An Investigation of the Iron-Ore Wheel Damages using Vehicle Dynamics Simulation

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1. Introduction

Background
In order to transport extracted iron-ore from Kiruna’s mines to Luleå in the north of Sweden and Narvik in Norway, the railway is used. The freight wagons are so called Fanoo wagons attached to three piece bogies which running on Malmbanan/Ofotbanen. The track history goes back to the 19th century and has been used to bear around 1 tonne load. Since then, due to market demands to increase the transportation capacity and decrease its maintenance costs, the track components and structure have been improved several times. The recent upgrade was changing the rail profile and replacing the wooden sleepers with concrete ones. Nowadays, the freight trains run with 30 tones axle load and 60 km/h velocity. Although these improvements increase the income, there are some obstacles and problems which raise the company’s costs such as maintenance. They are mostly mechanical wear (adhesive and abrasive) and fatigue wear (rolling contact fatigue, RCF) on the wheels; however, the later problem is now dominating especially in winters. To detect and predict the mechanisms of deterioration and sources of damages, besides many other benefits such as investigating the vehicle track interaction and studying the track forces, computer simulation is used with multibody simulation software GENSYS and the model results are validated against the experimental measurements.

Aim and Scope
This work tries to enhance and improve the knowledge of the vehicle-track interaction of the Swedish iron-ore freight wagons used at Malmbanan. The scope of the present work is to:

- Model and simulate the dynamic behavior of the freight vehicle on both tangent and curved track.
- Validate the model by comparing experimental data and simulation results.
- Investigate sources of rolling contact fatigue on the wheels via studying the following variables:
  - The effect of vertical track stiffness and viscous damping regarding the seasonal variation of the track conditions,
  - Influence of the wheel-rail friction coefficient because in winter time the climate is very dry along most parts of the Swedish iron-ore line,
  - The impact of track gauge, track quality and cant deficiency on RCF,
- Comparing the calculated and observed RCF locations on wheels,
- Calculating the wear number on the wheels on various track conditions,
- Finding a relation between wear number and RCF damage.

2. Vehicle

Three-Piece Bogies in General
One of the most common freight wagon running gear in the world with a history of 150 years is the so-called three-piece bogie. More than 2.5 million three-piece bogies are operating under North American freight lines. Also China, Russia, Australia, South Africa, Brazil and Sweden are using this type of bogie in their freight transportation fleet. Simple mechanical design, easy maintenance,
lightweight structure and low initial cost can be considered as advantages of the bogie. The word three-piece comes from two parallel side frames and a bolster beam. Since the first introduction of this bogie it has been improved and today various types of three-piece bogies are used in heavy haul operations. A review of the history of three-piece bogie trucks can be found in [1]. It is possible to categorize three-piece bogies into three main groups based on their design features. The simplest design is called standard three-piece bogie. To improve the ride stability and curving performance of three-piece bogies frame-braced and inter-axle linkage bogies are introduced and are widely used in heavy haul lines. Figure 1 (a) and (b) show the typical design of three-piece trucks with frame-braced and inter-axle linkage bogies respectively. A review on a comparison between various kinds of three-piece bogies is presented in [2] and a review on the historical background of developing inter-axial linkage bogies can be found in [3].

**Figure 1(a)** Typical frame-braced and (b) inter-axle linkage three-piece bogies

The primary suspension (the coupling between the wheelset and the bogie frame) is a thin rubber element, called *adapter plus* in Amsted three-piece bogies, that provides elastic couplings in all three dimensions and avoids metal-to-metal contact between the top of the bearing box and the side frame. The secondary suspension (coupling between the bogie and the carbody) consists of the *centre plate*, *the side bearings*, *the set of coil springs* and *friction wedges*. The frictional contact between the wedges and the side frame provides some damping in vertical and lateral direction. There are two main types of friction damping adopted for the suspension. The first type provides constant damping known as *Ride Control and* the second type provides load dependent frictional damping called *Barber stabilized*. The later version is more common in freight wagons as there is a significant weight difference between empty and loaded wagon. Moreover, there are two main different types of wedges used in three-piece bogies. The most common typical version is called *planar wedge*; however, to increase the critical speed and to avoid instability *spatial wedges* are recently developed [4], compare Figure 2 (a) and (b).
Figure 2(a) Planar and (b) Spatial wedge designs.

The design of the coil spring nest can also be different. Two typical types widely used in North America and Russia are shown in Figure 3. In the American design the height of the wedge springs (control springs) is slightly higher than that the load springs to compensate for wear of the wedges. In the Russian design this is considered via inclination of the side frame columns by 1-2 degrees towards the centre in the upper part [5]. The wedge inclination of the American design is higher which makes it possible to provide more friction damping [4].

Figure 3. Design of central suspension shown in the nominal positions without any load on the bolster beam: 1-bolster, 2-wedge, 3-load springs, and 4-wedge springs (control springs), [5].

The described suspension settings allow the side frames to have relative motions in longitudinal direction called the warping motion. Warping of the bogie causes instability at high speeds. Therefore, to run with 120 km/h the warping stiffness must have a lower bound [6]. It also leads to a higher angle of attack and flange contact in curves [7]. The results of modal analysis of non-truck hunting tests at TTC [8] show that the warping motion mostly comes from:

- Lateral displacement of the wheelset;
- Some longitudinal motion taking place at the adapter;
- Insufficient warping resistance against the longitudinal displacement of the side frames (between the bolster, wedges and coil springs, and the rotational constraints provided by the adapter pads);
- Insufficient rotational resistance inside bearings and centre plate against the yaw motion of the bolster.

The warping stiffness of the bogie is strongly non-linear, and it is much higher for small displacements of the side frames compared to larger displacements, see [4] and [9]. Investigations show that the warping stiffness is also load dependent and therefore, usually empty wagons have
lower critical speed than the loaded ones [6]. Several attempts have been made to measure the warping stiffness via a test-rig, cf. [6, 9, 10 and 11]. Frame braced bogies usually have higher warping stiffness as its design squares the truck. In [11] the benefits of using frame-braced truck are discussed. The inter-axle linkage bogie design decouples the equivalent shearing \( k_s \) and bending stiffness \( k_b \) of the bogie and therefore, it is possible to improve both curving performance and stability of the truck. One typical example of the inter-axle linkage bogies is the \textit{Scheffel} bogie with steering arms [13]. The so called shearing and bending stiffness are calculated as

\[
K_s = \frac{k_x k_y b_L^2}{k_x b_L^2 + k_y a^2}
\]

Respectively,

\[
k_b = k_x b_L^2
\]

cf. Figure 4.

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{figure4.png}
\caption{Schematic plot of the primary suspension couplings and (b) bending and shearing stiffness}
\end{figure}

**Dynamic Simulation of Three-Piece Bogies**

Many authors have used multibody simulations to analyze the dynamic behavior of wagons with three-piece bogies. Using the multibody simulation software GENSYS, Berghuvud developed simulation models for vehicles running on

- a standard three piece bogie with frictional contact in the primary suspension;
- a typical frame braced bogie;
- a typical inter-axle coupled bogie with cross braced design.

He compared the curving performance and calculated lateral contact forces of the mentioned three bogies [14] and [15]. Bogojevic also used GENSYS; however, he focused on the standard three-piece bogie with the elastic couplings in the primary suspension. He validated the model by comparing simulation results with on-track data [16]. In [17], Orlova presents a simulation model of the Russian 18-100 three-piece bogies using the software MEDYNA. The fundamentals of all mentioned modeling methods are more or less the same. Primary suspension usually is modeled as an elastic stiffness and damping in parallel, unless there is no elastic rubber pad between the axle box and the side frame which should be modeled as friction element; however, in this case one should consider the dither phenomenon as the frictional element is close to the wheel-rail contact. The model of the frictional
damping depends on its design and whether it is load dependent or not. For more details about modeling and validation of various types of friction damping in three-piece bogies see [5, 18, 19 and 20].

**Iron-Ore Wagons Characteristics and Dynamic Simulation**

The Iron-ore wagons are so called Fanoo040 wagons running on Amsted Motion Control M976 three piece bogies with load sensitive frictional damping. Fanoo040 wagons contain 2 units. One of the units is called master and the other is called slave unit. Controller of the breaking system is attached to the master wagon as shown in Figure 5.

![Figure 5, A two-unit iron-ore wagon](image)

The characteristics of the iron-ore wagons are given in Table 1.

