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Impact of Wheel Hollow Wear on Heavy-Haul Vehicle Components

P. Augusto De Paula Pacheco[1,2], M. V. Lopes[1], G. F. Mendonça Dos Santos[3], M. Costa[4] and A. A. Santos Jr[1]

[1]School of Mechanical Engineering, State University of Campinas, Brazil
[2]Railway Department, Federal Institute of the Southeast of Minas Gerais, Brazil
[3]Departament of Mechanical Engineering, Federal University of Espírito Santo, Vitória, Brazil
[4]Railways Engineering, Vale S.A., São Luís, Brazil

Abstract

The impact of worn wheel profile evolution on vehicle components' stress for heavy-haul wagons were investigated using multibody dynamic simulation. A broad gauge GDU-Ride Control wagon, a Brazilian heavy-haul three-piece bogie with 37.5 tons per axle, was modelled. Four wheel profiles were examined, with 1, 2 and 3 mm of hollow wear and the new wheel profile. The effects of wheel hollows on vehicle components in terms of vertical, longitudinal, and lateral forces, vertical and lateral acceleration of the pedestal and wagon body, and lateral displacement of the wheelset, were examined and discussed. The result shows that the hollow is critical for the forces in the primary suspension (pad) and centre plate. Increasing the hollow wear to 2 mm results in larger lateral displacement; however, a 3 mm hollow wear wheel does not enhance this behaviour due to the false flange. Besides, there was a 48% increase in longitudinal forces at the pad for the 1-mm hollow wheel, and 53% for the 2- and 3-mm, when compared to the effect of new wheels. Conversely, the forces of the secondary suspension were not affected. Finally, for the centre plate, increased hollows amplified the friction force and resulted in uneven wear.

Keywords: railways maintenance, worn wheels, component load, dynamic simulation, heavy haul, multi-body modelling.

1 Introduction

The contact conditions between wheels and rails impacts on dynamic performance, wheel/rail wear, train energy consumption, and vehicle/track maintenance costs [1]. As a result of contact, both wheels and rails will wear out. On the other hand, worn wheel profiles alter the wheel/rail contact relationship, causing dynamic problems such as hunting motions [2].

In Brazil, the freight railways operate a huge number of GDU-Ride Control wagons. They are three-piece bogie wagons, mostly with a capacity of 37.5 tons per axle. Wheel wear on these wagons is characterized by hollow wear [3, 4]. The hollow shape leads to a profile at the centre of the tread below the original profile (Figure 1). As a result, it affects vehicle running behaviour and contact damage mechanism [5].

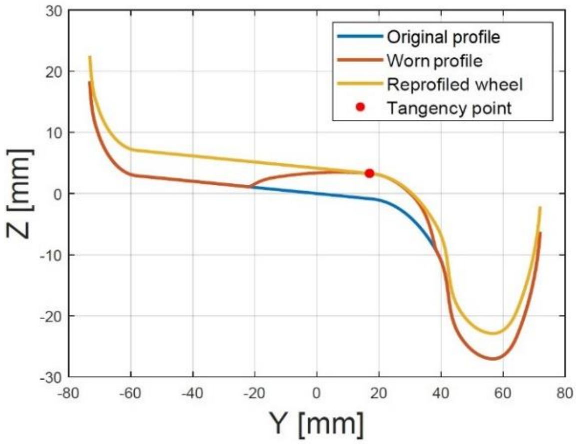

Figure 1. Hollow wear profile [4].

In recent years, many studies have been conducted about wheel wear on railways [6]. With dynamic analysis, Sawley and Wu [7] demonstrate that hollow wheel shapes impair the steering capabilities of bogies, causing increased rolling resistance and lateral wheel/rail forces. As a result, fuel consumption increases, and rail wear accelerates. Additionally, hollow wear can increase rail fatigue due to increased wheel/rail contact stress. The Transportation Technology Center [8] analyzed and tested the effect of hollow-worn wheels on wheel-rail interactions. According to both the modelling and the tests, hollow wear raises the standard deviation of lateral acceleration of the car body, which reduces the defined critical speed. Despite this, the tangent track simulations with loaded vehicles do not indicate that hollow-worn wheels lead to a high derailment coefficient (lateral force by vertical force - Y/Q). Simulation results suggest that hollow-worn wheels produce different instabilities than non-hollow-worn wheels. Kaiser, Poll and Vinolas [9] presented an enhanced simulation model of a vehicle-track system. They explored the impact of structural deformations on wheel-rail contact. According to the investigation, structural deformations of the wheelset can significantly affect wheel-rail contact and wear.

