

The influence of friction coefficient and wheel/rail profiles on energy dissipation in the wheel/rail contact

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ABSTRACT

This work investigates the energy dissipation in a wheel/rail system through friction work modeling. In order to identify the effect of the friction coefficient on the energy dissipation in the wheel/rail contact, several simulations were performed using a 3D multibody model of a railway vehicle implemented in the software package VI-Rail Adams, with a friction coefficient varying from 0.2 to 0.7. The energy dissipation and wear rates of the inner and outer wheels of the first bogie of the vehicle running over a curve of a metro line were calculated for different friction coefficients. The total frictional work was obtained from the resultant force and slip in a reference point. The wear was also analyzed according to the Ty method including the spin, in combination with Kalker's simplified theory Fastsim, assuming that the wear is proportional to the frictional work. Two sets of rail and wheel profiles were studied in order to determine the effect of the profile's quality on the energy dissipation and wear rates. To such an end simulations and energy calculations were performed with a friction coefficient of 0.4.

Keywords: Multi-body simulation; Energy dissipation; Frictional Work; Wear; Wheel and Rail Profiles Quality.

1 INTRODUCTION

The wear on the wheel/rail interface is one of the most critical problems in railway systems due to the high maintenance costs that it generates, which can reach U.S \$54 total annual per meter and U.S. \$1.5 million for a 30 km route [1]. The profiles of the wheels and rails vary significantly due to wear, changing the wheel/rail contact geometry, increasing vehicle-track dynamic forces and reducing the performance and dynamics characteristics of the vehicles such as the

stability or passenger comfort. The removal of material from the surface by wear is a function of the sliding and contact stresses, where the amount of sliding depends on the contact patch geometry, normal force, lateral force, and friction coefficient [2].

Friction modifiers (FMs) are considered a promising solution for wear and noise problems in railway systems. FMs are commonly adopted to decrease the friction coefficient between wheel flange and rail gauge face following reduction in wear and energy

dissipation [3-5]. The friction coefficient plays an important role in the dynamic characteristics of the vehicle, especially because it has effect in the tangential forces. So, an adequate friction coefficient value may improve the bogie curving performance [6-8].

Several researches have demonstrated that it is possible a reduction in wheel/rail wear rates and noise level, implementing effective lubrication strategies on small radius curves [9-12]. A previous work performed by the authors [13] demonstrated that using a friction modifier can result in a reduction of at least 9.3% in power consumption in the contact and 19% in the wheel wear rates. This means savings near to 1.2% in the total power.

This paper gives a further explanation of the effect of the friction coefficient in the energy dissipation in the wheel/rail contact. To such an end, several simulations were performed using a multibody model, including the torque of the wheels, and varying the friction coefficient from 0.2 to 0.7. The energy dissipation and wear rates were calculated using a similar methodology to that described previously [13]. In this work, the effect of the profile's quality is also analyzed in order to compare the power savings reached using friction modifiers against the profile's quality benefits.

2 METHODS

2.1 Multi-body dynamics model

2.1.1 Vehicle model

A 3D multibody model of a railway vehicle implemented in the software package VI-Rail ADAMS was developed in collaboration with the *Metro of Medellin*, who provided the technical documentation and experimental results. The vehicle is comprised of three passenger coaches, referred to as A, R and B, of which the first and the last host the engines and apply the traction, as Fig. 1 shows.

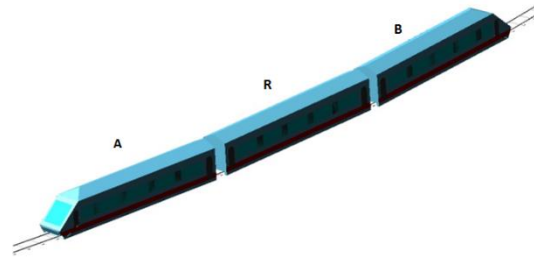


Fig. 1 Metro of Medellin vehicle model.

Fig. 2 shows a VI-Rail model of a bogie of the Metro of Medellin. The bogies are non-articulated bogies with two axles (Bo-Bo type) and its frame is a single rigid body in the form of “H”. The central pivot and the air springs support the car body loads, which are transmitted to the suspension element located in the axle box.