**Table 1, Iron-ore wagons characteristics**

<table>
<thead>
<tr>
<th>Characteristic</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length of wagons</td>
<td>10.29 (m)</td>
</tr>
<tr>
<td>Distance between centre plates</td>
<td>6.77 (m)</td>
</tr>
<tr>
<td>Total truck height</td>
<td>3.64 (m)</td>
</tr>
<tr>
<td>Basket width</td>
<td>3.49 (m)</td>
</tr>
<tr>
<td>Weight of empty wagon</td>
<td>21.6 (t)</td>
</tr>
<tr>
<td>Payload</td>
<td>102 (t)</td>
</tr>
<tr>
<td>Weight of loaded wagon</td>
<td>120 (t)</td>
</tr>
<tr>
<td>Maximum speed (empty)</td>
<td>70 (km/h)</td>
</tr>
<tr>
<td>Maximum speed (loaded)</td>
<td>60 (km/h)</td>
</tr>
<tr>
<td>Wheelbase</td>
<td>1778 (mm)</td>
</tr>
<tr>
<td>Track gauge</td>
<td>1435 (mm)</td>
</tr>
<tr>
<td>Wheel Diameter (max)</td>
<td>915 (mm)</td>
</tr>
<tr>
<td>Wheel Diameter (min)</td>
<td>857 (mm)</td>
</tr>
<tr>
<td>Weight wheelset</td>
<td>1341 kg</td>
</tr>
<tr>
<td>Weight bogie incl. wheelsets and mechanical-braking equipment</td>
<td>4 650 kg</td>
</tr>
</tbody>
</table>

The primary suspension is held via a rubber pad called *Adapter Plus* between the axle box and the side-frame and is modeled as an elastic stiffness and damping in parallel as it shown in Figure 6.
Figure 6, Primary suspension modeled as elastic damping and stiffness in parallel in three dimensions

The wedges are modeled as massless bodies, and the position of the wedge is calculated by solving the local equilibrium matrix. Normal contact forces acting on the surfaces of the wedge are used to calculate the friction forces, $F_{fr0}$, in the friction block. The coupling in the contact surface between the bolster and the wedge is modeled as one-dimensional friction block and the friction surface between wedge and side frame is modeled as two-dimensional friction block in lateral and vertical direction, as shown in Figure 7.

Figure 7, Couplings in the secondary suspension of a three-piece bogie

The friction coefficient between wedges and bolster is estimated to around 0.15 (hardened cast iron wedge to cast steel), between wedges and side frame it is estimated to around 0.38 (hardened cast iron and hardened steel plate), according to [5]. However, the friction level varies from morning to night and it is dependent on the weather condition, roughness of the surfaces and so on. A very low friction level saturates the friction force and decreases the warping stiffness significantly, while a very high friction level increases the risk of stick condition between the wedge and the side frame leading to very high peak vertical forces.

All friction contacts in the suspension system such as the couplings between side frame and wedge, wedge and bolster, side bearers and centre plate are modeled with Saint Venant elements as it is shown in Figure 8.
Figure 8, One dimensional friction block, Saint Venant element

The carbody basket and the bogie bolster are connected via centre plate and side bearers. Side bearers are placed at both ends of the bolster. They carry 10% of the vertical load when the wagon is loaded. The main coupling elements are the vertical nonlinear elastic stiffness and a longitudinal friction element. The connection between the carbody and the centre plate is set via five connection points at the centre, front, back, left and the right side of the plate which bears the remaining 90% of the load. The couplings are defined as two dimensional friction elements in the x-y plane. Figure 9 shows the carbody and bolster connections.

Figure 9, Connection between carbody and bolster

Some of the other model assumptions can be summarized as follows:

- Car body, bolster, side frames, wheelset and wheels are modeled as rigid bodies;
- Side bearers have always contact with the car body;
- Clearances between elements are implemented in the model such as bolster-side frame, axle-side frame etc.

In this study vehicle speed is considered to be 60 km/h in all simulated sections. Figure 10 summarizes the entire model in vertical direction.
Figure 10, Connection between masses in vertical direction

3. Track

The line Geometry
The length of the iron-ore line between Luleå and Narvik is around 470 km including around 50 percent of curves with radii below 1000 m. About 75% of the curves below 450 m radius are located on the Norwegian side. From Luleå to Narvik the line contains more left handed curves than right handed curves as shown in Figure 11.

Figure 11, The Iron-ore line
The details of the line geometry are presented in Table 2 and Table 3 regarding the distribution of the left and right handed curves respectively.

### Table 2, Details of the left handed curve sections

<table>
<thead>
<tr>
<th>Curve radius interval</th>
<th>Mean Radius (m)</th>
<th>Length (%)</th>
<th>Mean Cant (mm)</th>
<th>Mean Gauge (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>&lt;350</td>
<td>305</td>
<td>2.3</td>
<td>68</td>
<td>1445</td>
</tr>
<tr>
<td>350-400</td>
<td>386</td>
<td>0.7</td>
<td>66</td>
<td>1448</td>
</tr>
<tr>
<td>400-450</td>
<td>412</td>
<td>0.6</td>
<td>63</td>
<td>1446</td>
</tr>
<tr>
<td>450-600</td>
<td>559</td>
<td>8.4</td>
<td>54</td>
<td>1447</td>
</tr>
<tr>
<td>600-800</td>
<td>649</td>
<td>10.1</td>
<td>45</td>
<td>1446</td>
</tr>
<tr>
<td>800-1000</td>
<td>947</td>
<td>5.3</td>
<td>36</td>
<td>1444</td>
</tr>
</tbody>
</table>

### Table 3, Details of the right handed curve sections

<table>
<thead>
<tr>
<th>Curve radius interval</th>
<th>Mean Radius (m)</th>
<th>Length (%)</th>
<th>Mean Cant (mm)</th>
<th>Mean Gauge (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>&lt;350</td>
<td>305</td>
<td>2.2</td>
<td>68</td>
<td>1444</td>
</tr>
<tr>
<td>350-400</td>
<td>368</td>
<td>0.4</td>
<td>61</td>
<td>1443</td>
</tr>
<tr>
<td>400-450</td>
<td>409</td>
<td>0.8</td>
<td>66</td>
<td>1448</td>
</tr>
<tr>
<td>450-600</td>
<td>547</td>
<td>7</td>
<td>51</td>
<td>1447</td>
</tr>
<tr>
<td>600-800</td>
<td>648</td>
<td>8</td>
<td>46</td>
<td>1445</td>
</tr>
<tr>
<td>800-1000</td>
<td>934</td>
<td>4</td>
<td>37</td>
<td>1444</td>
</tr>
</tbody>
</table>

The model of the track comprises of ground, ballast, rails, stiffness and damping between these bodies as shown in Figure 10.

### Track Stiffness

In most track sections on the iron-ore line concrete sleepers are used. Therefore, in this study a track model representing concrete sleeper track is used within the simulation. To study more on track flexibility characteristics and its validation see [21]. Figure 12 is the track deflection measured in winter and summer on wooden and concrete sleepers by Banverket. The frozen track is stiffer than the non-frozen track; consequently, in winter tracks are less flexible and have less deflection under the wagons load. As the figure shows for high vertical force per rail, for the wooden sleeper track, the deflection in summer is 1.7 times of the track displacement in winter. And the deflection of non-frozen concrete sleeper track is twice as much as the frozen track deflection. Based on this fact and according to Hook’s law while considering that all couplings between the track components are linear, the vertical track characteristics are determined for the winter condition. The curves which are not tagged in the figure are the vertical displacement of the twin block concrete sleepers in
winter and summer respectively. The word winter refers to January, February, March and the word summer is used for June and July.

![Vertical rail displacement as a function of vertical static load measured by Banverket.](image)

**Figure 12.** Vertical rail displacement as a function of vertical static load measured by Banverket.

**Track Irregularities**

Any deviation from the designed track geometry is known as track irregularities. They have a great influence on contact dynamic forces and unwanted vibration which may cause severe RCF and noise. There are four types of track irregularities that can be captured by the Swedish track recording vehicle STRIX:

- Geometrical track errors in vertical direction
- Geometrical track errors in lateral direction
- Deviation from nominal track gauge
- Deviation from nominal cant

In case of dynamic analysis the vertical and lateral irregularities are calculated as the mean value of errors of left and right rail and they are often called longitudinal level and line irregularities. Figure 13 shows these four track irregularities.

Three classes of track qualities are defined with regard to the necessity of maintenance and to the applicability for acceptance tests of vehicles:

- **QN1** refers to the value which necessitates observing the condition of the track or taking maintenance measures as part of regularly planned maintenance operations.
- **QN2** refers to the value which requires short term maintenance action.
- **QN3** refers to the value which, if exceeded, leads to the track section being excluded from the acceptance analysis because the track quality encountered is not representative of usual quality standards.
Table 4, Permissible standard deviations for longitudinal level and line for track classes QN1 and QN2 according to UIC Code 518

<table>
<thead>
<tr>
<th>Standard deviation: longitudinal level</th>
<th>QN1 [mm]</th>
<th>QN2 [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>0 &lt; v &lt; 80km/h</td>
<td>2.3</td>
<td>2.6</td>
</tr>
<tr>
<td>80 &lt; v &lt; 120km/h</td>
<td>1.8</td>
<td>2.1</td>
</tr>
<tr>
<td>120 &lt; v &lt; 160km/h</td>
<td>1.4</td>
<td>1.7</td>
</tr>
<tr>
<td>160 &lt; v &lt; 200km/h</td>
<td>1.2</td>
<td>1.5</td>
</tr>
<tr>
<td>200 &lt; v &lt; 300km/h</td>
<td>1.0</td>
<td>1.3</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Standard deviation: line</th>
<th>QN1 [mm]</th>
<th>QN2 [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>0 &lt; v &lt; 80km/h</td>
<td>1.5</td>
<td>1.8</td>
</tr>
<tr>
<td>80 &lt; v &lt; 120km/h</td>
<td>1.2</td>
<td>1.5</td>
</tr>
<tr>
<td>120 &lt; v &lt; 160km/h</td>
<td>1.0</td>
<td>1.3</td>
</tr>
<tr>
<td>160 &lt; v &lt; 200km/h</td>
<td>0.8</td>
<td>1.1</td>
</tr>
<tr>
<td>200 &lt; v &lt; 300km/h</td>
<td>0.7</td>
<td>1.0</td>
</tr>
</tbody>
</table>

The values are calculated for every 100 m of the iron-ore line and the corresponding statistics for 2012 track irregularities measurement are presented in Table 5.

