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INFLUENCE OF SWITCHES AND CROSSINGS ON WHEEL PROFILE EVOLUTION IN FREIGHT VEHICLES

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Abstract

Wheel reprofiling costs for freight vehicles are a major issue in Sweden, reducing the profitability of freight traffic operations and therefore hindering the modal shift needed for achieving reduced emissions. In order to understand the damage modes in freight vehicles, uniform wear prediction with Archard's wear law has been studied in a two axle timber transport wagon, and simulation results have been compared to measurements. Challenges of wheel wear prediction in freight wagons are discussed, including the influence of block brakes and switches and crossings. The latter have a major influence on the profile evolution of this case study, so specific simulations are performed and a thorough discussion is carried out.

1. INTRODUCTION

Wheel reprofiling costs for freight wagons are a major issue for freight traffic operators. Among the reprofiling causes there are rolling contact fatigue (RCF), wheel flats, out of round wheels or uniform wear. In Sweden, a rough estimate would be that *ca.* 80% of the reprofilings are due to RCF shelling, *ca.* 15% due to wheel flats and *ca.* 5% due to other causes. So the wheel profile evolution due to uniform wear does not have a major direct influence on these reprofiling costs. However, these changes in wheel profiles will increase the equivalent conicity of each wheelset, which might influence riding characteristics that will affect the development of RCF. Moreover, these two damage modes, *i.e.* RCF and uniform wear, are not independent, as high uniform wear will avoid the development of the cracks generated by RCF that eventually deteriorate the wheelset up to a point when it needs reprofiling.

S1002 is a common wheel profile for freight wagons in Europe, which was originally developed for 1:40 rail inclination. In Sweden there is 1:30 rail inclination, so the wheel-rail contact geometry is not necessarily optimized: for negative lateral displacements of the wheelset, contact pairs are concentrated in a specific area of the wheel (Figure 1), which will increase local wear and might favour the appearance of RCF cracks. Also, the extreme weather conditions during winter increase the development of RCF. This leads to shorter wheelset life before reprofiling; some wagons are running less than 40kkm during winter between each reprofiling.

Figure 1: Contact pairs of a S1002 wheel profile against UIC60 rail with inclination 1:40 (black) and 1:30 (red)

There are other components to be considered, too. Switches are an essential component of rail vehicles operation,

as they enable the flexibility that parallel rails lack. They allow one track to be branched into two of them, allowing the vehicle to run in different directions. This flexibility comes at a cost, though: switches include nonlinearities in rail geometries, including rail gaps, which generate high dynamic contact forces that will eventually damage both wheels and switches. In fact, switches and crossings account for the highest proportion of all the maintenance costs in the Swedish network [1]. The usual damage mechanisms on a switch are wear, rolling contact fatigue and large accumulated plastic deformations. Probably due to the high maintenance costs, there are many studies on damage on switches, *e.g.* [2]–[4] to cite some of the latest; or on the optimization of switch geometry to reduce the damage level [5], [6]. However, there are no studies considering the threat that switches are for the wheels of the passing vehicle. Taking into account that the wheel-rail contact is a coupled system, the damage will most likely not only be on the switch components, but also on the wheel surface.

The deterioration of the wheel-rail contact interface is an intricate process involving contact mechanics, tribology and vehicle dynamics. Consequently, accurate numerical models of these three disciplines are needed for an accurate wear prediction. A bibliographic review on these three disciplines involving wheel-rail contact wear prediction has been performed by Enblom [7]. The research performed in this paper is applied to a macro-scale system, so the following literature review will deal with applications of different wear prediction methodologies.

The first created wear calculation models were single-variable, assuming that wear is proportional to the dissipated energy at the wheel-rail contact [8]–[10]. With the improvement of computational resources, more modern multi-parameter models appear. Jendel develops a numerical procedure based on Archard's wear model and Kalker's simplified contact theory [11]. This methodology has only been validated for wheels and rails of vehicles in passenger traffic, but not for freight wagons, which have evident differences in their operational conditions. The main differences between freight wagons and other rail vehicles are highly nonlinear suspension elements (usually friction dampers) and high axle load, both having notable influence on creep forces that will cause wheel wear [12]–[14].

The aim of the study is to assess the suitability of existing wear prediction methodologies, specifically Archard's wear model, when applying them to freight vehicles. In order to do so, the development of uniform wheel wear in a freight vehicle in Sweden is studied. Specific characteristics of the freight vehicles are carefully modelled in order to obtain significant results. The simulated profiles are compared with measured profiles in order to determine the validity of the methodology for the prediction of wear development in freight vehicles. The discrepancies between the experimental and the simulated profiles are discussed. Afterwards, the vehicle running through a switch is studied in order to confirm its influence on wheel wear, concluding that switches might have major influence for most switch configurations.

