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Influence of Vehicle Dynamics on Wear of Railway Wheel Profiles

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Abstract

The objective this work is to understand the effect of the vehicle dynamic on the wear generation on wheel and rail profiles, using a mathematical model to study the evolution of railway wheels profile due to wear. The dynamics of a two axle vehicle running on a realistic track was simulated using a Multibody code (Simpack), the contact points are located for each time step using a geometrical module developed in MATLAB, and considering equivalent Hertzian contact patches. The tangential forces are calculated using the Kalker's simplified theory (FASTSIM). The wear coefficients are calculated from the work of Bolton and Clayton. The contact area is divided according to the strip theory into finite elements. Hence, considering dry friction, the wear rate is calculated for each element in the presence of the slip by the proportionality law of wear and frictional work done and, accordingly the accumulative wear for the entire contact area can be obtained.

A short real track is adopted as a reference and the wear amount on the wheel profiles is obtained by cumulating the wear. Since the wear of the wheel is due to the wheel-rail interaction, evaluated on the reference track using a MBS code, it is important to know how the wear of the wheel affects the vehicle dynamic and how dynamic affects the wear respectively.

To investigate this problem, we use in this work a simple vehicle model (bi-axle vehicle) on a reference track. An overall track length of 13.000 Km is considered comparing different simulations. At first we consider as wear of profiles would not affect dynamics: in this case the wheel dynamic is evaluated once at the beginning using the MBS code.

Results obtained in this case are then compared with those obtained repeating the dynamic calculation every 1.300 Km and 260 Km.

Keywords: Wheel-rail wear; wheel-rail contact; vehicle dynamics; multibody simulation.

INTRODUCTION

In railway engineering, wear of wheel and rail profiles has become an important issue. With current tendency towards increasing axle loads and higher speeds, limitation of wear has become more important in vehicle design. Many researchers have worked in this field, either experimentally or numerically, in order to understand the mechanism of this phenomenon and, to understand the problems related to the presence of various types of wear, which are due to the various deterioration mechanisms.

Material loss through wear has been studied using three main categories: field measurements [1,2], laboratory research [3-8] and, theoretical wear models [9-11]. The numerical models were allowed to become more detailed and complex as the computers become more powerful. More attention was paid to finding a relationship between contact conditions and wear rate using different concepts. A particular feature of this kind of simulation is the difference in the time scales between the dynamic problem and wear accumulation. Some researchers have defined a mathematical relation between the two "time-scales", while others have adopted the concept of "wear step" that is used in the present work, whereas the modeling procedure is based on calculating wear depth, wheel profile shapes and, dynamic simulation of the vehicle-track interaction.

Since there are many parameters that have a direct effect on wear, different aspects require dedicated studies, one of which is vehicle dynamics, the aim of the present work.

It is well known that vehicle dynamic is investigated using Multibody codes, and those codes now also implement wear algorithms. Usually the wear of the profile is calculated considering a short track and multiplying the wear amount by a parameter (here indicated as wear multiplier) allowing to reproduce the desired mileage. If the adopted multiplier is too high, the effect on the shape of the profile may result unstable, but this depend also on the strategy adopted to augment the wear in function of the kinematical parameters, or in the adoption of particular smoothing algorithms. These aspects have been already investigated by the authors in [12]. Furthermore must be considered that when a vehicle runs with worn profiles, the dynamic forces exchanged with the track get higher and this can significantly modify the wear of the profiles. This aspect will be further investigated in this work also considering the interaction with the use o a wear multiplier.

WHEEL/RAIL CONTACT

In the current work, the contact model used for wear determination is a geometrically two-dimensional model and was developed in Matlab [12], while in order to estimate dynamic response of the vehicle, the contact problem is solved using Simpack MBS, as shown in the flow chart of figure 1.

