Influence of model parameters on the predicted rail profile wear distribution in a curve

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Abstract: A sensitivity study is performed to determine the effect of model parameters on the predicted wear distribution on rail profiles. The numerical framework uses multibody simulations to evaluate dynamic vehicle-track interaction for different traffic scenarios, and FASTSIM and Archard's wear law to evaluate rail profile wear. Vehicle and track parameters as well as different approaches to contact modeling are considered. A previous study has shown good agreement between predictions and measurements in overall wear, but deviations in wear distribution. The objective of this study is to obtain a better agreement for the profile wear distribution under operational traffic conditions compared to rail profile measurements. The study shows that the inclusion of freight vehicles, different wheel profile samples, and the consideration of a different contact model are some of the most important parameters.

1. Introduction and background

A model sensitivity study is performed to identify the most influential parameters that determine the local wear distribution on rail profiles in a curved track. The study is an extension of the work performed in [1] and assumes familiarity with the simulation methodology presented there. The objective of the study is to determine which parameters are most important when creating a set of simulations intended to represent the traffic situation in a curve and to achieve better agreement with the damage evolution of measured rail profiles. Figure 1a shows the final results of the simulated and measured profile change in the normal direction to the rail profile from [1]. Both the simulation and the measurement are for a curve radius of 1974 m and a traffic load of 10 Million Gross Tonnes (MGT). The profile change accounts for wear and plastic deformation, and it can be seen how the local damage differs between simulation and measurement, especially at the flange contact. Figure 1b shows five profile measurements along the track.

As discussed in [1], the most likely reasons for the discrepancies between measurement and simulation are: a) limitations in representing the traffic load and track compared to actual field conditions, and b) the choice of contact model. However, the discrepancy between the simulation and field measurements could also be caused by measurement noise, since only 10 MGT of traffic had passed between the two measurement occasions. The measured profile changes are on the order of a few tenths of a millimeter and are, therefore, sensitive to measurement errors.

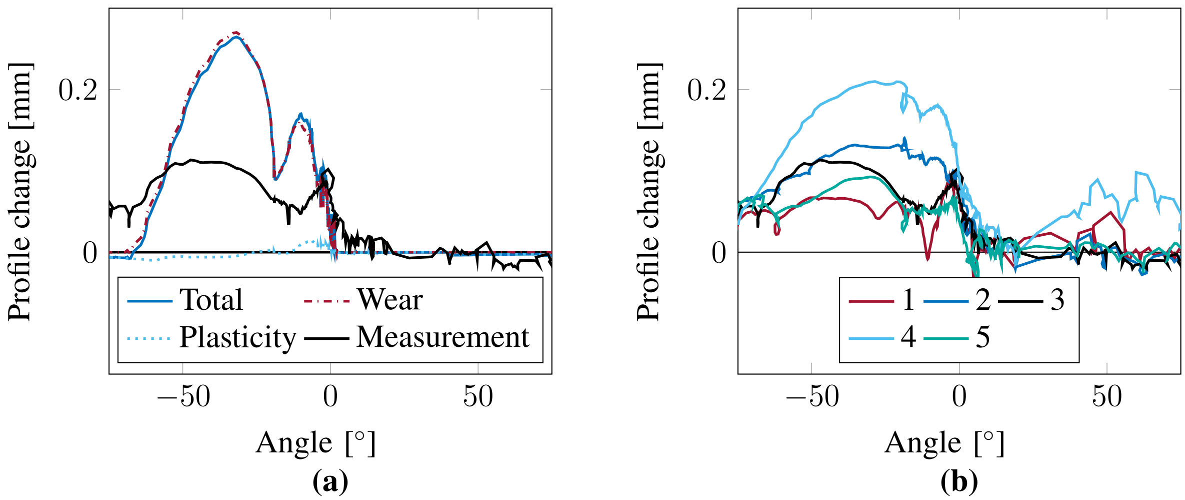

Figure 1: *Profile change in the normal direction of the rail profile relative to the reference cross-sections from the first measurement. The curve radius is 1974 m and 10 MGT of traffic ran on the tracks between the measurements. a) shows the profile change for both the measurement (cross-section three) and the simulation. b) shows the profile change for five different cross-sections along the rail.*

