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Adhesion Conditions at Wheel-Rail Interface: The Impact of Friction Modifiers on Curving Behaviour and Wear

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Abstract

The level of adhesion between the wheel and the rail, particularly under curving conditions, plays a crucial role in determining the vehicle dynamic behaviour and the wear of the wheel and the rail profiles. Therefore, one of the main objectives of infrastructure owners and train operators is to actively manage adhesion conditions, ensuring safe railway operations while minimizing maintenance costs for both rolling stock and infrastructure. This study investigates the influence of the friction coefficient on vehicle running behaviour and wheel-rail wear under curving conditions. Friction coefficient measurements with a portable tribometer are carried out on a curved section to assess the effectiveness of a track-side lubrication system for flange contact lubrication. Dry and lubricated track conditions are tested, with friction measurements on both the top of the rail and on the gauge face. Test results also show a reduced friction coefficient at the rail gauge face even before activation of the lubrication system, attributable to lubricant deposited by the wheels of trains equipped with onboard flange lubrication systems. The results of friction measurements are used as a reference for vehicle dynamics simulations which are carried out to assess the impact of friction modifiers on curving behaviour and wear. The use of friction modifiers is found to slightly increase the angle of attack and the lateral forces on the leading wheelset, resulting in a reduction in bogie self-steering capability, while still ensuring a significant decrease in wheel–rail profile wear.

Keywords: wheel-rail friction coefficient, tribometer, friction modifiers, vehicle dynamics, multibody simulations, wheel-rail interaction, wear.