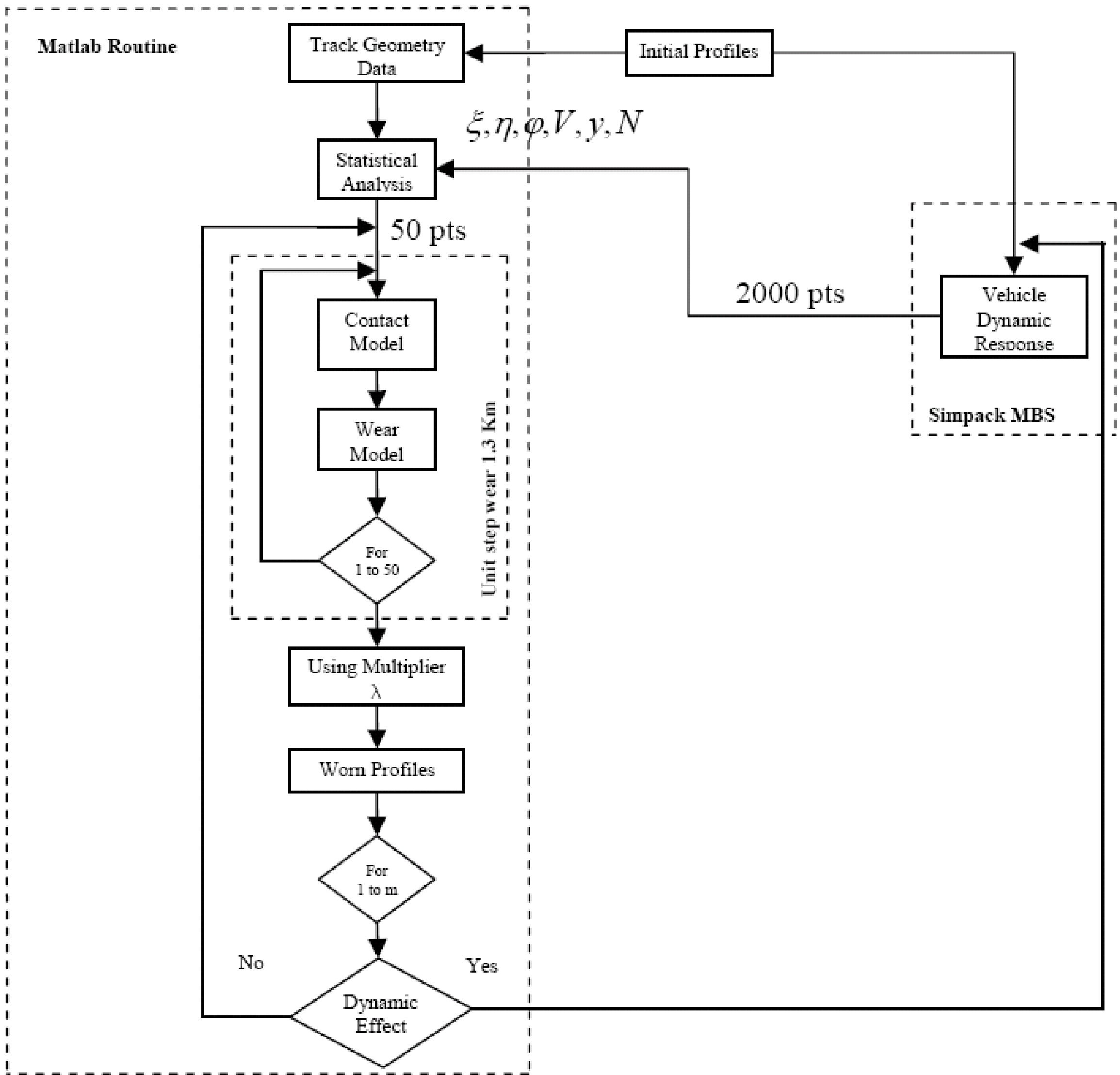

Figure 1. Wear prediction and vehicle dynamic response flow chart

Hence, the Matlab contact model has the following restrictions:

- Single point Hertzian contact.
- Rigid contact.

Therefore, when computing vehicle dynamic response in Simpack MBS, it is important to consider the single rigid contact in order to confirm the consistency in solving the problem. This has been verified in a previous work [13].

The wheel/rail profiles adopted for this work are the most common in Europe and in particular S1002 for the wheel and UIC60 for the rail respectively. Moreover, in the Matlab routine the normal and tangential problems are solved with Hertz's theory and Kalker's simplified theory (FASTSIM) respectively and, using Kalker's creepage coefficients (Cij), which are calculated using a look-up table. The Fastsim algorithm has been appositely modified in order to calculate the frictional work done and the corresponding wear rate for a number of wear steps.

WEAR CALCULATION

Generally, wear can be simulated according to two main approaches:

1. The energy dissipation method, which is a single parameter model [1, 14]. The wear is directly proportional to the work

done by the friction forces in the contact zone. Different proportionality factors can be used for different wear regimes.

2. Archard's wear model, which is a two parameter model [7, 8, 15]. The wear is directly proportional to the normal force times the sliding distance and, inversely proportional to the hardness of the material. The proportionality factor depends on the "wear maps" expressed in terms of contact pressure and sliding velocity [7, 12].

The energy dissipation model is used in the current study to predict worn wheel profiles. The following limitations are considered in modeling:

- Steady state rail profile;
- No abrasive wear and no lubrication present;
- Constant friction coefficient ($\mu = 0.36$);
- No plastic deformation;
- No braking;
- Only the wheel is subject to the wear process.