Lima et al [10] studied the effect of shear pads on the vehicle dynamic behaviour for heavy-haul vehicles. According to them, the use of shear pads can be beneficial to wheel-rail wear, reducing it up to 43%. For wear number reduction, only the longitudinal direction was relevant. These results agree with the findings of Corrêa et al [11] and Pacheco et al [12], who used multibody dynamic simulation to show that the longitudinal stiffness of the pad has a greater impact on wear volume, followed by longitudinal clearance. Correa et al [11] evaluated the effects of primary suspension parameters, friction wedges, and flange backspacing within operational ranges on Y/Q and wear index. Their results indicate that primary suspension longitudinal clearance and its stiffness significantly impact the derailment coefficient Y/Q and the wear index, resulting in higher operational costs and maintenance risks. A complement of the previous work was developed by Pacheco et al [12], to investigate the relationship between the condition of vehicle components and the wear volume of the wheel during regular service. For that, they used multiple linear regression and multibody simulation. The work showed that the longitudinal stiffness of the primary suspension had the greatest impact on the wear volume, followed by the longitudinal clearance. When the longitudinal stiffness is increased, the wear volume increases, while when the clearance is increased, the wear volume decreases.

Observing the wear from a different point of view and using advanced computational tools, Kuka et al [13] analysed the response of vehicles to changes in track defect levels. According to their research, the layout, irregularities, and degradation of the rails have the greatest influence on wear, safety, and maintenance costs associated with vehicle-track interaction loads.

As is much known, wheels represent one of the highest maintenance costs, which leads to a great interest in its wear [14]. Because of that, several studies have been published on predicting wear and on the effect of different parameters on wheel wear. Among them, Chen et al [5] investigated the impact of wheel profile evolution on wheel-rail dynamic interaction and surface-initiated rolling contact fatigue (RCF) in the turnout area. Their results indicate that the wheel transition position is shifted backwards with wheel profile evolution. When the degree of hollowing becomes medium, wheel profile evolution improves the conformal contact, which decreases the accumulated surface-initiated RCF in the switch panel. As the degree of hollowing becomes severe (over 2.5 mm), the false flange's contact with the wing rail is more prone to RCF in the crossing panel.

Because of the economic and safety importance of the wheel wear, there are not many works on the impact of the worn wheel profile evolution on vehicle components' stress, internal forces, and wear. The ones that do generally examine the impact of wheel wear on the dynamics of the vehicle as in Sawley and Wu [7], Bethel Lulu et al [15], Hou, Chen and Cheng [16], Simson and Pearce [17] and, Tavakkoli, Ghajar and Alizadeh [18]. Therefore, this work aims to understand such effects, obtaining the forces and accelerations from vehicle-track model simulations with different hollow wheel levels, and evaluating their effects.

In this text, Section II presents the methodology, including details about multibody simulations. Section III presents the results, while conclusions are presented in Section IV.

2 Methods

A description of the dynamic model developed in SIMPACK® is provided in this section. The steps of simulation are presented in Section 2.1. A description of the parameters employed (the track, speed, and wheel profiles) is given in Section 2.2.