Table 5, Track quality distribution on iron-ore line according to UIC 518. Maximum speed of 80 km/h assumed.

<table>
<thead>
<tr>
<th>Track quality classes</th>
<th>Definition of the classes</th>
<th>Distribution of each class</th>
<th>Recommended by UIC 518</th>
</tr>
</thead>
<tbody>
<tr>
<td>QN &lt; QN1</td>
<td>Sections with good track standard</td>
<td>29%</td>
<td>Should be &gt; 50%</td>
</tr>
<tr>
<td>QN1 &lt; QN &lt; QN2</td>
<td>Regularly planned maintenance operations</td>
<td>23%</td>
<td>Should be &lt; 40%</td>
</tr>
<tr>
<td>QN2 &lt; QN &lt; QN3</td>
<td>Short term maintenance action</td>
<td>43%</td>
<td>Should be &lt; 10%</td>
</tr>
</tbody>
</table>
The details of the track quality distribution for 2013 track irregularities measurement with respect to the curve section intervals are presented in Table 6 and Table 7 for the both left and right handed curves respectively. As seen in the tables the track quality becomes worst for tighter curves probably due to the wheel flange contacts with rails.

**Table 6, Left handed curves track quality distribution**

<table>
<thead>
<tr>
<th>Curve radius interval</th>
<th>Mean Radius(m)</th>
<th>QN1(%)</th>
<th>QN2(%)</th>
<th>QN3(%)</th>
<th>&gt;QN3(%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>&lt;350</td>
<td>305</td>
<td>40</td>
<td>17</td>
<td>43</td>
<td>0</td>
</tr>
<tr>
<td>350-400</td>
<td>386</td>
<td>41</td>
<td>11</td>
<td>47</td>
<td>0</td>
</tr>
<tr>
<td>400-450</td>
<td>412</td>
<td>74</td>
<td>11</td>
<td>15</td>
<td>0</td>
</tr>
<tr>
<td>450-600</td>
<td>559</td>
<td>79</td>
<td>7</td>
<td>13</td>
<td>1</td>
</tr>
<tr>
<td>600-800</td>
<td>649</td>
<td>81</td>
<td>10</td>
<td>9</td>
<td>0</td>
</tr>
<tr>
<td>800-1000</td>
<td>947</td>
<td>79</td>
<td>8</td>
<td>13</td>
<td>0</td>
</tr>
</tbody>
</table>

**Table 7, Right handed curves track quality distribution**

<table>
<thead>
<tr>
<th>Curve radius interval</th>
<th>Radius(m)</th>
<th>QN1(%)</th>
<th>QN2(%)</th>
<th>QN3(%)</th>
<th>&gt;QN3(%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>&lt;350</td>
<td>305</td>
<td>33</td>
<td>25</td>
<td>42</td>
<td>0</td>
</tr>
<tr>
<td>350-400</td>
<td>368</td>
<td>70</td>
<td>19</td>
<td>11</td>
<td>0</td>
</tr>
<tr>
<td>400-450</td>
<td>409</td>
<td>70</td>
<td>8</td>
<td>22</td>
<td>0</td>
</tr>
<tr>
<td>450-600</td>
<td>547</td>
<td>82</td>
<td>10</td>
<td>8</td>
<td>0</td>
</tr>
<tr>
<td>600-800</td>
<td>648</td>
<td>80</td>
<td>11</td>
<td>9</td>
<td>0</td>
</tr>
<tr>
<td>800-1000</td>
<td>934</td>
<td>71</td>
<td>8</td>
<td>21</td>
<td>0</td>
</tr>
</tbody>
</table>

4. Wheel-rail interaction

**Contact problem**

Briefly, the contact problem is to find the traction and normal pressure distribution (size and shape of the contact area) for either known deformations or known loads or a combination of these. There have been a large number of studies in the field of wheel-rail contact mechanics which are reviewed in [28]. However, in this section the two most common and widely used theories are presented.
which are also used in GENSYS (Hertzian and Kalker simplified theories). Considering the half-space assumption when the bodies of contact deform like infinite half-spaces we demand that:

- The size of the contact area is considerably smaller than a typical dimension of the bodies and
- The contact should be non-conforming (tread contact).

Boussinesq [23] and Cerruti [24] have presented the relation between the surface tractions and their corresponding displacements in the bodies of contact via the so called influence functions. The derivation of these formulae can be found in Gladwell [25] and Love [26]. Moreover, if the bodies in contact are quasi-identical:

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

where, \(G_1, G_2\) and \(\nu_1, \nu_2\) are the shear modulus and Poisson ratio of the first and second bodies, it can be shown that the displacements in tangential plane are not influenced by the normal traction and the displacement in the normal direction is not influenced by the tangential traction. Thus, in an approach called “the Johnson process” the normal and tangential contact problem is decoupled and the contact problem is formulated as follows:

- Determining the normal pressure distribution in absence of the friction and thus the tangential tractions considering perfectly smooth surfaces (Normal problem),
- Determining the tangential traction distributions when they are bounded by friction coefficient times the normal traction (Tangential problem).

Using the above assumptions Hertz presented his method to solve the normal contact problem (today known as Hertzian solution) [27]. He also had to use some other assumptions to simplify his problem such as:

- The bodies in contact are homogeneous, isotropic, and linearly elastic. Linear kinematic hardening is also assumed (constant range between yield in tension and compression with equal strain hardening rates).
- Displacements and strains are small.
- The curvatures of the surfaces in the vicinity of the contact area are constant to be able to calculate the shape and size of the contact area.

Considering quadratic functions for each surface near the contact (constant curvature assumption), elliptic contact area and a semi-ellipsoidal pressure distribution Hertz calculated the pressure distribution over the contact patch,

\[
p(x,y) = \frac{p_0}{2ab} \left( 1 - \frac{x^2}{a^2} - \frac{y^2}{b^2} \right),
\]

where, \(a\) and \(b\) are the semi-axes of the elliptic contact area and \(p_0\) is the maximum contact pressure,

\[
p_0 = \frac{3N}{2\pi ab},
\]

The semi-axes of the contact ellipse are a function of the geometries of the bodies and the applied force.
To solve the tangential contact problem, Kalker has among other theories developed, first, the linear theory [29]. In which it is assumed that except the trailing edge the entire contact patch is covered by the stick region. This is due to the fact that the creepages are so small that the tractions never violate the traction bound. Then, at the trailing edge the traction suddenly drops to zero. In this theory there is a linear relation between the creepages and creep forces as well as between the spin and the moment as shown in equations 6, 7 and 8.

$$F_x = -a \cdot b \cdot G \cdot C_{11} \cdot v_x,$$

(6)

$$F_y = -a \cdot b \cdot G \cdot C_{22} \cdot v_y - (ab)^{1.5} G \cdot C_{23} \cdot \varphi,$$

(7)

$$M_z = -(a \cdot b)^{1.5} C_{32} \cdot v_y - G(a \cdot b)^2 C_{33} \cdot \varphi,$$

(8)

where, $F_x$ and $F_y$ are the tangential creep forces, $v_x, v_y$ and $\varphi$ are the longitudinal, lateral and spin creepages respectively, $G$ is the shear modulus and $C_{ij}$ are the so called Kalker coefficients. Kalker coefficients are calculated and tabulated in [29] as they are only the functions of Poisson’s ratio and the contact ellipse semi-axes ratio $a/b$. Kalker then, introduced another theory called simplified theory. He used the Winkler bed theory to calculate the contact area, where an elastic foundation is introduced as a mattress with vertical springs independent of each other in a way that in compression they will not affect their neighboring springs.

![Figure 14, Winkler elastic foundation approach](image)

Therefore, due to the Hooke’s law the displacements are linearly proportional to the tractions,

$$u, v = L(p_x, p_y)$$

(9)

where, $u$ and $v$ are the displacement differences between the bodies in contact in longitudinal and lateral directions, $p_x$ and $p_y$ are tangential tractions and $L$ is the flexibility of the Winkler’s bed springs.

In steady state rolling the relative slip velocity in x and y directions, $(S_x, S_y)$, is calculated by,

$$(S_x, S_y) = (v_x - \varphi y, v_y + \varphi x) - \frac{\partial (u(x,y), v(x,y))}{\partial x},$$

(10)

In the full adhesion condition (the linear theory) no slip will occur,

$$(S_x, S_y) = 0.$$  

(11)

Thus,

$$u = v_x x - \varphi xy + k(y),$$

(12)

and likewise,

$$v = v_y x + \varphi \frac{x^2}{2} + l(y),$$

(13)

where, $k(y)$ and $l(y)$ are arbitrary functions of $y$ that arise after the integration. In the simplified theory to determine these functions, Kalker has used the fact that at the leading edge the traction
starts to build up from zero and grows with constant stress gradient through the contact until it reaches the bound and it stays at the bound level. Thus, the traction is zero only in the outside of the contact area and at the leading edge,

\[ p_x(a(y), y) = p_y(a(y), y) = 0, \]  

(14)

where, \( a(y) \) is the semi-axes of the elliptic contact area at \( y \) leading edge.