The wear model is built based on the strip theory, i.e. by dividing the contact patch into finite elements. Since the contact area consists of slip and adhesive regions, the most dangerous form of wear (adhesive wear [1]) will take place in the slip region. Hence, wear rate ($mg \times mm^{-2} \times m^{-1}$ rolled) is determined according to equation (1), which is based on the concept of energy dissipation. This wear rate calculation is done for each element in the slip region and the sum of wear depths for each strip in the rolling direction is calculated.

$$wear\ rate = K_1 \cdot (T\gamma / A) + K_2 \cdot (T\gamma / A)^2 \qquad (1)$$

As illustrated in the flow chart of fig. (1), the resulting dynamic response data obtained from Simpack MBS simulation for a certain track length (1.3 Km selected in this study) consists of 2000 points. This data will be used by the Matlab routine in order to calculate the wheel wear depth, but since the real time calculation is a time consuming operation, a statistical approach was developed to reduce it [12]. Alternatively the method could be applied considering a Multi-Core approach, that is possible using the proposed contact algorithm [16]. Briefly, a statistical class of lateral creepage was chosen as a reference class and all the other parameters were classified according to it; the median value for each class was considered, resulting in reduction of the data to a lesser number of points (50 points in this work). Then the accumulative wear depths obtained at the end of the wear step (1.3 Km), as illustrated in the flow chart of fig. (1), are multiplied by a certain multiplier, λ ,(1000 and 200) to artificially create a longer running distance, 1300 Km and 260 Km respectively. The calculated worn wheel profiles are used as input profiles for the next iteration step. In order to consider the dynamic response, the worn wheel profiles are used as input in Simpack MBS code to calculate the new dynamic response data. On the other hand, ignoring the dynamic response, the data obtained from Simpack in the first step is used, unchanged, for the whole simulation, while, obviously, the wheel profiles are updated due to the wear effect. For both cases, this procedure is repeated "m" times until a total running distance is reached (13000 Km considered in the work).

SIMULATION ENVIRONMENTS

In order to evaluate the performances of the algorithm to estimate the wear of the wheel profile, we have considered a single vehicle running on a reference track. The same track has been repeated several times in order to achieve the desired mileage. The strategy to use a simple track and vehicle model has been used in order to avoid to introduce further elements and to be able to give simple interpretation to the obtained results. The same process can be used for arbitrary tracks and vehicles.

Track data

The track adopted for the simulations is a portion of 1.33 Km of a real Italian secondary line, as shown in table (1).

Table 1. Track data used in simulations

No.	**Type**	**Length** (m)	**Radius** R1 (m)	**Radius** R2 (m)	**Superelevation** U1 (mm)	**Superelevation** U2 (mm)
1	Straight	200			0	
2	Clothoid	90	0	-1040	0	-90
3	Curve	200	-1040		-90	
4	Clothoid	90	-1040	0	-90	0
5	Straight	175			0	
6	Clothoid	75	0	-500	0	-150
7	Curve	29	-500		-150	
8	Clothoid	75	-500	0	-150	0
9	Straight	49			0	
10	Clothoid	65	0	588	0	130
11	Curve	129	588		130	
12	Clothoid	65	588	0	130	0
13	Straight	117			0	

Vehicle data

The method is applied to a single wheelset of a Gs213 bi-axle freight vehicle with an axle load of 15 tons, and a wheel radius of 460 mm at the beginning of the simulation. The vehicle moves along the track at a constant speed of 40 m/s (maximum forward velocity of the vehicle). The inertial characteristics of the vehicle are shown in table 2.

Table 2. Freight vehicle inertial data

Part	**Mass** [kg]	**Ixx** [Kg. m2]	**Iyy** [Kg. m2]	**Izz** [Kg. m2]
Wheelset	1500	750	100	750
Vehicle body	27000	23676	158805	160960

According to UIC standard the inertial data of the Gs vehicle and, the suspension data (stiffness and damping) for each wheel are illustrated in table 3.

Table 3. Stiffness and damping adopted for the suspension of the vehicle

Stiffness			**Damping**		
Cx [KN/m]	**Cy** [KN/m]	**Cz** [KN/m]	**Dx** [KNs/m]	**Dy** [KNs/m]	**Dz** [KNs/m]
1000	250	500	10	5	25

SIMULATION PLAN

A total running distance of 13000 Km is selected as a reference in the simulation of a single wheelset, as described in the wear calculation section. Since in the Matlab routine a statistical approach is performed in order to reduce the computational time, two (200 and 1000) different multipliers (λ) were selected to compare the statistical effect on the solution of the problem. The other important feature to be considered in the simulations is the effect of the dynamic response on wear evolution. Therefore, the simulation plan can be summarized as shown in table 4.

Table 4. Simulation plan

Simulation	**Running distance** [Km]	**Multiplier** (λ)	**Dynamic response**
1	13000	200	No
2	13000	200	Yes
3	13000	1000	No
4	13000	1000	Yes

RESULTS

The simulations performed on the model have considered two different aspects: at first the worn profiles resulting from the simulations, performed using different methods have been compared.

