PARAMETRIC MODELS OF RAILWAY SYSTEMS FOR REAL-TIME APPLICATIONS

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Abstract.

Hardware and Software In the Loop (HIL & SIL) techniques are widely used for fast prototyping of control systems, electronic and mechatronic devices. In the railway field, several mechatronic on board subsystems (for example WSP, traction controls, active/semi-active suspensions, ATP,ATC, etc.) are often tested and calibrated following HIL and SIL approach.

The accuracy of HIL and SIL tests is deeply dependent on how the simulated virtual environment approximates the real/physical experimental conditions. As the computational power available on real time hardware grows, the demand for more complex and realistic models of railway vehicles for real-time application increases.

In past research activities authors have worked with Trenitalia SPA on the implementation of simplified real time models in the Matlab-Simulink environment for several applications and in particular for a HIL test rig devoted to the type approval of WSP system (named MI6 and realized in Florence).

In the last year the research activity has been focused on the development of a three-dimensional dynamics model of a whole railway vehicle for the development of more complex applications. The paper summarize the features and the results of the study.

1   INTRODUCTION

A number of multibody models of railway vehicle has been developed by means of different dynamics simulation software, the model described in this paper has been developed with the objective of a real time implementation, in order to use it to control the actuators of Hardware In the Loop test rigs.

The authors have worked on the design and preliminary testing of a HIL test rig realized by Trenitalia SpA (named MI6) able to validate some on board subsystems (WSP, odometry units, ATP on board subsystems etc.). Figure 1 shows a block diagram of the main test rig components and their interaction.
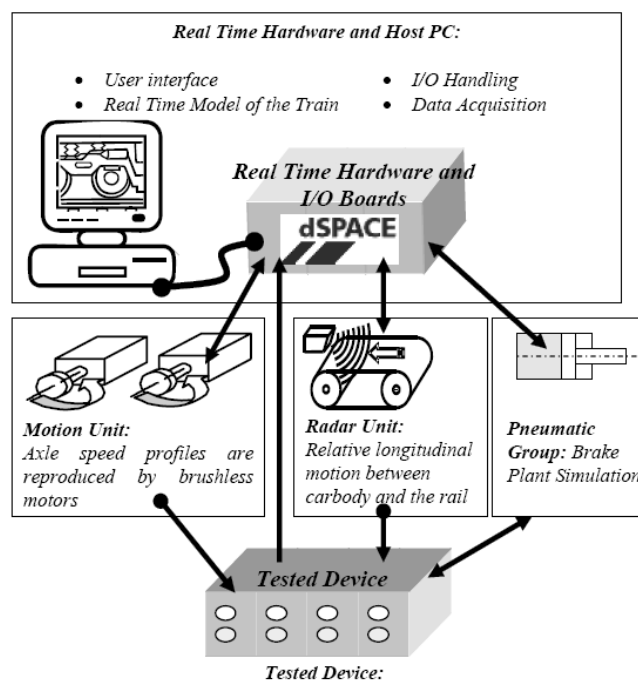

Figure 1: block diagram of the HIL test rig for the evaluation of WSP systems

The test rig is composed of a series of actuation units on which the tested system hardware parts are connected. The actuation units have to reproduce realistically the dynamics of the tested part (for example, the actuation units relative to the axle speed sensors have to reproduce the dynamics of the vehicle axles).

A software simulator is needed to calculate the reference signals used to control the actuation unit. Since the tested component may interact in operative conditions with the dynamics of the train (for example, WSP system modulates the braking force in order to control wheel sliding) its actions have to be measured and used as input for the software simulator, that then has to run in real time conditions.

The numerical model implemented in the MI6 test rig and those described in this paper have been realized by means of Matlab–Simulink™, that is one of the most frequently used environments for the fast simulation and prototyping of control systems and real time applications. It also supports trough Real-Time Workshop™ the creation of real time applications for different platforms, through the embedded coder it allows to build generic C-codes for custom applications from standard Simulink™ models.

The design of a numerical simulator used to control a HIL test rig involves the research of a tradeoff between result accuracy and calculation burden, that has to be compatible with the available computational resources. Furthermore the model has to be parametric in order allow the simula-

tion of different kind of railway vehicles with different wheelset, braking/traction systems, etc. The layout of the model have to be easily customizable according different kind of test done on the rig.

In order to satisfy this specifications the simulation model can be divided in an array of modular subsystems (mechanical layout, braking system, traction system, adhesion and line models etc.). Different versions of the same subsystem for example the traction system can be used different kind of equipments. Every subsystem has a parametric structure and can be calibrated by modifying its own set of configuration variables. Sub-models are standard Matlab-Simulink models while configuration variable are stored in Matlab format files so almost every part of the model is open and can be modified or customized using reliable and affirmed commercial products. Input and output of different versions of the same sub-model are standardized so the realization of different simulation models can be easily obtained by assembling different combination of sub-models and configuration variables. This operation can be completely automated using the features of Matlab-Simulink environment.

This numerical model of the vehicle can work on a standard PC so it can be used to test new or customized vehicle model/submodel (also in the design phase, when it has not yet been realized). Once this reference Matlab-Simulink model is produced, the end user of the rig has to decide the kind of test activities that have to be performed, than the preprocessor will automatically modify the reference model in order to produce a code tailored for the specific application.