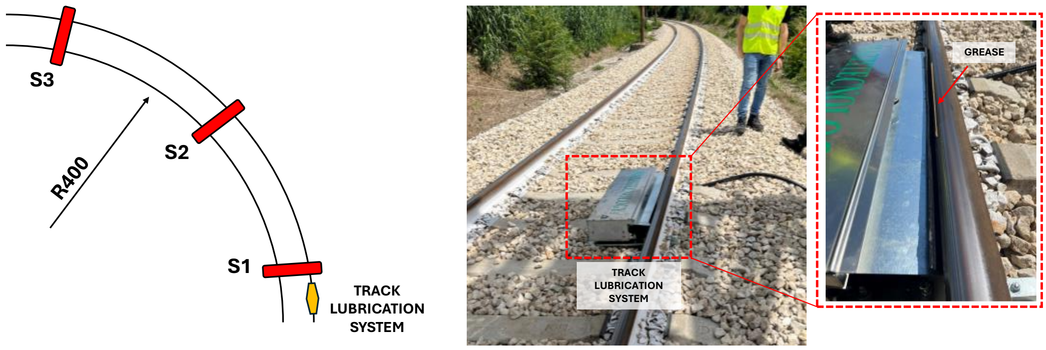
1. Introduction

In recent years, extensive research has been carried out on vehicle behaviour in curves, with a strong focus on optimizing wheel profiles, bogie design, vehicle layout, and suspension systems. In parallel, significant efforts have also been devoted to analysing wheel-rail contact conditions and adhesion mechanisms, since they affect significantly the vehicle running behaviour, as it directly governs vehicle stability, ride comfort, and degradation of rail and wheel profiles [1]. The overall objective of these studies is to reduce creepage and contact forces between the wheel and the rail, thereby minimizing wear of the wheel and rail profiles, as well as mitigating the risk of derailment [2]. In this context, it is well established that the level of adhesion plays a fundamental role in determining wheel-rail contact conditions, the forces transmitted at the wheel-rail interface, and consequently the wear of the wheel and the rails. While a high adhesion coefficient is beneficial during traction and braking, it is often desirable to reduce this value in the case of very tight curves, particularly when flange contact occurs. It is well known that high values of adhesion coefficient usually lead to higher tangential stresses at the wheel-rail interface, which promote rapid wear of the wheel and rail, rolling contact fatigue (RCF), and progressive deterioration of both wheel and rail profiles [1]. Moreover, high value of friction coefficient can also trigger unstable stick-slip phenomena, leading to rail corrugation [3] and squeal noise in curves [4,5]. Therefore, the study, and possibly the management, of the friction coefficient and consequently the adhesion condition plays a crucial role in controlling wear mechanisms at the wheel/rail interface [6], as well as vehicle running behaviour and passenger comfort and safety. Overall, the reduction of contact forces and wear rates translates into lower maintenance demands for both rolling stock and infrastructure, ultimately resulting in enhanced service reliability, improved ride quality, and reduced life-cycle costs.
Although the role of the friction coefficient in wear mechanisms is widely recognized, its influence on the overall adhesion conditions at the wheel-rail interface, and the consequent impact on vehicle curving behaviour, has not yet been extensively and systematically investigated. Therefore, adhesion management cannot be addressed solely from a wear mitigation perspective but must be evaluated within a comprehensive vehicle dynamic analysis. As noted by Berg et al. [7], the wheel–rail friction conditions during test runs for vehicle certification are difficult to manage, as the friction coefficient must be sufficiently high while also being inherently challenging to quantify. Furthermore, the importance of accurately defining the role of the friction coefficient in vehicle dynamics has grown in recent years, driven by the increasing reliance on vehicle dynamics simulations in certification procedures.
However, achieving a thorough understanding of the wheel-rail friction conditions and their impact on vehicle dynamics and wear cannot rely solely on experimental testing or numerical simulations; rather, it requires an integrated approach in which both activities play a fundamental role. In this work, the influence of the friction coefficient on vehicle running behaviour and wear is investigated. The numerical analysis is supported by on-track tests, with the aim of defining realistic values for simulations in the case of dry wheel-rail contact and when friction modifiers are applied. Section 2 describes the experimental campaign carried out to measure the

friction coefficient under both dry and lubricated track conditions. Section 3 then presents the simulated scenarios, while Section 4 reports the results obtained as the friction conditions vary. Finally, concluding remarks are provided.

2. Field measurements of friction coefficient

A series of experimental activities were carried out with the aim of estimating the friction coefficient at the wheel-rail contact under different operating conditions and at different locations along the track. Measurements were conducted along a curve with a radius of 400 meters, evaluating both the friction coefficient on the top of the rail and on the gauge face of the outer rail (i.e., at the contact between the wheel flange and the lateral surface of the rail head). The tests were repeated over three track sections spaced 100 m apart (see the scheme in Figure 1a). Test were carried out under dry track conditions and after activating a wayside lubrication system aimed at reducing the friction coefficient at the flange-rail interface on the outer rail. Thus, in addition to estimating the friction coefficient in dry conditions, a further objective of the tests was to assess the effectiveness of the lubrication system and to determine its range of influence along track.



(a) (b)

Figure 1: On track friction measurements: (a) scheme of the test sections and (b) trackside lubrication system installed on the outer rail.

The tests were carried out using the portable tribometer developed by Rivelin Rail [8,9] (see Figure 2). The portable tribometer is magnetically clamped to the rail head and uses a freely rotating steel wheel as the counterbody. The wheel rolls along a 300 mm section of rail at a speed of 200 mm per second. The wheel is set at a yaw angle of 30 mrad relative to the rolling direction, generating transverse creepage at the contact interface. The tangential force acting on the wheel is measured with a sampling frequency of 100 Hz. During the measurements, a normal load of 100 N is applied. For each test run, the friction coefficient is determined as the ratio between the tangential and the normal forces and by averaging all recorded data points. Additional averaging is then performed over three repeated measurement series for each test section.

(a) (b)

Figure 2: Portable tribometer designed by Rivelin Rail [8,9]: measurements on the (a) top of the rail and (b) gauge face.