The main contributions of the paper to the existing bibliography are i) the study on how switches affect wheel wear and ii) the comparison between wear calculation methodologies for extreme wheel-rail contact conditions.

2. MODELLING AND VALIDATION

2.1 Wear calculation

As previously stated, in order to calculate wheel profile evolution Archard's wear calculation methodology has been used. According to Archard's wear law (Eq. 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/m²).

$$V_w = k \frac{sN}{H} \quad (1)$$

To calculate the wear distribution inside the contact patch, the contact ellipse is divided in $m \times n$ elements, and the wear depth is calculated for each one (Eq. 2). It is proportional to the contact pressure p (N/m²), the sliding distance for that element Δs (m) and inversely proportional to the hardness of the worn material H (N/m²).

$$\Delta h = k \frac{p \Delta s}{H} \quad (2)$$

The proportionality factor is represented by the Wear Coefficient (k). This coefficient has been determined in laboratory experiments and depends on the sliding velocity (v_s) and the contact pressure (p) of the surfaces in

contact. The dependence of k with these magnitudes is represented by a chart with four different regions, which represent the different wear mechanisms occurring for different contact conditions (Figure 2).

Figure 2: Wear coefficient (k) for steel to steel contact [1].

2.2 Vehicle model

The studied vehicle is a two axle freight wagon with *Unitruck* running gear where each wagon is composed of two short coupled units with UIC standard side buffers [15]. It is used for carrying timber around central Sweden, with a 5.8 t axle load when unladen and 22.5 to 25 t when laden.

The *Unitruck* running gear is detailed in Figure 3. The design is optimized for 25t axle load, but usable up to 30t axle load. The suspension system is single stage, composed by four nested coil springs connecting the carbody (1) and the saddle (2). These nested coil springs allow having an optimized stiffness for both laden and unladen vehicle. Friction dampers provide the necessary damping in connection to the wedge (3), the piece that couples the vertical pre-load with longitudinal, lateral and vertical friction surfaces. The inner coil spring is connected to the wedge, transmitting vertical load to the carbody across an inclined friction surface (4). The vertical inclination of the surface generates a longitudinal force that is transmitted between saddle and wedge through a vertical friction surface (5). This coupling enables the generation of friction damping in longitudinal, lateral and vertical direction. The saddle (2) is mounted on the axle box (6) through a rocket seat coupling (7) that enables the saddle to have a sway angle with respect to the axle box. Displacements between carbody and saddle are limited by bumpstops in the different directions.

Figure 3: Vehicle suspension system with friction dampers (4 and 5) that connects axle box (6) with carbody (1)

The vehicles equipped with this suspension system suffered from high flange wear, and the running gear was modified in 2005 in order to solve the problem (Figure 4). The obvious modification consisted on softening the longitudinal suspension, for which the longitudinal friction plate was replaced with a rolling element (Figure 4A). Then, in order to avoid the buckling of the springs, a spring plate was introduced connecting the centre position of both of them (Figure 4B). With these modifications, flange wear was reduced up to 50%.

Figure 4: Modified Unitruck running gear [14].

The vehicle has been modelled in MBS tool GENSYS for the dynamic simulations [16]. The model consists of fifteen bodies: the carbody, two wheelsets and four saddles, plus eight massless wedges. No flexible bodies are considered. The vehicle has a total of 90 degrees of freedom. Track flexibility was modelled introducing two additional *dof* per wheelset, representing track deflection in vertical and lateral direction, for a total of 98 *dof*. Track irregularities obtained from measurements are introduced.

Wheel-rail contact geometry and its nonlinear functions are pre-calculated considering elastic deformation at the contact patch. Up to three simultaneous contact patches are considered in order to account for multiple contact points when studying worn profiles. These contact patches are modelled as linear springs, and creep forces are pre-calculated by FASTSIM and stored in a table.

The vehicle model has been validated with measurements carried out for the UIC approval of the vehicle. Details about the validation can be found in [14].

Simplification of the suspension

In order to include the modified running gear in the new model the plate has been included (Figure 4), 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 developed (Figure 5). The simplification consists of 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 Figure 5 comprise three dimensions, *i.e.* $\vec{k} = \{k_x, k_y, k_z\}$. The carbody is considered fixed to the reference system.

The following simplifications have been made:

- 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 have 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 3). Also, the roller element between carbody and wedge delimits the vertical displacement of the wedge (Equation 3).

- All the stiffness values of the four coilsprings connected to the plate are considered the same, with value $\vec{k} = \{k_x \ k_y \ k_z\}$.