Then, in order to understand the differences between various simulations, we have analysed the forces behaviour resulting by the use of different methods. Of course the two aspects are strictly connected each others.

Effect of vehicle dynamic response on wear evolution

As described before, the simulations were performed using different multipliers. Accordingly the wear evolution and, consequently, wear depths for both wheel profiles are shown for the different cases on figure 2 and 3.

It is evident that the effect of vehicle dynamic is very important both for the amount of material removed (which is higher considering the dynamic effect) and for the distribution of wear along the wheel profile.

Another important consideration is that when dynamic effect is neglected, using the statistical method proposed in [12], a multiplier equal to 1000 was enough to achieve good numerical results, in fact case 1 and 3 of table 4 give similar results. When considering the dynamic effect it is necessary to use a lower multiplier (200) and therefore more frequent re-evaluations of the wheel profile are required. If high multipliers are used, due to the dynamic of the vehicle, wear tends to concentrate in certain locations (in the simulation) and then to amplify the dynamic effect with further increase of wear in the same point. This phenomenon is not completely realistic and can be avoided using lower multipliers.

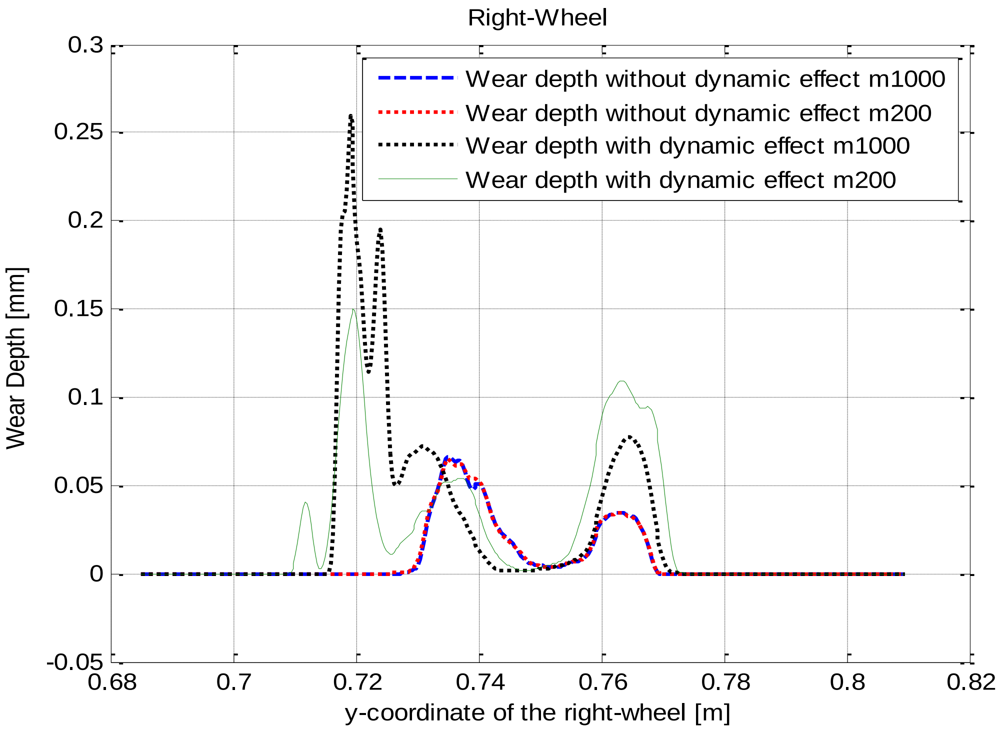

Figure 2. Wear depth comparison for the right wheel

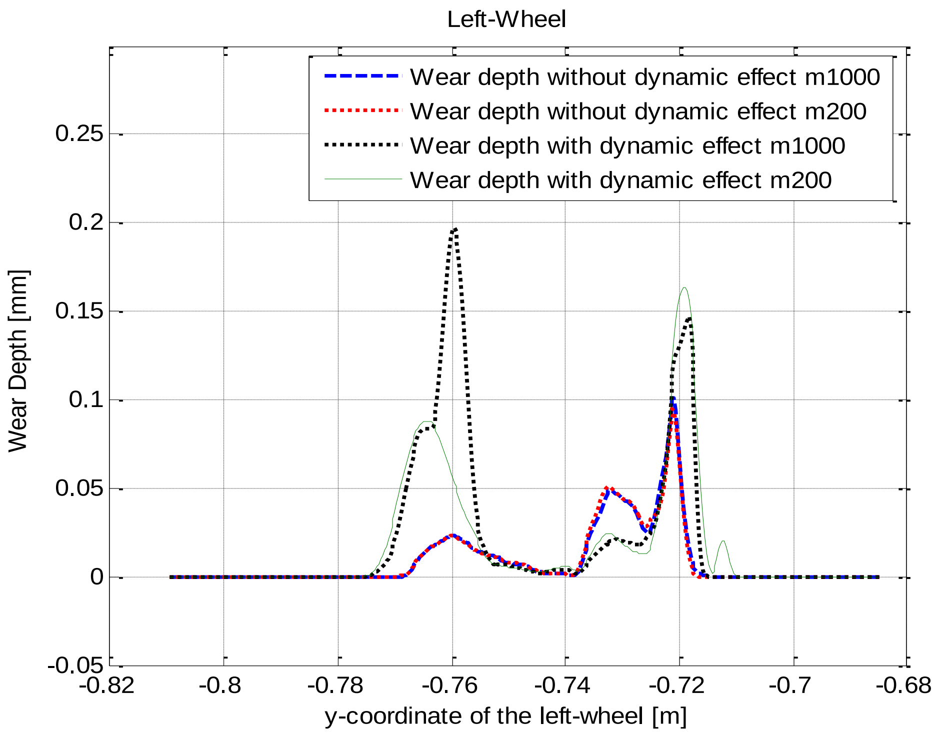

Figure 3. Wear depth comparison for the left wheel

Effect of worn wheel profiles on wheel/rail interaction forces and on vehicle dynamic response

The influence of dynamic effect can be evidenced considering the different lateral displacement of the wheelset (Figure 4) during a complete run on the track, using profiles worn by a 13000 Km mileage and considering different methods.

In the "delta" diagrams, graph D1 represents the difference between the results obtained using worn profiles without the dynamic effect and that of new profiles. Graph D2 represents the difference between the results considering the dynamic effect and that obtained on new profiles.