The test campaign was conducted over two days. On day 1, dry friction measurements were carried out on the outer rail on both the top of the rail and the gauge face. Immediately after day 1, the trackside lubrication system was activated, and the tests were repeated after 30 days. This time interval corresponds to the period recommended by the system manufacturer to reach steady-state conditions along the entire curve, as the grease is applied at each train passage and progressively distributed along the track by the train wheels across the full measurement section.
The measurement results are presented in Figure 3. The mean value of the friction coefficient obtained during each measurement and for each measurement section is reported. Figure 3a shows the results of the measurements on dry track (day 1), while Figure 3b presents the results of the measurements carried out after the activation of the track-side lubrication system (day 2). In the case with lubrication, measurements carried out on the head of the inner rail (which is in dry conditions since the lubrication system is installed on the outer one) are also reported to assess the influence of lubrication system in reducing the friction coefficient. The environmental conditions during the two test days were comparable, as both were characterized by clear skies and temperatures between 25 and 30 °C throughout the tests. From the results under dry rail conditions on day 1, the friction coefficient measured on the rail head falls within the range 0.45-0.60. However, during the tests carried out on day 1 (without the trackside lubrication system), a lower friction coefficient was observed on the gauge face of the rail, within the range 0.15-0.25 across all measurement sections. This difference has been attributed to the presence of on-board flange lubrication systems installed on the trains operating along the tested curve section. This hypothesis is further supported by the presence of an oily surface detected to the touch on the gauge face of the rail.
The tests carried out on day 2 showed a friction coefficient on the inner rail (which was not contaminated by the lubricant) comparable to that obtained on day 1. This further confirms that the environmental conditions during both test days can be considered consistent. Tests on the outer rail made it possible to investigate the effect of the trackside lubrication system. The lowest friction coefficient values were observed at the gauge face, with average values between 0.05 and 0.15. This is an expected result, as the system is designed to apply grease near the potential flange

contact region. However, a similar effect was also observed in the friction coefficient measured on the top of the outer rail head, which ranged between 0.10 and 0.15 across all test sections. Therefore, although the lubrication system is primarily intended to reduce friction at the flange contact, the grease appears to spread along the track and across the rail cross-section, also contributing to a reduction in the adhesion available at the tread.

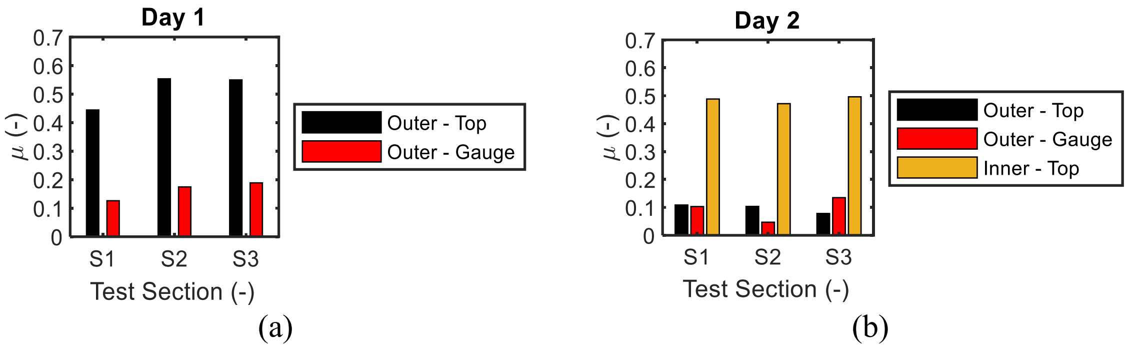


Figure 3: Results of the friction measurements: (a) day 1, dry track and (b) day 2, effect of lubrication on the outer rail.

3. Vehicle dynamics simulations

To analyse the effect of different wheel-rail contact condition on the running behaviour of a railway vehicle during curve negotiation, a typical passenger train, widely adopted on the Italian railway network, is modelled with SIMPACK multibody simulation software. The trainset includes three cars, with two Jacobs bogies shared between the end cars and the central car, all modelled as rigid bodies (see Figure 4). The wheelsets are connected to the bogies through a primary suspension system composed of springs and dampers arranged in parallel. The carbodies are connected to the bogies through air springs including dampers in parallel, bumpstops, an anti-roll bar, and yaw dampers. Additionally, inter-car yaw dampers connect the end carbodies to the central one.