In the New Model (Figure 5A) the displacements of the plate can be determined as a function of the displacements of the saddle. Considering the plate in quasi-static equilibrium, the following relations are obtained (Equation 4).

Figure 5: Simplification schemes of the plate suspension.

$$\begin{cases} x_w = x_s \\ z_w = x_w \cdot \tan \alpha = x_s \cdot \tan \alpha \\ \phi_w = \phi_s \end{cases} \quad (3)$$

$$\begin{cases} x_p = \frac{3}{4} x_s \\ z_p = \frac{z_s}{2} - x_s \frac{\tan \alpha}{4} \\ \phi_p = \frac{3}{4} \phi_s - x_s \frac{\tan \alpha}{4} \frac{k_z b_p}{k_\phi + k_z b_p^2} \end{cases} \quad (4)$$

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 new and simplified models are made equal, and the results are expressed the following equations.

$$\vec{K}_{cs} = \frac{\vec{k}}{2} \quad (5)$$

$$\vec{K}_{cw} = \left\{ - \frac{k_z}{2} \quad - \right\} \quad (6)$$

$$\vec{K}_{ws} = \left\{ \frac{k_x}{4} \quad \frac{k_z b_p}{4 b_w} \quad \frac{1}{4} [k_\phi + k_z \cdot b_p (b_p - b_w)] \right\} \quad (7)$$

Considering the simplifications that have been done, the calculation of K_{cw}^x and K_{cw}^ϕ 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 $\vec{K}_{cs}$ the values for $\vec{K}_{cw}$ are set as:

$$\vec{K}_{cw} = \frac{\vec{k}}{2} \quad (8)$$

The results of both suspension elements are compared in Figure 6. As it can be seen, the results of both models are extremely similar, which is a very good result 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 6: Comparison of the lateral force of the leading wheelset between the new model and the simplified model in a common curving case simulation.

2.3 Operational case

The studied vehicles run loaded with timber lodges in central Sweden, taking wood from the inland timber terminals to the industries at the east coast. All the existing routes have been reduced to ten distinctive runs that include *ca.* 90% of the total distance run by these vehicles, in order to simulate only the most significant ones. The operational case is modelled as follows:

- The network is represented by 17 curve intervals.
- Worn rail profiles are modelled with three levels of wear.
- Three levels of track irregularities are considered, depending on curve radius.
- Friction coefficient at the wheel-rail interface is considered constant.
- Traction or braking is not considered.

In order to calculate the total wear, 17 curves are simulated, the total generated wear is calculated and the wheel profile is updated. The process is repeated with the updated wheel profiles until the desired wear calculation loops are achieved, or until the desired mileage is reached. In order to account for the different rail wear levels the following process has been followed: each curve interval has a statistical spread of these worn levels, which have been obtained from track measurements. In order to minimise time simulations, each curve interval is randomly assigned a rail profile on each wear loop calculation. This way, in the long run each curve interval has the correct spread of worn rail profiles without increasing the number of simulations performed in each wear loop.

2.4 Experimental validation

Experimental profile measurements were performed in Green Cargo installations in Borlänge 10th and 11th of November 2011. Eighty wheelsets were measured and the statistics of these measurements are represented using three different scalar variables: flange height (h_f), flange thickness (t_f) and flange inclination (q_f).

In the following graphs (Figure 7 to Figure 9) the simulated values for flange height, thickness and inclination are depicted along with the experimental measurements. From these results the following can be deduced:

- Flange height (h_f) agrees quite well with the experimental results.
- Flange thickness (t_f) is correctly predicted for the first few thousand km. After *ca.* 10 000 km it starts to detach from the measurement trend and around *ca.* 20 km it is no more an accurate prediction. However, it should be taken into account that the experimental data is not very consistent, with a span of up to 4 mm

for specific mileages. Even though, the predicted t_f values are always within the experimental span.

- Flange inclination (q_r) does not agree with the experimental results. Its values are equivalent to the ones of an original S1002 profile.
- 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 7: 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 8: Experimental flange thickness (t_f) 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 9: 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)

Figure 10 depicts individual profile measurements and simulations for different mileages. Looking into these specific profiles, a big difference is detected both at the tread end and the flange between the simulation and the measurement. This heavy wear at the tread end is produced even for very low mileages (Figure 10), where running for 100km has generated wear at the tread end that can be detected by plain sight. This behaviour is consistent for all the measured profiles and cannot be detected when using the scalar variables for wear (h_f , t_f and q_r) as neither of them accounts for the geometry at the field side of the wheel. Also, for all the measured profiles there were not visible RCF cracks. Measurements were carried out in November, before the cold season where RCF is developed at a faster rate; that might be the reason for the low occurrence of RCF.

