STUDY OF THE LONG TERM EVOLUTION OF LOW-RCF WHEEL PROFILES FOR LKAB IRON-ORE WAGONS

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SUMMARY
The Swedish iron-ore company LKAB is using three-piece bogies with 30 tones axle load to transport pellets from the mines in Kiruna and Malmberget to the ports in Luleå and Narvik. LKAB provided a new wheel profile that is expected to perform better with regard to rolling contact fatigue and results in a longer wheel life than the currently used WP4 profile (RCF).

A simulation model of the freight wagon has been built at KTH using the multibody simulation code GENSYS. However, in order to study the long-term stability of the new profile, monitoring the evolution of the shape of the wheel profile is inevitable: there is a risk that the optimised profile might wear in an unfavourable way and, after a while, cause even higher RCF or wear than the original one.

The current study predicts the wheel profile evolution as function of running distance using the uniform wear prediction tool developed at KTH, see e.g., Jendel and Berg [1]. It includes an analysis of when, during the wheel life, the highest RCF probability occurs. The procedure contains of a number of so-called wear steps with a set of time-domain simulations reflecting the actual rail network. As the wagons are equipped with pneumatically actuated tread brakes, the influence of braking on material removal is also estimated via the calculation of the braking force for each simulation case; this is, considering the topography of the line, the amount of braking by the locomotives and the running resistance from curving, aerodynamics and mechanical resistance. The calculation is first performed for the current wagon wheel profiles, and the results are compared with field measurements in order to validate the simulation procedure. There is a great potential of saving time and money if an optimal wheel-rail profile combination can be selected with the help of computer simulation.

The results did not show any major difference between the wear of WP4 and WPX profiles. Although the WPX profile produces less RCF on the tread, it may be at a risk of developing RCF at the flange area.

Since there are no major differences observed in the simulation results, LKAB has not yet decided if they use the suggested WPX profile. A field test with the new profiles on wagons would be beneficial to verify the simulation results.

This work has been performed and continuously validated in the Malmbanan Wheel-Rail group to try to solve the RCF problems with the wagons and loco wheels.

INTRODUCTION
In order to transport extracted iron-ore from Kiruna’s mines to Luleå in Sweden and Narvik in Norway, railway is used. The freight units are so-called Fanoo wagons with three-piece bogies running on Malmbanan-Ofotbanen. The history of the line goes back to the 19th century. Since then, track components and structure have been improved several times due to market demands to increase capacity and decrease maintenance costs. In a recent upgrade, the rail profile was changed and the wooden sleepers were replaced with concrete ones. Today, the freight trains run with 30 tones axle load and 60 km/h velocity in loaded condition. Although these improvements in
general decrease the transport cost, the maintenance costs, for e.g., wheels, may increase. Issues are mainly mechanical wear (adhesive and abrasive) and fatigue wear (rolling contact fatigue, RCF) on the wheels; the latter problem is today the dominating one.

Wear and RCF affect the wheel profile in a way that is not desirable for both railway operators and infrastructure owners. Moreover, wheel profile geometry is an important factor affecting derailment safety.

With severe wheel flange wear, the flange inclination gets too high and the top of the flange may hit the switch blades. Also, too high flange inclination increases the conicity of the wheels. This affects the train ride stability by reducing the critical speed. Generally wheel wear, besides the environmental and metallurgical perspectives, is a function of creepages and creep forces in the wheel-rail contact. Therefore, flange wear usually depends on the flexibility of the running gear, the curve radii, and the wheel-rail friction level, while the wheel tread wear is more a function of axle load, tread braking and, for the locomotives, traction. Larger creep forces also increase the risk of surface initiated RCF on both wheels and rails.

LKAB uses several measurement stations on Malmbanan in order to detect high vertical impact forces, possibly due to wheel flats, where urgent maintenance actions are needed. Besides, regular inspections for wagon wheels are carried out every 80'000 km. These inspections are made daily at the yards and more thoroughly after 80'000 km in the Kiruna workshop and they are mainly for detecting cracks on the wheels, and thus, a possible need for re-profiling. The average mileage interval between two consecutive wheel turnings for a wagon wheel is around 250'000 km. The rails are checked by the infrastructure owner Trafikverket and are ground once a year. The maintenance policies have led to a total service life of the wagon wheels of 1'000'000 km. To further enhance the service life of wheels, LKAB is investigating the possibility of changing the wheel profiles to achieve a better curving performance for the vehicle, and consequently, reduce the creep forces. LKAB is also improving the lubrication strategy by moving from flange lubrication towards top of the rail lubrication in order to control and reduce the friction level between wheel and rail.

