Development of an integrated design methodology for a new generation of high performance rail wheelsets

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Abstract

An integrated design methodology, based on the use of numerical simulation tools (MultiBody-FEM), is being established and applied for the development of innovative railway wheels and axles having outstanding reliability performances. The advanced modelling of the dynamic vehicle mission, the service loads determination taking into account rail-wheel contact and the axle flexibility, along with a procedure to compute the load spectra for a number of representative running conditions encountered by the wheelset during standard service, offer a comprehensive framework for the early design stages of an innovative, high performance wheelset within the European Community funded HIPERWHEEL project.

1. Introduction

Wheelset engineering is facing new severe specifications on wear (to decrease wheel reprofiling operations), weight (to decrease aggressiveness on the track) and reliability, aiming at improving running safety and reducing significantly total life cycle costs.

At the same time, the railway vehicle mission is changing due to:

- the need to operate the rolling stock both on tracks with low radius curves and on high speed tracks;
- the increase of commercial speeds on conventional tracks (allowed by tilting technology, as well as by increased bogie performances);
- the decrease of rail tracks quality due to maintenance reduction.

Therefore, new design methodologies are needed to meet the above mentioned requirements. In particular, in the modern development of new wheelsets, it is necessary to use, in an integrated way, advanced numerical tools for the mathematical modelling of railway vehicle dynamics, suitable models for the prediction of the damage undergone by the wheelset assembly under mission loads, and appropriate test rigs for the final design validation.

This paper illustrates the interim results achieved within the European Community funded project *"HIPERWHEEL"*, which features the participation of major European research institutes, railway vehicle components suppliers and railway operators. The main goals of this 4-years project, started in the early year 2000, are, among others:

- to determine, via multi-body modelling of the railway vehicle dynamics, the mission loads acting on the wheelset, taking into account the variety of operational manoeuvres and track conditions;

- to develop an integrated CAE procedure for wheelset durability assessment, which takes into account all damage mechanisms experienced by the wheelset assembly during service conditions (i.e. metal fatigue, rolling contact fatigue, wear and fretting);
- to develop a numerical methodology for assessing the vibro-acoustic behaviour of wheels;
- to specify the wheelset test conditions (bench test simulation of track conditions) to evaluate the durability of the assembly and its vibro-acoustic performance;
- to design, using the new developed methodologies, innovative wheelset demonstrators, having outstanding performances in terms of low weight, durability and low noise emission.

The present paper deals, in particular, with the set up of a Multi-Body (MB) model to simulate the dynamic behaviour of a railway vehicle running in tangent and curve track, with the flexible axle effects on contact forces and with the numerical techniques to generate representative load spectra for the wheelset design.

The MB model, used to estimate wheel rail contact forces under realistic operating conditions, has been set up referring to a long distance / high speed train which can be used for both high speed service (e.g. in Italy between Florence and Rome) and for service on standard lines at high values of cant deficiency.

The MB vehicle model has been validated comparing the numerical simulation results with the available measurements concerning the dynamic behaviour of the vehicle in tangent track and curve.

2. Multi-Body modelling in ADAMS/Rail environments

A complete model of a passenger coach has been developed in ADAMS/Rail 10.1. Three different sub-assemblies compose the model: the carbody, the front bogie and the rear bogie.

The carbody and the bogie frame are treated as rigid bodies and defined giving their mass characteristics, which are obtained taking into account the presence of auxiliary elements.

The front and rear bogies are equal except for the position of the yaw dampers, which is symmetrical with respect to the middle of the carbody.

The single bogie is basically composed by the bogie frame, two wheelsets, suspensions and dampers connecting the bogie frame to the wheelsets and to the carbody. Masses of the components constituting the bogie such as auxiliary elements, suspensions and dampers, are reduced to the bogie frame except those of the wheelset and those of arms and axleboxes connecting bogie and wheelsets.

For the wheelsets two alternative schematisations have been defined, in which they are considered either as rigid or as flexible bodies.

Primary and secondary suspensions are represented with elastic linear elements while the vertical and lateral dampers are treated as viscous non-linear elements.

The deformability of the bogie frame-wheelset connection is represented by the primary suspension vertical springs and by arms and elastic bushing elements representing the real bogie-axlebox-wheelset connections. The yaw damper is represented with a viscous non-linear element.

The carbody-bogies bumpstops, anti-roll bar and traction forces were also modelled, while the possible presence of a car body tilt system was not considered, as it is believed that this feature has negligible effect on the calculation of wheel-rail contact forces, which are the main goal of the model.

Figure 1 and figure 2 show the complete model of the coach and the bogie as represented in the ADAMS/rail environment.