2.1 Dynamic Model

For the present work, a broad gauge GDU-Ride Control wagon was selected. This is a Brazilian heavy-haul three-piece bogie wagon with 37.5 tons per axle. The simulation model and parameter values are referred to Pacheco et al [3], Pires et al [4] and Lima et al [10]. A simplified view of the model is shown in Figure 2

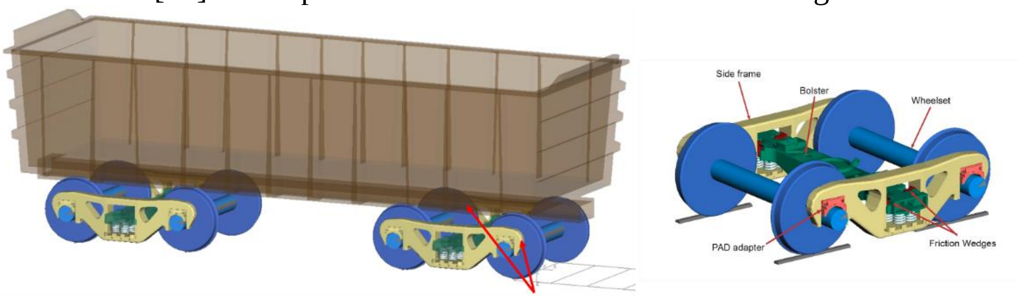

Figure 2. GDU-Ride Control model developed with SIMPACK® (adapted from [19]).

The primary suspension is based on a rubber pad adapter modelled using three force elements, bushing type, connecting side frames to wheelset. Stiffness and damping properties are considered in all directions of translation and rotation. The stiffness in all directions were obtained from Lima et al [10].

The secondary suspension is composed of 5 springs, four of these are arranged such as forming a square. Each element occupies one of the corners of the square, and the last spring is in the centre, as shown in Figure 3. Secondary suspensions are modelled using bushing springs with stiffness and damping properties. In this bogie, discrete contact elements were used to model the wedge's constant damping.

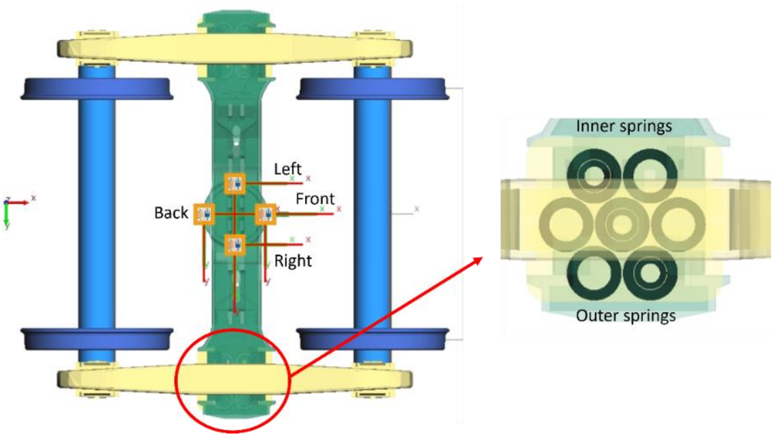

Figure 2. Ride Control bogie with its centre plate force elements and secondary suspension springs viewed from the top.

The modelling of the centre plate element was divided to represent two functions. The first function refers to the support of the box and friction on the base of the platter, which is modelled using 4 universal point of contact force elements. They are positioned as: 1 in the front, 1 in the back, 1 in the left and 1 in the right (Figure 3). The second function refers to the contact between the edges of the centre plate with the circumference of the plate in the radial direction. For that, we employed a unidirectional spring-damper element and a non-linear friction element, which establishes tangential friction on the circumference of the centre plate. In the roller side bearing, a spring (bushing type) was used for vertical stiffness, with a gap, since this type of side bearing does not have constant contact.

FASTSIM (Kalker Theory) was used to calculate the wheel-rail contact forces. It is a fast contact force calculator, widely used in W/R contact programs [20]. It is based on the simplified assumption that the deformation on the surface is linearly dependent only on the stress [21]. SIMPACK® offers two methods for locating the contact patches between rail and wheel: equivalent elastic and discrete elastic. In this work, the discrete method was chosen. This method uses the actual contact patch shape for normal and tangential force calculation. It is generally slower but more accurate than the alternative [22].

The effect of worn wheel profiles on the vehicle components will be evaluated by force results on the components and by three points of measurement of accelerations, represented by accelerometers. These were positioned on the side frame pedestal, right and left, and on the wagon box, as shown in Figure 2.