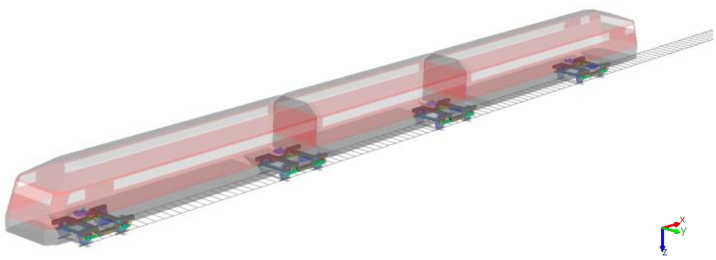


Figure 4: The multibody vehicle model.

Track flexibility is considered using the default SIMPACK co-following track model, whereas the track irregularities are not included in the simulations since the aim of this investigation is solely to highlight the effect of the different adhesion conditions

in curve on curving behaviour and wear. The wheel and rail profiles adopted in the simulations are the nominal S1002 and UIC60 profiles, respectively. For all simulations presented, a multi-Hertzian model is adopted to solve the normal contact problem while FASTSIM algorithm is used to compute the tangential contact forces [10]. Finally, simulations are carried out considering a curved track with the radius of 400 m and the track cant equal to 130 mm. The vehicle runs at 87 km/h, resulting in a non-compensated lateral acceleration (a_{nc}) equal to 0.6 m/s^2.
Three cases are analysed to investigate the effect of different adhesion conditions at the wheel-rail interface:

- The first case compares two different track conditions with a constant friction coefficient and equal to μ = 0.30 and μ = 0.60.
- The second case simulates the presence of a flange lubrication system. This scenario is based on the friction coefficient measured on the gauge face during friction measurements. The friction coefficient on the wheel tread is set equal to μ = 0.60 while the one at the wheel flange is progressively reduced up to μ = 0.18 as shown in Figure 5.

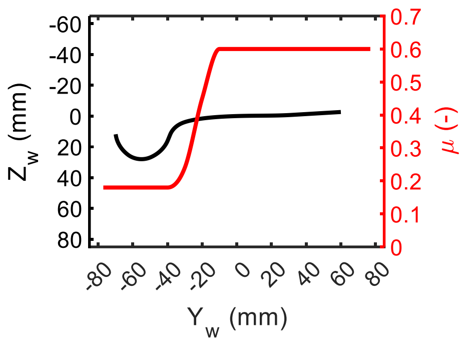
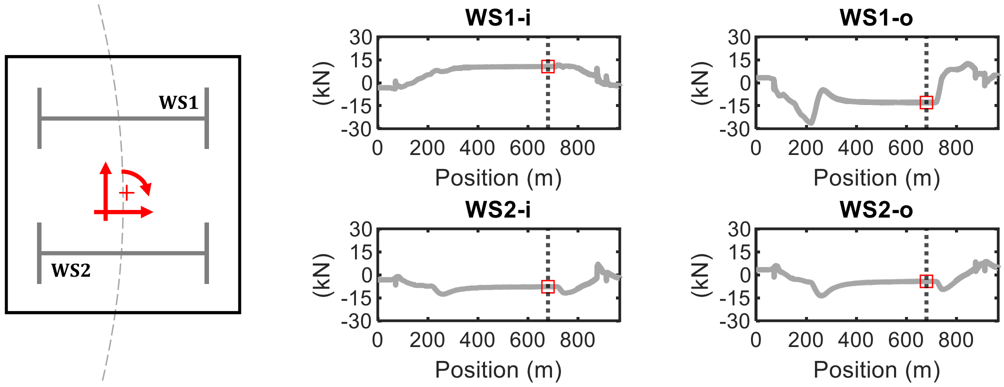


Figure 5: Modelling the effect of on-board flange lubrication system.

- The third case represents the effect of the activation of the track-side lubrication system on the outer rail in curve. In this scenario, the friction coefficient on the inner rail is kept at μ = 0.60 (both on the tread and the flange), while a reduced friction level μ = 0.10 is imposed on the outer rail along the entire rail profile.