Figure 10: Experimental validation of wheel wear. Original S1002 (black) simulated (red) and measured (blue) for ca. 22 000 km (1L) and 100 km (1R)

In order to detect where these differences are produced, an analysis of the wheel-rail contact geometry is performed. Figure 1 depicts the contact pairs for a S1002 wheel with UIC60 rail profile with inclinations 1/40 and 1/30. As it can be seen, contact points never reach the field side of the wheel for ideal track. Simulations include track irregularities, and even though gauge widening up to 20mm might occur for very poor track quality, this would only happen for short periods of time, so it would not affect the heavy tread end wear. Even measured worn out profiles as the one depicted in Figure 10 have similar contact pair distribution which never reaches tread end. For any conventional vehicle contact so far out on the tread only occur at one operational condition, *i.e.* when running through switches and crossings.

It has to be mentioned that, for mileages over 80kkm, measured wheel profiles had plastic deformation at the field face of the wheel. This plastic deformation was detected by hand; no measurements could be performed as the MINIPROF device cannot be used to measure the outer face of the wheel. Therefore, we can assume that high tread-end wear might be generated by both uniform wear and plastic deformation towards the field side; however, as this plastic deformation was only detected for very high mileages and wheelsets about to be reprofiled, it can be assumed that its influence is small for the low mileage studies that have been performed.

2.5 Modelling of a switch

A typical switch geometry is depicted in **Error! Reference source not found..** During the switch run, the wheel running through the stock rail does not have any special behaviour, but the opposite wheel is first leaving its own stock rail and running onto the blade, and at the crossing section is jumping over a rail gap when the wing rail disappears, impacting into the nose. So in every switch the separation of the diverging rails (stock rail at the blade section and wing rail at the nose section) might cause contact points at the tread end and at the flange at the same instant, having high creepages and dynamic contact forces at both contact points. According to Archard's wear law, these will generate high volume of wear at these positions, which might explain the discrepancies between the developed wear calculation model and the profile measurements. Thus, the influence of generic switch geometry on the uniform wheel wear of this specific freight vehicle has been studied.

Figure 11: Generic switch geometry and components of the system [1].

For the study of the dynamic behaviour of the vehicle running through a switch, the capabilities of GENSYS for simulation of variable rail profiles have been used. The modelled turnout is a medium size turnout about 60 m long and with standard track design UIC60-760-1:15 (UIC60 rails, curve radius 760 m and crossing angle 1:15) [1]. The rail profiles that describe the switch are measured in 30 different sections and assigned to a specific longitudinal position, for a total length of 60 m. Wheel-rail geometrical properties have been calculated for all these sections, and during the calculations these functions are interpolated according to the longitudinal position of each wheelset. The switch is ideal, so there are no irregularities or wear on the track.

Figure 12 depicts the wheel-rail contact pairs of some sections at the nose, when the outer rail leaves the central position (to the left side) and the nose starts to rise and develop into a full rail profile. There is a succession of contact pairs with a concentration of these points close to the field end of the profile; this might generate small contact areas constantly applied in the same spot, which in addition to the high tonnage of freight vehicles, create very high normal pressure values. Also, when two point contact conditions arise, the contact points are so far from each other that the rolling radius difference is extremely high; this will generate very high creepages at one or both contact points, as the longitudinal and rolling speed is the same for both of them, equal to the speed of the wheel. As it was stated at the description of Archard's wear law (Eq. 2), the wear volume is proportional to the contact pressure and sliding distance, so a very high volume of wear can be expected both in tread-end and flange.

Figure 12: Wheel-rail contact pairs (left) for specific longitudinal sections of the switch at the nose (right)

3. RESULTS AND DISCUSSION

The two axle freight vehicle has been simulated running through the curved path of the switch at 50km/h. Two wheel profiles have been used (Figure 13), new S1002 and a measured S1002 after 51.4 kkm running through the network. First, the volume of wear generated at the different lateral positions of the wheel has been estimated by an energy approach ($T\gamma$). The calculation is straightforward as an output of the dynamic simulation, and is depicted in Figure 14, where the energy is plotted for each contact point position during the simulation (positive positions indicate flange contacts).

Figure 13: Wheel profiles used for the simulations: new S1002 profile and measured worn S1002 profile.