2.2. Simulation parameters

In this study, a track with a curve to the left (Figure 4) was employed (Table 1). It is the same used by Pacheco et al [3] and Pires et al [4]. This represents the sharpest curve of the ore railway where this wagon runs in Brazil. The speed is 70 km/h for all simulations, representing the actual average speed in that railway.

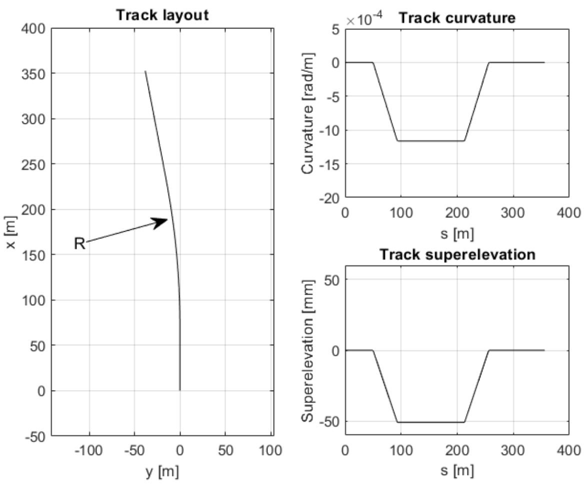

Figure 3. Layout of the track used in the simulations.

Radius (m)	860.00
Tangent (m)	50.00
Transition (m)	43.10
Length of const. radius (m)	120.00
Tangent (m)	100.00
Superelevation (mm)	51.00
Total length (m)	356.20

Table 1. Track description [3].

The rail profiles were obtained from measurements performed in the field by the railway company. They use CPC (contact point in the centre) to tangent sections and low rail in curves, and High Sharp for the high rails (Figure 5).

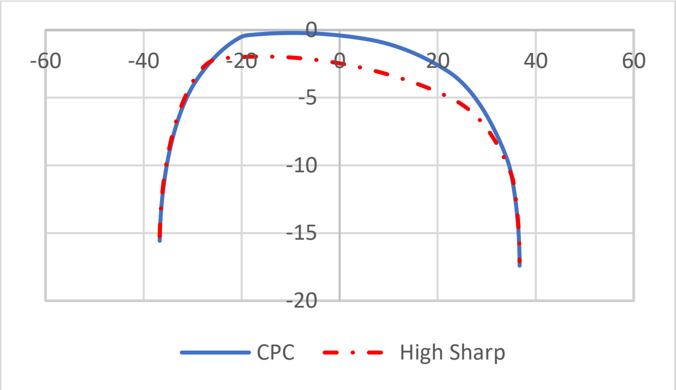

Figure 4. Rail profiles.

As showed by Bosso et al. [23], it is important to consider the railway track irregularities in vehicle dynamic simulations, since they can compromise the vehicle running safety and stability. The effects of irregularities can also contribute to failures of the main components of the railway system. In this way, the analysis will be made considering such irregularities.

The track irregularities in the vertical and lateral direction are defined as Federal Railroad Administration (FRA) Class 4 irregularities given by Power Spectral Density (PSD) functions, as shown by [3] and represented by Equations (1) and (2) [24].

$$S_L(\Omega) = \frac{A_a.\Omega_c{}^2}{\Omega^2.(\Omega^2+\Omega_c{}^2)} \tag{1}$$

$$S_V(\Omega) = \frac{A_v.\Omega_c{}^2}{\Omega^2.(\Omega^2+\Omega_c{}^2)} \tag{2}$$

The coefficients for this class of irregularities are shown in Table 2. According to railway operator, this classification is the one which best represents their track.

Class	$A_v\left[cm^2\frac{rad}{m}\right]$	$A_a\left[cm^2\frac{rad}{m}\right]$	$\Omega_s^2\left[\frac{rad}{m}\right]$	$\Omega_c^2\left[\frac{rad}{m}\right]$
4	0.5376	0.3027	1.1312	0.8245

Table 2. PSD parameters for irregularities FRA 4 [25].