2. Evaluating model parameters

For each parameter configuration described in [1], a load sequence of traffic is simulated using the commercial multibody simulation software Simpack [2]. The load sequence can be summarized as an effort to capture different vehicle types described by different axle loads, different amounts of wagons, and different variations of vehicle speed depending on the vehicle type, which may vary along the track. In total, the load sequence consists of 20 vehicles, with each vehicle having multiple wagons as listed in Table 2 in [1].

The load sequence used in [1] only includes different types of passenger vehicles. These vehicles are represented in the simulation by utilizing the Manchester Benchmark model [3], with primary spring stiffness values of 26 MN /m and 32 MN /m in the longitudinal and lateral directions, respectively. Each wagon in the sequence has a different worn wheel profile measured from a Bombardier Regina passenger train from [4]. The simulation assumes a standard Swedish 1:30 rail inclination and a standard track gauge of 1435 mm, with no cant or track irregularities considered. To solve the normal wheel-rail contact, an equivalent elastic contact model based on Hertz's contact theory [5] is used in the Simpack multibody simulation software. For the tangential contact, Kalker's FASTSIM algorithm [6] is used.

To study the damage distribution on a rail cross-section, only wear is considered, whereas in [1] where both wear and plastic deformation was simulated. Since wear is the dominant damage mechanism for the curve where the field measurements were taken, plasticity can be excluded to save simulation effort. Distributed wear is computed using Archard's wear law [7] in conjunction with a FASTSIM [6] discretization of the contact area. Table 1 outlines the changes made to the nominal load sequence considered in the parameter study. These include increasing the lateral track plane acceleration for X2000 [8], introducing a calculated equilibrium cant based on [9], doubling the primary spring stiffness values of the vehicles in the longitudinal and lateral directions, and replacing 10 % of the traffic with freight. To assess the impact of the contact model on the damage distribution, the contact formulation in Simpack can be changed to a discrete elastic contact using the contact modeling approach STRIPES [10, 11] which uses a modified FASTSIM algorithm [6] for the tangential contact. An ideal elastic-plastic material can be used for this contact formulation [12, 13]. In addition, the Hertzian-inspired metamodel is used in conjunction with the equivalent elastic contact model and FASTSIM to compute the distributed wear, using the contact pressure and sliding distance from Simpack and the contact area from the Hertzian-inspired metamodel.

Table 1: *Changed model parameters between the original setup and modified one to study the most influential parameters.*

Number	Model Parameter	Original setup in [1]	Updated setup
1	Lateral track plane acceleration for X2000	1 m/s^2	1.6 m/s^2
2	Equilibrium cant	0 mm	117 mm
3	Primary longitudinal and lateral stiffness	[26,32] MN/m	[52,64] MN/m
4	Vehicle types	Passenger	Passenger + 10 % freight
5	Measured wheel profiles	Bombardier Regina passenger train profiles from [4]	Y25 bogie profiles from [14]
6	Contact model	Equivalent elastic	Discrete-elastic
7	Contact model	Equivalent elastic	Discrete-elastic-plastic
8	Contact model	Equivalent elastic	Equivalent elastic + metamodel

An increased lateral track-plane acceleration for X2000 is considered since in [1], the same lateral acceleration of 1 m/s^2 was used for X2000 and all other passenger vehicles. However, since X2000 is a tilting train it is according to Banverket BVF 586.41 allowed to have a lateral acceleration up to 1.6 m/s^2 [8]. This should result in increased lateral contact forces.