Left wheel is the outer wheel at the curve, and is also dealing with the blade and the nose, while right wheel is the

inner wheel at the curve. The energy dissipation at the left wheel is much higher, especially at flange contact (green, *ca.* 30 to 40 mm). It should be noted that this position, high at the flange, is one of the discrepancy points at the experimental comparison (Figure 10-1L). High energy dissipation is also detected at the tread centre (blue, *ca.* -25 to 0 mm) when running through the blade. However, there are no contact points at the far end of the tread (*ca.* -60 mm) that would justify high wear at the very end, as it can be seen in the profile measurements. For the right wheel, there is very high energy dissipation when running through the blade sections at the centre of the tread (*ca.* -10 to 0 mm) and two point contact is never achieved.

Figure 14: Energy dissipation ($T\gamma$) at the wheel rail contact relative to the lateral position of the contact point at the wheel, considering original S1002 wheel profiles (left) and measured worn profiles (right). cpt – tread contact point; cpf – flange contact point; r – right wheel; l – left wheel.

For the vehicle with worn profiles the behaviour is very similar, except for two details: i) contact points reach tread end when running both through the blade and the nose (green and blue respectively, *ca.* -60mm). However, the energy dissipation level is below 100J/m except for two simulation points just after the wheel hits the nose, where dynamic forces and creepages increase momentarily for less than 50ms. So even if contact at the tread end is detected, energy dissipation does not justify the high wear level observed in the measurements; and ii) the overall energy dissipation level is slightly lower than for the original profile, except for isolated points. Also, the transitions from low to high wear levels are much smoother, especially at the blade (blue, *ca.* -10mm). This is an interesting result, as it means that the shape of the worn wheel profile is better suited to run on the switch than the original profile. To put it in other words, it seems that the geometry evolution of the worn profile is driven by the geometry of the switch, but the $T\gamma$ model is unable to predict the wear at the tread end.

Afterwards, Archard's wear law was implemented in the simulation of the vehicle running on the switch. Energy dissipation did not fit experimental measurements, so this multi-variable approach was tried to check if it predicted the uniform wear more accurately. Figure 15 depicts the wear depth against the lateral position of the wheel, calculated with this method. In order to compare the wear generated by the switch with the one generated by the network geometry (operational case), the number of switches has been calculated, resulting in an average of one switch every 3.14 km of the network. In long-dash the wear for 3.14km run is depicted, while solid and short-dash depict the wheel wear when running through a switch with new and worn S1002 profiles respectively.

Figure 15: Radial wear d_z (m) generated by running through the network (long dash), new S1002 profile over a switch (solid line) and worn S1002 profile over a switch (short dash)

First, wear depth from running through S&C is *ca.* 10 times higher than the one of the operational case. The shape of the wear is obviously different, as it is concentrated at the flange while running on the network results in more evenly distributed wear. Comparing wear volumes, the wear for the S1002 original profile vehicle is ten times higher than the operational case; for the worn S1002 profiles, wear is 17.3% higher than for the original S1002 profiles.

As it can also be seen in Figure 15, the original S1002 profile running through this specific switch does not cause tread-end wear. However, the already-worn-profile is able to predict a very high level of wear at this position below -40mm. According to these simulations, once that wear at the tread-end starts, it is easier to develop further wear.

Also, this analysis was performed considering that the wear is evenly distributed along the perimeter of the wheel, this is, for any angle that we measure the profile will be the same. Although this might be a good assumption for the operational case, it is not for the switch simulation, as the high wear patterns occur when running through the switch blade (*ca.* 40mm length) and through the nose crossing (*ca.* 40mm length). Considering a wheel diameter of 0.955m, one wheel turn is 3m; thus, the influence of each of these two parts can never be considered as uniform wear and Figure 15 should be treated as a simplification.

Lastly, the scaling of the track generated wear has been performed considering that the vehicles run through all the switches on the curved section, which is not very accurate. Actually, most of the switches are crossed through the tangent track section, so this scaling might over-estimate the relative wear between the operational case and the switch run, but only if the straight run through the switch has a lower wear level. In order to account for this behaviour, a detailed analysis on different switch conditions has been performed in order to assess the relative influence of the different situations that can be encountered when running through a switch.

Non-uniform wear

In order to understand the wear pattern created on the wheel when running on the switch, wear is depicted on the running plane. Figure 16 shows a contour plot of the wear generated at the wheel for the longitudinal position of the switch. However, the different wear scales that appear in the contact patch, ranging from $1\text{e-}10\text{m}$ to $1\text{e-}8\text{m}$ in a few square millimetre contact area, do not allow a good representation of these results. The following figures will help to explain different characteristics of the generated wear.

Figure 16: Wheel wear depth at the wheel developed on the horizontal plane. Isolines depicted each $0.5e-10$ m. Reference switch, curved section; vehicle with S1002 wheel profiles.