Pacheco et al [3] and Pires et al [4] analysed how wheel wear can affect wheel service life and examined whether their profiles will maintain the performance over time. Wear simulations were developed by them, resulting in worn profiles and distance travelling. The simulations were run until 3 mm of tread wear was reached, which is the standard reprofiling criteria used by the selected railway.

Their results show that the reduction of hollowing criteria increases the service life, i.e. reducing the current hollowing criteria can increase wheel service life by up to 6%. Furthermore, the shakedown diagram was applied to the worn profiles to evaluate how critical is the working conditions. If the hollow wear exceeds 3 mm, the profile is in the ratcheting region, which is not recommended. As a result, the stopping criterion should not exceed 3 mm and so this limit is used in the current study.

To investigate the relationship between wheel hollow and vehicle component stresses, the effects of 1, 2, and 3 mm of hollow wear were examined along with the new wheel profile. The profiles are shown in Figure 6.

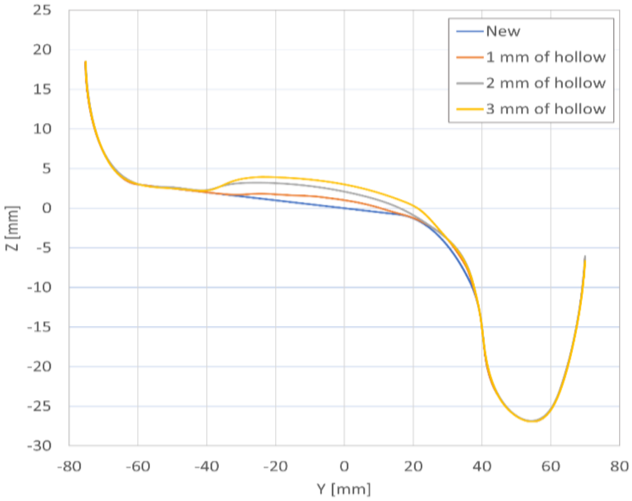

Figure 5. Wheel tread profiles employed in the current study.

3 Results

Figure 7 shows the lateral displacement of the front wheelset for different hollow wheels in the 860 m radius curve described in section 2.2.

In the curve, between 93.1 m and 213.1 m in Figure 7, up to 2 mm, as the hollow increases, the lateral displacement decreases (Table 3). The increase in the hollow increases the wheel conicity, which leads to more rapid rolling radii variation, i.e., the difference between the left and right wheel contact diameters due to lateral displacement. As a result, the lateral displacement needed for the curving is reduced [26].

The 3-mm hollow wheel does not show the same behaviour of the 1-mm and 2-mm hollow wear. This can be related to the false flange caused by the severe wheel wear. According to Spangenberg et al [27] and Chen et al [5], false flange wheel wear reduces equivalent conicity. There is an increase in lateral displacement. As a result of the decrease in the wheel's equivalent conicity, combined with the lowered gauge corner of the rail, the outer wheel (right) contacts the rail over a wider area, while the inner wheel (left) has a false flange that encloses the contact (Figure 8).