4. Results

The results of the simulation scenarios described in Section 3 are now presented. The analysis focuses on the leading bogie of the trainset and is carried out by considering the angles of attack and lateral displacements of the leading and trailing wheelsets of the leading bogie, together with the same kinematic quantities of the leading bogie frame. In addition, the longitudinal and the lateral contact forces at the four wheels are analysed. These parameters are evaluated after the steady-state curving condition is obtained, despite the vehicle dynamics is simulated also including curve entrance and curve exit. An example of the lateral contact forces on the wheels during the curve negotiation (μ = 0.6) is reported in Figure 6b. The steady state curving condition is highlighted with a square mark (□). The leading and trailing wheelsets are named "WS1" and "WS2," respectively, while the letters "i" and "o" denote the inner and outer wheels.



(a) (b)

Figure 6: Analysis of the leading bogie behaviour in curve: (a) schematic overview and (b) lateral forces and steady-state identification (□).

As far as wheel and rail wear prediction is concerned, several methods have been proposed in the literature, generally based on two main approaches, namely semi-empirical formulations and tribological models integrated within multibody dynamics simulations [11,12]. Among the available approaches, wear laws based on dissipated energy are widely adopted. These methods rely on the evaluation of the wear number T_γ introduced by Elkins et al. [13], which is computed as the sum of the dot product of creep forces and creepages:

$$T_\gamma = |F_x \nu_x| + |F_y \nu_y| + |M_z \phi_z| \tag{1}$$

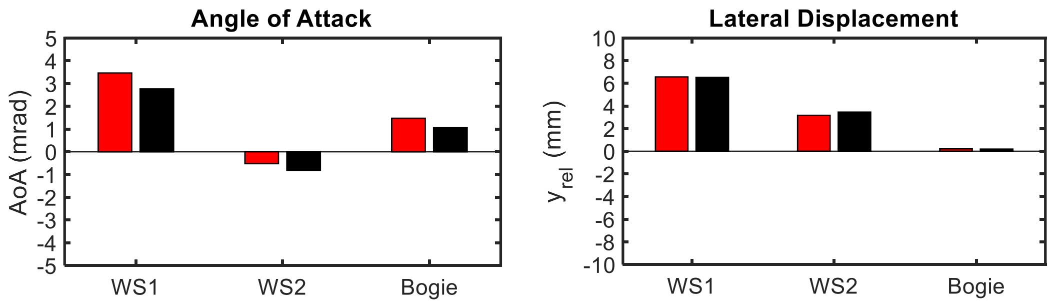
where, F_x, F_y and M_z are respectively the longitudinal creep force, the lateral creep force and the spin moment on the contact patch area and ν_x, ν_y and ϕ_z are respectively the longitudinal, lateral and spin creepages.

4.1. Effect of the friction coefficient

On-track measurements carried out in previous research (see [1,14]) indicate a significant variability of the friction coefficient under dry conditions, mainly driven by environmental factors (e.g. temperature and humidity). Accordingly, the first investigation is carried out considering two friction coefficient values that can be obtained in dry track conditions, namely μ = 0.30 (■) and μ = 0.60 (■). In fact, despite it is common to carry out vehicle dynamics simulations with μ = 0.30-0.40, the on-track tests presented in Section 2 reveals friction values systematically higher than

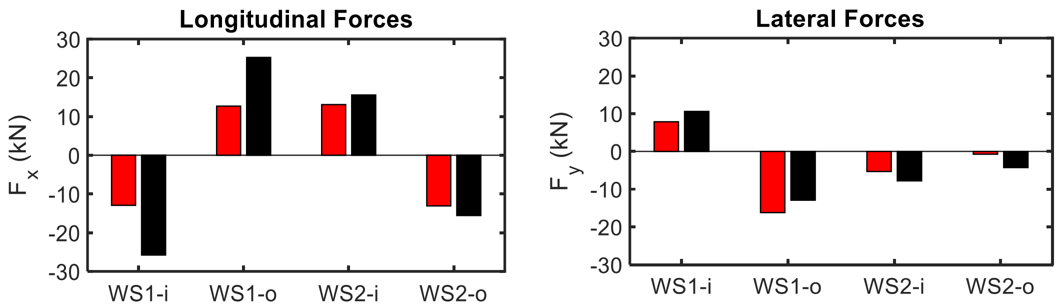
0.45-0.50. Thus, a first analysis is needed to assess the potential impact of such a variation in the friction coefficient on the same curve.