The primary and secondary suspension has been modeled in detail including non-linearities such as in the bumpstops and the bushings. The primary suspension is composed by two nested coil springs, two bushings, one damper and two guide bars. The guide bars were modeled using a suspension element. The vertical loads are supported by the coil springs and the longitudinal and lateral loads by the guide bars and bushings, the vertical damping is provided by one damper.

The secondary suspension is composed by two air springs located in the middle of the frame, which were modeled using a suspension element and one damper. For the central pivot, two bumpstops were used to restrict the displacement of the car body in the longitudinal and lateral direction. Moreover, the frame is connected with the car body through two vertical dampers and one lateral damper. The values of the stiffnesses and damping have been validated using eigenmodes identification in [14] and [15]. For the wheels profiles a new ORE S1002 was used.

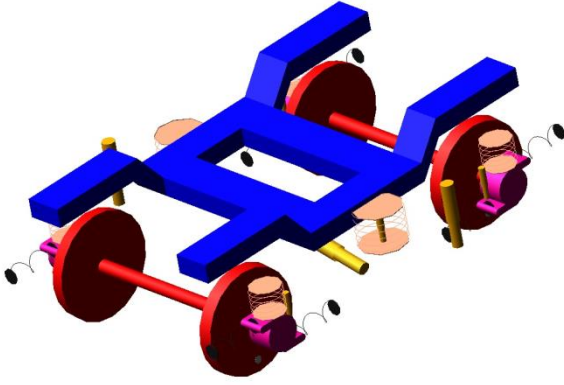


Fig. 2 Bogie model in VI-Rail Adams [13].

Due to the important role traction plays in the tangential forces and creepages, a moment of force was included in each wheel in order to reproduce the vehicle's acceleration curve in the field. To such an end, velocity and time measurements of a vehicle running in the analyzed section of track were performed using a waveform recorder, WR300® from Graphtec. The equation of the velocity curve was derived and the applied moment per wheel was calculated as:

$$M = \frac{ma(t)R}{16} \quad (1)$$

Where:

m : Vehicle mass R : Wheel radius
 $a(t)$: vehicle acceleration

2.1.2 Track model

The section of the line B between the San Javier and Santa Lucía stations was modeled. This section is composed of two straight lines and one curve with a curvature radius of 300 m; the track gauge is 1435 mm and the cant is 150 mm, the total length of the track is 765.4 m and the vertical difference between the end points is 13.48 m, the track irregularities were not taken into account for this work. *Metro of Medellín* worn profiles (CPC and HRC, inner and outer rail respectively) measured using a MiniProf Rail instrument were used for the model.

2.2 Energy and wear calculations

2.2.1 The wheel/rail contact problem

During the multibody simulation of the vehicle passing the curve, a contact module of VI-Rail ADAMS evaluates the contact forces that are needed in the dynamic equations of the wheelsets. In this study a multi-Hertzian-Fastsim approach is employed for this purpose.

In the first step the exact position of the wheelset is used to determine the interpenetration regions between the wheel and the rail, considering both wheel and rail as rigid objects. In each interpenetration region (up to three per wheel/rail pair) Herztian theory is used to obtain a normal contact force and a contact pressure distribution based on the penetration between wheel and rail and the local curvature of the contacting bodies [16].

The second step is to evaluate the tangential contact forces. To such end the simplified theory of Kalker [17,18] is used. It assumes that the tangential stress in the contact area is proportional to the tangential displacement and limited by Coulomb's law. The details of this theory as well as a *Fortran* code of the implementation, Fastsim, are given in [17]. In multibody analysis the moment around the normal of the contact point is usually neglected as it has only a very small influence on the wheelset's dynamics. However, for the frictional energy considered in this study this moment is relevant, therefore Fastsim has been modified so that this moment is also calculated.

2.2.2 Energy calculations

In order to identify the effect of the friction coefficient on the energy dissipation in the wheel-rail contact, several simulations were performed varying the friction coefficient from 0.2 to 0.7.

The local energy dissipated at a point of the contact area is the scalar product of the local slip and the local tangential stress, so the total energy dissipated in the

whole contact patch is equal to the surface integral of the local energy. Accordingly, the power dissipated as a consequence of frictional work is often approximated by ignoring the contribution of the spin as:

$$P_{frict} = (T_x\gamma_x + T_y\gamma_y)V \quad (2)$$

Where V is the velocity of the vehicle, T_x and T_y are the tangential force in the longitudinal and lateral direction, and γ_x and γ_y are the creepage in the longitudinal and lateral direction (normalized slip). When the contribution of spin is included it becomes:

$$P_{frict} = (T_x\gamma_x + T_y\gamma_y + M\varphi)V \quad (3)$$

Where M is the spin moment and φ is the spin creepage.

