Hjulprofil för godsvagnar i Sverige
Wheel profiles for freight wagons in Sweden

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1 INTRODUCTION

“Hjulprofil för godsvagnar i Sverige - Wheel profiles for freight wagons in Sweden” is a project funded by Trafikverket with the collaboration of Green Cargo AB (GC), Tikab Strukturmekanik AB (Tikab) and Kockums AB.

1.1 BACKGROUND

The most commonly used wheel profile for freight vehicles in Sweden is the S1002, which came out from the investigations of the ORE Working Group in the early 70s. This profile is not a de facto standard and some countries are using their own modified profiles adapting better to their own conditions. S1002 was designed for a rail inclination of 1:40, which differs from the Swedish rail inclination of 1:30 and leads to unfavourable contact conditions between wheel and rail that may cause additional wear on both elements as:

- The radial steering capability can be significantly deteriorated, even completely absent in particular cases.
- Has initial high wear of rails and wheel flange.

Moreover, normal and tangential tensions in the wheel-rail contact area significantly contribute to the emergence of different types of surface damage both on wheels and rails. Therefore it is desirable to reduce contact stresses, which can be particularly difficult for the increased axle loads of freight vehicles. The contact geometry can also generate the so-called low-frequency hunting instability on tangent track or large radius curves, which leads to decreased comfort and increased track forces.

1.2 OBJECTIVES

The objectives of the project were defined in the original project plan as follows:

- To reduce the maintenance cost of the wheels and rails (wear, track degradation and rolling contact fatigue) by optimizing wheel profiles.
- Explore how the new running gear and traffic arrangements for freight vehicles with increased axle loads and higher speeds generate different requirements for wheel profiles.
- Modify wheel or rail profile so that hunting stability is reduced for Swedish wagons.

The results of the project will lead to a longer wheel life in freight traffic.

2 METHODOLOGY

The wear calculation methodology that will be used has been developed at KTH Rail Vehicles Division and KTH Tribology Group since 1997. For further information about the methodology we refer to references [1–4].

Validation of the wear calculation methodology

This wear prediction methodology has been validated only for passenger vehicles and locomotives, but not for freight trains. Both the use of the vehicles and the lines where they run are completely different. Specifically, the main differences between freight trains and other rail vehicles are highly nonlinear suspension elements and high axle load, both having notable influence on creep forces that will cause wheel and rail wear [5], [6].
Thus the wear calculation methodology will first be validated for freight vehicles, and the influence of freight traffic specific characteristics will be analysed; i.e. nonlinear suspension elements as friction dampers, high maximum axle load (>25 t/axle) or high axle load difference between laden (>25 t/axle) and unladen (~5 t/axle). To fully understand the behaviour in different networks, an analysis of the influence of rail conicity (1/20, 1/30 or 1/40), track irregularities and different rail profile wear levels will be carried out. The methodology will be the following:

- First, a selection of vehicles will be done according to the following requirements:
  - Availability of wheel profile measurements, or vehicles to carry out such measurements.
  - The vehicle must run on a line that can be characterized.
  - Possibility to build a multibody model of the vehicle and validate its dynamic behaviour.
- Second, the wheel wear model will be integrated with the multibody model of each vehicle under study.
- Then, the results of the simulations will be compared to the profile measurements in order to validate the wear prediction methodology for freight vehicles.
- Once the prediction model is validated, different parametric studies will be carried out in order to check how the specific characteristics of freight vehicles influence wheel damage generation.

### Optimisation of the wheel profile

The wheel profile will be optimised, minimizing the uniform wear, the material to be removed in each repolishing, and the low frequency instability of different vehicles. The actual freight vehicle fleet is not necessarily the same that will be running the next 50 years, and optimisation must be carried out with a view to long term usage. So first, the evolution of the freight train fleet during the last years will be investigated, in order to determine the different type of vehicles that will be used in the future.

To fulfil this objective, the so called Genetic Algorithm optimization technique will be used to optimise the wear, which will lead to the optimised wheel profile.

### Validation of the new profiles

The optimized wheel profile will be installed in some freight vehicles in order to experimentally validate its benefits and detect further improvements. Different kind of freight vehicles will be used for the validation to achieve a broad feedback on the wear results.

#### 2.1 SIMULATION CASES

In order to validate the wear calculation methodology for freight vehicles we need a vehicle that runs in a restricted area and that we can obtain or perform experimental measurements of wheel profiles. In a first meeting Green Cargo suggested two operational cases that could be studied:

- Operational case 1: steel slab traffic between Luleå and Borlänge. Single line that takes steel slabs from Borlänge to Luleå and goes back unladen. Y25 bogie vehicles. Wagons leased by AEE to SSAB.
- Operational case 2: timber wagons in the Trätåg line. Laaps wagons with Unitruck running gear that run in a network of around 10 stations transporting timber lodges.

Neither of the lines has experimental profile measurements available, so permission was asked in order to perform our own measurements. In the first case the leasing company was not prepared to give the experimental wheel profiles away. According to GC their concern was that we could theoretically determine that their hollow profiles were not good enough for operation and that they could get into trouble because of these results.
2.1.1 OPERATIONAL CASE 1

This case was dropped when we realised that there was no possibility of obtaining experimental measurements of the wheel profiles. However, some data was obtained on the route and wear simulations.

2.1.1.1 Route

Route between Luleå and Borlänge (Figure 1). A small section of the route is double track, while most of it is single track.

Route data was obtained from Trafikverket and the operational case was built. The route comprises c.a. 1000 km of which c.a. 440 km are tangent track, c.a. 275 km right hand curves and c.a. 265 km left hand curves. In Figure 2 curve radius density is depicted. The symmetry of the network is confirmed, as right- and left-hand curves have almost the same length and the density of curve radii is equivalent for both of them. Thus, simulations will be carried out considering a symmetric network.

![Figure 1: OC1 route, Luleå-Borlänge.](image)

![Figure 2: Curve length vs. radius of OC1.](image)
2.1.1.2 Vehicle
The Y25 bogie vehicle model has already been validated in a previous project [7]. In order to calculate wheel wear it has been introduced in the wear calculation methodology and preliminary theoretical worn profiles have been obtained.

2.1.2 OPERATIONAL CASE 2
This Operational Case has a route more complicated than the first one, and a vehicle whose multibody model has not been validated with experimental results. Thus, it was a secondary option until we realised that no experimental results could be obtained from the first OC.

2.1.2.1 Route
This operational case is far more complicated than the first one. There are 186 vehicles that run in the Trätåg system. If routes that occur less than 1% are discarded, we have 19 different routes left (Table 1). Some of these routes are actually part of a bigger route, so we can check it manually (Table 2). The map of the lines is depicted in Figure 3.

Figure 3: Significant locations inside the operational case.

Table 1: Number of runs in the most common routes for vehicles running in the Trätåg network. 1 to 2 is laden run and 2 to 1 is unladen run. Names of the stations are Trafikverket codes.