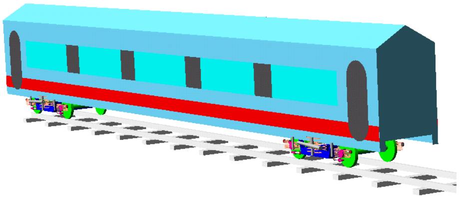


Figure 1: ADAMS/Rail coach model

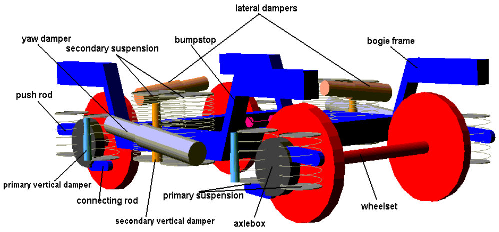


Figure 2: ADAMS/Rail bogie model

In the model the active lateral suspension (hold off system) was introduced. The HOD is a slow-active pneumatic positioning device, which reduces quasi-static lateral carbody displacements to avoid bumpstop contact during curve manoeuvres reacting the centrifugal force. Its force is proportional to lateral bogie non-compensated acceleration.

In the real vehicle the signal of the bogie accelerometer is processed and elaborated by an Electronic Central Unit and from these information the HOD force command is given to pneumatic actuators acting between the carbody and the bogie.

In the MB model a simplified HOD device was introduced. The lateral bogie non-compensated acceleration was conditioned by a transfer function to filter high frequencies and the output acceleration is introduced in a proportional control (ECU). The force is delayed in order to reproduce the pneumatic system and is actuated between the carbody and the bogie.

3. Validation of the Multi-Body model with experimental data

Several validation activities were performed in order to verify the accuracy of the model in reproducing the dynamic behaviour in tangent track, with particular regard to the capability of the model to reproduce the stability threshold of the vehicle, and in curve, considering the steady-state and dynamic components of wheel-rail contact forces during curve negotiation.

The experimental data used for performing the validation of the multi - body model are relative to measurements of vehicle stability threshold, vertical and lateral contact forces, vertical and lateral bogie accelerations and carbody vertical acceleration. The data referring to tangent track running were measured on a high speed line, at medium to high vehicle speeds and with different vehicle configurations (mainly for the vehicle without or with yaw dampers). The data concerning curve negotiation refer to the running of the vehicle through curves of different radii, with cant deficiency ranging from 0.5 to 1.5 m/s2.

3.1 Vehicle Stability threshold analysis

Two different models are considered in order to reproduce the contact forces: the tabular model, which leads to low computation time but actually allows to describe only one contact point, and the general model, which allows to consider multiple contact points. The capability to consider multiple contact points is very important when dealing with problems such as profile wear or local fatigues besides a more accurate description of contact phenomena.

Starting from the analysis of the behaviour in tangent track, two different vehicle configurations are considered that is a vehicle with and without yaw dampers. The comparisons performed on the vehicle without yaw dampers allow to define a sort of “intrinsic” stability threshold of the vehicle, which is mainly influenced by some stiffness and geometric parameters of the bogie. On the other side, the comparisons on the vehicle with yaw dampers allow to verify that the model is able to predict the improvement in vehicle stability produced by the use of this very critical passive control device.

Figures 3 and 4 report the variance of the lateral bogie acceleration at the axleboxes as function of vehicle speed for a vehicle without and with yaw dampers. This quantity can be assumed in order to assess vehicle stability (UIC fiche 518). As reference values the experimental critical speed without yaw dampers and the corresponding variance of axle box acceleration are assumed.

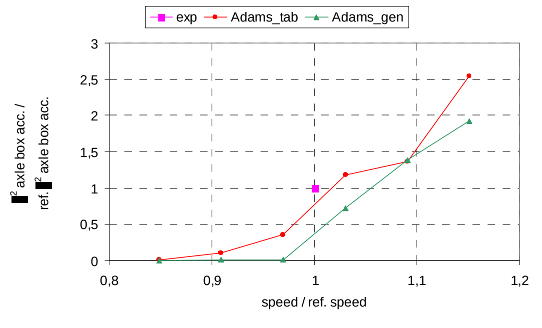


Figure 3: axlebox acceleration variance (without yaw dampers)

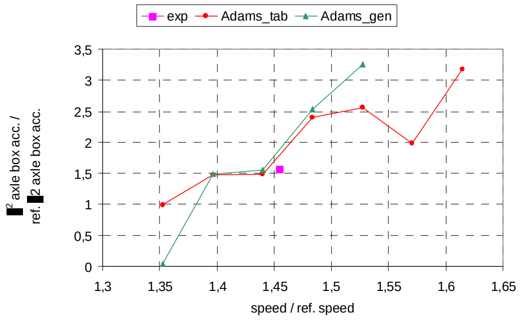


Figure 4: axlebox acceleration variance (with yaw dampers)

For both configurations of the vehicle the model is able to reproduce the transition from stability to instability conditions apart from the contact model considered.