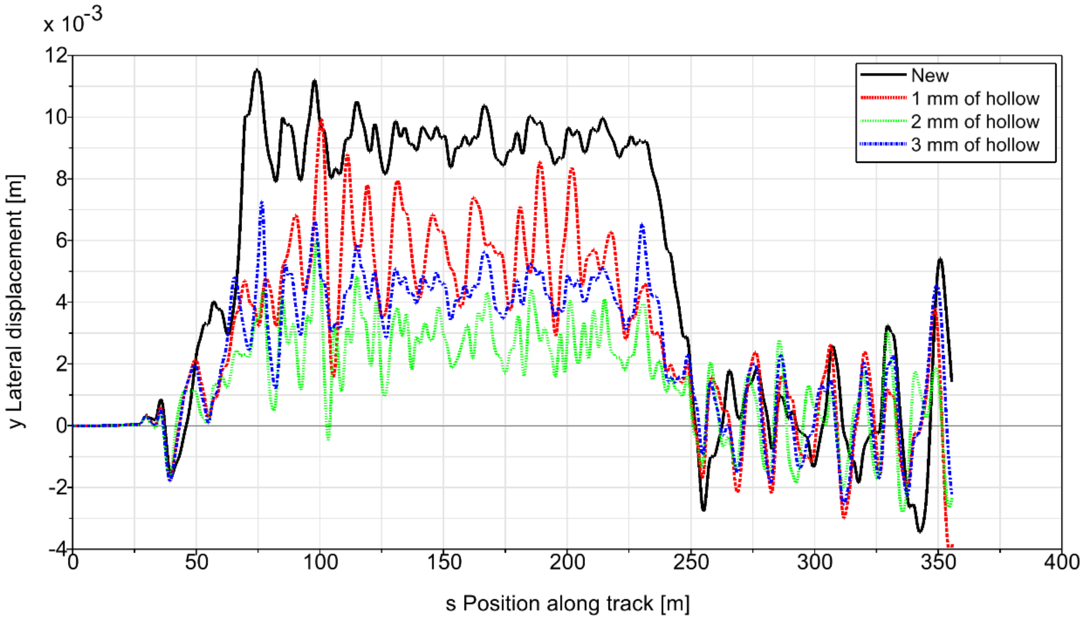

Figure 7. Lateral displacement of the first wheelset.

	Lateral displacement RMS [mm]
New	9.2
1-mm hollow	5.7
2-mm hollow	2.7
3-mm hollow	4.3

Table 3. Lateral displacement RMS (root mean square) in the curve (93.1 m to 213.1 m) for different hollow wheels.

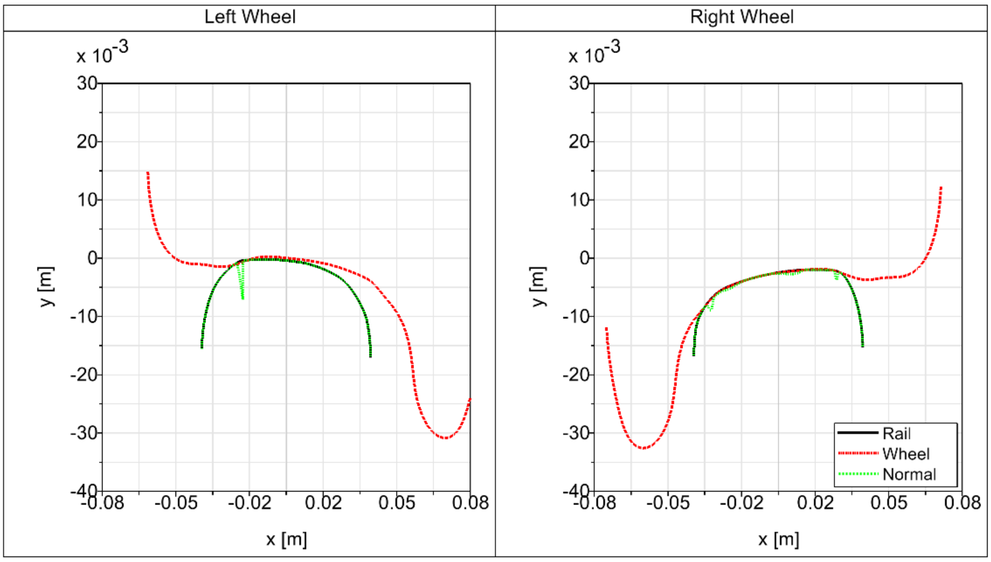

Figure 6. False flange contact in the 3 mm hollow wheel.

Table 4 presents the acceleration RMS (root mean square) in the lateral and vertical direction obtained in the curve. Since the simulation occurs at a constant speed, applied directly to the rigid body joints, the longitudinal results are not presented. The results show that the hollow has a stronger effect in the Bogie Pedestal than in the Wagon Body. Such damping can be explained by the dissipation of energy from the elements between those components.