As shown in Figure 6, the angle of attack (AoA) and the relative displacement of both the leading and trailing wheelsets, as well as that of the bogie, vary depending on the friction coefficient adopted in the multibody simulations. For a friction coefficient μ = 0.6, the AoA of the leading wheelset is reduced, whereas that of the trailing wheelset increases, resulting in an overall reduction of the bogie AoA (see Figure 6a). A similar result is obtained for the relative lateral displacement (see Figure 6b). This behaviour is a consequence of the different longitudinal and lateral forces exchanged between the wheels and the rails, as illustrated in Figure 8. In general, a reduction in lateral forces is expected for lower friction coefficients, assuming the same bogie configuration relative to the track. However, an increase in the friction coefficient leads to higher longitudinal forces on all wheels, which slightly alters the relative position of the wheelsets with respect to the track. In particular, the increase in the longitudinal forces acting on the left and right wheels induce an additional deformation of the primary suspension in longitudinal direction, which in turn reduces the wheelset AoA [1]. This, in turn, decreases the lateral creepage and the lateral creep forces exchanged between the leading wheelset and the rails. Conversely, the trailing wheelset experiences an increase in AoA, resulting in higher lateral forces. However, the overall increase in longitudinal forces generates a positive yaw moment on the bogie, which slightly reduces its AoA with respect to the track.



(a) (b)

Figure 7: Curving behaviour of the leading bogie: (a) AoA and (b) relative lateral displacement ((■) μ =0.3 and (■) μ=0.6.).



(a) (b)

Figure 8: (a) Longitudinal and (b) lateral contact forces: (■) μ =0.3 and (■) μ=0.6.

In general, the highest wear number levels are observed on the leading wheelset, with the leading outer wheel exhibiting the most critical condition. Although an increase in the friction coefficient contributes to a reduction in both the angle of attack (AoA) and the magnitude of the lateral forces acting on the leading wheelset, this effect is not leading to a reduction in the wear number T_γ. On contrary, the total energy dissipated at the contact is significantly amplified due to the increase in longitudinal forces (see Figure 9). This finding confirms that, despite a marginal improvement in the curving behaviour of the bogie, a higher adhesion limit has an overall detrimental impact on wheel-rail wear, particularly with regard to the wear number T_γ on the leading wheelset.

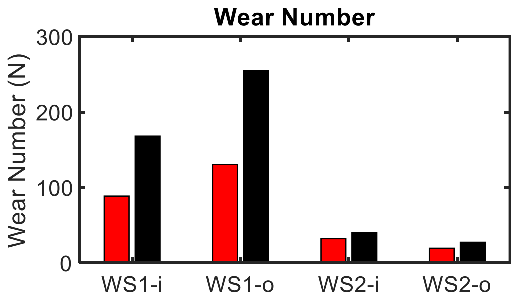


Figure 9: Total wear number: (■) μ =0.3 and (■) μ=0.6.

4.2. *Effect of friction modifiers*

The previous section showed that the choice of friction coefficient in vehicle curve-dynamic analysis can lead to considerable variability in the results, emphasizing the need to account for actual contact conditions whenever possible to accurately predict vehicle behaviour in curves and the forces exchanged with the track.
This section presents a focused analysis of real wheel-rail adhesion conditions, based on the experimental measurements reported in Section 2 related to the adoption of a wheel flange lubrication system mounted on the vehicle and track-side lubrication system. The first objective is to assess the effect of local variations in the friction coefficient on running dynamics. The second is to evaluate the potential benefits of the use of friction modifiers on contact forces and wear. For this purpose, the wear number of the wheels on the first bogie is analysed, as before, while also monitoring the number and positions of wheel-rail contact points. The reference case with μ = 0.60 presented in the previous section is adopted as a benchmark.
The AoA and the relative lateral displacement of the leading bogie and its wheelsets are shown in Figure 10. The dry track conditions (■) are compared to the case simulating the presence of the on-board flange lubrication system (■) and the use of a track-side lubrication device (■). The reduction in the friction coefficient at the wheel-rail interface leads to an increase in the AoA of the leading wheelset and a reduction in the AoA of the trailing one. Overall, the AoA of the bogie increases.