The results obtained after applying Eqs (2) and (3) are compared, so that the consequence of neglecting the spin contribution was quantified.

2.2.3 Wear calculations

The Derby wear index used by Pearce and Sherratt [19, 20], which adopts an energy concept in the analysis of the relationship between wear rate and contact conditions, is used to calculate the wear rates at a specific point of the track. It is assumed that the wear rate (expressed in $\mu\text{g}/(\text{m mm}^2)$) is related to frictional work done at the wheel/rail contact as:

$$Wear_{Rate} = KT\gamma/A \quad (4)$$

Where, K is a wear coefficient and A is the contact area.

The wear rate represents the mass of removed material by unit distance traveled by the vehicle (expressed in m) and by unit surface (expressed in mm^2).

Braghin et al. [21] conducted wear tests in the case of metal-metal contact with dry surfaces using a twin disc machine in order to find correlations between the wear rates and the wear index. The relationship between wear rate and wear index adopted from [21] for the

wear model was split into three regions seen in Fig. 3, where K_1 , K_2 and K_3 refer to the slope in each region, and a wear coefficient was defined for each of these regions as the Eq. (5) shows:

$$Wear_{Rate} = \begin{cases} 5.3I_w & I_w < 10.4 \\ 55.0 & 10.4 \leq I_w \leq 77.2 \\ 61.9I_w & I_w > 77.2 \end{cases} \quad (5)$$

Where I_w is the wear index.

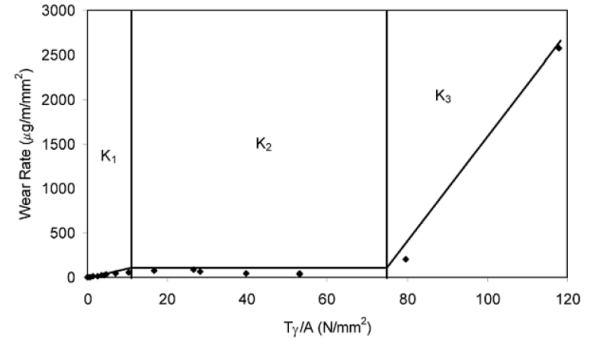


Fig. 3 Wear rate for different values of the wear index $T\gamma/A$. K_1 , K_2 and K_3 refer to the slope in each region [21].

2.3 Effect of profiles on the energy dissipation and wear rates

The quality and type of profiles play an important role in the energy dissipation and wear rates. As Table 1 shows, two pairs of rail and wheel profiles were studied in order to know the effect of the profiles quality on the energy dissipation and wear rates. The measured profiles are composed of the *Metro of Medellin* rail profiles (worn CPC and HRC, inner and outer rail respectively) and a new Ore S1002 profile for the wheels, these profiles are not designed optimally to work together. On the other hand, the standard profiles are composed by a new UIC 60 and New Ore S1002 for the rail and the wheel respectively, which are designed to work optimally.

Simulations and energy calculation were performed using the profiles presented in Table 1 with a friction coefficient of 0.4.

Table 1. Measured profiles and standard profiles

Couple	Rail profiles	Wheel profiles	Designation
1	Worm CPC-HRC	New Ore S1002	Measured Profiles
2	New UIC 60	New Ore S1002	Standard Profiles

3 RESULTS AND DISCUSSION

3.1 Traction moment on the wheels

Fig. 4 shows the vehicle velocity measurement in the field, the tendency curve is described by a 6th grade polynomial, which is derived in order to get the acceleration equation for the traction moment. Once the acceleration is calculated, the moment is obtained using Eq. (1), which is applied to each wheel using a force actuator in VI-Rail Adams.

