight track)

**Figure 11.18:** Lateral track shift force $S$, low-pass filtered at 10 Hz

**Figure 11.19:** Vertical load $Q$ per wheel, low-pass filtered at 10 Hz
### Appendix B - Detailed information about the test cases

Figure 11.20 above shows a table with detailed information about the test cases. The column "Test" represents the test number in the general protocol from the on-track measurements. The column "Plot-pos" is the track range that needed to be exported from SiView to retain the entire curve of interest, and "Kurv-pos" is the position of the curve itself. The column "Rf" is the cant deficiency for each case. "Simtime" is the simulation time that was used in Simpack, which is basically the time it takes for the train to pass through the range given in "Plot-pos" at the constant speed "v".

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<th>Bandel Spår</th>
<th>Plot-pos [km + m]</th>
<th>Kurv-Pos [km + m]</th>
<th>Kurvradi</th>
<th>Rf</th>
<th>simtime</th>
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Figure 11.20: Detailed information about the test cases.
Chapter 12

Bibliography