1. VEHICLE AND TRACK MODELLING

A review of the history of three-piece bogies can be found in Terrey and Ceng [2]. The features of various types of three-piece bogies designs are described in Hossein [3]. Here, a short description of the simulation model of the iron-ore three-piece bogies and the corresponding track which is used in the simulations in this paper is given. The present Iron-ore wagons are so-called Fanoo wagons running on Amsted Motion Control M976 three-piece bogies with load sensitive friction damping. Fanoo wagons contain two units: one unit is called master and the other is called slave. The controller of the braking system is attached to the master wagon as shown in Figure 1.

Figure 1: A two unit iron-ore wagon

The primary suspension is a rubber pad called Adapter Plus located between the axle box and the side-frame, and it is modelled as elastic stiffness and damping in parallel as shown in Figure 2.

Figure 2: Primary suspension modelled as elastic damping and stiffness in parallel in three dimensions

The wedges are modelled as massless bodies, and the position of the wedge is calculated by solving the local equilibrium equations. Normal contact forces acting on the surfaces of the wedge are used to calculate the friction forces in the friction block. The coupling in the contact surface between the bolster and the wedge is modelled as a one-dimensional friction block, and the friction surface between wedge and side frame is modelled as a two-dimensional friction block in lateral and vertical directions, as Figure 3 shows.

Figure 3: Couplings in the secondary suspension of a three-piece bogie

The friction coefficient between the 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 be around 0.38 (hardened cast iron and hardened steel plate), see Orlova and
However, the friction level varies and it is also dependent on weather conditions, roughness of the surfaces, etc. 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 sticking between the wedge and the side frame and leads to very high peak vertical wheel-rail 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 modelled with Saint Venant elements, as shown in Figure 4.

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 percent 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 sides of the plate, which bear the remaining 90 percent of the load. The couplings are defined as two-dimensional friction elements in the x-y plane. Figure 5 shows the carbody and bolster connections.

In this study, the vehicle speed is considered to be 60 km/h for all simulated track sections. Figure 6 summarizes the entire vehicle model for the vertical direction.

In most of the 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 Chaar and Berg [5]. The model of the track comprises of ground, ballast, rails, and stiffness between these bodies as marked with a blue line in Figure 6.

Characteristics of the model are tuned and validated against both the track-based and vehicle-based measurements [3].

2. WEAR MODELLING

The methodology of the wear calculation in this paper is based on Archard’s definition of the wear [6]. He found that the volume of material removed by wear per sliding distance ($W$) is proportional to the quotient of the pressure ($p$) and the hardness ($H$) of the softer material. The proportionality factor is called the wear coefficient ($k$),

$$W = k \frac{p}{H}$$  \hspace{1cm} (1)

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-disk tests. Olofsson and Telliskivi [7] investigated the evolution of rail profiles on a commuter line track within two years and performed several laboratory tests with two different testing
machines: a two-roller (disc on disc) and a pin-on-disk machine. The tests resulted in a simplified wear map with the wear coefficients depending on sliding velocity and contact pressure, as shown in Figure 7.

Figure 7: Wear map for wheel and rail steel with typical regions of tread and flange contacts; \( H \) is the hardness of the material.

The wear prediction methodology is based on a load collective concept, which determines a set of dynamic time-domain simulations [1]. These simulations reflect the actual rail network for vehicles in question, including for example, track 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. A summary of the methodology is presented in Figure 8.

![Figure 8: Methodology of the wheel wear prediction tool developed by Jendel & Berg [1].](image)

3. RCF CALCULATION

Johnson [8] 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 determining the RCF probability in railway wheels and rails. The horizontal and vertical axes of the shakedown diagram are functions of friction values and the normal load in the contact area, respectively.

An engineering model for RCF risk assessment is developed by Ekberg, Kabo, Andersson in [9] based on the shakedown theory. The method calculates the distance between the simulated results from a boundary curve within the shakedown diagram and it is called the surface initiated fatigue index (\( F_{\text{surf}} \)). The boundary curve is one divided by the calculated traction coefficient.

\[
F_{\text{surf}} = T - \frac{1}{v}
\]  

where \( T \) is the traction coefficient and \( v \) is the vertical load normalized by the material yield stress in shear. Positive values of \( F_{\text{surf}} \) represent the ratchetting part of the shakedown diagram resulting in surface initiated fatigue. The higher the values of \( F_{\text{surf}} \), the higher the tangential forces, and thus, the higher the risk of RCF. A simpler shakedown diagram from what Johnson [8] introduced is shown in Figure 9.