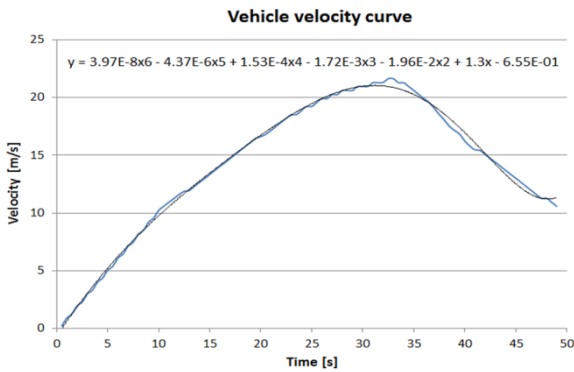


Fig. 4 Vehicle velocity curve calculated from field measurements, made with the aid of waveform recorder WR300® from Graphtec.

Fig. 5 and Fig. 6 show the torque generated on a wheel and the velocity curve of the vehicle calculated from the multibody simulation. It is seen that the velocity from the field measurement is similar to that from Adams (compare Fig. 4 and Fig. 6), meaning that the calculated vehicle displacement over the track is in very good agreement with the field measurements.

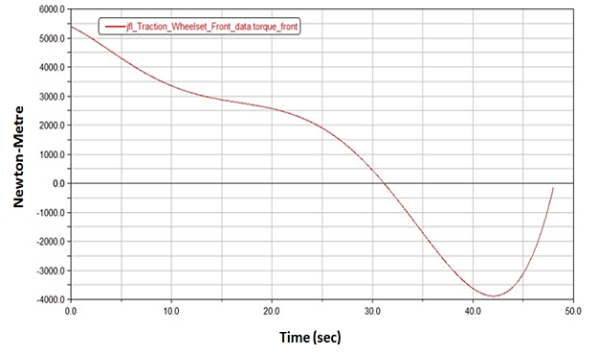


Fig. 5 Traction moment applied to each wheel of the leading and the trailing vehicle in the vehicle simulation.

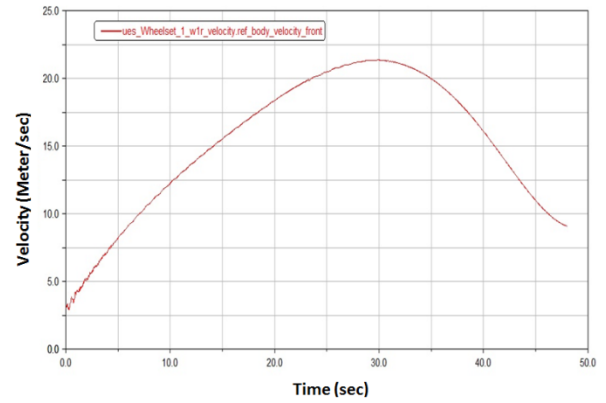


Fig. 6 Velocity curve of the vehicle model calculated from the multibody simulation.

3.2 Power calculations

This section shows the power dissipated in the wheel rail contact based on the outputs from the dynamic simulations at time $t=16s$ in the simulation. This is when the vehicle is in the middle of the curve and at that time the traction moment is 2780 Nm. Fig. 7 and Fig. 8 show the results of the total energy dissipation under different friction coefficients for the inner and outer wheels of the first bogie of the vehicle. The results were obtained using the Tgamma method and Fastsim (Eqs (2) and (3)). The latter allows determining the contribution of the spin energy dissipation since the Ty approach does not take into account this term. So the energy dissipated in spin is the difference between the two curves.

Fig. 7 shows that the inner wheels energy dissipation rises as the friction coefficient increases. However at a

friction coefficient near to 0.4, the energy dissipation starts to decrease. For the outer wheels this behavior is not observed and the energy dissipation continuously increases with the friction coefficient, see Fig. 8. When the Fastsim and Tgamma results are compared it is seen that the energy dissipation generated by spin increases with the friction coefficient in both inner and outer wheels. The energy dissipation in the outer wheels is around ten times higher than in the inner wheels.

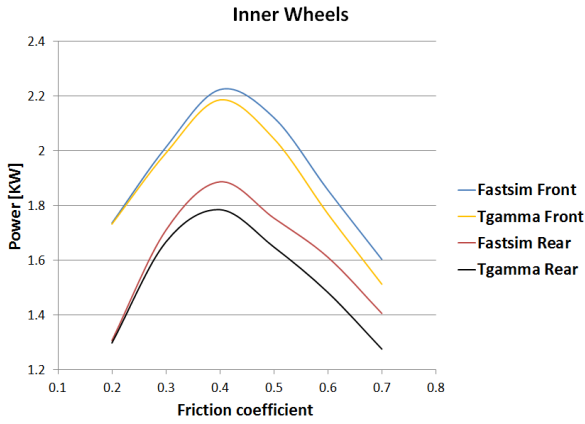


Fig. 7 Effect of the fiction coefficient in the total energy dissipation for the inner wheels.