The contact positions and creep forces on the rail depend to a large extent on how the wheelset is positioned on the track. If the longitudinal and lateral primary spring stiffness in the bogies are increased (doubled in this case), the angle of attack of the leading wheelsets in each bogie should increase [9]. This should result in more gauge contact for the load sequence and higher tangential forces.

The simulations in [1] did not include a cant, which is typically used in curved tracks to allow for higher speeds while limiting lateral track plane acceleration. Therefore, the presence of cant affects the velocity profile of the vehicles. According to Andersson et al. [9], the equilibrium cant h_q can be calculated based on the given speed v, curve radius R, the running tread position $b_0 = 0.75$ m, and the constant of gravity g as

$$h_\mathrm{q} = \frac{2b_0}{g}\frac{v^2}{R}. \tag{1}$$

For $R = 1974$ m and a nominal speed of $v = 140$ km/h, the equilibrium cant in the curve becomes $h_\mathrm{q} = 117$ mm. The lateral acceleration at the chosen cross-section is reduced to 0 m/s^2 when a cant is utilized, as opposed to 0.7 m/s^2 when no cant is used. The velocity profile is not affected by the cant because the limiting factor for the considered track section is the regulation on longitudinal acceleration rather than the maximum lateral track plane acceleration. However, since the aim of the simulation is to find model parameters that result in more flange contact with a larger distribution, utilizing a cant for this track setup will have the opposite effect of what we want to achieve. Despite that, it is important to investigate its effects since the absence of a cant does not accurately represent operational conditions for curved tracks.

Considering the model parameters from 1-3 listed in Table 1, and the original setup of the load sequence used in [1], a scaled wear distribution for one load sequence can be seen in Figure 2. It should be noted that the rail geometry is not updated in this simulation, which is why the wear affects a wider region in Figure 1 compared to the present results. From Figure 2, there is no evident change in the distribution using the updated model parameters compared to the original setup, but rather the magnitude of the wear. An increased spring stiffness increases the damage at the flange contact, whereas utilizing a cant and increasing the lateral acceleration for X2000 results in less wear compared to the original setup.

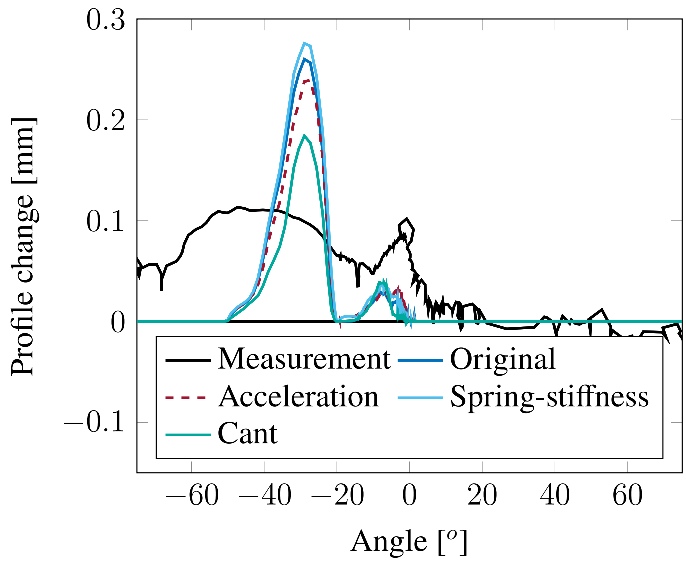

Figure 2: *Profile change comparing the measurement to the wear for one simulated load sequence using different model parameters.*

The traffic at the measurement site is a mixture of both 90 % passenger and 10 % freight vehicles, but the load sequence in [1] did not include any freight vehicles in the simulations. To investigate if the presence of freight vehicles can impact the results, a load sequence is created where 10 % of the load sequence is exchanged to freight vehicles. This load sequence is modeled in two different ways: (1) by representing a freight vehicle with a passenger vehicle using the same speed and axle load as the freight vehicle and (2) by utilizing the Manchester Benchmark Freight Vehicle [3] specifically designed for freight trains. Figure 3 shows that including freight vehicles does have an impact on the wear distribution, with the running gear of the vehicle being the primary influencing factor rather than the increased axle load which has a similar damage distribution to that of the passenger vehicles.