3.2 Validation of MB model in tangent track mission

The comparison between experimental data and numeric results in tangent track was performed for a 1Km line, running at 210Km/h in stability conditions. The irregularities of the track were obtained using ORE formulation. Numerical results and experimental data were compared in terms of lateral and vertical acceleration for the bogie (Figure 5-6) and in terms of vertical acceleration for the carbody (Figure 7):

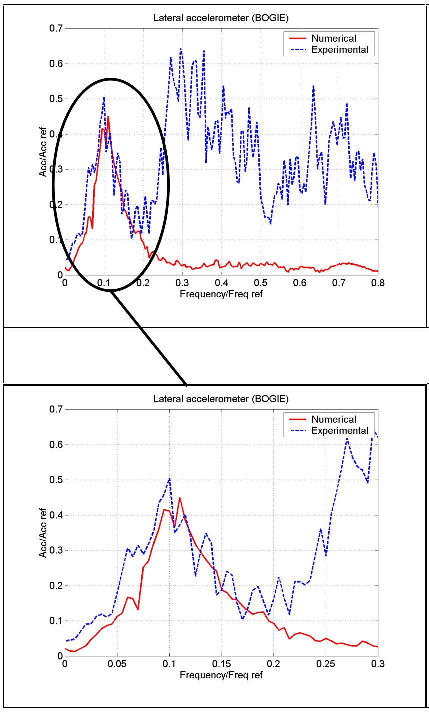


Figure 5: Bogie lateral acceleration

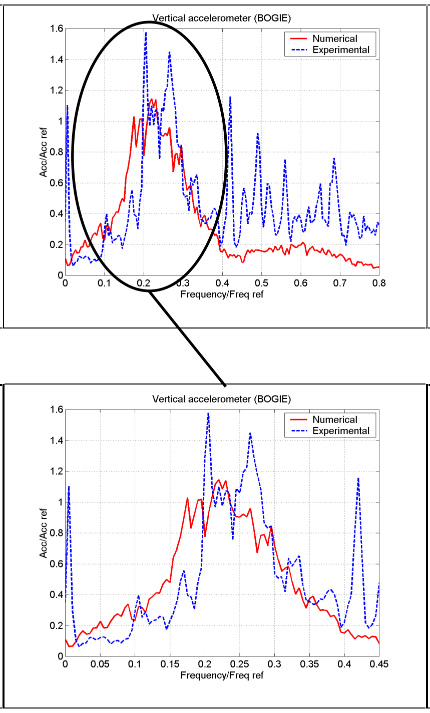


Figure 6: Bogie vertical acceleration

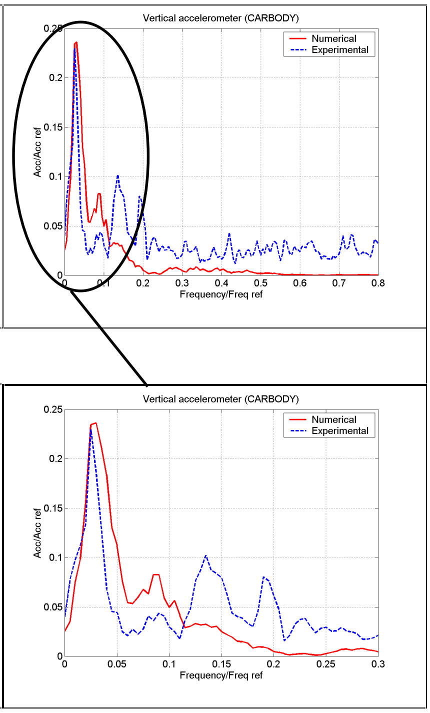


Figure 7: Carbody vertical acceleration

The differences between numerical results and experimental data are very small and some differences (especially at higher frequencies) could be due to the following factors:

- track and wheel profiles of the testing vehicle are unknown, so in numerical simulation standard profiles are used (UIC60 and S1002);
- the irregularities are determined stochastically (ORE defects);
- the inertia and the position of the CM of the carbody were not measured in the operative conditions;
- in numerical simulations a single coach was modelled;
- real position of the accelerometers was unknown;
- flexibility of the carbody and others structural elements was neglected.

It can be concluded that the Multi-Body model can be used to predict the dynamic behaviour of the vehicle and it can be used to evaluate the forces acting on the wheelsets in tangent track.