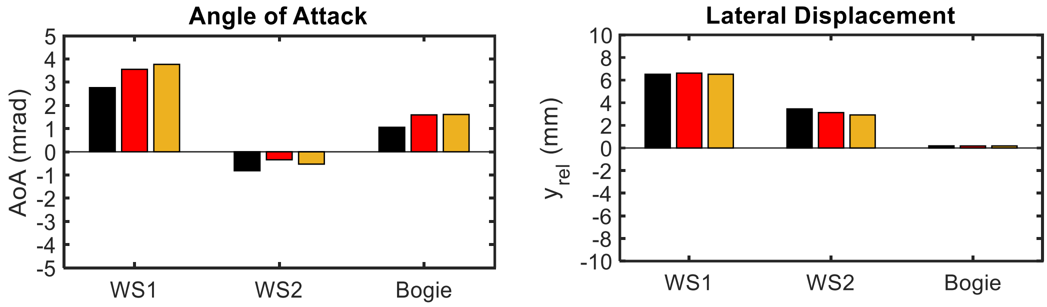


Figure 10: (a) AoA and relative (b) lateral displacement on the leading bogie: (■) dry track, μ = 0.6, (■) effect of flange lubrication and (■) effect of track-side lubrication.

The results are similar to the case investigated in the previous Section, since the reduction of the friction coefficient results in a significant decrease in the longitudinal forces acting on the leading wheelset (see Figure 11a), which in turn reduces the overall self-steering capability of the bogie, leading to a larger angle of attack. This condition results in an increase in the lateral forces acting on the leading wheelset, particularly on the outer wheel (see Figure 11b).

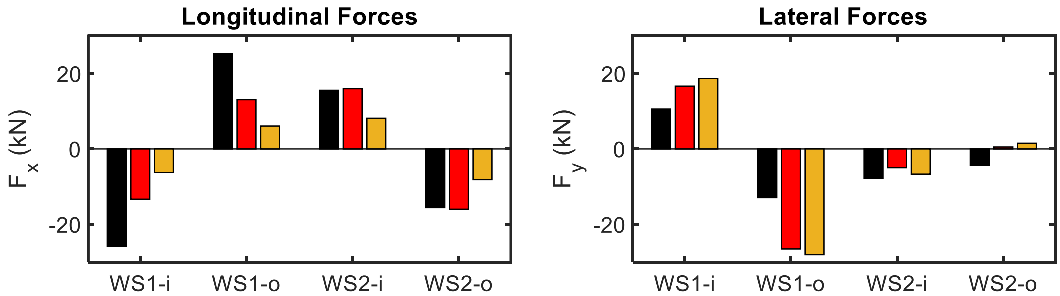


Figure 11: (a) Longitudinal and (b) lateral contact forces: (■) dry track, μ = 0.6, (■) effect of flange lubrication and (■) effect of track-side lubrication.

As far as the wheel-rail wear is concerned, the use of friction management systems leads, as expected, to a reduction in the wear number T_γ. This is reported in Figure 12 for the four wheels of the leading bogie under the different friction conditions. The use of the flange lubrication system (■) results to be beneficial considering the wear number of all the wheels, with the most significant T_γ reduction on the leading wheelset, which was the most critical one in the dry track case. An even greater reduction is achieved with the application of the trackside lubrication system on the outer rail (■). In this case, the wear number of the leading outer wheel is drastically reduced also compared to the flange lubrication cases. This is due to the lower value of the friction coefficient at the contact between the wheel and the rail (μ = 0.1 on the whole profile). A similar, although less pronounced, trend can be observed for the other wheels.

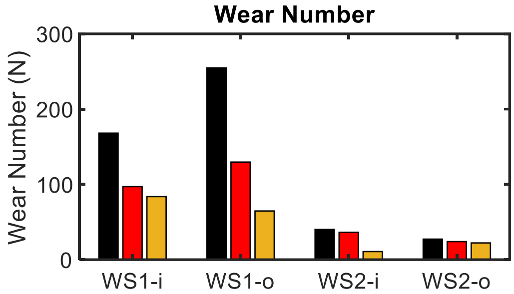


Figure 12: Total wear number

The position of the contact points between the wheels and the rails was also investigated (see Figure 13). The main difference is observed on the leading outer wheel, where two simultaneous contact points are activated in full curve when the presence of the track lubrication system is included in the simulation. This shows that variations in the friction coefficient due to the use of friction modifiers, can influence the evolution of wheel and rail profiles wear over time by modifying the number and/or the position of the contact patches.