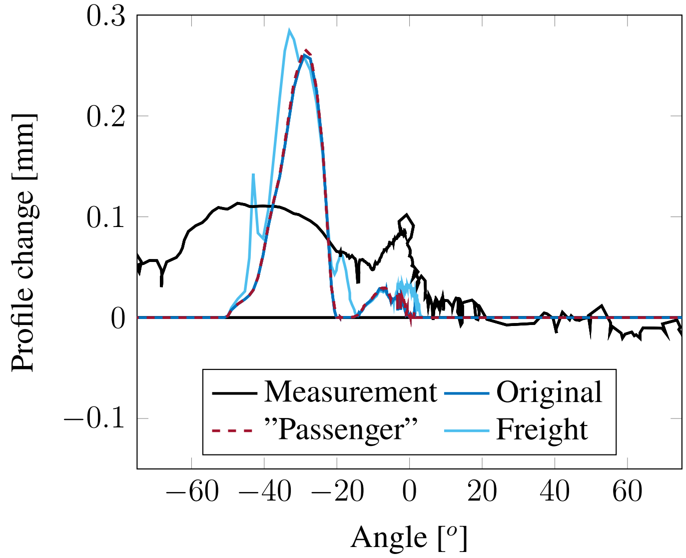

Figure 3: *Profile change comparing the measurement to the wear for one load sequence using different vehicle types.*

The location of contact points on the rail is influenced by the wheel profiles of the vehicles. Thus, a different set of measured wheel profiles from Y25 bogies used in freight vehicles from [14] are incorporated into the load sequence to assess their impact on the contact situation. As shown in Figure 4, the use of measured Y25 bogie wheel profiles results in more flange contact and a larger distribution of damage as compared to the original load sequence that used the measured Bombardier Regina passenger wheel profiles from [4].

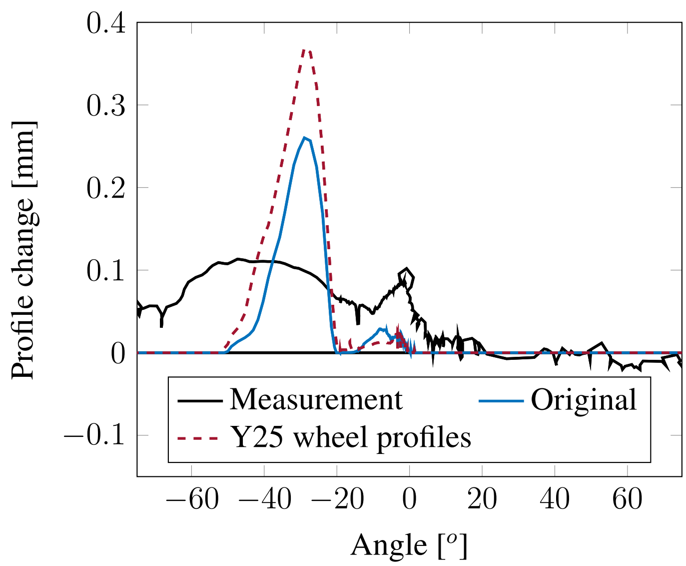

Figure 4: *Profile change comparing the measurement to the wear for one load sequence using different sets of measured wheel profiles. The original simulation uses measured Bombardier Regina passenger wheel profiles from [4].*

The accuracy of pressure magnitude and distribution, and contact area solved from a contact model can influence the distribution and magnitude of damage. In Simpack, two contact models are available: (1) the equivalent elastic contact method solving the normal contact, which is based on Hertz's contact theory [5], and FASTSIM algorithm for solving tangential contact, and (2) the discrete elastic contact method based on the semi-Hertzian approach called STRIPES [10, 11]. The latter method considers tangential contact using a modified FASTSIM algorithm and allows the determination of contact forces at each discrete strip. This method is preferred when the half-space assumption is violated due to flange contact of the wheel or a varying curvature of the bodies in contact.