3.3 Validation of MB model in curved track mission

The capability of the model to reproduce the dynamic behaviour of the vehicle during curved track was then studied. Figures 8, 9, 10 and 11 report a comparison between experimental and numerical time histories relative to ripage forces on the wheelsets of the rear bogie and to vertical force variations on the wheel of the second wheelset of the same bogie. Only the results obtained with the general contact model are considered. The comparisons mainly concern the steady state value of contact forces in full curve.

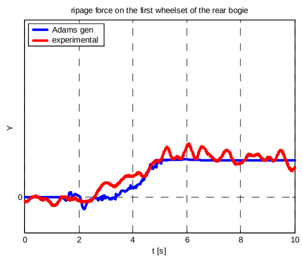


Figure 8: ripage force on the first wheelset of the rear bogie

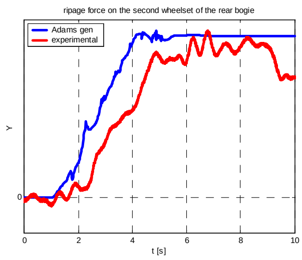


Figure 9: ripage force on the second wheelset of the rear bogie

Figure 10: vertical force variation on the left wheel of the second wheelset of the rear bogie

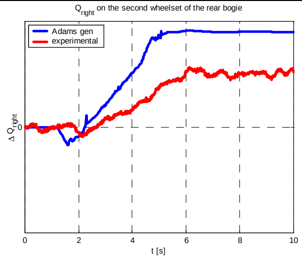


Figure 11: vertical force variation on the right wheel of the second wheelset of the rear bogie

The ripage forces are well predicted by the model. Much greater discrepancies between measured data and simulations are obtained for the vertical forces, as shown by Figures 10 and 11. These differences are at least partially due to the facts that for many geometric data in the MB model only nominal values are known. These are probably different from real ones, especially considering that nominal values are referred to a vehicle in standard operating conditions, which may differ from test train used for the measurements. Among these geometric data, the height of the carbody centre of gravity above the rail level has a very high impact on the entity of load transfers between the inner and outer wheels.

4. Introduction of flexible axle

The main aim of the numerical simulations is to generate load-histories spectra for fatigue analysis evaluations and for this reason the wheelset/axle deformability effects have been considered on forces in straight and curve missions. The forces were analysed in terms of Power Spectral Density and Range Pair counting method.

4.1 Wheelset and axle deformability modelling

The wheelset Finite Element (FE) model developed was obtained from CAD data provided by AF and it is composed of two wheels, an axle and three brake discs.

The modal analysis of the wheelset showed that wheels local modes are excited at higher frequencies than the axle frequencies and considering that low-medium frequency range are of interest, it has been decided to use only a FE model of the axle with brake disks.

The axle FE model developed was modelled by 8-nodes brick elements and the total number of nodes was approximately equal to 27000. Figure 12 show wheels and axle main mode shapes in free-free condition.

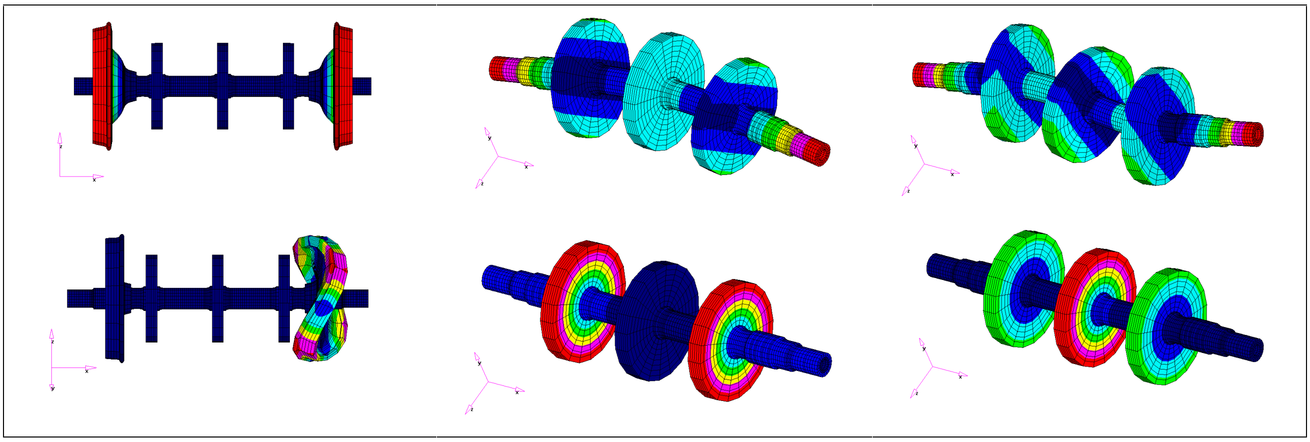


Figure 12: Wheels and axle mode shapes in free-free condition

The flexible axle was introduced in the MB vehicle model only on the front bogie in order to estimate its effects on the resulting forces acting on the wheels when simulating running conditions.