Figure 16 depicts the wear depth at different levels, with respect to both the longitudinal position of the switch where they occurred, and the lateral position on the wheel, showing the contact patch position and width. This case represents the reference switch, curved path, with the vehicle running with new S1002 profiles. At $x=0$ m the switch blade appears and the stock rail starts to move laterally towards the field side, thus the contact point is also displaced towards the field side. A few meters later, the wheel finally contacts the blade, and two point contact arises. Between these two contact points there is around 40mm separation, so the rolling radius difference is quite high. Also, the initial contact with the blade generates very small contact patches, so contact pressure increases drastically generating very high wear values. After ca. $x=10$ m the fully developed blade supports the wheel, going back to low wear values.

After the curve, the crossing section appears. First, the wing rail diverges quite drastically, generating a great displacement of the contact point towards the field side, until the wheel contacts the nose. At that instant and for a very short time, there is two-point contact with extremely high wear at the tip of the nose, due to the high contact pressures and sliding speeds. This high wear is reduced to low values in a very short period, when the nose develops into a full rail.

Figure 17 focuses on the blade section, where it can be seen that the high wear sections on top of the switch blade have a very thin contact patch, and there is a very high variation on the maximum wear value. This means that the conditions through the blade are very close to a border between wear mechanisms in Archard's wear map.

Figure 17: Wheel wear depth at the wheel developed on the horizontal plane, blade section close-up. Reference switch, curved section; vehicle with s1002 wheel profiles. Flange section in red ($y > 40\text{mm}$)

Changes in the contact conditions around the frog are much more abrupt. Figure 18 depicts the behaviour around the nose gap. First, at ca. $x=45.5\text{m}$ the contact point on the wing rail has a very pronounced lateral displacement. Then the nose appears and two point contact arises for a short period, with higher wear at the wing rail due to the high sliding speeds generated by the rolling radius difference, and very high wear at the nose tip due to both high sliding speeds and high pressures in the small contact patch. In this case the maximum wear ($1.047 \cdot 10^{-6}\text{m}$) is two orders of magnitude out of scale. After a short period the nose widens and the contact patch size increases, while the second contact point is lost. This way, both pressure and sliding speed are reduced, going back to a mild wear zone in Archard's wear map.

Figure 18: Wheel wear depth at the wheel developed on the horizontal plane, nose section close-up. Reference switch, curved section; vehicle with s1002 wheel profiles. Flange section in red ($y > 40\text{mm}$)

For the other wheels of the vehicle, the inner wheel of the leading wheelset has wear values one order of magnitude lower than the outer wheel, and the lateral displacement of the contact patch is below 30mm at any time. The rear wheelset has a very similar behaviour, delayed by the wheelbase with respect to the leading wheelset and with some differences in the wear depth level: because of the soft longitudinal suspension, the angle of attack of the rear wheelset is close to the front one, leading to similar behaviours; it lacks further interest, so a graphic representation

is not included.

Also in this case, field side contact points never reach tread end, but a minimum of -40mm when running through the nose Figure 16. That is the reason for the limitation of the lateral spread of contact points in Figure 14. Worn wheels have a much more interesting behaviour in this aspect, with the wheel running on the edge of the tread.

Worn wheels

As it was noted in Figure 15, worn wheels have particularly heavy wear at the field side, including the very end of the tread. Figure 19 depicts the wear results for the same conditions as the previous one, but with worn profiles for 51.4 kkm run.

Figure 19: Wheel wear depth at the wheel developed on the horizontal plane. Isolines depicted each $0.5e-10$ m. Reference switch, curved section; vehicle with worn S1002 wheel profiles.

In this case, both the transitions in the blade and the nose sections have much higher lateral displacements in the contact points, and more multi-point contact conditions. Extremely heavy wear can be detected between $y=-0.04$ m and $y=-0.06$ m for both transitions, and in the case of the blade, the flange is also heavily worn (Figure 20). Again, as in the previous case, wear is highly discontinuous, meaning that it is jumping from mild to severe wear zones in Archard's wear map.

Figure 20: Wheel wear depth at the wheel developed on the horizontal plane, blade section close-up. Reference switch, curved section; vehicle with worn S1002 wheel profiles.

Focusing on the nose (Figure 21), the behaviour is broadly similar to the unworn profiles. When the wing rail starts to diverge, the contact point on the wheel rapidly moves towards the field side; from this point, some differences arise. First, the wheel is running sitting on the tip of the tread end for more than one meter, with enormous wear depths at these points. Second, the contact point lateral displacements are abrupt, meaning that at these points wheel and rail profiles are quite conformal. And third, flange contact with the nose is delayed: due to the geometry of wheels and rails, there is wing rail running for a longer time until the nose makes contact with the wheel.