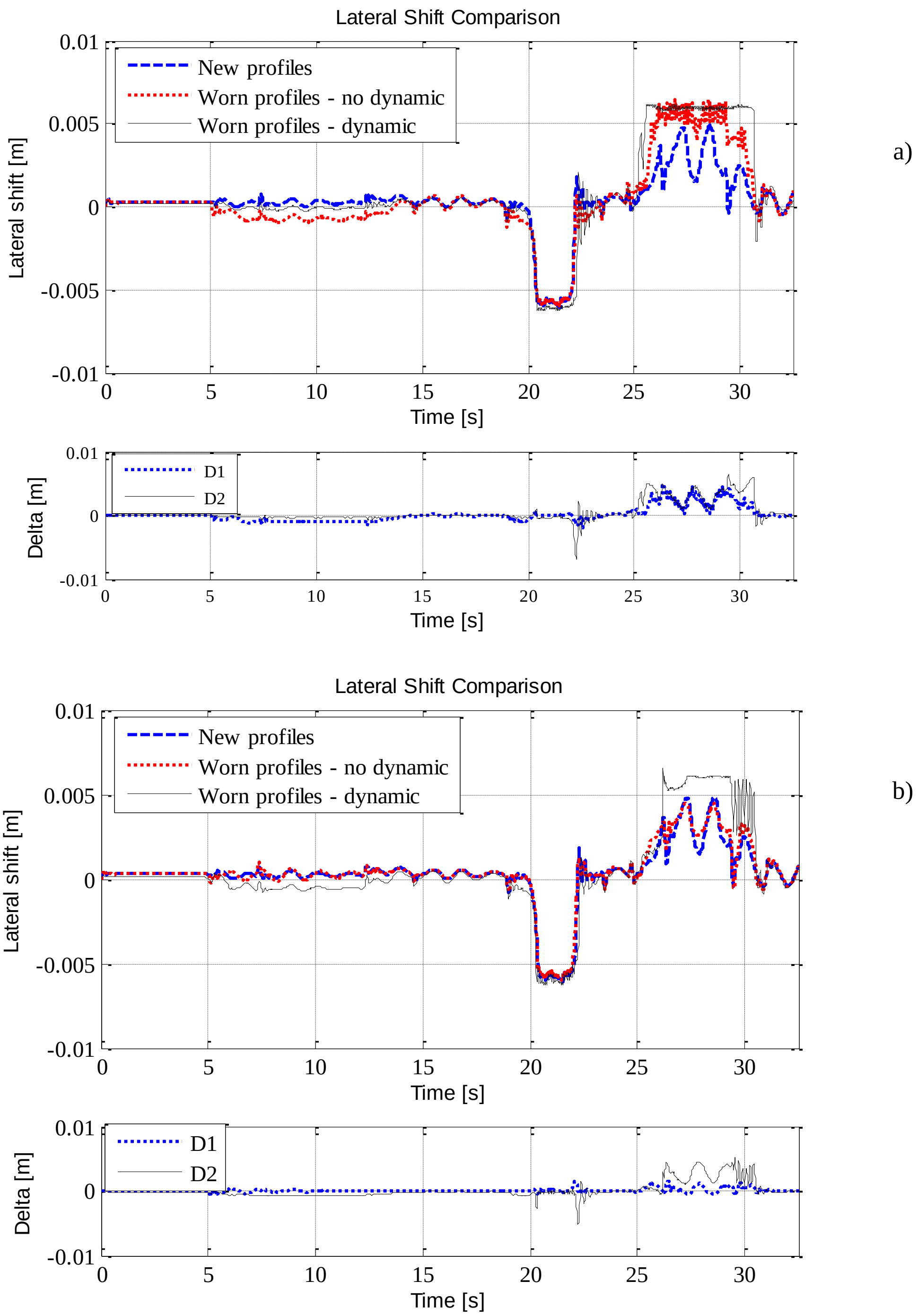

Figure 4. Lateral creepage (shift) comparison using multiplier equal to 1000 (a) and to 200 (b).

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The effect of dynamic is more evident when analysing the behaviours of the tangential forces acting between wheel and rail. For example analyzing the tangential forces in lateral direction (figure 5 and 6), it is evident that dynamic effect gives higher forces for the guiding wheel during curving, and that neglecting the dynamic effect, the worn profile behaves in the same way as the new one (figure 6).

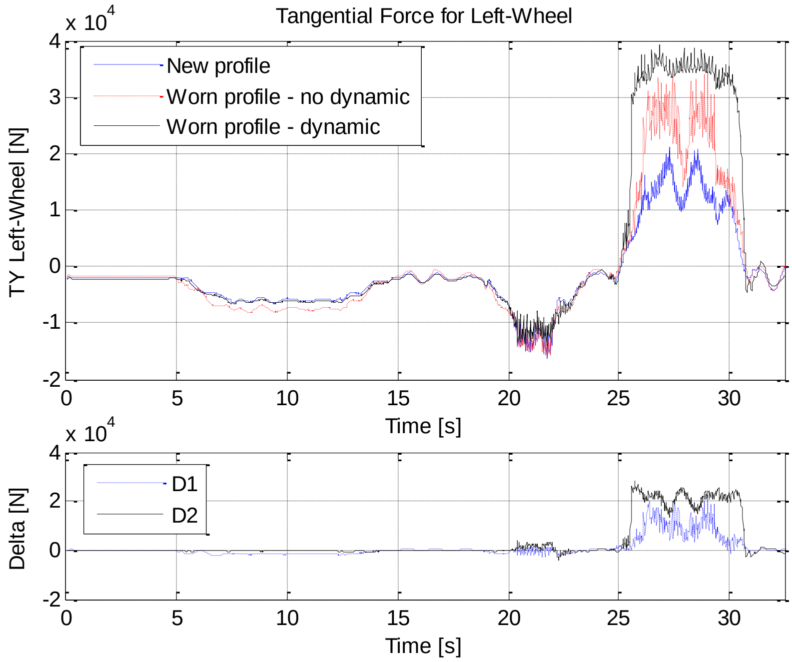


Figure 5. Variation of tangential force in y-direction for right wheel, using a multiplier equal to 1000.