<table>
<thead>
<tr>
<th></th>
<th>2</th>
<th>1 to 2</th>
<th>2 to 1</th>
</tr>
</thead>
<tbody>
<tr>
<td>LS</td>
<td>GKR</td>
<td>8412</td>
<td>8810</td>
</tr>
<tr>
<td>HLF</td>
<td>GMS</td>
<td>6432</td>
<td>6661</td>
</tr>
<tr>
<td>VO</td>
<td>TA</td>
<td>6591</td>
<td>5548</td>
</tr>
<tr>
<td>LOM</td>
<td>SUR</td>
<td>4011</td>
<td>5625</td>
</tr>
<tr>
<td>LS</td>
<td>LSE</td>
<td>3746</td>
<td>3739</td>
</tr>
<tr>
<td>LOM</td>
<td>TA</td>
<td>4082</td>
<td>2956</td>
</tr>
</tbody>
</table>
From these results it can be deduced that in each line the vehicle always runs laden in one direction and unladen in the other, except for two lines that have both load cases in both directions (Table 3). The connection between HLF and TÅ is a fairly symmetric load run, and will be treated so for the sake of simulation simplicity. The number of runs between GKR and TÅ is very small in comparison to the reverse route, so these will be ignored in order to obtain a simpler operational case.

Table 3: Stations that have both laden and unladen runs in both directions. 1 to 2 is laden run and 2 to 1 is unladen run. Names of the stations are Trafikverket codes.
Considering these simplifications, the individual lines that will be studied in this operational case are gathered in Table 4. These routes comprise 90% of the total km run by the total number of vehicles of the fleet that works at Trätåg. As it can be seen, HLF-TÄ and TÄ-HLF are two separate routes (#2 and #3 in Table 4) as the laden and unladen run are in both directions.

Table 4: Simulated routes in the operational case, including the laden and unladen proportion of km run in these lines.

<table>
<thead>
<tr>
<th>#</th>
<th>From</th>
<th>To</th>
<th>Laden Run (%km)</th>
<th>Unladen Run km (%km)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>HLF</td>
<td>GMS</td>
<td>12.48%</td>
<td>12.59%</td>
</tr>
<tr>
<td>2</td>
<td>HLF</td>
<td>TÄ</td>
<td>5.74%</td>
<td>5.35%</td>
</tr>
<tr>
<td>3</td>
<td>TÄ</td>
<td>HLF</td>
<td>3.98%</td>
<td>3.63%</td>
</tr>
<tr>
<td>4</td>
<td>VO</td>
<td>TÄ</td>
<td>11.02%</td>
<td>10.82%</td>
</tr>
<tr>
<td>5</td>
<td>LOM</td>
<td>TÄ</td>
<td>14.59%</td>
<td>15.02%</td>
</tr>
<tr>
<td>6</td>
<td>TÄ</td>
<td>GKR</td>
<td>19.13%</td>
<td>19.10%</td>
</tr>
<tr>
<td>7</td>
<td>LS</td>
<td>GKR</td>
<td>26.27%</td>
<td>26.63%</td>
</tr>
<tr>
<td>8</td>
<td>GKR</td>
<td>NDT</td>
<td>2.24%</td>
<td>2.28%</td>
</tr>
<tr>
<td>9</td>
<td>NDT</td>
<td>LSE</td>
<td>1.98%</td>
<td>2.01%</td>
</tr>
<tr>
<td>10</td>
<td>GKR</td>
<td>SUR</td>
<td>2.55%</td>
<td>2.57%</td>
</tr>
</tbody>
</table>

A total of 39 curve intervals have been used for the simulation (19 curve intervals, x2 because of the unsymmetry and +1 because of the straight track). The curve radii and lengths used for the simulations are plotted in Figure 4 and a list of the values can be found in listed Annex 1.

Figure 4: Simulation values of curve radii and length for the operational case. $R=0$ m is straight track.

2.1.2.2 Vehicle

The proposed vehicle for this study is a two axle vehicle with Unitruck running gear that transports timber logs in central Sweden. The axle load is 5.8 tons for an unladen vehicle and between 22.5 and 25 tons for a laden vehicle.

The Unitruck running gear is detailed in Figure 5. The design is optimized for 25 tons axle load but can be used up to 30 tons per axle. It is a single stage suspension system, composed by four nested coil springs.
which connect the carbody (1) and the saddle (2). The nested coil springs allow having different stiffness values between laden and unladen, so its behaviour has been optimized for those two cases separately. The damping is provided by friction elements attached to the wedge (3), a small component that enables to couple the vertical preload with longitudinal, lateral and vertical friction. The inner coil springs are connected to the wedge, which transmits vertical load to the carbody through an inclined friction surface (4). This angle generates a longitudinal force that is transmitted between wedge and saddle through a vertical friction surface (5). This coupling creates friction damping in longitudinal, lateral and vertical directions. The saddle is mounted on the axle box (6) through a rocket seat coupling (7) that enables it to have a relative sway angle with respect to the axle box. Displacements between carbody and saddle are limited by different bumpstops.

This suspension model was used in the studied vehicles until year 2005. The vehicles suffered from high flange wear, so the running gear was modified in order to solve the problem (Figure 6). The first modification consisted of softening the longitudinal suspension by replacing the longitudinal friction surface with a roller (Figure 6A). In order to avoid spring buckling due to this rolling couple, the nested springs were shortened and a plate was introduced connecting the centre position of both springs (Figure 6B). Once these modifications were introduced, flange wear was reduced up to 50%.

Along the report, the vehicles will be called Old Model for the original running gear and New Model for the vehicle with modified running gear.

![Figure 5: Unitruck running gear and expanded view of its components.](image)

![Figure 6: Modifications in the Unitruck running gear.](image)

**COMPUTATIONAL MODEL**

The multibody simulation software GENSYS is used for the analysis. The old model is composed of fifteen bodies: carbody, two wheelsets, four saddles and eight massless wedges. All bodies are rigid. The model has a total of 90 degrees of freedom (dof).
The flexibility of the track in lateral direction is also considered for a total of 94 \textit{dof}, and track irregularities based on experimental measurements are introduced.

The new model also includes eight spring plates for a total of 142 \textit{dof}. This increase of the \textit{dof} of the model and the low mass and inertia values of the spring plate are prone to negatively influence the computational efficiency of the dynamic model.

The nonlinear geometry functions that describe wheel-rail contact geometry are pre-calculated considering elastic deformation in the contact patch. Multiple contact patches are considered, depending on the wheel and rail geometry. These are modelled as linear springs. The creep forces are calculated in strips by FASTSIM.

The suspension systems of both old and new models are detailed in Figure 7. Nested coil springs are modelled as linear elements, including buckling stiffness. Friction elements are modelled as 2 dimensional series stiffness-friction elements, plus a perpendicular stiffness element to calculate the normal force on the friction surface. The rocket seat enables a displacement in the tilt angle of the saddle with respect to the axlebox.

The old model has been validated with measurements carried out for the UIC approval of the vehicle (Figure 8), but no data could be obtained for the validation of the new model.