![Figure 9: Shakedown diagram](image)

4. TRADE OF BETWEEN WEAR AND RCF

Many attempts have been made to combine wear and RCF effects in the prediction of wheel and rail damages. At a certain level of wear depth the initiated cracks could be polished away from the surface. In the UK a model has been proposed and calibrated on six intermediate radius curves, where RCF on rails are observed, see Burstow [10]. In the UK model it is assumed that the wear depth is linearly proportional to the wear number, i.e., energy dissipation in the contact patch. However, the spin effect is not included in this calculation. The energy dissipation per unit distance area \( E \), or wear number, is the creep forces times the creepages added to the moment \( M \) times the spin \( \varphi \) in the contact patch.
\[
\bar{E} = F_x v_x + F_y v_y + M \varphi
\]
where \(F_x\) and \(F_y\) are longitudinal and lateral creep forces, respectively \(v_x\) and \(v_y\) are the corresponding creepages.

In the current paper, we adopt the idea of the influence of energy dissipation on RCF, as introduced in the model from Burstow, and assume that RCF is more pronounced below a value \(\bar{E} = 50 \, J/m\). Beyond that value, wear starts to grow and finally, for values higher than \(\bar{E} = 100 \, J/m\), wear is totally dominating and no RCF will be propagated.

With this concept, we introduce a factor to correct the effect of RCF. This factor is multiplied with the total number of simulation points that passed the boundary curve in the shakedown diagram and that are subjected to RCF.

The schematic figure of this correction factor is shown in Figure 10. The values \(\bar{E}_1\) and \(\bar{E}_2\) are chosen in such that the calculated crack locations on the wheels match the observations from the field.

![Figure 10: RCF correction factor according to the energy dissipation per meter rolling distance.](image)

### 5. OPERATIONAL CASES

To reflect the actual rail network, a set of time-domain simulation cases called load collective is prepared. Some of the parameters for designing a load collective are track design geometry and irregularities, rail profiles, speed, friction level, and wear coefficient. Based on the statistics of the measured track geometry of the iron-ore line, nine track sections are finally chosen for each wear step.

Details of each section are presented in Table 1.

![Table 1: Details of track design geometries for each simulation case concluding a wear step](image)

<table>
<thead>
<tr>
<th>Radius (m)</th>
<th>Transition Curve (m)</th>
<th>Constan t Curve (m)</th>
<th>Length (m)</th>
<th>Cant (mm)</th>
<th>Gauge (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>306</td>
<td>80</td>
<td>270</td>
<td>4.4</td>
<td>68</td>
<td>1444</td>
</tr>
<tr>
<td>368</td>
<td>100</td>
<td>210</td>
<td>0.9</td>
<td>61</td>
<td>1443</td>
</tr>
<tr>
<td>409</td>
<td>175</td>
<td>140</td>
<td>1.5</td>
<td>66</td>
<td>1448</td>
</tr>
<tr>
<td>547</td>
<td>145</td>
<td>227</td>
<td>13.9</td>
<td>51</td>
<td>1447</td>
</tr>
<tr>
<td>648</td>
<td>140</td>
<td>210</td>
<td>16.1</td>
<td>46</td>
<td>1445</td>
</tr>
<tr>
<td>834</td>
<td>170</td>
<td>200</td>
<td>7.9</td>
<td>37</td>
<td>1444</td>
</tr>
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<td>110</td>
<td>210</td>
<td>5.4</td>
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<tr>
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<td>110</td>
<td>190</td>
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<td>-</td>
<td>500</td>
<td>48.1</td>
<td>0</td>
<td>1435</td>
</tr>
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</table>

Track irregularities are also chosen based on the measurement data. Note that for quantification of the irregularities, the European track quality standard is used [11]. Various types of rail profiles are used on the iron-ore line, depending on curve radius and on whether the track is located in Norway or Sweden. As around 75 percent of the tight curves (below 600 m radius) are located on the Norwegian side of the line, only the Norwegian standard is used for the tight curve sections. Likewise, for 88 percent of the wide curves (above 600 m radius) it is reasonable to use the Swedish standard for the wide curve sections. The speed of the train is also chosen according to the sections’ curve radii. For the tight curves 50 km/h, for medium curves 55 km/h and for wide curves and straight tracks 60 km/h are chosen. Note that the weight of the wagons changes between loaded wagon (axle load: 30 tons) and empty wagon (axle load: 5.4 tons). This means that the weight of the train changes every 470 km of running distance.