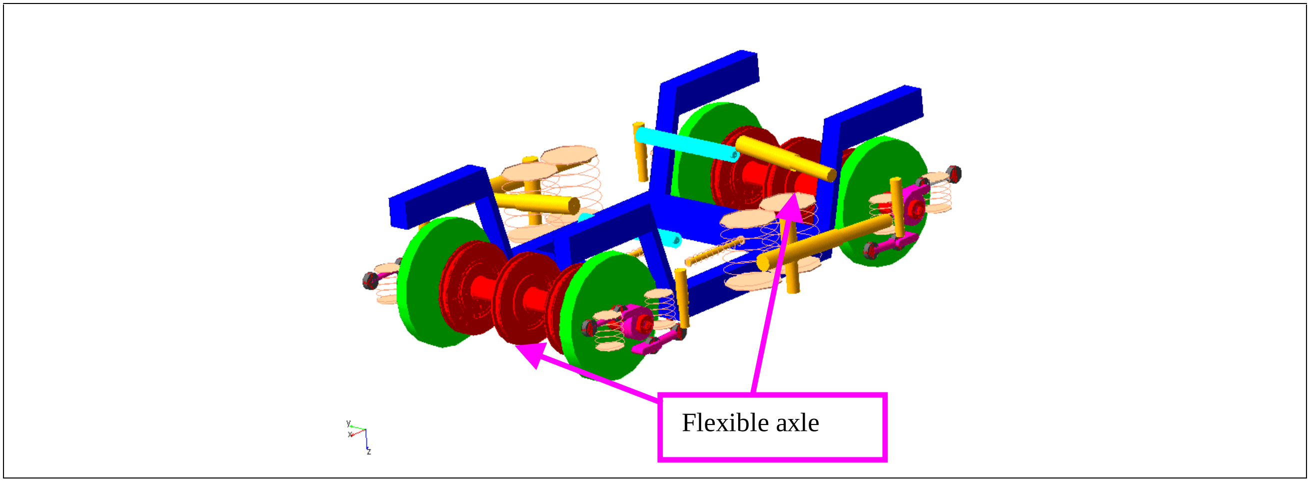


Figure 13: Bogie with flexible axles

4.2 Effects of axle flexibility

In tangent track the forces were analysed in terms of Power Spectral Density to understand dynamic behaviour of the vehicle with flexible axle and Range Pair counting method to evaluate the effects on the inputs for fatigue analysis.

Considering longitudinal contact forces the main difference in the PSD results in an higher peak amplitude at low frequencies. This effect is due to different lateral stability behaviour for the flexible wheelset in straight track (Fig. 14). This difference reflects on force spectra as a scale factor between the two curves (Fig. 15).