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**Figure 7:** Modelling of old (A) and new (B) Unitruck running gear. OCS - Outer Coil Spring; ICS - Inner Coil Spring; FXZ - Friction element between carbody and wedge; FYZ - Friction element between wedge and saddle; RS - Rocket Seat of the saddle onto the axle box; RE – Rolling Element. ⊗ - connection element with vertical, lateral and angular-buckling stiffness. ∥∥ - coupling element with normal stiffness and lateral friction.

**Figure 8:** Validation of the multibody model (gray) with experimental results (black). For experimental results minimum (....) average (---) and maximum (___) values are represented. Lateral and vertical acceleration in leading bogie for tangent track (S), R<400m curves (A), 400m<R<600m curves (B) and R>600m curves (C).
Even if there are no experimental results for the validation of the new model, its curving performance has been analysed in order to check that its behaviour is similar to the behaviour of the real vehicle. The dynamic simulations are carried out under the following conditions: 480m radius curve with 0.08m cant deficiency, 500m long and at 80km/h speed. Figure 9 depicts the yaw angle of both old and new models. The steering behaviour of the old model is very bad, leading to high values of the angle of attack and thus, high values of flange wear. The new model, instead, has a good curving performance with, presumably, much lower values of flange wear.

As it has been previously stated, the modification of the running gear resulted in the reduction of flange wear up to 50% compared to the old model, so it can be concluded that the new model has a reasonable behaviour.

SIMPLIFICATION OF THE SUSPENSION

In order to include the modified running gear in the new model the plate has been included (Figure 7), an element with very low mass and inertias. This element will cause difficulties during the integration of the equations in the model that will slow down the simulation, thus an equivalent simplified model has been calculated (Figure 10). The simplification consists in a new element between carbody and wedge that will represent the coupling of these two elements via plate.

The stiffness values of all the elements in the Figure 10 comprise three dimensions, i.e. \( k = \{ k_x, k_y, k_z \} \). The carbody is considered fixed to the reference system.

The following simplifications have been done:

- It is a 2D model, considering only the XZ plane (longitudinal and vertical). The influence of the plate will be almost entirely in this plane.
- The plate mass and inertia has a minimal influence on the dynamic behaviour of the vehicle, so the full element is removed.
- The wedge is coupled to the saddle with a sliding coupling which represents the sliding of the friction surface. Thus, longitudinal and angular displacements are equal between the two bodies (Equation 1). Also, the roller element between carbody and wedge delimits the vertical displacement of the wedge (Equation 1).
- All the stiffness values of the four coilsprings connected to the plate are considered the same, with value \( k = \{ k_x, k_y, k_z \} \).

In the New Model (Figure 10A) the displacements of the plate can be determined as a function of the displacements of the saddle. Considering the plate in quasi-static equilibrium, these relations are obtained (Equation 2).
Figure 10: Simplification schemes of the plate suspension.

\[
\begin{align*}
    x_w &= x_s \\
    z_w &= x_w \cdot \tan \alpha = x_s \cdot \tan \alpha \\
    \varphi_w &= \varphi_s \\
    x_p &= \frac{3}{4} x_s \\
    z_p &= \frac{3}{2} x_s - x_s \frac{\tan \alpha}{4} \\
    \varphi_p &= \frac{3}{4} \varphi_s - x_s \frac{\tan \alpha}{4} \frac{k_z b_p}{k_p + k_z b_p} \\
\end{align*}
\]

In order to check the influence of the plate in the interaction of the carbody and saddle, the forces acting on each solid have been calculated. The forces in each solid of both models A and B are made equal, and the results are expressed in Equations 3 to 5.

\[
\begin{align*}
    K_{cs} &= \frac{k}{2} \\
    K_{cw} &= \left\{ - \frac{k_z}{2} \right\} \\
    K_{ws} &= \left\{ \frac{k_x}{4} \frac{k_z b_p}{4 b_w} \frac{1}{4} \left[ k_{\varphi} + k_z \cdot b_p (h_p - b_w) \right] \right\} \\
\end{align*}
\]

Considering the simplifications that have been done, the calculation of \( K_{cw}^{x} \) and \( K_{cw}^{\varphi} \) is not possible as these degrees of freedom are equal in the solids connected by them, i.e. wedge and saddle. Following the behaviour of \( K_{cs} \) the values for \( K_{cw} \) is set as:

\[
\overline{K}_{cw} = \frac{k}{2}
\]

The results of both suspension elements are compared in Figure 11. As it can be seen, the results of both models are extremely similar, which is a very good approximation taking into account the nonlinearities introduced by the friction elements.

Computational time of the simplified model is 1/5 of the time of the new model for favourable contact geometry, and up to 1/50 for unfavourable contact geometry, i.e. geometries with two or three point contact conditions. During the wear calculation process, wheel profiles are modified according to the wear model, so it is very common to run simulations with unfavourable contact conditions. Thus, this simplification is very useful for reducing the total calculation time.
Figure 11: Comparison of the lateral displacement of the leading wheelset between the new model and the simplified model in a common curving case simulation.

**Coupling Model**

One vehicle is composed by two wagons coupled with a flexible element called Safety Bar. In order to couple two vehicles, pre-stressed Buffers and Drawgear are used. This couplings are not modelled in a first approach, but a latter analysis is carried out after the discussion of the results.

### 3 RESULTS AND DISCUSSION

#### 3.1 OC1

The operational case of the steel slab traffic was ditched in a very early stage, so the simulation results are from preliminary simulations that were meant to test the model and the wear calculation methodology. These are not significant enough for a discussion so they won’t be commented here.

#### 3.2 OC2

For the second operational case, first some preliminary studies were carried out so the behaviour of the model was determined before starting with the long simulations of wheel profile evolution.

#### 3.2.1 PRELIMINARY RESULTS

In order to carry out preliminary simulations, wear was calculated as the volume of material removed for a certain amount of time \( m^3/s \), so that the influence of the different parameters can easily be compared. According to Archard’s wear law described in Equation 1, the volume of material worn out is proportional \( k \) to the sliding distance \( s(m) \) and normal force \( N(N) \), and inversely proportional to the material hardness \( H(N/m^2) \).

\[
V_w = k \frac{SN}{H} \tag{1}
\]

To enhance the smoothness of the wear distribution, the contact ellipse is divided in \( m \times n \) elements, and the wear depth is calculated in each of them as given by Equation 2. It is proportional to the contact pressure \( p(N/m^2) \), the sliding distance for that element \( \Delta s(m) \) and inversely proportional to the hardness of the worn material \( H(N/m^2) \).

\[
\Delta h = k \frac{p\Delta s}{H} \tag{2}
\]

The proportionality is represented by the so called “wear coefficient” \( k \). This wear coefficient has been determined in laboratory measurements and depends on sliding velocity and contact pressure. It is represented by a wear chart (Figure 12) with four different regions that represent different wear patterns.
The following dynamic simulations are carried out under the same conditions: 480m radius curve with 0.08m cant deficiency, 500m long and at 80km/h speed. All the simulations are performed for unladen, laden and exceptionally laden vehicle (6t, 25t and 30t axle load respectively).

First, a comparison between the different models is carried out. Figure 13 shows the massive reduction in uniform wheel wear for the New Model, which confirms the experimental results obtained with the 2005 modification of the running gear.