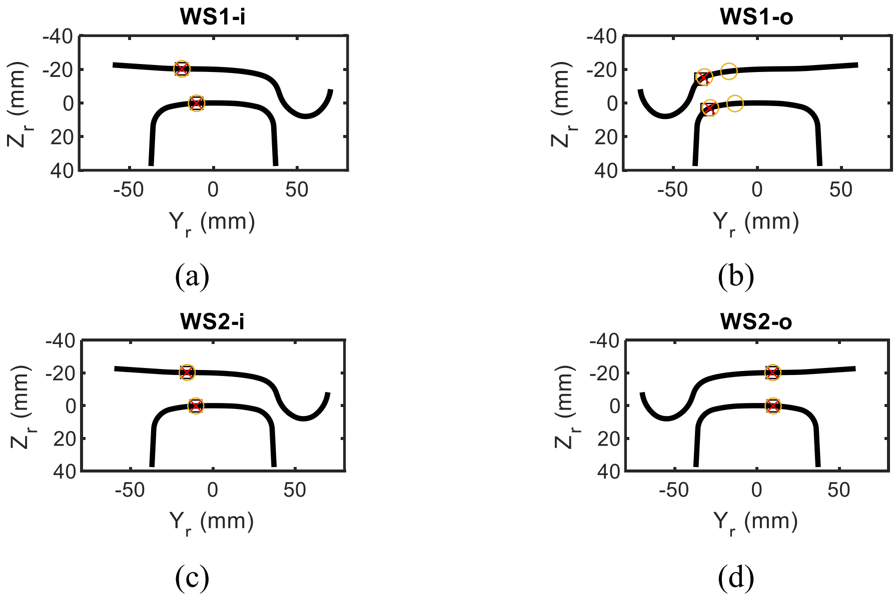


Figure 13: Position of the wheel-rail contact points: (a) leading left, (b) leading right, (c) trailing left and (d) trailing right ((□) dry track, (**x**) flange lubrication system and (**o**) track-side lubrication).

5. Conclusions

Despite the extensive literature on wear prediction, relatively limited attention has been devoted, so far, to the impact of the use of friction modifiers on vehicle running behaviour in curve. This paper investigated the role of the friction coefficient in wheel-rail interaction under curving conditions, focusing on both vehicle dynamic behaviour and wear. A multibody model of a passenger train is adopted to analyse

different adhesion scenarios, including dry and lubricated conditions derived from on-track friction coefficient measurements with a portable rail tribometer.
The availability of friction coefficient values measured in real operating conditions represents a key strength of this research work, allowing the definition of realistic adhesion levels and improving the reliability of the numerical simulations. In particular, the experimental campaign enabled the identification of a significant variability of the friction coefficient under dry and lubricated conditions, which was directly incorporated into the simulation framework.
The numerical simulations clearly show that the friction coefficient plays a key role not only in wear processes but also in governing vehicle running dynamics. Reduced friction levels were found to reduce the self-steering capability of the leading bogie, resulting in increased angles of attack and higher lateral contact forces. At the same time, the application of friction modifiers significantly reduces the wear number, especially for the leading outer wheel, confirming their effectiveness in mitigating wear in severe curving conditions. The trackside lubrication provides the most beneficial effect in severe curving conditions, particularly for the outer wheel of the leading wheelset, which is typically the most critical in terms of wear. It should be noted that this work represents a preliminary investigation. Future developments will address a broader range of operating conditions and friction management scenarios, as well as the adoption of more advanced wear prediction models. Overall, these findings highlight a fundamental trade-off between adhesion, curving performance, and wear, which must be carefully investigated to support informed design decisions regarding the application of friction modifiers.

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