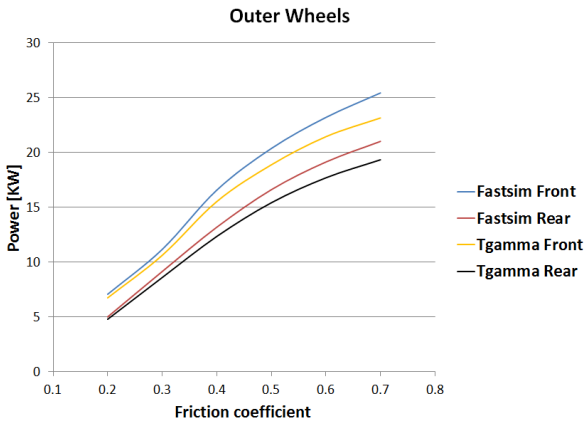


Fig. 8 Effect of the fiction coefficient in the total energy dissipation for the outer wheels.

With the aim of explaining the behavior seen in the previous results, Fig. 9 and Fig. 10 show the energy dissipation in the lateral and longitudinal direction respectively and Fig. 11 to Fig. 14 show the creepage and creep force for the inner and outer wheels. From Fig. 9, it is seen that both the inner and outer wheels

show the same behavior for the lateral energy dissipation. In other words, the lateral energy dissipation goes up until a friction coefficient between 0.4 and 0.5 is reached and then goes down.

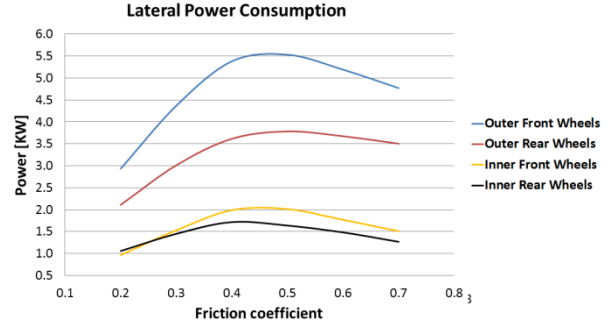


Fig. 9 Power consumption of the inner and outer wheels in lateral direction.

However, for the outer wheels the longitudinal energy is clearly much more significant than the lateral component, see Fig. 10. Accordingly, when the total energy is calculated in Fig. 8 for the outer wheels, it is not considerably influenced by the lateral behavior. On the other hand, in the inner wheels the lateral energy dissipation is more significant and the longitudinal component decreases as the friction coefficient increases, see Fig. 10. Hence, the behavior of the lateral component dominates in the total energy calculation for the inner wheels as Fig. 7 shows.

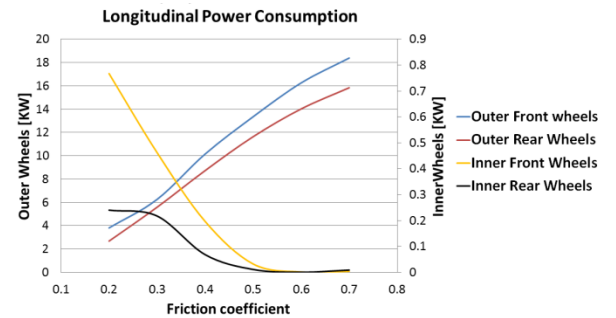


Fig. 10 Power consumption of outer and inner wheels in longitudinal direction.

Fig. 11 shows the contribution of the longitudinal force slip to the energy dissipation in the inner and outer wheels (respectively the first and second terms of Eqs (2) and (3)). It can be seen that at a friction coefficient of 0.2 both inner and outer wheels have a similar longitudinal creep force. As the friction

coefficient increases, however, the inner wheel loses traction while the outer wheel increases it, keeping the total traction constant. These results explain the behavior of the energy dissipation shown in Fig. 10, where the energy dissipation decreases in the longitudinal direction for the inner wheel and increases for the outer wheel. Although the longitudinal creepage decreases in both inner and outer wheels (see Fig. 12), this is not dominant for the outer wheel.