The main reason for this reduction on the wear volume is the improvement of steering conditions. In Figure 9 it was shown that the steering behaviour of the Old Model is very bad, leading to high values of the angle of attack and thus, high values of flange wear. The New Model, instead, has a good curving performance with much lower values of wear volume.

Figure 14 shows the comparison of the wear volume for different wheel-rail contact conditions. For higher friction coefficient in the wheel-rail contact, we get a higher wear volume. The same applies for higher axle load. However, some unexpectedly high values are observed, such as $\mu = 0.2$ with 25T and 30T per wheelset.
The cause of these anomalies can be deduced from Figure 15. There, the wear volume $V_w$ is depicted for three different track irregularity parameters: ideal track and two equivalent low-level track irregularities (Tr. Irr. 1 and Tr. Irr. 2). The three curves show discrete jumps in $V_w$ values, which are caused when the wear coefficient $k$ of part of the contact patch switches from a mild wear region to a severe wear region (Figure 12). In this case, contact pressures in the sliding contact area vary around $2 \cdot 10^9 N/m^3$, while Archard’s pressure limit for severe wear is placed at $2.08 \cdot 10^9 N/m^3$. When part of the sliding area surpasses this severe wear limit, the wear volume increases drastically. However, due to the behaviour of vehicles with friction dampers on a non-ideal track, this discrete jump in the coefficient makes it extremely difficult to analyse the influence of different parameters on the behaviour of heavy laden freight wagons.

According to the manufacturer, the friction coefficient of the friction dampers is $\mu_d = 0.3$. However, suspension systems based on friction elements tend to have a wide range of possible values for their characteristics. In order to analyse the influence of this uncertainty, wear calculation has been performed with different friction coefficient values for the friction dampers (Figure 16). The friction coefficient in the wheel-rail contact is $\mu = 0.3$ for all the cases.
Figure 16: Wear volume of the leading wheelset, influence of the friction coefficient of the dampers 

\[(\mu_d = \mu d)\].

The behaviour of friction dampers has two contradictory characteristics: for higher friction coefficient, there will be more damping when a relative displacement occurs. However, the Coulomb force necessary for starting this relative displacement of the damper grows proportional to the friction coefficient, and thus damping cycles occur more seldom. The overall result is that the higher the friction coefficient in the dampers, the higher the wear volume in the wheel-rail contact. The locking of the dampers for loads smaller than the Coulomb force stiffens the suspension and increases lateral wheel-rail creep and forces, increasing the wear volume on the wheel.

It can also be seen some anomalies in the trend for 30t axle load and friction coefficients \(\mu = 0.3\) and \(\mu = 0.4\) with a worn volume slightly higher than expected. The explanation to this variation is equivalent to the wheel-rail contact friction case: the nonlinearity of the wear coefficient for pressure values close to the severe wear limit makes the prediction very sensitive to track irregularities and friction element behaviour.

3.2.2 WEAR PROFILE VALIDATION

3.2.2.1 Experimental measurements

Experimental profile measurements were performed in GC installations in Borlänge 10th and 11th of November 2011. 80 wheelsets were measured and the statistics of these measurements are presented in the following graphs. Wear is depicted with three different variables: flange height \((h_f, \text{Figure 17})\), flange thickness \((t_f, \text{Figure 18})\) and flange inclination \((q_r, \text{Figure 19})\). All these figures have four graphs representing the values for the different wheels in the vehicle: 1l, 1r, 2l, 2r (1 for leading wheelset, 2 for rear wheelset, r for right wheels and l for left wheels). The graphs show the experimental values, the linear trendline and the quality of the adjustment \(R^2\) value.

As it can be noticed, the trend for all the scalar variables that define the wear is clear: the more mileage flange height increases while flange thickness and inclination decrease. However, if we try to fit it with a trendline, values are very spread \(R^2\) values are between 0.4 and 0.7 in most cases, the best adjustment is \(R^2=0.833\). This variability will make the experimental validation much more difficult and imprecise, as we will have to use these values for the comparison with the simulated profiles.
Figure 17: Experimental flange height ($h_f$) for different mileages, for each wheel on the vehicle.

Figure 18: Experimental flange thickness ($h_t$) for different mileages.
This variability on the scalar values is caused by the behaviour of the suspension elements, specifically the friction dampers. The way friction dampers work is very particular, as there is no energy dissipation until a relative displacement is reached, which occurs when the friction force reaches the Coulomb limit. This means that the dampers have a tendency to get stuck in a specific position until a tangential force high enough is able to move them. This point where the damper is stuck is not necessarily the nominal position, which reduces the symmetry of the system. There are eight friction dampers in the suspension, so depending on how this non-symmetry gets stacked different wear patterns could be reached, as it actually occurs.

In order to check for the effective symmetry of the system measurements in right and left wheels have been compared in Figure 20. It can be noticed that the behaviour of right and left wear is very similar, so a symmetric wheel wear model can be used. There can also be seen that there is some difference between lead and rear wheelset, so vehicle couplings should be correctly modelled in order to account for this difference.

The most interesting parameter for railway engineers is the equivalent conicity. In Figure 21 the equivalent conicity of these experimental profiles running on UIC60 rails with inclination 1/30 is depicted. As the run distance increases, the conicity shape is flattened, but the minimum value is constantly rising. This means that wear can actually improve the dynamic behaviour of these vehicles, as the equivalent conicity gradient is smaller.
3.2.2.2 Simulation of profile wear evolution

Using the wear calculation methodology, wheel profiles for a specific mileage have been simulated in order to carry out the experimental validation of the methodology. As can be seen in the following figures, the comparison is not satisfactory at all. Figure 22 depicts the original S1002 profile (black), the simulated profile (red) and the measurement (blue) for c.a. 22kkm. It can be observed that flange and tread end wear is underestimated while wear in the flange-tread transition shows a good agreement between the experimental measurement and the simulated profile.

Figure 23 shows the same three profiles for a very low running distance, about 100km. At this low run mileage, it can clearly be seen that simulations do not vary much the original profile, i.e. the wear is
negligible; meanwhile the measurements show that for this small running distance a heavy tread end wear pattern appears. In order to discard design errors, it was checked which one was the original profile for the experimental profiles, and it was confirmed that it was the S1002. This means that there is a behaviour not considered in the wear prediction which highly affects the material removal of the wheel, especially at the tread end.

The discussion then shifted to what are the differences between the real case and the simulated case. A number of options arose, the strongest ones being the following:

- The difference at the tread end and at the flange top are due to switches and crossings (S&C). When the blade raises and the outer rail shifts outwards, two point contact may appear in very separated points at the wheel. Thus, high sliding can occur in the tread end and flange top, which including the high axle loads because of the freight transport can cause heavy wear. Moreover, during the experimental measurements it was observed that for heavily worn profiles there was plastic deformation both in the tread end and at the flange top (Figure 24).

- The difference in wear level can be because of the friction coefficient on the wheel-rail contact. In the original operational case an average value of friction coefficient of $\mu=0.3$ was used. However, the friction coefficient in a generic track can vary from $\mu=0.1$ to $\mu=0.7$ as shown in Figure 25.