### 6. BRAKING

As the wagons are equipped with shoe brakes, their influence on wheel wear has to be considered. This is done via calculating the total amount of resistance force for each simulation case. The total running resistance (\(D_l\)) can be divided into four parts

- Mechanical resistance (\(D_m\));
- Aerodynamic resistance (\(D_a\));
- Curve resistance (\(D_c\));
- Gradient resistance (\(D_g\)).

For details of the calculation of the mentioned resistance forces see Andersson and Berg [12]. It is known that a maximum 500 kN of the needed braking force can be produced by the locomotive. Therefore, the rest of the required braking force is taken care of by the pneumatic tread braking system of the wagons. Note that a negative total resistance force (\(D_l\)) means that the vehicle needs to accelerate, while a positive value means that the vehicle needs to brake to keep the speed constant. As the effect of braking is the only one that matters in wagons, negative values are disregarded. Table 2 shows an example of the estimated resistance forces for a curve section with 648 m radius in Sweden and a section with 409 m radius in Norway. The negative values are filled in grey colour.
Table 2: An example of the distribution of the estimated resistance forces for various track and traffic conditions

<table>
<thead>
<tr>
<th></th>
<th>Norway</th>
<th>Sweden</th>
<th>Gradiente</th>
</tr>
</thead>
<tbody>
<tr>
<td>13.61 -13.61</td>
<td>6.84</td>
<td>-6.84</td>
<td></td>
</tr>
<tr>
<td>-1120 -563</td>
<td>0 563</td>
<td>Loaded</td>
<td></td>
</tr>
<tr>
<td>-244 -122</td>
<td>0 122</td>
<td>Empty</td>
<td></td>
</tr>
<tr>
<td>-154 -92</td>
<td>-20 -20 Empty</td>
<td></td>
<td></td>
</tr>
<tr>
<td>-24 -26</td>
<td>-26 -26 Loaded</td>
<td></td>
<td></td>
</tr>
<tr>
<td>-6 -7</td>
<td>-7 -7 Loaded</td>
<td></td>
<td></td>
</tr>
<tr>
<td>936 -888</td>
<td>-125 438 Loaded</td>
<td></td>
<td></td>
</tr>
<tr>
<td>-308 -175</td>
<td>-35 69 Empty</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 2 shows the maximum estimated braking force is 936 kN, which is located in one of the tightest curves in the Norwegian part of the iron-ore line. As mentioned earlier in this section, 500 kN should be subtracted from this value to estimate the braking force which is handled by the wagons. As there are 68 wagons attached to the each locomotive, the amount of braking force per wheel is calculated to (68 x 4 x 2). Therefore, for the specific case of the example presented in Table 2, the amount of braking force remaining for each of the wheels is 0.8 kN. This value is almost less than 2 percent of the tangential force created in the wheel-rail contact area in a similar operational condition as the example, Hossein [13] and Hossein, Jönsson, and Stichel [14]. Consequently, the effect of the braking force on the initiation of RCF can be neglected. However, it may affect the propagation of the cracks by slightly polishing the surface of the wheel in the vicinity of the nominal wheel and rail contact position.

7. RESULTS

The available measurement data for worn wheel profiles are limited to 150'000 km running distance. Therefore, the maximum simulation running distance is also limited accordingly. It is assumed that the wear is symmetric on both the left and the right wheels. Therefore, only the results of the left wheels are shown in this section. To compare the calculated worn wheel profile and the measured wheel profile, one of the few available measured profiles is chosen. Figure 11 shows the agreement between the measured and simulated worn wheel profiles is very good. However, there are three ranges with deviations between the measured and simulated worn profiles, which will be discussed here. The first range is close to the field side corner between -0.06 m and -0.04 m. The deviation is mostly due to the effect of wheel-rail contacts in switches and crossings. This is neglected in the simulations due to the simplicity and reducing the simulation time. However, in [15] the problem is investigated and the possibilities of considering the effect of switches and crossings on the wheel profile evolution are shown.