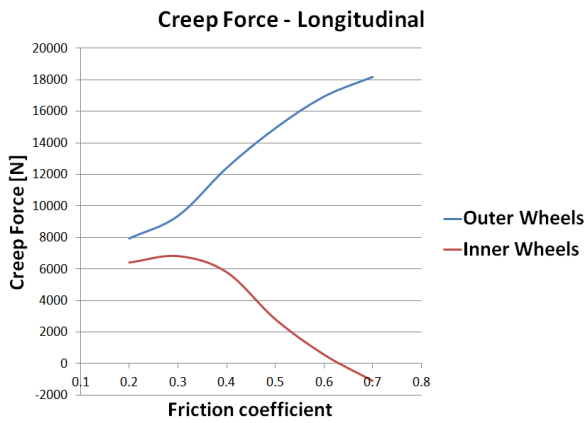


Fig. 11 Creep forces in longitudinal direction.

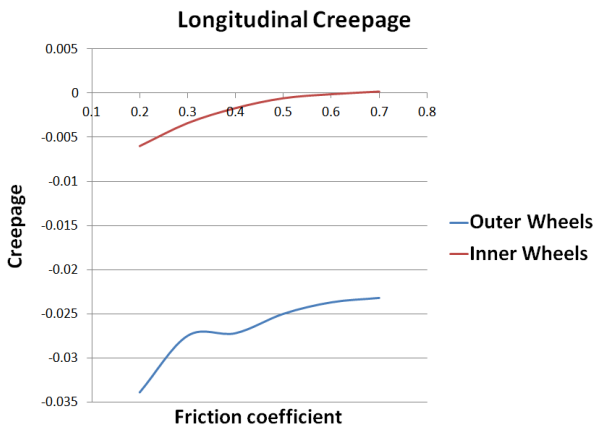


Fig. 12 Creepages in longitudinal direction.

The lateral creepage for the inner and outer wheels is shown in Fig. 13. It decreases linearly as the friction coefficient grows. This behavior is explained because the difference in the longitudinal creep force for the inner and outer wheels produces a rotational moment of the wheelset in clockwise direction, reducing the yaw angle of the wheelset. On the other hand, the lateral creep forces shown in Fig. 14 for the inner and

outer wheels, increase as the friction coefficient grows, however after a friction coefficient of 0.4 - 0.5 the creep force tends to stabilize, this condition produces that the energy dissipation reaches a peak and later starts to decrease as shown in Fig. 9.

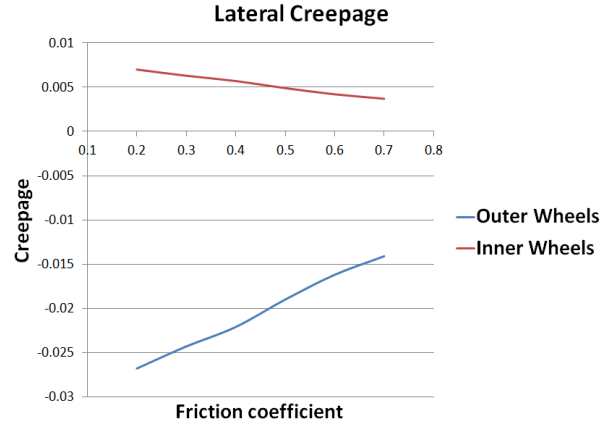


Fig. 13 Creepages in lateral direction.

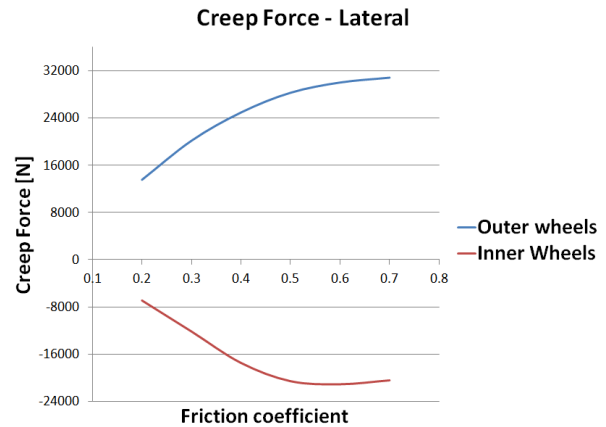


Fig. 14 Creep forces in lateral direction.