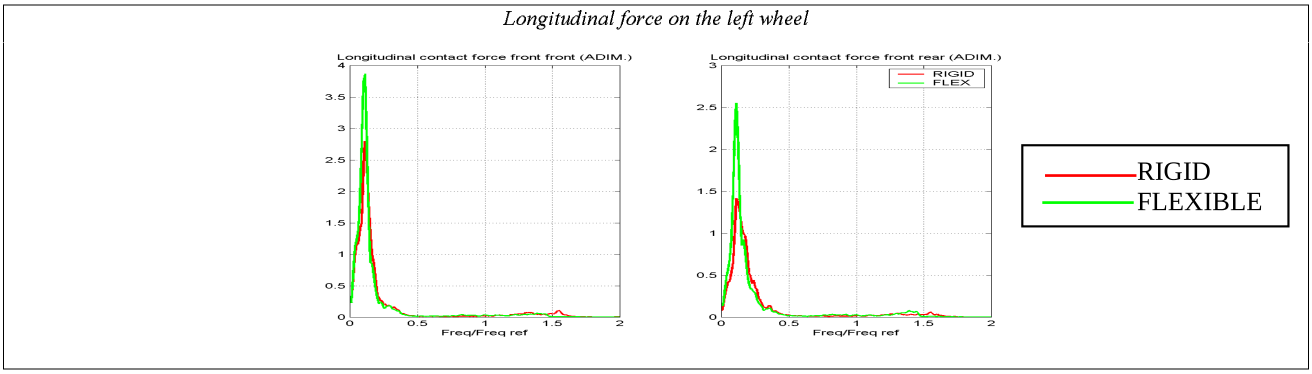

Longitudinal force on the left wheel

Figure 14: PSD spectrum

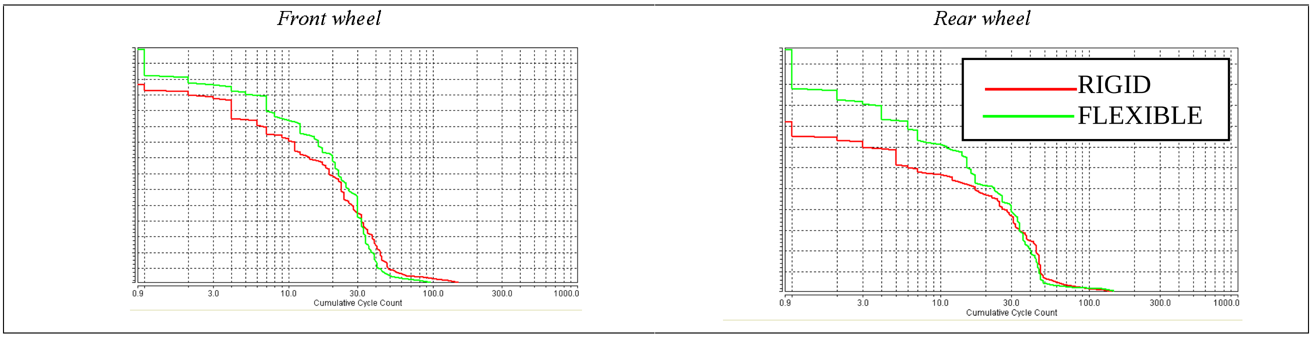

Front wheel *Rear wheel*

Figure 15: Range Pair spectrum

With regard to lateral contact forces, the main differences in the PSD are higher peak amplitude at low frequencies (as for longitudinal input) and the excitation of axle flexible modes due to input displacements in lateral direction at high frequencies (no input after 60Hz) (Figure 16). As a consequence the shapes of the spectra appear to be quite different (Figure 17).