Figure 11: Comparison between the measured and the simulated wheel profiles after 150'000 km of running distance for the original WP4 wheel profile

The second difference is due to the block brakes effect and it is located around the nominal wheel and rail contact positions. Figure 12 shows the difference between the measured and simulated worn profiles divided by the running distance. As it is seen in the figure, there is a 0.002 micrometers difference between the simulated and measured worn profiles due to the braking effect. This small difference could be due to the very low braking force which is described in Section 6.

Figure 12: Difference between simulated and measured wheel wear per kilometre of running distance
The third difference is at the flange root. Due to the lack of variety in the data of measured rail profiles it was not possible to use extremely worn rail profiles in the simulations. Therefore, it was not possible to get wheel-rail contact points in the flange root often, which probably is the case in reality.

The simulations are repeated for the new suggested wheel profile. Figure 13 shows the starting new wheel profile and the corresponding simulated worn profile after around 150'000 km.

Figure 13: The evolution of the proposed new wheel profile due to wear after 150'000 km.

The total depth of the material removal after this running distance is compared for both the suggested and current profiles in Figure 14.

Figure 14: Total calculated material removal of the leading axle after 150'000 km for the current WP4 and the suggested WPX wheel profiles.

As the figure shows, there is not much difference between the amount and shape of the wear for both wheels. However, the flange wear is expected to be a bit higher for the WPX than for the current WP4 profile.

To see the evolution of RCF on the wheel profiles, we need to first sum up the weighted RCF indices for the nine simulation cases shown in Table 1. Second, the procedure needs to be repeated for all consecutive wear steps until the desired travelling distance is achieved to get the accumulative RCF crack locations on the wheel profiles. In Figure 15 the accumulated risk for RCF is shown as function of the position on the wheel profile and of the rolling distance.

Figure 15: Evolution of RCF on the first axle for the WP4 current wheel profile, the darker the area the more severe the RCF.

The location with a higher risk of RCF, shown in Figure 15, matches the reported crack locations from the observations in the Kiruna workshop [16], as shown in Figure 16.

Figure 16: Observed RCF locations on the iron-ore wagon wheels [16].

Figure 17 shows the RCF evolution of the suggested WPX wheel profile. The RCF locations on the tread are almost the same for both the wheel profiles. However, the WPX profile evolves extra RCF cracks on the flange side after a long running distance. Note that although there is risk to initiate cracks on the flange side, but the propagation of the cracks is unlikely, occurs at this location. This is due to the direction of the longitudinal forces which trap any lubricants (like water) into the crack and cause widening the cracks on the inner wheel (the contact area is in the wheel tread) and outer rail (the contact area is in the wheel flange) [17].
Figure 17: Evolution of RCF on the first axle for the WPX suggested wheel profile, the darker the area the more severe the RCF

The probability of RCF is also calculated for both wheel profiles by dividing the number of ratchetting simulation points by the total number of points cf. Figure 9. The mean value of the RCF risk probability for all simulation cases is around 41 percent for WP4 profile and 38 percent for the WPX profile. The calculations also show that the suggested WPX profile produces less RCF than the current WP4 profile during the initial wear steps, with a mean value of around 30 percent. For the latest wear steps, RCF probability for the WPX profile is worse than for the actual profile. This means that the wheel profile evolution due to wear changes the geometry of the WPX profile into a shape with higher long term RCF development than with the profile used today.

8. CONCLUSION & RECOMMENDATIONS

In this paper the long term stability of wheel profiles of the iron-ore wagons running in northern Sweden and Norway is analysed. Wear and RCF evolution of the profiles as function of rolling distance are studied. The effect of shoe brakes is estimated first by calculating the amount of braking force required for each wheel and then by comparing the simulated worn profile with the measured one. The methodology is first validated by comparing the simulated and measured worn profiles for the actual wheel profiles. Simulated wear and RCF evolution of the current wheel profiles shows good agreement with measurement and observation data; minor differences are mainly due to the wear at the corner of the field side due to switches and crossings and due to the lack of measured extremely worn rail profiles in the simulations. The total wear depth of both current WP4 and suggested WPX profiles is calculated and the results show high similarities between the worn profiles. The RCF calculation process reveals that although the suggested wheel profile produces less RCF in the beginning of its operation, it may be subjected to more RCF damage in the long run. Due to the mentioned conclusions and the similarity of the profiles, LKAB has not decided to change the wheel profiles of the wagons. However, a field test with the new WPX profiles on wagons would be beneficial to verify the simulation results.

9. ACKNOWLEDGMENTS

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10. REFERENCES

[11] UIC: Testing and approval of railway vehicles from the point of view of their dynamic