- Track irregularities (TI): the simulated TI were obtained for a passenger line, thus TI for the actual case needed to be obtained from Trafikverket database. If these TI have a higher level than the simulated ones, the contact point distribution along the wheel profile can be influenced and thus modify the overall wear profile.
The wear map used in the simulations was obtained in 1999 for a specific case, a passenger X1 vehicle wheel with UIC 900 graded rails. Axle loads of freight vehicles can be twice as high as the ones of passenger vehicles, and thus the material used in the wheelsets and rails might be different (harder) than the one in passenger trains. This will have some effect on the tribology of the contact, modifying Archard’s wear map in a way that cannot be predicted. In order to perform accurate simulations this wear map should be re-calculated.

Most freight vehicles have block brakes, and these wagons are not an exception. This braking system consists in the application of pressure with two brake blocks on the wheel tread, generating heat loss due to friction between the block and the wheel. The friction in the contact area and the high temperatures caused by the energy dissipation might have a high influence on the wear pattern at the wheel tread, so it must be studied in detail.

Some of these topics are important enough to develop them in further sections of the text.

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**3.2.2.3 Friction coefficient in the wheel-rail contact**

As it has been demonstrated in the preliminary results (section 3.2.1), the friction coefficient in the wheel-rail interface has a major influence on the worn out volume of material. In order to analyse the influence of the friction coefficient in the developed profiles, four different simulations have been carried out ($\mu=0.2$, $\mu=0.3$, $\mu=0.4$, $\mu=0.5$, $\mu=0.6$). Figure 26 depicts the differences on the worn profiles for c.a. 49 kkm run distance.

In Figure 27 wear of the tread and flange is separated. Tread wear depth at $y=0$mm is c.a. 0.0143mm/kkm both for $\mu=0.4$ and $\mu=0.5$ with wear volume rather constant, and it increases for $\mu=0.6$ up to c.a. 0.214mm/kkm, 50% higher than the previous two and increasing the total wear volume value c.a. 25% (Figure 28). It is noteworthy that for lower values of friction coefficient wear volume increases, i.e. for $\mu=0.3$, wear level at the tread c.a. 30% higher than for $\mu=0.4$ and $\mu=0.5$; and for $\mu=0.2$ the wear depth in the tread center is the highest among all the cases, 1.2mm.

At the flange, the influence of the friction is clear as the maximum wear depth increases with the friction coefficient.
The trend seems not very clear, but there is a reason for this behaviour. For higher $m$ at the wheel-rail contact creep forces can be higher, so the wear tends to increase. However, for low friction values, the low creep forces in the contact decrease the curving performance of the whole vehicle, increasing the angle of attack and thus increasing the wheel wear. This behaviour was not detected in the preliminary simulations (section 3.2.1), although these are results for a full profile evolution simulation, so they are more detailed than the preliminary studies.

Figure 28 depicts the wear volume of the five cases. The analysis is basically the same as in previous paragraphs, but with some extra detail. The overall wear volume has a minimum value for $\mu=0.4$, with a similar trend for tread wear. At the flange, wear volume increases with the friction coefficient.
In order to extend the analysis, in Figure 29 the wear volume is depicted normalized with the friction coefficient. Normalized wear volume decreases with the friction coefficient, but is somehow stable after $\mu = 0.4$.

![Figure 28](image1)

**Figure 28:** Wear volume per contour unit (m\(^3\)/m) on the whole profile (1) tread (2) and on the flange (3), c.a. 49kkm.

![Figure 29](image2)

**Figure 29:** Wear volume per contour unit (m\(^3\)/m) on the whole profile (1) tread (2) and on the flange (3), normalized with the friction coefficient, c.a. 49kkm.

In the following graphs (Figure 30 to Figure 32) the simulated values for flange height, thickness and inclination are depicted along with the experimental results previously obtained. From these results the following can be deduced:

- Flange height ($h_f$) and flange thickness ($t_f$) agree quite well with the experimental results.
- Flange inclination ($q_r$) does not agree with the experimental results. Its values are equivalent to the ones of an original S1002 profile.
- The values for the rear wheelset profiles show much lower agreement with the experimental results. Simulated wear on rear wheelset profiles is smaller than the one on the leading wheelset profiles.
- The order of magnitude of the variations of the simulations with the wheel-rail contact friction coefficient is the same as the variations in the measured profiles. For flange thickness and inclination this variation is even much lower than the one of the experimental results.
Figure 30: Experimental flange height ($h_f$) for different mileages, for each wheel on the vehicle. Comparison to the simulation results for different friction coefficient values in the wheel-rail contact: $\mu=0.3$ (lightest gray), $\mu=0.4$ (light gray), $\mu=0.5$ (dark gray) and $\mu=0.6$ (darkest gray)

Figure 31: Experimental flange thickness ($h_t$) for different mileages. Comparison to the simulation results for different friction coefficient values in the wheel-rail contact: $\mu=0.3$ (lightest gray), $\mu=0.4$ (light gray), $\mu=0.5$ (dark gray) and $\mu=0.6$ (darkest gray)
Figure 32: Experimental flange inclination ($q_r$) for different mileages. Comparison to the simulation results for different friction coefficient values in the wheel-rail contact: $\mu = 0.3$ (lightest gray), $\mu = 0.4$ (light gray), $\mu = 0.5$ (dark gray) and $\mu = 0.6$ (darkest gray)

It must be mentioned that this simulated values come from laden simulations, while experimental values come from both laden and unladen traffic, c.a. 50% each. This means that the mileages of the simulated cases and the experimental values are not equivalent as they don’t represent the same operational case. Simulations with unladen vehicle are depicted in Figure 33, compared to a laden vehicle, both of them with $\mu = 0.4$ at the wheel-rail-contact. Wear depth on the tread is c.a. 10 times lower for the unladen case, while c.a. 2 times lower at the flange. Performing a rough approximation, wear depth on the tread in a mixed operation simulation would be c.a. 0.55 times that of the laden case; while at the flange it would be c.a. 0.66 times that of the laden case. Of course these are approximate values obtained from a rough approximation, but show that the validation carried out in Figure 22 is even worse than the original discussion.

Thus, the analysis of the influence of the friction coefficient can be useful to show its theoretical influence, but given the enormous variability of the experimental results its use is rather limited in order to validate the vehicle model or the running conditions.
3.2.2.4 Influence of track irregularities

Initial simulations were carried out with track irregularities from the commuter network in Stockholm. In order to analyse wear with the proper irregularities, these have been asked to Trafikverket. These simulations have still not been carried out.

3.2.2.5 Influence of couplings between wagons

As previously stated, Laaps vehicles are two wagons coupled with a flexible element that transmits longitudinal loads and moments between both wagons. These two wagons are coupled to other units with pre-stressed draw-gear and buffers. Stiffness values of the buffers, draw-gear and safety bar are depicted in Figure 34 and Figure 35.