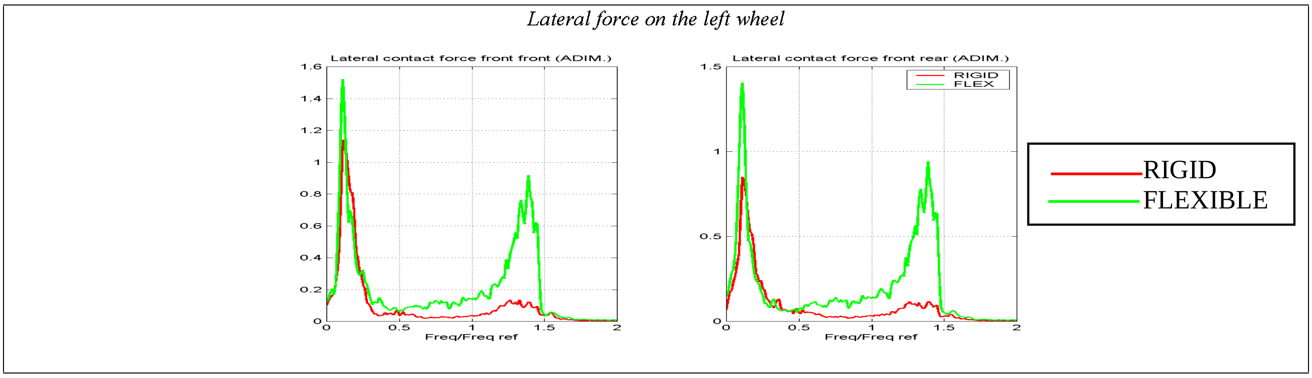

Lateral force on the left wheel

Figure 16: PSD spectrum

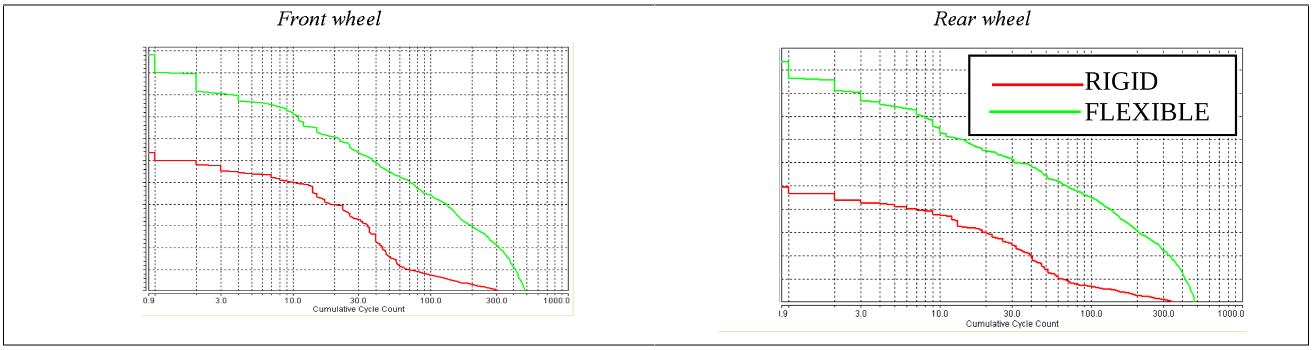

Front wheel *Rear wheel*

Figure 17: Range Pair spectrum

For vertical forces the main difference in the PSD is a different amplitude at high frequencies. This effect is a consequence excitation of FE axle flexible modes due to the displacements in vertical direction imposed by track irregularity. Also in this case the difference appears as a scale factor between the forces spectra.

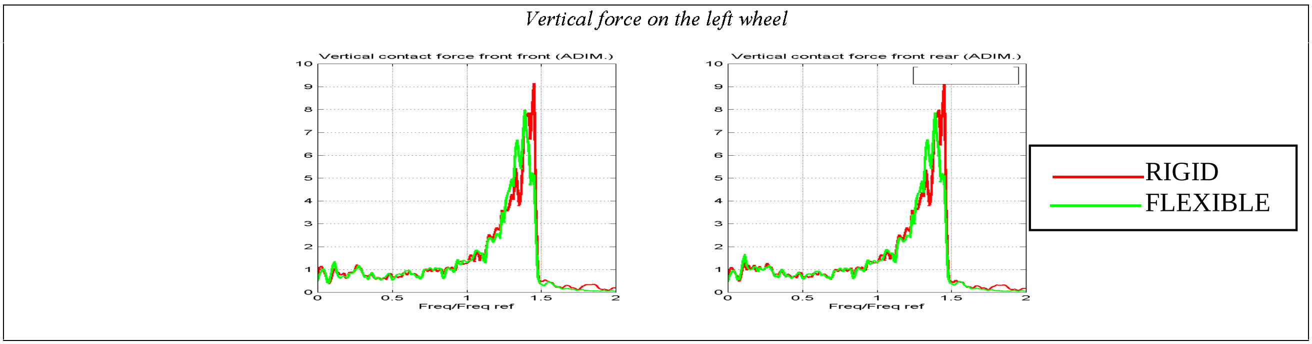

Vertical force on the left wheel

Figure 18: PSD spectrum

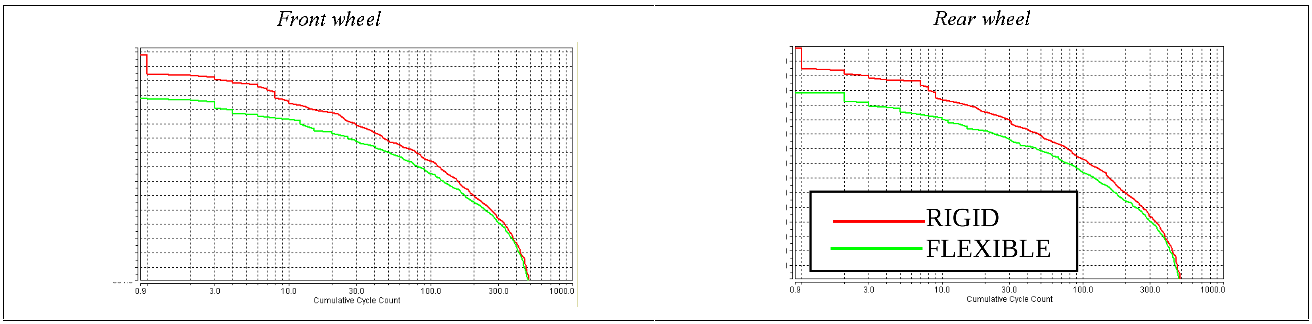

Front wheel *Rear wheel*

Figure 19: Range Pair spectrum

Table 1 summarises the principal effects of axle flexibility on the forces acting on the wheels in tangent track mission:

	PSD	**Range Pair**
Longitudinal force	Higher value of the peak amplitude at low frequencies	Scale factor at low cycles and high amplitudes
Lateral force	Higher value of the peak amplitude at low frequencies and excitation of high frequencies	Different shape of the spectrum
Vertical force	Different excitation of high frequencies	Scale factor at low cycles and high amplitudes

Table 1

In curve track axle flexibility has no effects on the steady-state components of the contact forces.