Figure 21: Wheel wear depth at the wheel developed on the horizontal plane. Reference switch, curved section; vehicle with worn S1002 wheel profiles.

Different switch conditions

These results were obtained for specific switch geometry, running through the curve. It might be the case that for other conditions, *i.e.* running through the switch in tangent track or running through worn S&C, the wear level might be different, so in order to account for different variables that might affect the result, the following simulations have been carried out:

- Straight track
- Reduction of track stiffness of the switch blade
- Worn nose
- Tare vehicle

First of all, the vehicle running through the straight track path of the switch is considered. Switch geometry does not change, except for the fact that there is no curve between the blade section and the crossing section. This generates much more favourable contact conditions, as flange contact does not occur and two point contacts have very similar rolling radius, so sliding speeds at the contact patch are lower than the curve case. Wear volume for the curve case is $21.83 \cdot 10^{-10} \text{ m}^3$, while for the tangent track case is $1.79 \cdot 10^{-10} \text{ m}^3$, one order of magnitude lower.

Secondly, track stiffness for the blade should be much lower than for the stock rail. The first one is a moving part of the switch mechanism, while the second one is permanently attached to the sleepers. According to Nicklisch *et al* [17] the vertical stiffness in the tip of the blade is up to 70% lower than at the end of the switch rail, depending on the maintenance of the system.

In order to study this difference in a MBS software it is not straightforward to differentiate between these two rails and including different track stiffness. Therefore, in order to somehow account for this vertical stiffness and determine if there is any influence at all in the developed wear, a simplification has been applied: the contact stiffness between wheel and rail has been reduced for the second contact point. Vertical stiffness of the switch rail is $1200 \cdot 10^6 \text{ N/m}$, whereas the stiffness of the stock rail is $2400 \cdot 10^6 \text{ N/m}$.

The results are very similar to the initial case, except that there is some wear transference between flange and tread: being the switch rail softer, most of the vertical load will be supported by the stock rail, also reducing the load on the switch. The overall wear is extremely similar to the initial case, but the share of wear located in the outer part of the flange is increased, while flange wear is decreased.

Third, the switch that has been considered has an ideal theoretical geometry. However, the nose of the switch is one of the most damaged components in railways, so it can be assumed that its geometry will not stay the same for long time. In order to account for a worn nose, the studies of Wies *et al* [2] have been used as a reference, because there were not experimental measurements of a worn switch available for this study. According to them, a difference between 2mm and 3mm is expected after the nose starts to wear out and deform. Nose profiles have been modified in order to account for this deformation.

The main result is that, as the encounter of the wheel and the nose occurs later, the contact with the wing rail is maintained for a little longer time, getting a contact patch further out to the field side and generating larger sliding speeds when the nose appears. The larger gap creates larger lateral displacements of the contact patch and higher wear values close to the nose. As a result, wear appears in a wider region on the wheel surface. However, total wear volume is increased only 2%, from $2.183 \cdot 10^{-9} \text{ m}^3$ on the original case to $2.227 \cdot 10^{-9} \text{ m}^3$ on the worn nose case.

Finally, Figure 22 depicts the difference between the laden (22.5 tons per axle) and tare case (6 tons per axle) for a curve switch run. Cumulated wear volume for laden vehicles is $3.707 \cdot 10^{-10} \text{ m}^3$, while its $21.83 \cdot 10^{-10} \text{ m}^3$ for the laden case. Considering the wear volume per axle ton and one switch ride (normalized volume $\bar{V}$, Equations 9 and 10) it is obvious that the relationship is not linear: for higher loads, the wear is more severe. This is why freight vehicles are especially critical compared to passenger vehicles when analysing wheel wear.

Figure 22: Cumulated wheel wear at the wheel perimeter. Reference switch, curved section; vehicle with s1002 wheel profiles. Laden (—) and tare (---).

$$\bar{V}_{laden} = 0.970 \cdot 10^{-10} \text{ m}^3/\text{ton} \quad (9)$$

$$\bar{V}_{tare} = 0.618 \cdot 10^{-10} \text{ m}^3/\text{ton} \quad (10)$$

Figure 22 also shows another issue with the different plots that have been used in this work. Each plot has a deficiency in showing the whole magnitude of the problem. In this case, wear is summed up for each of the wheel's angular positions, and plotted along the wheel perimeter. In this case the cumulated wear in a specific angular position of the wheel might be a deceiving parameter, as it might occur that at one specific angular position there is *e.g.* flange wear generated at the switch blade and also tread end wear generated at the crossing nose. Summing up these two might give the incorrect impression that there is a wear peak at some point. However, it is very useful for comparing the cumulated damage at the wheel for the different cases.