Buffers and draw-gear are preloaded with 20kN, so it is rather unusual to act with a bump-stop-like behaviour, unless there are extremely tight curves. In order to get the big picture of the influence of these couplings, first a simplified study is carried out. Figure 36 depicts the yaw angle between wagons depending on the curve radius. The full demonstration is not presented, but the results for this specific vehicle are...
displayed in Equation 7. The value of the yaw angle for each wagon is proportional to the inverse of the radius.

\[
\psi = 400R^{-1} (°)
\]  

(7)

For the following equations we consider linear buffers. The longitudinal forces are depicted in Figure 37. The moment introduced in the vehicle by the whole coupling system is developed in Equation 8. Then the moments introduces in the couplings by this elements are depicted in Figure 38 and Figure 39.

\[
M_{bf} = (F_{bf}^l - F_{bf}^r) \cdot 2b_{bf} = k_{bf} \cdot \psi \cdot (2b_{bf})^2
\]  

(8)
Figure 38: Quasistatic moment introduced by the buffers ($M_{bf}$) vs. curve radius ($R$) taking into account the non-linearity of the buffer stiffness.

Figure 39: Quasistatic moment introduced by the safety bar ($M_{sb}$) for different pre-load values vs. curve radius ($R$). Depending on the longitudinal pre-load the yaw stiffness can vary from 5kNm/rad to 85kNm/rad.

As it can be seen in these last two figures, the yaw stiffness of the safety bar is extremely small compared to the one of the buffers, and thus there is a yaw moment unbalance that will affect the angle of attack of the whole vehicle (Figure 41). Essentially, the leading vehicle will reduce its angle of attack and the rear vehicle will increase it; or at least, if the yaw angle is not modified, the lateral forces at the contact will be modified, reducing them in the outer wheelsets and increasing them in the inner ones.

Couplings have been modelled in the computational model in GENSYS in order to assess their influence on the behaviour of the vehicles, particularly their influence on wheel wear. However, the dynamic behaviour of the friction elements is so chaotic that obtaining significant results was impossible. In order to make simulations simpler, the following steps were taken:
- Friction elements were removed ($\mu=0$) so their influence was cancelled. A priori this was a good simplification, however the vehicle became extremely unstable and it behave in limit cycles range. Thus, consistent results could not be obtained.
- Quasi-static analyses were carried out, both with and without friction. In this case there were issues with the model, as allegedly there are some unconstrained degrees of freedom for the QUASI analysis to be successful. These simulations did not converge and thus, significant results could not be obtained.
- In order to try to solve this issue, a very low speed simulation was carried out in order to perform a time simulation that resembled a quasi-static simulation. However, the length of the simulation in order to arrive to a quasi-static regime was extremely long, so this option was not valid either.

We were not able to calculate the influence of coupling elements in vehicle dynamics or wheel wear. However, some qualitative comments can be made based on Figure 41. The high stiffness on the buffers introduces a moment proportional to the curvature of the curve. This moment will place the vehicle in a position that, a priori, is better for the outer wheels and worse for the inner wheels. In practice, leading wheelset needs to steer anyway, regardless of buffer moment, so the influence on outer wheelsets won’t be so significant. However, lateral creep forces on inner wheelsets will probably increase, increasing the amount of wear at the wheels.

As future work, it is planned to add dampers to the vehicle model in order to simulate the quasistatic behaviour and understand the influence of the couplings on vehicle dynamics.

3.2.2.6 Influence of switches and crossings

In tangent or curved tracks the wheel-rail contact point usually stays between $y_{\text{min}}=-30\text{mm}$ and $y_{\text{max}}=30\text{mm}$ in the wheel (Figure 42). This means that, in these cases, contact point will never reach tread end. But in experimental measurements it can be seen that there is heavy wear in the end of the tread, so there is some situation in which the contact point reaches the outer end of the wheel. In order to achieve that in tangent or curved track there must be a lateral shift of the rail of more than 20mm, and even that does not assure that the contact point will reach the corner of the wheel. The only case where the contact points reach the tread-end of the wheel is when running on turnouts or switches and crossings (S&C).
An S&C is composed by two point blades that can be laterally displaced so the vehicle is diverged to one or another track. However, the most critical contact conditions come when wheels are running on the crossing between the left rail of the right track and right rail of the left track, also known as “nose” (Figure 43).

When running along the nose, the incoming rail raises close to the flange and its shape progressively becomes that of a proper rail (Figure 45), while the original rail is laterally shifted until it does not support the wheel anymore. In this process there is two-point contact in the different rails, so when the support rail is reaching the end of the wheel profile there can be two contact points separated by c.a. 80mm (sections 18-21). This can cause huge sliding speeds with high contact pressure values that might heavily erode or cause plastic deformation on the outer part of the wheel.
Figure 45: Contact pairs for a new S1002 wheel profile through different sections of a switch (sect. 17 to 22) comprising the ‘nose’.

In order to demonstrate this wear simulations have been carried out in a model that includes varying rail profiles which represent an S&C. The approximate geometry of the track is depicted in Figure 43 and the geometry of the rails is handled by GENSYS considering 30 different sections of the S&C. The results are discussed below.

Figure 46 depicts the maximum stress in the different contact patches along the switch and the lateral position of the contact point in the profile. Several conclusions can be drawn from this graph: i) maximum normal stress is high above uniaxial yield stress; ii) the highest stress values occur mostly for high lateral displacements towards the flange (positive lateral displacement of the contact point), for \textit{cpf} \_11l (green) and \textit{cpf} \_12l (yellow) contact points while running through the point blades, and for \textit{cpt} \_11l (green) and \textit{cpt} \_12l (yellow) contact points while running through the nose.

It must be mentioned that the lead and rear wheelsets have roughly the same behaviour, having the rear wheelset a delay equal to the distance between axles (\textit{c.a.} 10m).

The initial supposition was that contact conditions when running through the nose (x=42m) would generate a huge amount of wear at the tread end. However, it can be seen that although the contact point almost reaches tread end (\textit{posw} = -45mm) the stress values for those points are not very high.

It is extremely common to have normal stress values in the contact patch above the yield stress of the material, and is not a particular behaviour of the switch crossing. In our case the heavy plastic wear observed in the tread end (Figure 24) could be caused by the high creepages or creep forces acting on the wheel when it is running close to the tread end. The influence of these parameters is usually represented by the energy dissipated at the wheel-rail contact ($T \gamma$ model) as the product of the creepage and the creep force. Figure 47 represents this energy dissipation against the position of the contact point at the wheel. There it can be observed that the maximum energy dissipation occurs when the contact point is placed in the flange, \textit{posw} = 0.03m to \textit{posw} = 0.04m (A). This occurs when the wheels enter the point blades, and the two point contact generates high energy dissipation also in the tread area (B).
Figure 46: Maximum stress ($\sigma_{\text{max}}$) for the different contact patches along the switch; and lateral position of the contact point in the profile ($\text{pos}_w$). $\text{cpt}$ and $\text{cpf}$ are two contact points of the same wheel. $1l$, $1r$, $2l$ and $2r$ are codes for leading ($1$) and rear ($2$) wheelsets and right ($r$) and left ($l$) wheels. Original S1002 wheel profiles.