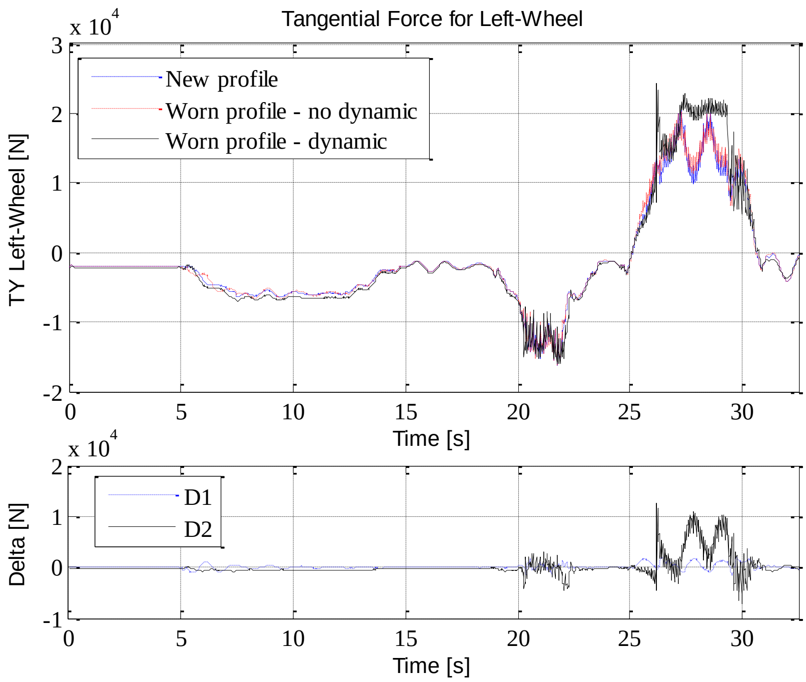
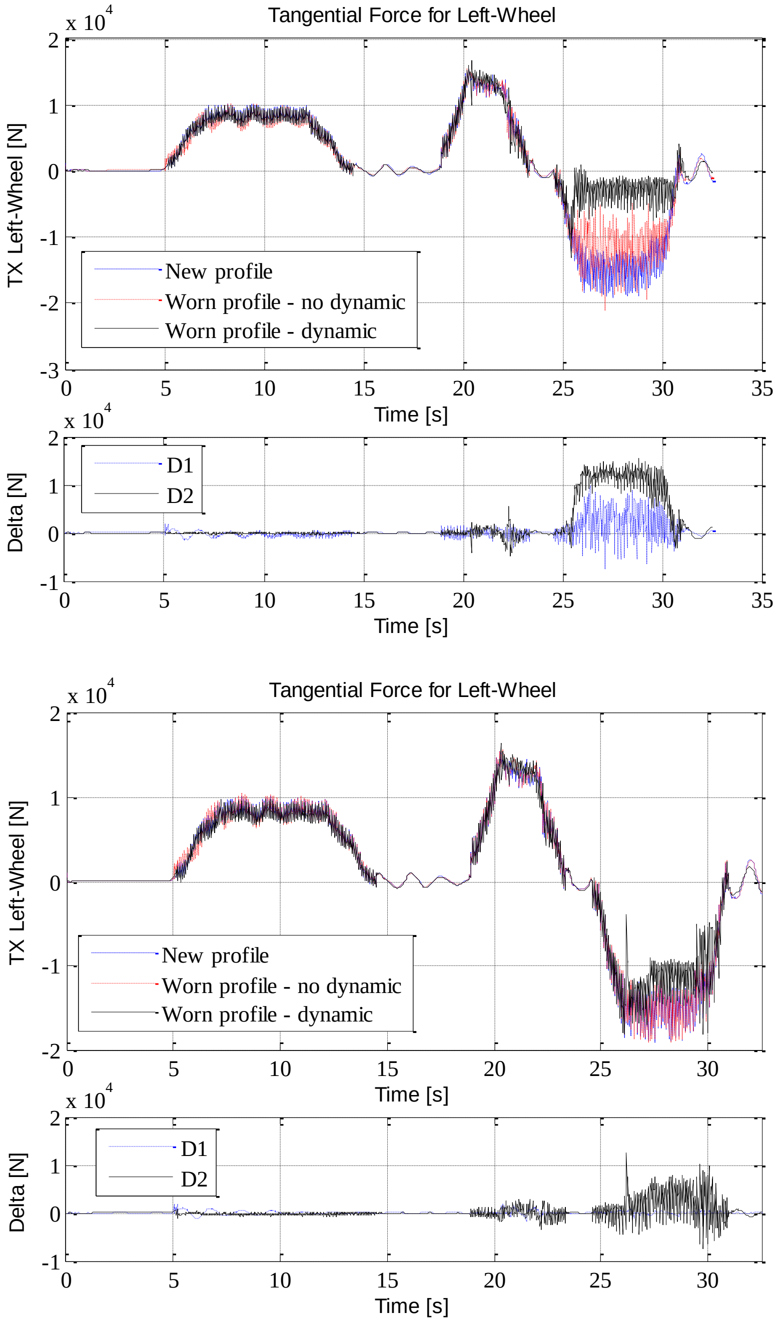


Figure 6. Variation of tangential force in y-direction for right wheel, using a multiplier equal to 200 .