Fig. 15 shows the total energy dissipation and the energy dissipated by spin as a function of the friction coefficient of the first bogie. The outer wheels are responsible for 89.8% of the total energy dissipated. According to these results, it is possible to conclude that the total energy dissipation increases roughly linearly as the friction coefficient grows. The energy dissipation by spin increase also linearly with the friction coefficient, reaching a maximum value of 4.2 kw at a friction coefficient of 0.7.

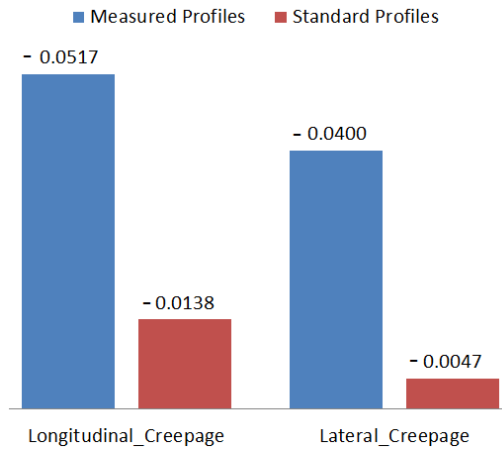


Fig. 18. Longitudinal and lateral creepage in the outer wheels.

Fig. 19 and Table 2 show the results of total energy dissipation of the first bogie. Energy savings of 69%, 81% and 61% are observed in the longitudinal, lateral and spin directions respectively, when optimal profiles are used. This can be more influential than the effect of the friction modifiers, which according to previous reported results [13] only reaches a maximum energy saving of 36%.

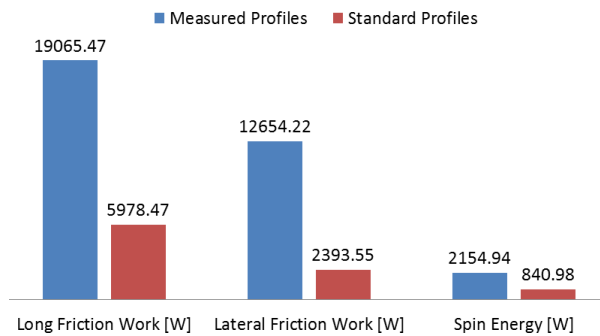


Fig. 19 Longitudinal, lateral and spin energy dissipation in all the wheels of the first bogie.

Table 2. Wheel/rail profiles effect in the total energy dissipation of the first bogie.

	Measured profiles	Standard Profiles
Total Energy [W]	33874.6	9174.4

Table 3 shows the differences between measured profiles and standard profiles regarding the wear rates.

An increase of 81% is possible using new optimal profiles, which is 2 times higher compared to the friction modifier reduction [13].

Table 3. Wheel/rail profiles effect in the wear rates of the first bogie.

	Measured profiles	Standard Profiles
Wear Rate [$\mu\text{g}/(\text{m mm}^2)$]	158.1	30.6

4 CONCLUSIONS

The energy dissipation generated by spin increases as the friction coefficient grows. It is only recommended to neglect the effect of the spin moment in the energy calculations for low friction coefficient conditions since for friction coefficients higher than 0.4 the energy loss is equivalent to more than 2kW and reaches a maximum of 4kw at a friction coefficient of 0.7 for the leading bogie.

As the friction coefficient increases, the inner wheel loses traction while the outer wheel increases it, this condition produces a rotational moment of the wheelset in clockwise direction, reducing the yaw angle of the wheelset. This means that higher friction coefficients improve the vehicle navigation over the curve, this condition is also achieved when the profiles are new and designed to work together.

The energy consumption increases linearly as the friction coefficient grows on the outer wheels and because they provide 89.8% of the total energy dissipated in the contact, it is possible to conclude that more energy is saved for a low friction coefficient in the outer rail. However, it must always be guaranteed that there is enough friction for the required traction and braking.

The results of simulations with different profiles are important since they show the necessity for optimizing profiles and re-profiling procedures. An optimized wheel/rail profile combination greatly reduces the energy dissipation and the wear rates. This reduction is

estimated to be twice the reduction expected by applying a friction modifier. Accordingly, it is recommended to combine the use of friction modifiers with an optimum reprofile procedure in order to reduce energy dissipation and wear rates.

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