5. *Simulation of reference mission and prediction of load spectra*

Besides the definition of the rail vehicle multi-body model, a procedure to compute the load spectra for the different components of wheel-rail contact forces has been set-up.

The procedure is schematically resumed by Figure 23. As a first step, based on the available service measurements, a number of representative running conditions are identified for the vehicle. These are selected in order to be sufficiently representative of the different loading conditions encountered by the wheelset during standard service.

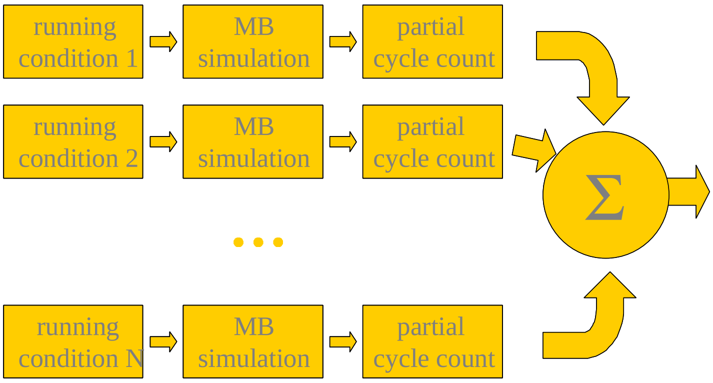


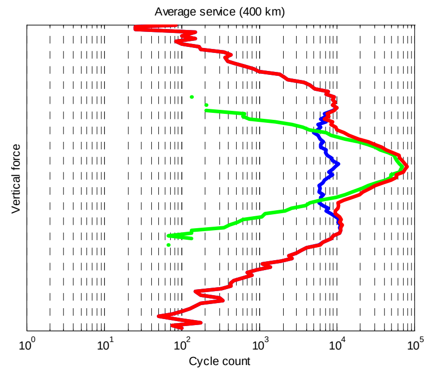
Figure 23: outline of the procedure for the numerical estimation of the contact force load spectra

A limited number of running conditions is therefore defined as representative of the whole vehicle mission profile; for the conditions with curved track, one curve to the left and one curve to the right are simulated, including reasonable lengths of entry and exit spirals. As represented in figure 23, for each condition a simulation is performed using the mathematical model described and a cycle count is performed on the different force components obtained from the simulation.

In this way, different separate load spectra are computed, one for each running condition, and the cumulative load spectrum is then obtained as a weighted sum of these “elementary spectra”. Different weights can be used in this sum, in order to reproduce different kinds of mission profile, i.e.:

- an “ordinary line” mission profile, where no speed higher than 190 km/h is allowed, and a high percentage of sharp curves is assumed;
- A “mixed” track, composed partially by an ordinary line and partially by a high speed line, in which case speeds up to 250 km/h and a higher number of curves with large radius are considered.

As an example the total load spectra obtained by this analysis for the right wheel in the front bogie leading wheelset in the case of the “ordinary line” mission profile are reported in figures 24 and 25 respectively for the vertical and lateral force components.



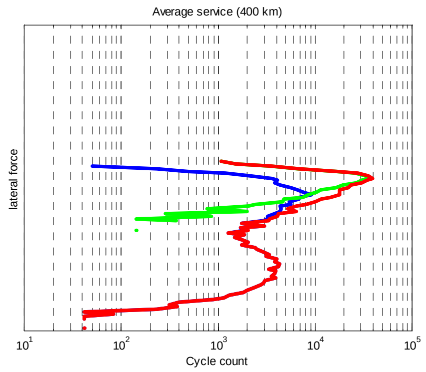


Figure 24: numerical load spectrum, vertical force

Figure 25: numerical load spectrum, lateral force

6. Conclusions

In this paper, some of the interim results of the HIPERWHEEL project, presently in progress, have been outlined. In particular, a description of an integrated procedure MB-FEM to generate the load spectra needed for the design of an innovative, high performance wheelset has been provided.

The results have shown that the mathematical model is able to reproduce with good accuracy the behaviour of the real system and in particular allows a reliable and accurate prediction of the contact forces acting on the wheel, which are the main input of the subsequent research steps.

The results have also shown that axle flexibility has an important effect on the wheel-rail forces in tangent track and so it permits to calculate more accurately the load spectra needed for durability analysis.

Besides the definition of the rail vehicle multi-body model, a procedure to compute the load spectra for a number of representative running conditions has been set-up, in order to be sufficiently representative of the different loading conditions encountered by the wheelset during standard service.

This numerical methodology is an important step within the HIPERWHEEL project, that aims at significant improvements in the design of railway wheelsets in terms of reduced noise and vibration impact, improved durability and reduction of life cycle costs.

Acknowledgements

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