The discrete elastic contact method offers the option of modeling ideal elastic-plastic material behavior of the rail [12, 13]. This results in reduced contact stresses as the contact area changes due to plastic flow. However, if the stresses are truncated at the yield limit, no accumulated plasticity would occur in the simulation. Another way to account for the effects of plasticity on the contact is to use a Hertzian-inspired metamodel [1, 15] to solve the normal wheel-rail contact, which considers the elastic-plastic material of the rail using the cyclic plasticity material model Ohno-Wang [16]. The output of the metamodel provides information on the size of the contact area while the normal pressure and sliding distance are given from Simpack using contact model 1.

Figure 5 presents that the magnitude of damage changes when using the discrete elastic contact method compared to the equivalent elastic contact method, but there is not much difference in the distribution. When considering an ideal elastic-plastic material in the discrete elastic contact model, the damage distribution becomes wider. The same effect is observed when using the contact area output from the metamodel but to a lesser extent due to the different material model accounting for the hardening of the rail material. The wider damage distribution is caused by an increased contact area when plasticity is considered. Despite achieving better agreement with the measurement, the simulations and the measured profile changes did not show a significant amount of plasticity.

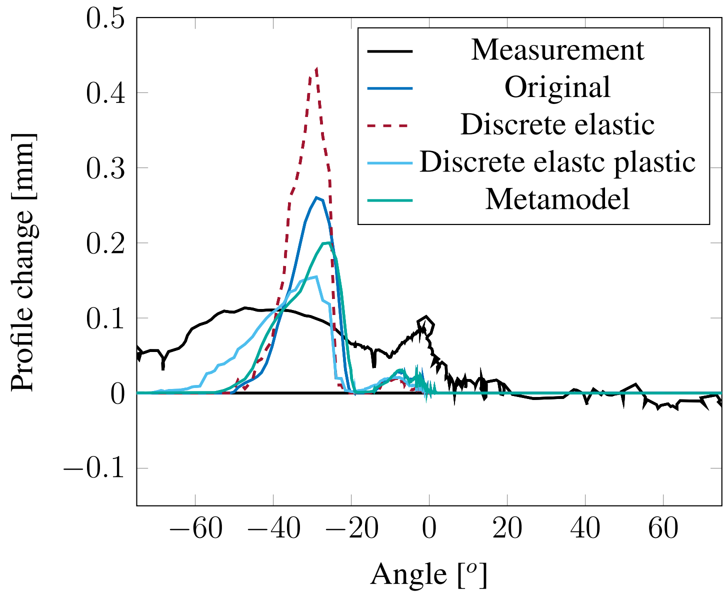

Figure 5: *Profile change for one loading sequence for different model parameters. The result displays the profile change for the measurement and different contact models.*

3. Conclusions and outlook

The aim of the parameter study was to identify the most influential model parameters causing discrepancies between field measurements and simulated results of long-term rail evolution in curved tracks under operational traffic. The study found that all tested parameters have some influence on the profile wear. In some cases, the change only influenced the wear magnitude. However, to improve agreement with measurements, a wider wear distribution is needed. Promising results are obtained by adding freight vehicles to the load sequence, changing wheel profile samples, and using a different contact model.

To further investigate, additional parameters such as (1) track irregularities, (2) other sets of measured wheel profiles, and (3) the inclusion of the rail roll degree of freedom could be considered. The inclusion of track irregularities could explain the variation in profile change along the track (cf. Figure 1b). Other sets of measured wheel profiles could also be explored as they were found to have an influence on this study. Finally, roll rotation of the rail head due to the wheel-rail contact loads could also impact profile change as it can alter the contact point locations.

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