The RMS of acceleration in the lateral direction (y-direction) for the Bogie Pedestal and Wagon Box increases up to a maximum at 2-mm hollow, then it slightly decreases. As already mentioned, the increase of the lateral acceleration for 3-mm hollow compared to the 2-mm hollow is explained by the conformal contact in the hollow, limiting the lateral movement of the wheelset. In the vertical direction (z), the acceleration RMS value decreases with the hollow wear for the Bogie Pedestal, while for the Wagon Box, it seems to not impact the acceleration.

	Bogie Pedestal [10^{-2} m/s²]		*Wagon box* [10^{-2} m/s²]	
	Lateral (y)	**Vertical (z)**	**Lateral (y)**	**Vertical (z)**
New	32.30 100%	-1.29 100%	23.20 100%	-0.94 100%
1-mm hollow	39.20 121%	-1.14 88%	29.80 128%	-0.93 99%
2-mm hollow	59.70 185%	-0.73 57%	40.50 174%	-0.96 102%
3-mm hollow	48.90 151%	-0.19 15%	34.40 148%	-1.00 107%

Table 4. Accelerations RMS in the curve.

Figure 9 shows the longitudinal and lateral forces at the pad as a function of the hollow wear. The lateral force rises just a little bit with the hollow wear, less than 19%. The greater impact is on the longitudinal forces, in which a huge increment of 48% from the new to the 1-mm hollow wheel can be noted. Notwithstanding, from 1 to 2-mm the force increases just a little bit more (5%) and remains constant for 3-mm. Increased longitudinal forces on the pad contribute to increased shear stresses in that direction and, consequently, to more pad wear. A worn pad is stiffer, which is detrimental to the wheel for it reduces the bogie steering capability. In fact, pad stiffness is the most influential parameter on $T\gamma$ [11] and wear rate [12] in the wheel-rail contact. These results underscore a cyclical relationship between the wheel and pad, highlighting their interdependence. The primary function of the pad is to safeguard the wheel from wear and damage. However, an increasingly worn wheel demands more support from the pad, causing gradual wear and deterioration of the pad itself. As the pad wears down, it loses its ability to effectively protect the wheel, ultimately leading to a detrimental feedback loop, where the deteriorating pad exacerbates the wear on the wheel. This cycle of wear and dependency results in a mutually reinforcing, and detrimental, relationship between the wheel and pad.

Figure 9. RMS of the pad forces as a function of hollow wear.

There is a low influence of hollow wear on the lateral force of the secondary suspension, less than 5%, for both inner and outer springs. Longitudinal force is not affected by hollow wear either.

Table 5 presents the RMS of wedge friction force obtained in the curve section as a function of hollow wear. There is no significant difference between the front and back wedges, i.e. the friction force is not affected by hollow wear.

	FRICTION FORCE [KN]	
	Front	**Back**
NEW	6.35 (100%)	6.38 (100%)
1-mm hollow	6.48 (102%)	6.55 (103%)
2-mm hollow	6.52 (103%)	6.54 (102%)
3-mm hollow	6.39 (101%)	6.51 (102%)

Table 5. Wedge friction force as a function of hollow wear.

The plate friction forces are shown in Figure 10. The highest value occurs in the right position, Figure 3, while the smallest is in the left point, in the opposite position. This difference happens due to the curve direction since the curve is to the left. An increase of the friction force with the hollow until 2 mm can be observed, reaching up to 85% increase from the new wheel, followed by a reduction from 2 to 3 mm hollow.

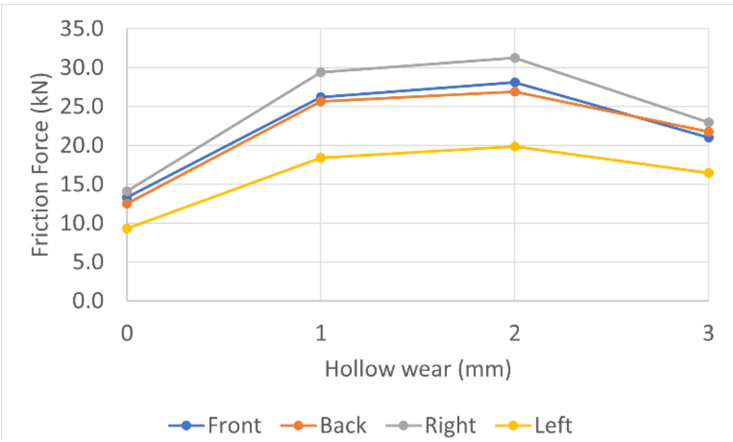

Figure 10. Plate friction force RMS as a function of hollow wear.