Consequently, the forces calculated by simplified Kalker theory will be,

\[ F_x = -8a^2 b v_x / (3L) \]  

(15)

\[ F_y = -8a^2 b v_y - \pi a^3 b \varphi / (4L) \]  

(16)

Note that the sign of the creep forces is always opposite to the sign of the corresponding creepages. The flexibility \( L \) then is calculated by comparing the results of the simplified Kalker equations (15) and (16) and the linear (Exact) theory calculated by the true theory of elasticity equations (6) and (7). Note that the effect of spin is considered in the lateral creep force. Moreover, Kalker presented a mathematical algorithm called FASTSIM to calculate the tangential problem in the general case where there is no need of pre-calculated tables or neglecting the spin based on the simplified theory which is widely used in vehicle dynamic simulations.

**Dynamic calculation of a wheelset (creepage and spin)**

As it is seen in equations (15) and (16) spin and creepages have great influence on the contact forces. The contact forces then affect the contact position on the wheel and rail and consequently the size and shape of the contact patch. Here is useful to briefly mention how these creepages and spin are calculated via the dynamics of a rolling wheelset on a rail. Note that the details of how to derive these equations and terms are presented in [30] chapter 8.

![Figure 15, The schematic of a wheelset on rails](image)

If \( \alpha \) is a small angle between the \( y \)-axis and the wheelset, the lateral creepage depends on

- \( \alpha \),
- Lateral velocity of the centre of the wheelset \( \dot{y} \).

Therefore, the dimensionless lateral creepage would be

\[ v_y = \frac{\dot{y}}{V} - \alpha, \]  

(17)

where, \( V \) is the rolling velocity

\[ V = R \omega, \]  

(18)
$R$ is the radius and $\omega$ is the angular velocity of the wheel.

The Longitudinal creepage is determined by

- Difference of the rolling radii
- Rotation of the wheelset about z axis
- Braking or acceleration

Note that the signs of the longitudinal creepages are different on the left and the right wheels (in absence of braking or acceleration) cf. equations 19 and 20.

\[
\nu_x^{\text{left}} = \frac{-\omega (R_{\text{left}} - R_{\text{right}})}{V} a + (V - \omega R_{\text{mean}}),
\]

\[
\nu_x^{\text{right}} = \frac{\omega (R_{\text{left}} - R_{\text{right}})}{V} a + (V - \omega R_{\text{mean}}),
\]

(19)  (20)

Here $b$ is the half distance between the contact points on the left and right wheel.

Finally the spin is a function of Kinematic and geometric spin as shown in equations (21) and (22).

\[
\varphi^{\text{left}} = \frac{\dot{a}}{V} - \frac{\lambda_{\text{left}}}{R},
\]

\[
\varphi^{\text{right}} = \frac{\dot{a}}{V} - \frac{\lambda_{\text{right}}}{R},
\]

(21)  (22)

where, $\lambda$ is the equivalent conicity of the wheel and rail.

**Wheel and rail profile**

GENSYS pre-calculates the six wheel-rail geometry functions contact position, wheel radius, contact angle, wheel lift, lateral curvature, lateral position of the contact point on wheel and lateral position of the contact point on rail by calculating the distance between wheel and rail and then calculating the contact point for the actual displacement between wheel and rail via a program called KPF (Kontakt Punkt Funktion). The program steps forward from right to left on the wheel profile and calculate the contact point function for all relative displacements between wheel and rail.

The wheels of iron-ore wagons have the so called WP4 profile. The rail profile distributions, however, are more complicated. On the Norwegian side of the line for the curve sections with radii below 600m the higher rail has the MB1 and the lower rail has the MB4 rail profiles. The MB4 rail profile is the standard UIC60 rail with a slight gauge corner relief. This moves the contact point slightly towards the field side and leads to a higher rolling radius difference and better steering performance. A comparison between the implemented rail profiles of the line is shown in Figure 16. The MB1 rail profile is more ground at the gauge corner to avoid any contact and consequently head checking on the corner of the rails. For the track sections having radii above 600m including the straight track sections, the MB4 rail profile is used for both the high and low rails. On the Swedish side of the line for all curve sections the MB1 and UIC60 rail profiles are used for the high and low rails respectively. On straight lines both rails have either MB4 or UIC60 rail profiles. The distribution of the rail profiles along the entire line is shown in Table 8.
Table 8, The rail profiles along the iron-ore line

<table>
<thead>
<tr>
<th></th>
<th>Curve radius &lt;600</th>
<th>Curve radius &gt;600</th>
<th>Straight line</th>
</tr>
</thead>
<tbody>
<tr>
<td>Norway</td>
<td>High rail Low rail</td>
<td>High rail Low rail</td>
<td>High rail Low rail</td>
</tr>
<tr>
<td>Sweden</td>
<td>MB1 assymetric MB4</td>
<td>MB4 MB4 MB4 MB4</td>
<td>MB4 MB4</td>
</tr>
<tr>
<td></td>
<td>MB1 UIC60 MB1 UIC60</td>
<td>UIC60 MB4 UIC60</td>
<td>UIC60</td>
</tr>
</tbody>
</table>

Figure 16, Comparison between the rail profiles

Any combination of the WP4 wheel profile and each of the mentioned rail profiles produces a unique wheel-rail geometry functions. Here we compare the equivalent concities for each combination mentioned in Table 8 for the nominal track gauge (1435mm), and the average track gauge of the line (1445mm). As it is shown in Figure 17, at nominal track gauge the highest equivalent conicity is achieved while both rails have the UIC60 rail profile while at the average track gauge of the line (1445mm) using MB4 rail profiles for both rails give the highest equivalent conicity. Generally higher equivalent conicity leads to greater rolling radius difference and a better steering performance in curves. While a wheel-rail profile combination with low values of equivalent conicities usually helps the stability of the vehicle in higher speeds.
Figure 17, The Equivalent conicities of the wheel and rail profiles match as a function of lateral displacement of the wheelset

It is also possible to see the contact positions on the wheel and rail depending on the lateral displacement of the wheelset. Figure 18, Figure 19 and Figure 20 show The WP4 wheel profile on UIC60, MB4 and MB1 rail profiles, respectively. As it is seen in the figures UIC60 gives the most distributed contact positions on the wheel and rail and the gauge corner relief MB1 rail profile gives the most condensed contact positions. Moreover, both the MB4 and MB1 rail profiles give two point contacts at the outer wheels in tight curves, since there is no material at the corner of the rails. This, on the one hand, will divide the load on two different contact points and avoids the concentrated load on one point leading to safer transferring the load through the fastenings and to the ballasts, especially when the fastenings are not strong enough or not positioned correctly. Having two point contacts on the outer wheel, on the other hand, decreases the steering ability. Since the directions of the longitudinal creep forces would be different on the two contact patches the total resulting steering forces decreases. Consequently, higher attack angle is predicted in these cases.
Figure 18, WP4 wheel and UIC60 rail profiles; Contact points as function of relative displacement dy
Figure 19, WP4 wheel and MB4 rail profile; Contact points as function of relative displacement $dy$
Friction coefficient

The proportionality factor of the normal load and the resistance force in sliding is called the friction coefficient. The dynamic behavior of the vehicle, among many other things, strongly depends on the coefficient of friction. On one hand a high friction coefficient is needed for adequate adhesion conditions and better radial steering. On the other hand higher friction leads to larger creep forces and will increase the risk of RCF. The coefficient of friction greatly depends on the weather, rail temperature, passing axle number and tribological surface condition like roughness, hardness etc. It may vary dramatically during a day from morning to night. In [31] the proportionality of the friction level to the rail temperature (among other parameters like humidity of the weather) is experimentally investigated. Usually from very humid to extremely dry weather, the friction coefficient can vary from 0.2 to 0.75. In cold dry climate conditions, as the water content in air drops significantly, the wheel-rail friction coefficient increases.

5. Validation

In this chapter we tried to validate the simulation model against the available measured forces and accelerations. There are two types of measurement data available from the iron-ore line. One is the track based strain gauge measurement mostly for online monitoring the vehicle performance in order to detect wheel failures. And the other one is the on track measurement via accelerometers
attached to the measurement vehicle. The later type of measurement has been performed by a company called Interfleet Technology Limited during summer of 2004 and winter of 2011.

**On track measurement**

Bogojevic [32], compared the calculated and measured carbody accelerations in vertical and lateral directions for both empty and loaded conditions. Here some more results are presented. Note that the friction level is assumed to be 0.4 for all the calculations. The 2004 measurement data is used.