Figure 47: Energy dissipated in the wheel-rail contact per run distance vs. position on the wheel of each contact point ($\text{pos}_w$).

When the blade at the nose is raised and a two-contact point is obtained with these contact points very far from each-other, it is expected to have high creepages, and thus high energy dissipation. However, although we do get contact points very close to the tread end ($\text{pos}_w$<45mm) the energies dissipated at the contact points are not very high: there is some energy dissipation for the flange (C), while the contact point at the tread end does not even have dissipated energy peaks (D).

Worn Profiles

Additional simulations have been carried out with worn profiles in order to analyse different wheel-rail geometry conditions. A measured worn wheelset has been used, but the original MINIPROF measurement has been filtered to avoid errors in the KPF files. Also, some variables in KPF files ($\text{cp1}_\text{rofn}_l$ and $\text{cp1}_\text{rofn}_r$) have been simplified so the total memory usage does not surpass the limit in GENSYS.
Figure 48 depicts values of maximum contact stress ($\sigma_{\text{max}}$) and position of the contact point at the wheel ($\text{pos}_w$) for the different contact points of the leading wheelset. The results are similar to the simulations with original S1002 profiles, with the following differences:

- Both when running through the nose ($x \approx 45\text{m}$) and trough the switch blades ($x \approx 5\text{m}$) there are contact points placed at the end of the tread ($\text{pos}_w < -60\text{mm}$). This positioning is maintained for a short period in both cases, so the vehicle is running on the corner of the wheel for some meters. This might increase the amount of wear at the tread end, depending on the dissipated energy; but also generates a very small contact patch with, as a consequence, a rise on the maximum contact stress (c.a. 7GN/m²), which is almost twice the one in the previous case (c.a. 4GN/m²).

- When entering the blades two-contact point starts, as in the original case. However, contact point distribution and positioning has bigger discontinuities. This means that the worn wheel profile is more conformal with the shape of the switch blades rail section, and thus when the two conformal sections coincide, several contact points appear within a wide range of lateral positioning on the wheel. When any profile wears out due to the wheel-rail contact geometry, wheel and rail profiles eventually tend to become conformal with each other, so having a wheel profile rather conformal with the switch blade geometry means that it might have much influence on the wear generated at the wheel.

Figure 49 depicts the energy dissipation at each contact point in the first wheelset, plotted against its positioning in the wheel. High energy dissipation ($E > 2500\text{J/m}$) is detected when the wheel is running on the tread corner; however, this energy is maintained for a very short period so its influence might be not so high.

In Figure 50 a detail ignoring this peak is presented. The behaviour is very similar to the previous model, except for some extra contact point positioning at the tread end ($\text{pos}_w < -60\text{mm}$). The energy dissipation in the tread is still very low, but there is an interesting phenomenon at the edge of the wheel: there is a concentration of contact points with energy dissipation slightly higher than the surrounding contact points. Again, this energy dissipation is very low compared to the one at the flange.
It can be concluded that turnouts might have a great influence in the evolution of wheel wear, especially at the flange. However, the simulations failed to predict the high wear at the tread end that can be seen in experimental measurements, which might be due to the following reasons:

- The model of the vehicle is not very good.
- The geometry of the turnout is a reference geometry from a passenger line. The switches and turnouts that this specific vehicle runs on might be different from the utilized geometry; even worn switches might have different rail profiles, thus not obtaining significant results.

The overall results are very promising: it can be concluded that, unlike passenger vehicles, uniform wear of freight wagons cannot be studied without accounting for the switches and crossings.

**Statistical data on turnouts, switches and crossings**
Once the need for studying switch-generated wear is settled, the question remains if this wear is significant enough compared to the wear caused by tangent tracks and curves. Table 5 shows statistical data on the number of turnouts in the Trätåg system, not taking into account secondary turnouts or switches that can be encountered when the train is parked in a track yard, deviated in a station to avoid the main lines, or running through a shunt yard. The highest rate of switches is in the line from Lomsmyren (LOM) to Skutskär (SUR), with one switch each 2.27km of track. The average value for all the system is one switch each 3.14km.

Table 5: Number of turnouts in the different routes of the Trätåg system.

<table>
<thead>
<tr>
<th>Route</th>
<th>Turnouts</th>
<th>Total mileage</th>
<th>km/turnout</th>
</tr>
</thead>
<tbody>
<tr>
<td>GÄ-NDT</td>
<td>15</td>
<td>35km</td>
<td>2.33</td>
</tr>
<tr>
<td>GMS-HLF-BLG</td>
<td>54</td>
<td>244km</td>
<td>4.52</td>
</tr>
<tr>
<td>HFJ-LSE</td>
<td>11</td>
<td>38km</td>
<td>3.45</td>
</tr>
<tr>
<td>LOM-BLG-GÄ-SUR</td>
<td>106</td>
<td>241km</td>
<td>2.27</td>
</tr>
<tr>
<td>LS-GÄ</td>
<td>51</td>
<td>165km</td>
<td>3.24</td>
</tr>
<tr>
<td>VO-BLG</td>
<td>21</td>
<td>86km</td>
<td>4.10</td>
</tr>
<tr>
<td>All (average)</td>
<td>258</td>
<td>809km</td>
<td>3.14</td>
</tr>
</tbody>
</table>

Figure 51 depicts the wear comparison for the reference switch previously used (black and blue) and simulated wear on tangent and curves (red, called “track wear” from now on), calculated with Archard’s wear law. As there is an average of one switch per 3.14 km, track wear has been calculated as the wear simulated for 71.8 kkm scaled down to 3.14 km. Then, comparing this wear value to the wear caused by the vehicle running through a switch, the dominant wear cause can be determined.

In this particular case, the wear caused by the reference S&C is much higher than the track wear:

- The vehicle with unworn S1002 profile running through the switch has much higher flange wear than the operational case wear, about 10 times higher; in the tread area the difference is small.
- The vehicle with worn S1002 profile running through the switch has much higher flange wear than the operational case wear, and higher volume than the original S1002 profile running through a switch; the tread area is the most interesting part, as the wear volume obtained in the tread end is extremely high.
This result is very promising, as it shows that with some profiles there is a concentration of wear in the tread-end, as it has been observed in experimental measurements. In order to improve the wear calculation methodology, switch simulations must be included. These simulations have still not been carried out.

The results also confirm that there is actual difference between the $T\gamma$ model and Archard’s wear model in the wear results: while the energy dissipation model could not predict high wear at the tread end, Archard’s model successfully predicts it.

So for already worn S1002 profiles, running through a switch creates heavy tread-end wear, which creates a feedback loop generating more wear in the already worn out zones; while, in case of original S1002 profiles, it does not wear the tread-end. At this point, one question appears: if the original profile does not generate wear in the tread end, how does wear in this area start? One answer could be that there is a specific switch geometry different than the reference S&C analysed, which generates a wear pattern with heavy wear in the tread-end. Once this wear starts it gets retro-fed, as it has been demonstrated that a worn profile generates more wear than the original one. This has still not been studied.