An uneven friction force on the side may result in an elliptical worn centre plate, e.g., the longitudinal diameter is less than the lateral one. Figure 11 shows the ratio of the friction force based on the front force, which represents 100% for each hollow case.

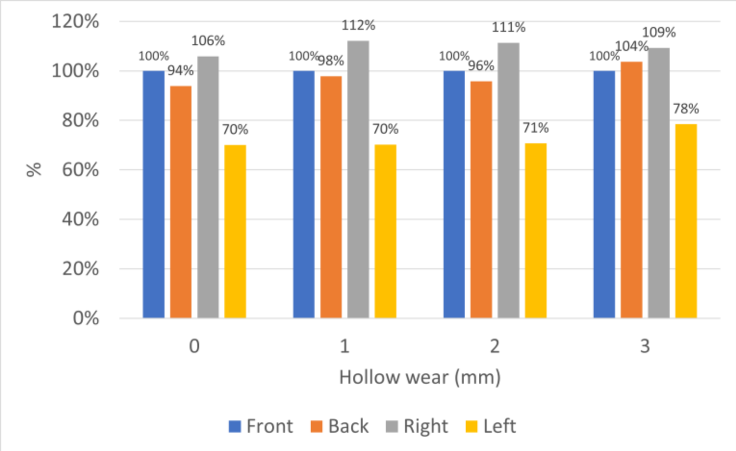

Figure 11. Correlation between the force points and the front in the centre plate.

As shown in Figure 11, the ratio increases for the back force when the hollow increases to 1-mm, then it decreases for 2-mm, but is still greater than the new profile. For 3-mm, the ratio surpasses the front friction force.

Considering the right and left positions, it is noticed that the forces between the sides increase until 1-mm hollow, followed by a slight decrement until 3 mm, but still higher than for the new profile. This uneven wear can contribute to increase the derailment risk and wheel wear. According to Olshevskiya et al. [28] significant centre plate wear reduces the strength of the unit, disturbs normal coupling conditions between the car body and the bogie, and leads to unspecified loading conditions for the bolster. Furthermore, Shvets [29] showed that a worn centre plate can contribute to the centre pivot wearing out, bending, and even breaking, crushing the hole where it is inserted during operation.

4 Conclusions and Contributions

Since the wear of rail wheels is related to the maintenance parameters, the study of the effect of hollow wheels on the bogie components has been proposed to identify the force, leading to interesting results. Multibody dynamic simulations of a heavy-haul wagon with three different hollow wear wheel profiles and a new profile were performed to address the problem.

Analysis of lateral displacement showed that increasing the hollow results in a sharper variation of rolling radii and, therefore, larger lateral displacement. The 3-mm hollow wear wheel, however, did not exhibit this behaviour due to the false flange.

Hollow wheels showed a critical impact on the longitudinal forces of the primary suspension (pad). Putting in numbers, an increment of 48% for the 1-mm hollow wheel compared to the original profile, plus 5% for the 2- and 3-mm hollow wheels. Increasing longitudinal forces on the pad can result in higher pad wear. The wear on the pad aggravates the wear on the wheel as it wears out. The wheel and pad are thus able to reinforce one another, and this is often detrimental.

There was little effect of hollow wear in the vertical and lateral forces of the secondary suspension, comprised of springs and friction wedges.

Increased hollows mean higher friction force in the centre plate and modify its distribution, which may result in uneven wear.

As the wheel profile during its service life spends more time in worn condition, new maintenance limits based on hollow wear values could extend the life of the wheels and bogie components. The analyses presented in this work contribute to the decision about these limits.

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