Figure 23 compares the wear for the reference case (vehicle running through the curved section of the switch with new S1002 wheel profiles) to all the different cases that have been analysed: worn wheel profiles, worn crossing nose and reduced blade stiffness. First, worn wheel profiles have c.a. 15% higher wear rate than any other analysed condition, $2.547 \cdot 10^{-9} \text{ m}^3$. Second, the wear is not uniform along the perimeter, and there are sections where flat-like shapes appear. However, the maximum value is 1/200 the depth of a typical 20mm wide wheel flat, so it will hardly behave like one. Also, previous figures have shown that the heavy wear is concentrated in both tread-end and flange, so flat-like surfaces in these areas will not be a problem for the vertical dynamic behaviour of the vehicles.

Figure 23: Angular position of wear for the front left wheel after running through the diverging part of the turnout in different conditions.

Figure 24 depicts the results for the tangent track section in the turnout. Overall damage is much smaller, but worn wheels are still the condition that most induces severe wear, which is one order of magnitude higher than for the rest of the cases. So in case the contact geometry is favourable, wear is ten times lower than for more conformal profile shapes, at a level which is comparable to the curve switch case. The maximum wear peak for the worn wheels case is $9.04 \cdot 10^{-9} \text{ m}^3/\text{m}$, Figure 24 focuses below $2 \cdot 10^{-9} \text{ m}^3/\text{m}$ for better detail on all the cases.

Figure 24: Angular position of wear for the front left wheel after running through the straight part of the turnout in different conditions.

A special characteristic of straight track runs in switches is that there is no limitation in the speed, compared to the curve run. Thus, simulations have been carried out with different fictitious speeds in order to understand its behaviour. Figure 25 depicts the cumulated wear on the wheel for an extremely high speed, $V=160 \text{ km/h}$. For this fictitious speed, the worn off volume is roughly half the volume of the curve run with 50 km/h , pointing out the relative importance of the straight track run with no speed limitation.

Figure 25: Angular position of wear for the front left wheel after running through the straight part of the turnout in different conditions, $V=160 \text{ km/h}$.

4. CONCLUSIONS

It is possible to calculate uniform wear of the wheel on freight vehicles, but specific components of the freight traffic system must be considered as they might have a significant influence on wheel profile evolution.

In this study, it has been demonstrated that the wear generated by running over switches and crossings with the high axle-loads of freight vehicles cannot be disregarded; actually, it is the predominant cause for wear, one order of magnitude higher than the wear generated by running through tangent track and curves of the network.

Regarding wear calculation methodologies, wear estimation with the $T\gamma$ model is very convenient, as it is only

applying some algebraic operations to some output variables. However, it has a limited usefulness regarding uncommon contact conditions such as switches: it is able to predict high wear levels at the top of the flange, but not at the tread end. This is due to the fact that it only considers one variable ($T\gamma$) and not both of them separately.

Archard's wear law, on the other hand, was able to predict high wear at the thread end of the wheel as expected from the measurements, even if its implementation in a wear prediction code is not so straightforward and wear calculations take much longer. This is because, in order to account for the wear, Archard's model uses two variables which must be evaluated in the wheel rail contact, which needs a discretization of the contact patch that $T\gamma$ does not need to perform. Thus, Archard's wear prediction methodology seems much more useful when predicting wear for extreme conditions. It should be taken into account that the experimental wear map has not been obtained for the specific materials in the switch, which might introduce further uncertainties in the prediction. In this specific case Archard's model did not detect high wear for the original S1002 profile, but that can be explained by the dynamic simulations, as the contact points never reach the tread end.

To address the different conditions in a switch, several parameters were considered: straight track run, reduced stiffness for the switch blade and worn nose geometry were simulated. The results show that wear for these different conditions is rather uniform, except for one: the most dangerous case is having a wheel profile which is conformal to the geometry of the switch, either in curved or straight run. In that case the contact conditions lead to extremely heavy wear high at the flange and at the field side of the tread.

Other operational cases with vehicles different to the studied one might include substantial block brake usage. In these cases the brake energy dissipation is in the tread area, so its influence on wheel wear must also be carefully studied to account for hollow wear. In the studied case there is no heavy use of block brakes, so wear in the tread area is correctly predicted.

The main conclusion is that the variable that most influences wheel wear at switches is the degree of conformity between the wheel profile and any of the critical sections of the switch (switch blade or crossing nose). In this case, high contact forces and sliding speeds will appear in these switch sections, generating more wear. This effect is specially critical when the proportion of wear produced by switches is much higher than the one caused by the network or block brakes, as they will evolve into even more conformal profiles.

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