There are some extra things to consider:

- This analysis is for a laden vehicle. The wear caused by an unladen vehicle is expected to be negligible, as this behaviour has not been detected for passenger vehicles, with a lower load per axle.
- This analysis is for the vehicle running through the curved track of the switch. For a vehicle that runs over the switch without changing track, the wear is expected to be different, as the lateral dynamics will be much less aggressive. However, if the track flexibility of the appearing rail is lower than the one in the curved stock rail, then similar contact conditions will appear: tread end contact with heavy contact conditions. The influence of track flexibility or worn switch profiles should be studied.

3.2.2.7 Influence of other elements

While considering the differences between the experimental and the simulated profiles, several differences arose. The least important of them are listed here.

Influence of the angle of attack in the dynamic simulations in Gensys

The angle of attack moves the contact point in the wheel, but this amount is negligible for small angles, as and in Gensys both track-piece and wheelset belong to the same local coordinate system the angles can be considered small. However a high value of the angle of attack leads to a widening of the track gauge. This is taken into account in Gensys if the value of stiffness of the rail fasteners ($kyrt=42e6N/m$) is similar to the stiffness of the ballast ($kytg=30e6N/m$).

This way of accounting for the angle of attack is enough for the small angles that we have in the laaps vehicle.

Influence of wheel load on kpf files

The geometry of the wheel-rail contact in GENSYS is calculated by the so-called kpf pre-processor. This calculation takes into account the normal load in the wheel-rail contact in order to adequate the contact parameters to each vehicle.
The influence of this parameter has been studied and the conclusion is that, for the studied loads (70kN and 122.5kN per wheel respectively) the differences in the wheel-rail contact geometry are minimal between these two cases, so its influence can be disregarded.

4 CONCLUSIONS

The work done until this point comprises the following tasks:

- Study of curving cases for the Trätåg operational case.
- Modelling and validation of the Laaps vehicle with Unitruck running gear, including the modifications carried out in order to reduce heavy flange wear.
- Simulation of uniform wheel wear of these vehicles. Optimization of the wear calculation procedure to reduce computational time consumption.
- Experimental measurement of Laaps wheel profiles. Statistical analysis of the discrete wear parameters of the measurements. These discrete values have a great variability, which makes the experimental validation of the simulations much more difficult. When comparing some measured profiles with simulated results, these were not validated regarding both the shape and the amount of wear.
- In order to use the wear calculation model for the validation of wear in freight wagons, several extra analyses have been studied, the most important ones being:
  - Variation of the friction coefficient at the wheel-rail contact has been studied with no clear results.
  - Couplings between cars have been modelled and its influence on both dynamic vehicle behaviour and wheel wear has been studied. From these analyses no definitive conclusion has been obtained, due to the chaotic behaviour that friction dampers induce in the vehicle model.
  - Laaps vehicle running through a turnout has been simulated. The results show some of the high wear causes at the flange, but not at the tread end as it was expected. However, when a vehicle with experimental worn profiles is simulated, high wear values appear both at the tread-end and the flange, which qualitatively confirms the experimental measurements.

One conclusion is that the Laaps vehicle may not be the most suitable for the validation of the wear prediction methodology because i) it is not a common vehicle in the freight rail fleet, ii) its worn profile measurements have shown a high variability on the wear parameters and iii) the validation of the model is quite limited due to the variability of both experimental results and simulations. Thus, even if the results obtained from this vehicle are very promising, different vehicles should be analysed in order to continue with the project, e.g. the Y25 bogie vehicle, which is widespread and has an experimentally validated model for dynamic simulations.

The most important conclusion is that the uniform wear calculation on heavy haul vehicles cannot be approached like passenger vehicles. Heavy loads in the wheel rail contact make the rails with special geometries have much more influence on the total amount of uniform wear. Moreover, if the vehicles have block brakes, this wear will be further increased. In order to have an accurate prediction of the wheel profile evolution the influence of several components must be studied.
5 FURTHER WORK

Deepen the analysis of the influence of wheel-rail friction

The variation of the friction coefficient at the wheel-rail contact has been studied with no clear results, so further analyses should be carried out in order to determine if there is a trend in the wheel profile evolution with the wheel-rail contact friction coefficient.

Include unladen simulations and turnout simulations in the full simulation procedure

Wheel wear has been simulated only for laden vehicles running on some characteristic tracks of the network. In order to fully validate the model, unladen runs and switch crossings should also be included in the simulation procedure, so all the major contributions to profile evolution are considered.

Analyze additional vehicles

As we have seen, the actual model is somehow limited as the model is not very accurate and the experimental measurements have a very high variability. The new model, with Y25 bogies, has already been described along the report as one of the initial proposals for study.

Tribological analyses

Independently of which vehicle is analysed, new tribological analyses must be carried out:

- Archard’s wear map has been used for the calculation of the simulated worn profile. However, this wear map was obtained for a specific set of wheel and rail materials which are different from the ones present in freight vehicles. In order to carry on with the calculation of wheel profile evolution new experimental wear maps should be calculated for the actual conditions of our system.
- In order to study the influence of block brakes other wear maps must also be experimentally obtained. Block brakes have different materials (sintered, composite...) that might vary the wear rate of the wheels.

In order to carry out these analyses there have been preliminary contacts with Prof. Ulf Olofsson (Department of Machine Design) on how to carry out these new experimental analyses.

Plastic deformation on the wheels

The influence of plastic material reallocation in wheel tread and flange is a very interesting problem regarding wheel profile evolution. However, it is also a very complex problem which involves tribology, continuum mechanics and dynamic vehicle simulations. In order to continue with this area, first experimental wheel measurements should be obtained from the most common freight vehicles to check for this behaviour.

6 BIBLIOGRAPHY


Annex 1 – Numeric values of the curves in operational case #2 (Trätåg)

\( V \) is the speed of the vehicle, with a maximum lateral acceleration of 0.65m/s\(^2\).

\( L \) is the characteristic total length of the curve, including curve transitions.

\( R \) is the characteristic radius of the curve, negative radius means left-handed curve.

\( h \) is the characteristic cant for the curve interval.

<table>
<thead>
<tr>
<th>( V(km/h) )</th>
<th>( L(m) )</th>
<th>( R(m) )</th>
<th>( h(m) )</th>
</tr>
</thead>
<tbody>
<tr>
<td>80</td>
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Annex 2 – Experimental wheel profile plots

Experimental profile plots measured with the Miniprof device (black), and original S1002 profile (red). Wheel number has a codename e.g. ‘w1660-1l’ with the following meaning:

- ‘w’ for wheel.
- ‘1660’ for the wagon number, 4 digit number that uniquely identifies the Laaps wagon.
- ‘1’ for the wheelset number, ranges from 1 to 4.
- ‘r’ for the wheel, can be r or l for right of left wheels respectively, considered with wheelset 1 as leading wheelset.

There are some missing profiles. The cause is that, despite the measurements were done for all the wheelsets in each wagon, the high number of wheels forced the plotting to be automated by a script. Some of these graphs had problems being plotted and were discarded. If considered interesting, the information is available upon request.
w0977-1l
18506km

w0977-1r
18506km

w0977-2l
18506km

w0977-2r
18506km