![Simulated lateral acceleration (top) and vertical acceleration (bottom) of empty wagon above leading bogie on tangent track](image)

**Figure 21.** Simulated lateral acceleration (top) and vertical acceleration (bottom) of empty wagon above leading bogie on tangent track
Figure 22, Measured lateral acceleration (top) and vertical acceleration (bottom) of empty wagon above leading bogie on tangent track.
An Investigation of the Iron-Ore Wheel Damages using Vehicle Dynamic Simulation

**Figure 23**, Measured lateral acceleration (top) and vertical acceleration (bottom) of loaded wagon above leading bogie on tangent track

**Figure 24**, Simulated lateral acceleration (top) and vertical acceleration (bottom) of loaded wagon above leading bogie on tangent track
Figure 25. Simulated lateral acceleration (top) and vertical acceleration (bottom) of empty wagon above leading bogie in the curve

Figure 26. Measured lateral acceleration (top) and vertical acceleration (bottom) of empty wagon above leading bogie in the curve
An Investigation of the Iron-Ore Wheel Damages using Vehicle Dynamic Simulation

Figure 27, Simulated lateral acceleration (top) and vertical acceleration (down) above leading bogie of loaded wagon in the curve

Figure 28, Measured lateral acceleration (top) and vertical acceleration (down) above leading bogie of loaded wagon in the curve

Some results of the 2011 on track measurements are presented in Appendix a Figure 59 to Figure 76. As can be seen most of the test results are corrupted. Therefore it is not possible to use them for the validation process. For each test lateral (Y11) and vertical (Q11) force of the leading axle, lateral
acceleration of the carbody $a_y$ and the speed (km/h) signal are shown. Some of the problems and issues with the graphs are shown with red marks.

**Strain gauge based measurement**

There are two measurement stations along the iron-ore line. One is in Sweden near Sevast outside Luleå, and the other one is located in Norway near Narvik. The measurement system consists of several strain gauge sensors. These sensors measure the strain in the rail in lateral and vertical directions. The deflection is converted to the track forces after proper calibration. Figure 29 shows waterproof strain gauge measurement equipment and its sensor locations on the rail at Sevast measurement station. For more about the details of the measurement station see [33].

![Figure 29, Measurement equipment and the sensor locations on the rail](image)

To compare simulation results and the measurement data from Sevast the actual track geometry information of the section is used as the simulation inputs. The curvature of the section as a function of the section distance is shown in Figure 30.

![Figure 30, Sevast station track curvature](image)

As it is seen in the figure the length of the section is around 550 m and the curve is left handed. The rail profiles in the simulation are the profiles measured with the Miniprof equipment before and after grinding together with the nominal UIC60 and MB1 profiles are used as shown in Figure 31.
Figure 31, The rail profiles used in high and low rails for the simulation section (Sevast measurement station)

Both the worn and nominal WP4 wheel profiles are considered in the simulations. Figure 32 shows the difference between the profiles.

Figure 32, New and worn WP4 wheel profiles

The characteristics of the wheel profiles are shown in Table 9, which are in the range of the measured profiles characteristics that passed the measurement station during a year [33].

Table 9, Wheel profiles characteristics

| Flange thickness | Flange eight | Flange gradient |
The minimum, maximum and nominal values of the speed and the load of the vehicles passed the section during a year is chosen for the simulations inputs.

<table>
<thead>
<tr>
<th>New wheel</th>
<th>26.89[mm]</th>
<th>29.08[mm]</th>
<th>10.07[mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Worn wheel</td>
<td>26.78[mm]</td>
<td>30.42[mm]</td>
<td>9[mm]</td>
</tr>
</tbody>
</table>

Figure 33, The speed of the loaded wagons (left), The vertical load of the loaded wagons (right)

We also carried out a parameter study on the friction coefficient that varied between 0.2 and 0.5 to simulate the weather conditions during a year.

As the measurement results contain also high frequent dynamic forces (cut of frequency is 100 Hz), we have estimated the contribution of the high frequent dynamic forces as it is suggested in [34] and the results are compared with the measurements.

The vertical Q and lateral Y forces are calculated within the conditions mentioned above and the results are shown in Figure 34 and Figure 35.
Figure 34, Calculated vertical forces for all 4 wheels of a wagon for friction coefficient 0.4. The red lines stand for the results of the nominal case (Load: 30ton, Speed: 60km/h, Wheel profile: new WP4, Rail profiles: MB1 and UIC60); L: load and S: speed

Figure 35, Calculated lateral forces for all 4 wheels of a wagon for friction coefficient 0.4. The red lines stand for the results of the nominal case (Load: 30ton, Speed: 60km/h, Wheel profile: new WP4, Rail profiles: MB1 and UIC60); L: load and S: speed

The variations of the lateral forces with various friction coefficients are shown in Figure 36.
Figure 36, Calculated lateral forces of all 4 wheels of a wagon as function of friction coefficient nwr (new wheels, new rails), nwwr (new wheels, worn rails), wwr (worn wheels, new rails), and wwwr (worn wheels, worn rails).

Figure 37 shows the range of all calculated Q forces and Figure 38 shows the range of the calculated Y forces in comparison with the measured ones.
**Figure 37**, Comparison between the measured and calculated vertical forces. a) left wheel, b) right wheel, the blue and red lines show the maximum and minimum of the range of the calculated forces while the background gray diagrams show the distribution of the measured forces in a year.

**Figure 38**, Comparison between the measured and calculated lateral forces. a) left wheel, b) right wheel, the blue and red lines show the maximum and minimum of the range of the calculated forces while the background diagrams show the distribution of the measured forces in a year.
As can be seen in the figures there is a fairly good agreement between the simulation results and the measurements. The highest deviation occurred in the outer wheel of the leading bogie. The vehicle usually runs with flange contacts at these conditions. And usually the flange contact (especially for worn wheels and rails with more conforming contact) violates the half space in the Hertzian theory assumption which may lead to non-realistic results.

6. Wheel damage mechanisms
From the tribology point of view, any damage to a solid surface, involving progressive loss of material and relocation of material, when two surfaces are interacting via a relative motion, is called wear. How a material wears depends not only to the nature of the material but also depends on other elements of the tribo-system such as geometry of contacting pairs, surface topography, loading, lubrication and environment. The mechanisms causing such damages usually are complicated and most of the time it is not possible to distinguish one from another. In [35], approximately, 60 terms describing wear behavior and mechanisms are listed. Some of the most important wheel-rail related mechanisms are listed below.

- Abrasive wear: wear caused by rough and hard surfaces sliding on each other or wear caused by hard particles trapped between two surfaces like hard oxide debris.
- Adhesive wear: wear caused by shearing of junctions formed between two contacting surfaces, sometimes used as a synonym for dry sliding wear.
- Chemical wear (Corrosive wear): wear caused by formation of any oxide or other components on surfaces due to chemical reaction of the surfaces with the environment.
- Erosive wear: Wear due to relative motion of contact surfaces while a fluid containing solid particles is between the surfaces
- Rolling contact fatigue (RCF): caused by cyclic stress variations leading to fatigue of the materials. Generally resulting in the formation of sub-surface and deep-surface cracks, material pitting and spalling.

Kimura [36] studies adhesive wear and RCF and concludes that both phenomena have elemental processes in common. In this study the term “wear” is used for adhesive wear otherwise the complete term of the mechanism is mentioned. Note that the term “mechanical wear” is also used for abrasion, erosion, adhesion and surface fatigue.

Wear modeling
Published research and studies in wear modeling usually have three approaches:

- field measurements,
- laboratory tests and
- theoretical prediction models.

Most of the field measurements have addressed the effect of lubrication on wear such as [37] and [38].
Archard [39] took the idea of the adhesive wear definition and found that the volume of material removed by wear per sliding distance \((W)\) is proportional to the quotient of the load \((p)\) and the hardness \((H)\) of the softer material. The proportionality factor is called the wear coefficient \((k)\).

\[
W = k \frac{p}{H}
\]

(23)

The wear coefficient depends on the governing wear mechanisms. Archard validated his model by determining the wear rates for different material pairs by pin-on-ring tests.

Lim and Ashby [40] have performed large amounts of laboratory tests and introduced wear maps where the wear coefficient is plotted as a function of sliding velocity and the nominal pressure (normal load divided by the nominal contact area). The wear map corresponding to medium carbon steel, based largely on pin-on-disc data, is generally divided into two main regions of mechanical and chemical wear, cf. Figure 39. The mechanical wear occurs at low sliding velocities where the wear coefficient is more a function of nominal pressure than the speed. And the chemical mechanism occurs in higher sliding velocity (above 1 m/s). The mechanical part contains three regions of mild and severe wear together with a transition area in between. Childs [41] has suggested a wear map for the mechanical wear mechanism where the wear coefficient is a function of the asperities attack angle and the relative strength of the interfaces. The chemical part of the map, however, contains of two regions: mild and sever oxidational wear. As it is seen in the figure mild oxidation could be even protective as at a given level of load and high sliding speed the wear coefficient drops to low values. This mild oxide material behaves like a lubricant in between the surfaces.

**Figure 39, Lim and Ashby wear map for medium carbon steel based largely on pin-and disc data.**

Olofsson and Telliskivi [42] have investigated the evolution of the rail profiles of a commuter track within two years together with performing several laboratory tests with two different testing machines: a two-roller (disk on disk) and a pin-on-disk machine. The results of the tests were a simpler wear map with the wear coefficients depending on sliding velocity and the contact pressure, cf. Figure 40.
Figure 40, Wear map wheel and rail steel with typical regions of tread and flange contacts; $H$ is the hardness of the material.