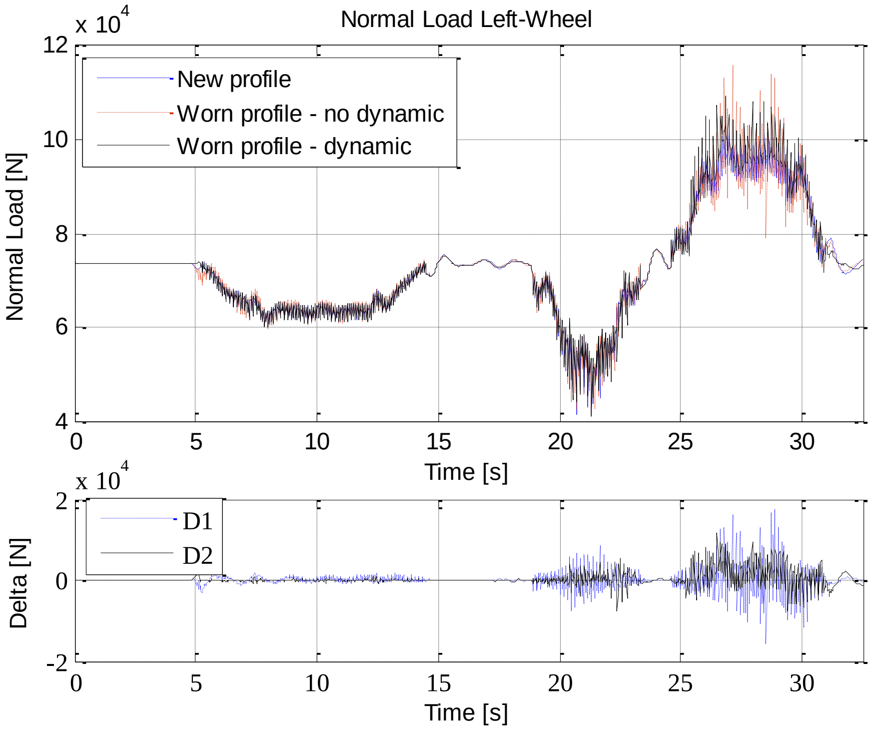


a)

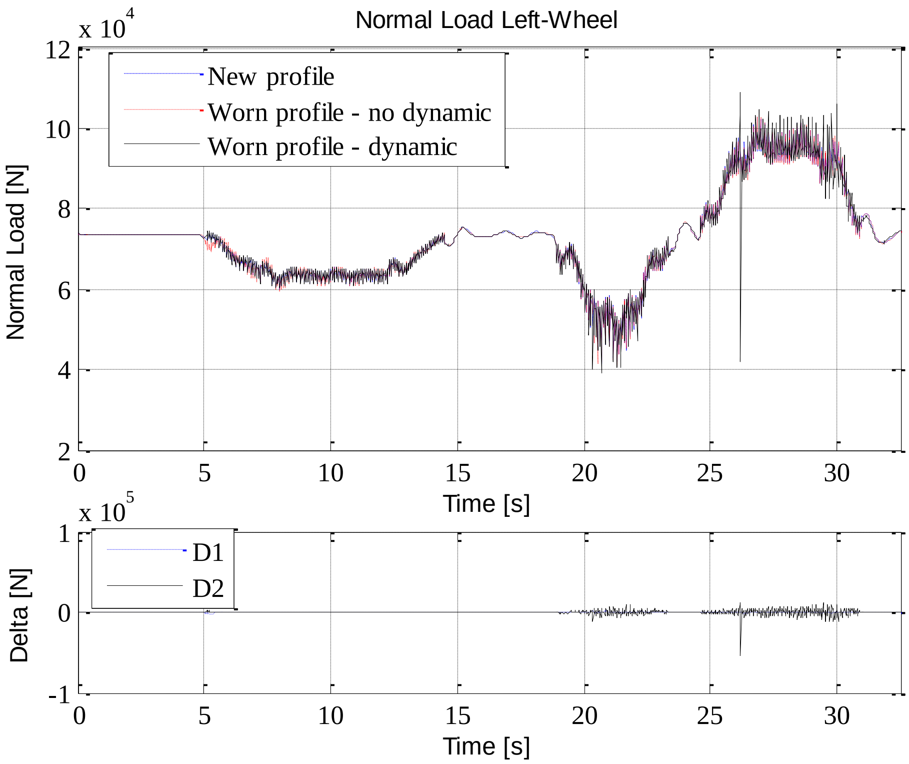
b)

Figure 7. Variation of tangential force in x-direction for right wheel, using a multiplier equal to 1000 (a) and to 200 (b).

The same behaviour can be observed considering the longitudinal tangential force, shown in figure 7, where the differences obtained using the multiplier 1000 respect the case of multiplier 200 are also evident, especially when considering the dynamic effect, in this case the multiplier 1000 is clearly inadequate.



a)



b)

Figure 8. Normal load variation for right wheel, using a multiplier equal to 1000 (left) and to 200 (right).

CONCLUSIONS

The effect of vehicle dynamic is important to simulate both the shape and the amount of material removed during the wear process of wheel profiles. This is mainly due to the fact that the worn profile changes the dynamic forces acting between wheel and rail, also affecting the contact point location. For this reason it is not possible to neglect the effect of vehicle dynamic when predicting the shape of worn profiles.

Furthermore, when using statistical methods, with "multipliers" used to repeat the track several times in order to reduce the computational time, it is necessary to perform an accurate sensitivity analysis to find the maximum multiplier which is possible to use without affecting the results of the simulation.

Competing Interests

The authors declare that there is no conflict of interests regarding the publication of this paper.

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