In a different approach McEven and Harvey [43] using a full-scale wheel-on-rail-wear rig have proposed a linear relation between the wear rate and the dissipated energy per unit distance rolled $\bar{E}$, per unit area $A$, adjusted with a constant off-set term, $K$. The energy dissipation per unit distance area is the creep forces times the creepages added to the moment times the spin in the contact patch. They also have predicted two wear regimes of mild (tread contact) and severe (flange contact) wear.

\[
\text{Wear rate Per unit rolled distance} = k \frac{E}{A} + K, \tag{24}
\]

\[
\bar{E} = F_x v_x + F_y v_y + M \phi. \tag{25}
\]

\(\frac{E}{A}\) sometimes is called the wear number. In another attempt [44] the authors suggested a simple relationship between the material loss and the energy dissipation:

\[
\begin{align*}
\bar{E} < 100\text{N} & \quad \text{Material loss}(mm^2) = 0.25 \frac{E}{D}, \tag{26} \\
100 \leq \bar{E} \leq 200 \text{N} & \quad \text{Material loss}(mm^2) = \frac{25}{D}, \tag{27} \\
\bar{E} \geq 200\text{N} & \quad \text{Material loss}(mm^2) = \frac{(119E - 154)}{D} \tag{28}
\end{align*}
\]

where, $D$ is diameter of the wheel in millimeters.

Krause and Poll [45] reviewed several other tests results and investigations regarding the proportionality of the wear rate to the longitudinal creepage, normal force and sliding distance. They concluded that the material loss is proportional to the friction work $W_R$

\[
W_R = \bar{E}.S, \tag{29}
\]

where, $S$ is the sliding distance. The proportionality factor is shown by $I_w$ and it depends on the environmental conditions such as the humidity, the material of surfaces and the temperature of the contact patch. The temperature itself is proportional to the frictional power $P_R$

\[
P_R = \bar{E}.V, \tag{30}
\]
where, $V$ is the speed. They also concluded that it is difficult to derive a simple mathematical wear law because:

- different parameters are affecting each other; like frictional work affects the surface temperature which changes the tribology of the surfaces and material behavior.
- different mechanisms are involved in a wear process.

**Prediction of profile evolution**

Several authors have investigated and developed wheel wear prediction tools. Some of them are reviewed in this section. Note that the studies which are mentioned here have neglected the plastic deformation of the material and focused on the uniform wear.

Kalker [46] has developed a method to predict the wheel profile evolution considering that the material loss is proportional to the frictional work and proportionality gained from field studies. His method was only applied on the tread contact and no flange wear is reported. As he has used the Hertzian contact and the simplified theory for his calculations it was not possible to predict any severe wear on the wheel profile as the wheel gets the hollow shape and the contact becomes more conforming. No comparison between measurements and calculations are mentioned.

Pearce and Sherratt [47] have developed their prediction method using the energy approach where the amount of material loss is proportional to the energy dissipation. The values of the wear coefficients are taken from [44]. In order to validate the model they have compared the development of the equivalent conicity over the running distance from both simulation and measurement; the conclusion was that only a qualitative judgment is possible.

Ward et al. [48] have also followed the energy approach. To find the wear coefficients they have carried out twin disk testing. The wheel profile is discretized into longitudinal strips and each contact patch is divided into several cells. The numbers of cells are equal for each strip. The wear depth is then, calculated separately in each of the cells and at the end integrated along the longitudinal strips as shown in Figure 41.

**Figure 41**, Summation of the wear per strip to give total wear depth; $t$ is the duration of the contact and $C_{pw}$ is the position of the center of the contact point
Jendel [49] has used Archard’s approach instead where he developed his method. The used methodology is based on a load collective concept, which determines a set of dynamic time-domain simulations. These simulations reflect the actual rail network for vehicles in question including for example, design geometry, track irregularities, rail profiles and the vehicle operating conditions. Hertzian theory is used for the normal contact and the FASTSIM method is applied for the tangential contact problem. The simulation results are compared with measurements and good agreement was observed. Summary of the methodology is presented in Figure 42.

![Methodology diagram](image)

**Figure 42, Methodology of the wheel wear prediction tool developed by Jendel [49]**

Enblom [50] has used the same methodology as Jendel. However, he also included the elastic strain in the sliding velocity assessment *cf.* Equation (10). He also increased the simulation set with simulation of braking.

**Rolling contact fatigue**

The current wear models do not include the role of RCF. However, the trade of between wear and RCF is of great interest. In [51], the implementation of emerging technologies for the prediction of wheel surface deterioration in an engineering environment is summarized. In this section we try to touch the concept and theories of RCF.

Every material subjected to repeated rolling contact loads responses in the following four ways as shown in Figure 43:

a. Perfectly elastic; if the load is sufficiently small so that no part of the material can reach the elastic limit, the response will be perfectly elastic.

b. Elastic shakedown; if the load exceeds the elastic limit in the first few cycles there will be small plastic deformation. However, due to the changes brought by the plastic flow (residual stresses, strain hardening and more conforming contact) the steady state would remain elastic. The maximum load which allows the material to be in this regime is called shakedown limit. Note
that the material is subjected to high cycle fatigue (HCF) and it will be expected to have cracks in long term.

c. Plastic shakedown; the steady state consists of a closed plastic stress-strain loop. In this regime the material is subjected to so called low cycle fatigue (LCF).

d. Ratcheting; the plastic deformation will remain unstable and there will be an incremental strain growth at each load cycle. Here, the surface of the material soon will be subjected to cracks.

Figure 43, Material response to cyclic loading

Johnson [51] has used the Von Mises yield criterion and developed a shakedown map in line contact for a perfectly plastic and kinematically hardening material which is widely used in detecting the RCF probabilities in railway wheels and rails. The shakedown map is presented in Figure 44.

Figure 44, Shakedown map
As it is seen the figure for friction levels approximately, above 0.3 in high levels of loads surface initiated fatigue will be developed and below this level subsurface flow will occur. An engineering model for RCF risk assessment is developed by Ekberg in [53] based on the shakedown theory cf. Figure 45 (left). Surface initiated fatigue is often characterized as low cycle fatigue and it usually happens when the plastic deformation remains unstable and the strain accumulates until the ductility of the material is exhausted (ratchetting). The location of the points in the shakedown diagram is a function of, firstly, the traction coefficient ($T$)

$$T = \frac{F_T}{F_z} = \frac{F_x^2 + F_y^2}{F_z},$$  \hspace{1cm} (31)

where $F_z$ is the normal contact force, $F_x$ and $F_y$ are longitudinal and lateral creep forces respectively. Secondly, it depends on the normalized vertical load ($v$) which is the maximum contact pressure ($P_0$) divided by the material yield stress in shear ($k$)

$$v = \frac{P_0}{k}.$$  \hspace{1cm} (32)

The boundary curve for surface plasticity is denoted by BC and calculated as

$$BC = 1/T.$$  \hspace{1cm} (33)

The horizontal distance between the working points and the BC line yields

$$FI_{surf} = T - 1/v = T - \frac{2a.b.k}{3F_z},$$  \hspace{1cm} (34)

where $a$ and $b$ are the semi-axes of the elliptic contact area in the Hertzian contact. Positive values of $FI_{surf}$ represent the ratchetting part of the shakedown diagram resulting in surface initiated fatigue. The area below the $FI_{surf}$ curve represents the depth of the ratcheted working points and it is particularly used as an index of RCF severity in the study of the influence of the wheel-rail friction coefficient. Here the number of the ratcheted points over the whole number of working points is called RCF probability. Figure 45 (right) shows the typical values of $FI_{surf}$ as a function of time while the vehicle is negotiating a curve, in the figure the positive values are indicated as “Area with risk of RCF”. The negative values are valid for straight track, while the positive values occur when the vehicle is running in the curve.

The longitudinal and lateral traction coefficients are also presented in the following form:

$$T_x = \frac{F_x}{F_z},$$ \hspace{1cm} (35)

$$T_y = \frac{F_y}{F_z}.$$ \hspace{1cm} (36)
Track geometry set for the simulations
Based on the statistics of the line the following curve radii, Figure 46, track gauge span, Table 10 and track cant span, Table 11, are used for the simulation sets.

<table>
<thead>
<tr>
<th>Character</th>
<th>Track gauge span (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>G1</td>
<td>1430 - 1435</td>
</tr>
<tr>
<td>G2</td>
<td>1435 - 1440</td>
</tr>
<tr>
<td>G3</td>
<td>1440 - 1445</td>
</tr>
<tr>
<td>G4</td>
<td>1445 - 1450</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Character</th>
<th>Track cant span (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>C1</td>
<td>25 - 50</td>
</tr>
<tr>
<td>C2</td>
<td>50 - 75</td>
</tr>
<tr>
<td>C3</td>
<td>75 - 100</td>
</tr>
<tr>
<td>C4</td>
<td>100 - 130</td>
</tr>
</tbody>
</table>
Influence of cant deficiency

For typical operational speeds of the iron-ore trains, equilibrium cant in the small radius curves R1 is in the range of C4 track cant. However, also the highest probability of RCF occurs for equilibrium cant. When running at cant deficiency centrifugal forces push the vehicle towards the outer rail and therefore the wheelset is able to move further laterally and achieve a higher rolling radius difference. This helps the wheelset to reduce the attack angle and steer better in curves. Consequently, less attack angle is predicted. Thus the RCF risk on wheels will be lower. Due to low generated steering forces in curves with larger radius, this is not seen in R2 cases. Figure 47 shows the RCF probability of both inner and outer wheel for various cant deficiencies.

One should note that the higher the cant deficiency \( h_d \) the higher the track plane acceleration \( a_y \)

\[
a_y \approx \frac{g}{2b_0} \cdot h_d ,
\]

(39)

where, \( 2b_0 \) is the distance between the nominal wheel-rail contact points on left and right side and \( g \) is the acceleration of gravity.

To compensate the high generated centrifugal force, lateral forces arise on the high rail, where wheel flanges are in contact with the rail. In extreme cases and in very high lateral track forces severe damage on the rail can occur. This may also result in flange climbing and derailment. In conclusion there are two effects that work against each other.

Figure 47, RCF probability vs. Track Cant; the track Gauge for both R1 and R2 cases is G3
Influence of gauge and track irregularities

Larger vertical track irregularities increase the dynamic contribution of the vertical track forces. In combination with unfavorable contact geometry that causes small contact patches, the vertical load can cause high stresses. On spots where the material has some locally low fatigue resistance the risk of sub-surface initiated fatigue is high. Lateral track irregularities, however, have more complicated impact on the RCF risk. Karttunen has studied the influence of lateral irregularities on rails [54]. He concludes that the irregularities result in a shift of the contact point either towards the gauge corner or towards the field side of the rail. The risk of RCF on rails depends then on the contact area size.

In this section to study both the effect of lateral track irregularities and track gauge on RCF on the wheels, based on the statistics in section “Track” four track sections are picked to cover the whole range from good to very bad track. All four sections are in the same range of the track cant (C4). The track gauge is changed manually from 1435 mm to 1455 mm. Table 12 shows data of the chosen track sections.

<table>
<thead>
<tr>
<th>Track Section #</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Track Quality</td>
<td>QN=1</td>
<td>QN=2</td>
<td>QN=3</td>
<td>3&lt;QN</td>
</tr>
<tr>
<td>Length(m)</td>
<td>308</td>
<td>358</td>
<td>327</td>
<td>662</td>
</tr>
<tr>
<td>Radius(m)</td>
<td>300</td>
<td>300</td>
<td>300</td>
<td>300</td>
</tr>
<tr>
<td>Lateral track error, standard deviation(mm)</td>
<td>1.22</td>
<td>1.53</td>
<td>1.74</td>
<td>3.83</td>
</tr>
<tr>
<td>Vertical track error, standard deviation(mm)</td>
<td>1.98</td>
<td>2.49</td>
<td>2.69</td>
<td>3.08</td>
</tr>
</tbody>
</table>

In the track sections with larger track irregularities a higher probability of RCF on wheels is predicted, as shown in Figure 48. The presented results are calculated for the inner leading wheel.

![Figure 48](image)

**Figure 48**, RCF probability as a function of track quality level and track gauge on the inner leading wheel

The main reason is the higher generated traction coefficient. It seems that in poorer track sections, higher steering forces are needed to steer the vehicle and keep it stable, see Figure 49. However, in wider track gauges the contact point on the inner wheel gets closer to the field side of the wheel and
the contact patch area decreases down to 20% of the value at 1435 mm track gauge, as shown in Figure 50. Consequently, the working points of the shakedown map move further towards the north and the risk of surface initiated fatigue will be higher.

**Figure 49**, Calculated 99.85 percentile of traction coefficients as a function of track quality level and track gauge on the inner leading wheel

![Traction Coefficient Graph](image)

**Figure 50**, Calculated 99.85 percentile of contact patch area as a function of track quality level and track gauge on the inner leading wheel

**Influence of friction coefficient**

As the friction coefficient increases, the adhesion condition improves and higher steering forces (longitudinal creep forces) are provided. This is true until the longitudinal forces get high enough to affect the angle of attack and, subsequently, the lateral creep forces decrease. As Figure 51 shows, the lateral traction coefficient \( T_y \) saturates at the friction level of 0.5; however, the longitudinal traction forces continue to grow linearly. As can be seen the total traction coefficient saturates with
the value of 0.4 at a friction level of around 0.65. The curve section studied has 300 m radius, G3 gauge interval and C4 track cant. In wider curves the traction coefficient saturates at lower friction levels as less creep forces are needed to steer the vehicle. Note that with the current load at a traction coefficient of around 0.3 surface initiated rolling contact fatigue is predicted to occur.

**Figure 51.** 99.85 percentile of calculated longitudinal, lateral and resulting tangential traction coefficient

Figure 52 shows the area below the $F_{l_{surf}}$ curve ($A_i$) for each simulation case with varying friction coefficient and the maximum possible area at friction coefficient 0.75 ($A_{0.75}$). Figure 53 shows the quotient between $A_i$ and $A_{0.75}$ for each simulation case. As it is seen in Figure 53 with the same normal force, an increase in the friction coefficient leads to a shift of the shakedown diagram working points to the ratchetting part and results in higher probability of RCF.

**Figure 52, Depth of the ratcheted points in $F_{l_{surf}}$ curve**
Influence of the seasonal variations of track stiffness

The track stiffness is not constant and varies along the track. Moreover, the properties of the track components, like rails, rail pads, fastenings, etc., can also differ and influence the fluctuation in the track forces. To investigate the effect of seasonal variation of the track stiffness on RCF and wear the vertical rail-track stiffness and viscous damping are reduced and increased by a factor of ten, see section “Track model”. Changing the vertical track stiffness not only affects the vertical forces but also has an impact on the lateral track forces due to the roll motion of the vehicle. In [55] it is shown that the amplitude of lateral forces is higher in the vertically stiffer track in the frequency range 30-60 Hz. The results of the present work support this conclusion. However, no significant difference is observed regarding RCF and the wear number. Moreover, there has not been any study showing that the frequency of the forces affects RCF of the wheels. Therefore, the seasonal variations of track stiffness should not be the main contributor to RCF of the wheels.

Energy dissipation and RCF

Wear is not always unfavorable. At a certain level of energy dissipation the initiated surface cracks can be worn away before they start to propagate through the surface. However, at higher levels of energy loss, wear can be dominating and become problematic. A model has been proposed and calibrated in UK on six intermediate radius curves, observing RCF on rails. Good agreement between simulation and observation has been achieved with regard to the location of RCF, see [56]. No such measurements have been made on Malmbanan. However, the RCF locations on wheels are known. A field study [57] indicates that only the tread of the inner wheel is affected by RCF; No cracks on flanges are reported. The simulation results show a very good agreement with the observations with regards to the RCF positions on wheels. 43 curves with various conditions are used as inputs. The average dissipated energy of the positions with risk of RCF is also calculated, see Figure 54. Comparing the calculated results and the observations on crack locations on wheels indicates that for \( \bar{E} \) values below approximately 65 /m there is a low risk of wear and high risk of RCF. For 65 < \( \bar{E} \) ≤ 90 /m, wear starts to dominate and for higher values of energy dissipation the wear rate is high enough to be a problem itself. Note that the mean values of the calculated energy dissipation are used here.
Figure 54, Calculated RCF positions on the wheel with corresponding average wear numbers. The very left line is also reported as the observed approximate location for RCF initiation.

7. Optimized wheel profiles

Within a project the consultant company MiW Rail Technology suggested three wheel profiles for the wagons to reduce the amount wheel damages. In this section it is tried to compare those three wheel profiles and choose the best one. The wheel profiles are compared in Figure 55.

Figure 55, Wheel profiles cross section

The track section is chosen to be the same that the consultant company used for its optimization calculations and is shown in Figure 56.

Figure 56, Curvature of the line
Safety
As in the process of profile generation the safety factor for derailment is considered it is not studied here.

Statistical values
The RMS, 99.85 percentile and the maximum values of the calculated results for the first axles of the first and second cars are shown in Table 13.

Table 13 depicts the generated lateral forces. As can be seen Wheel 2 generates the lowest dynamic forces; however, the differences of the RMS values are small.

Table 13, Lateral forces [kN]

<table>
<thead>
<tr>
<th></th>
<th>Wheel 1</th>
<th>Wheel 2</th>
<th>Wheel 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Y111r</td>
<td>4-18-19</td>
<td>4-14-15</td>
<td>5-20-21</td>
</tr>
<tr>
<td>RMS</td>
<td>99.85%</td>
<td>99.85%</td>
<td>99.85%</td>
</tr>
<tr>
<td>Max.</td>
<td>6-28-29</td>
<td>5-23-23</td>
<td>7-31-31</td>
</tr>
<tr>
<td>Y111l</td>
<td>6-28-29</td>
<td>5-23-24</td>
<td>7-31-31</td>
</tr>
<tr>
<td>RMS</td>
<td>99.85%</td>
<td>99.85%</td>
<td>99.85%</td>
</tr>
<tr>
<td>Max.</td>
<td>6-28-29</td>
<td>5-23-24</td>
<td>7-31-31</td>
</tr>
<tr>
<td>Y211r</td>
<td>5-18-19</td>
<td>4-14-14</td>
<td>5-20-21</td>
</tr>
<tr>
<td>RMS</td>
<td>99.85%</td>
<td>99.85%</td>
<td>99.85%</td>
</tr>
<tr>
<td>Max.</td>
<td>6-28-29</td>
<td>5-23-24</td>
<td>7-31-31</td>
</tr>
<tr>
<td>Y211l</td>
<td>6-28-29</td>
<td>5-23-24</td>
<td>7-31-31</td>
</tr>
<tr>
<td>RMS</td>
<td>99.85%</td>
<td>99.85%</td>
<td>99.85%</td>
</tr>
<tr>
<td>Max.</td>
<td>6-28-29</td>
<td>5-23-24</td>
<td>7-31-31</td>
</tr>
</tbody>
</table>

Table 14 and Table 15 are showing the calculated longitudinal forces on the tread and the flange respectively. It is observed that Wheel 2 generates the highest values of longitudinal forces which can explain why it produces lower lateral forces. However, again the RMS values do not differ much.

Table 14, Longitudinal forces generated on the tread [kN]

<table>
<thead>
<tr>
<th></th>
<th>Wheel 1</th>
<th>Wheel 2</th>
<th>Wheel 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>X111r_tread</td>
<td>7-21-21</td>
<td>8-29-31</td>
<td>7-21-23</td>
</tr>
<tr>
<td>RMS</td>
<td>99.85%</td>
<td>99.85%</td>
<td>99.85%</td>
</tr>
<tr>
<td>Max.</td>
<td>6-15-14</td>
<td>6-15-15</td>
<td>6-14-15</td>
</tr>
<tr>
<td>X111l_tread</td>
<td>6-15-14</td>
<td>6-15-15</td>
<td>6-14-15</td>
</tr>
<tr>
<td>RMS</td>
<td>99.85%</td>
<td>99.85%</td>
<td>99.85%</td>
</tr>
<tr>
<td>X211r_tread</td>
<td>7-21-21</td>
<td>8-29-30</td>
<td>7-21-23</td>
</tr>
<tr>
<td>RMS</td>
<td>99.85%</td>
<td>99.85%</td>
<td>99.85%</td>
</tr>
<tr>
<td>RMS</td>
<td>99.85%</td>
<td>99.85%</td>
<td>99.85%</td>
</tr>
</tbody>
</table>

Table 15, Longitudinal forces generated on the flange [kN]

<table>
<thead>
<tr>
<th></th>
<th>Wheel 1</th>
<th>Wheel 2</th>
<th>Wheel 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>X111r_flange</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>RMS</td>
<td>99.85%</td>
<td>99.85%</td>
<td>99.85%</td>
</tr>
<tr>
<td>Max.</td>
<td>8-35-36</td>
<td>6-29-30</td>
<td>7-31-31</td>
</tr>
<tr>
<td>X111l_flange</td>
<td>8-35-36</td>
<td>6-29-30</td>
<td>7-31-31</td>
</tr>
<tr>
<td>RMS</td>
<td>99.85%</td>
<td>99.85%</td>
<td>99.85%</td>
</tr>
<tr>
<td>Max.</td>
<td>8-35-36</td>
<td>6-29-30</td>
<td>7-31-31</td>
</tr>
</tbody>
</table>
Table 16 shows the attack angle of the first axle of each car. Although Wheel 2 creates the highest attack angle but the maximum values are generated on the straight line section while the maximum attack angle corresponded to wheel 1 and wheel 3 happened in a curve section. This shows that the vehicle runs on Wheel 2 probably experiences some extreme hunting motion on straight track sections.

Table 16, Attack angle of the first axles of each car [E-3 rad]

<table>
<thead>
<tr>
<th></th>
<th>Yaw_111</th>
<th>Yaw211</th>
</tr>
</thead>
<tbody>
<tr>
<td>RMS</td>
<td>99.85%</td>
<td>Max.</td>
</tr>
<tr>
<td>Wheel 1</td>
<td>839-2912-2956</td>
<td>837-2915-2961</td>
</tr>
<tr>
<td>Wheel 2</td>
<td>1369-4409-3959</td>
<td>1364-4393-3941</td>
</tr>
<tr>
<td>Wheel 3</td>
<td>861-2717-4054</td>
<td>859-2715-4148</td>
</tr>
</tbody>
</table>

The square root of the elliptic contact area’s axes are calculated and shown in table 5. The results are representing the first axle’s tread and flange contact area. It is seen that Wheel 1 has the largest contact area especially on the flange part. This can result in lower values of stresses and decrease the risk of RCF.

Table 17, Geometrical average radii of the contact ellipse, calculated as: \( c = \sqrt{a \cdot b} \)

<table>
<thead>
<tr>
<th></th>
<th>( \text{Area}_{111r}_\text{tread}(E-6) )</th>
<th>( \text{Area}_{111l}_\text{tread}(E-6) )</th>
<th>( \text{Area}_{111r}_\text{flange}(E-6) )</th>
<th>( \text{Area}_{111l}_\text{flange}(E-6) )</th>
</tr>
</thead>
<tbody>
<tr>
<td>RMS</td>
<td>99.85%</td>
<td>Max.</td>
<td>99.85%</td>
<td>Max.</td>
</tr>
<tr>
<td>Wheel 1</td>
<td>7074-8361-8739</td>
<td>6927-7909-8864</td>
<td>0</td>
<td>1274-5283-5289</td>
</tr>
<tr>
<td>Wheel 2</td>
<td>6927-8033-8716</td>
<td>6865-7737-8907</td>
<td>0</td>
<td>550-2637-2732</td>
</tr>
<tr>
<td>Wheel 3</td>
<td>7036-8056-8914</td>
<td>6933-7797-8819</td>
<td>0</td>
<td>844-3653-3682</td>
</tr>
</tbody>
</table>

The higher values of energy dissipation generate more wear. Table 18 lists the sum of the \( E \) values calculated on the flange and tread of the wheels of the first axles. The values are almost the same for all cases; however, wheel 2 generates creates slightly more dissipated energy on the tread side and wheel 3 on the flange side.

Table 18, Sum of the dissipated energy on the tread and flange (Wear number) [J/m]

<table>
<thead>
<tr>
<th></th>
<th>( \bar{E}_{111r} )</th>
<th>( \bar{E}_{111l} )</th>
<th>( \bar{E}_{211r} )</th>
<th>Tgamma111l</th>
</tr>
</thead>
<tbody>
<tr>
<td>RMS</td>
<td>99.85%</td>
<td>Max.</td>
<td>99.85%</td>
<td>Max.</td>
</tr>
<tr>
<td>Wheel 1</td>
<td>16-73-74</td>
<td>128-566-569</td>
<td>16-73-74</td>
<td>128-566-570</td>
</tr>
<tr>
<td>Wheel 2</td>
<td>19-87-95</td>
<td>125-568-580</td>
<td>19-87-92</td>
<td>125-568-577</td>
</tr>
<tr>
<td>Wheel 3</td>
<td>16-71-73</td>
<td>133-595-602</td>
<td>16-71-73</td>
<td>133-596-603</td>
</tr>
</tbody>
</table>
Behavior on straight track
The track section is around 600 m long. Figure 57 shows the lateral forces of the first axle of the leading car. It can be seen that Wheel 2 would cause severe hunting of the vehicle. This motion is not severe enough to result in derailment and continues with lower amplitudes in the following wide curve; however it is not favorable.

![Figure 57, Lateral forces on straight track](image)

Shake down diagram and RCF
All three wheel profiles only suffer from RCF in the tightest curve of 400 m radius. However, Wheel 2 seems to be worst. Figure 58 shows the RCF index calculated for the tightest curve. Note that the positive values correspond to the ratchetting area of the shake down diagram and indicates surface initiate fatigue.

![Figure 58, Surface initiated fatigue index](image)
Position on the wheels

Generally, the contact position span on the wheel 1 is between -40 to 25 for Wheel 2 is -30 to 25 and for Wheel 3 is between -35 to 20 mm. It seems that wheel 1 has the largest and Wheel 3 has the shortest contact position span. Rolling contact fatigue occurs on the tread somewhere between -25 to -35 mm and on the flange somewhere between 20 to 25 mm. However, the average wear number on the flange side is considerably high and can clean the initiated surface fatigue before they even start to propagate.

As a conclusion to this section we can say that all the three wheel profiles seem to be fine. However, Wheel 1 and Wheel 3 have slightly better performance on both curves and straight tracks than wheel 2. Comparing Wheel 1 and Wheel 3 and considering generated lateral and longitudinal creep forces, one can say that **Wheel 1** has slightly better performance while it generates the lowest angle of attack and lateral forces in the tightest curve.
References


Appendix a
Results from 2011 on track measurement.

Figure 59, Results of on track measurements 2011, Test 112
Figure 60, Results of on track measurements 2011, Test 110
Figure 61, Results of on track measurements 2011, Test 109
Figure 62, Results of on track measurements 2011, Test 106
Figure 63, Results of on track measurements 2011, Test 106
Figure 64, Results of on track measurements 2011, Test 104

Figure 65, Results of on track measurements 2011, Test 102
An Investigation of the Iron-Ore Wheel Damages using Vehicle Dynamic Simulation

Figure 66, Results of on track measurements 2011, Test 99
Figure 67, Results of on track measurements 2011, Test 23
Figure 68, Results of on track measurements 2011, Test 22
Figure 69, Results of on track measurements 2011, Test 12
Figure 70, Results of on track measurements 2011, Test 11

The average high values are around 50 kN, however; the static load is 150 kN.
Figure 71, Results of on track measurements 2011, Test 9
Figure 72, Results of on track measurements 2011, Test 8
**Figure 73**, Results of on track measurements 2011, Test 7
**Figure 74**, Results of on track measurements 2011, Test 5
Figure 75, Results of on track measurements 2011, Test 3
Figure 76, Results of on track measurements 2011, Test 3