During the real time simulation the standard GUI interface for offline calibration test is substituted by virtual instruments panels built on dSPACE Controldesk environment that is used to control the rig and manage the data acquisition. Once the test is successfully executed, data stored by the real time application on the disk of an host PC are converted in standard Matlab Simulink format.

The numerical model implemented in the MI6 test rig allows to reproduce the dynamics of the train in the longitudinal direction. In other words, the model describes the bi-dimensional dynamics of a railway vehicle in the longitudinal plane.

The study described in this paper is devoted to extend the numerical model to the three dimensional dynamics. This upgrade would allow to simulate a wider set of tests (curves, switches, etc.) and to obtain a more complete description of the vehicle dynamics (including tilting and yaw motions).

One of the key point in this study was the definition of a reliable and efficient model describing the forces arising in the wheel/rail interaction areas, that can be represented as the sum between a component normal to the tangent plane in the contact point and a component in the tangent plane, due to the friction between the contact bodies. This components, named creep forces take place in the contact area when a traction or a braking torque is applied to the axle. These forces play an important role in the wheelset dynamics and their value depends on a number of parameters, including the position of the contact point between the bodies.

The creep forces investigation and modeling is essential in order to obtain a realistic simulation of the railway vehicle dynamical behavior.

The introduction in a multibody model of the constrains that allows to model the contact between two three-dimensional surface is not easy and may sensibly increase the complexity and the computational burden of the numerical procedure.

In the presented model the contact point position is calculated offline by means of a procedure based on the simplex methods. This procedure is used to generate a three-dimensional lookup table used in the real time simulation to find the position of the contact points. The procedure is numerically sufficiently efficient and allows to manage multiple contact points.

The developed numerical procedure can be schematized as shown in Figure 2. As it can be seen, the model of the axles is the part of the whole vehicle multibody model on which the contact forces act. The dynamics of each wheelset depends on the external actions (trac-

tion/braking torques) and on the contact forces. The solution of the differential equations describing the wheelset dynamics allows to calculate the kinematical parameters (displacement and rotation) from which the position of the contact points can be defined. The creep forces are then calculated taking into account the position of the contact points and the wheel creepages, obtained from the wheelset kinematics.

In this paper, the main features of the developed numerical procedure are described. Then its application to a benchmark vehicle (the Manchester wagon) is summarized and the comparison between the obtained results and those obtained using a commercial multibody software (ADAMS™) are shown.
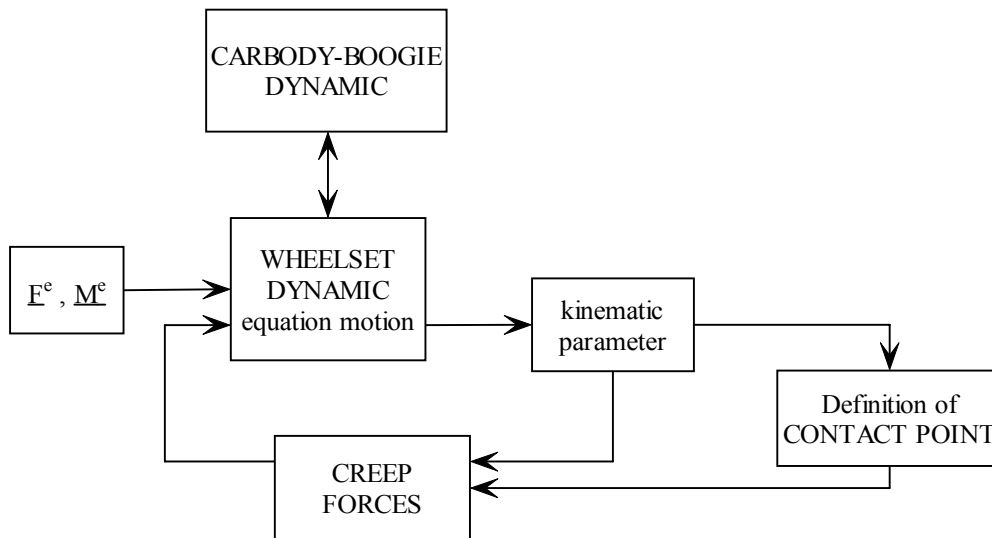
Figure 2: numerical procedure block diagram.

2 THE MULTIBODY MODEL

The multibody model of the railway vehicle was realized in Matlab-Simulink environment, and is shown in Figure 2.
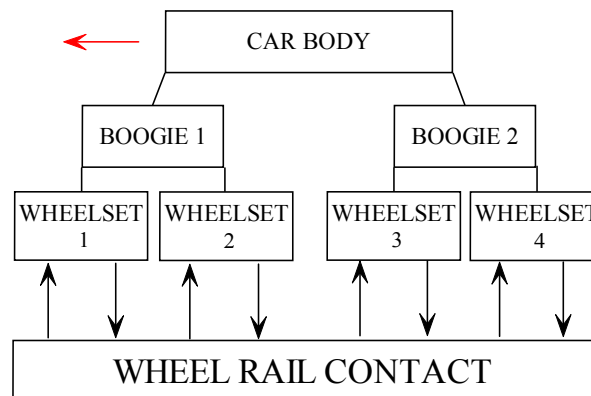
Figure 3: the multibody model.

The multibody model is composed of seven rigid bodies (the car body, two bogies, four wheelsets) connected by three-dimensional non linear elastic-viscous force elements (used to

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model the contact elements between the bodies, for example the vehicle suspensions). The wheel-rail interaction model considers a fully three dimensional rolling contact and can manage multiple contact points.
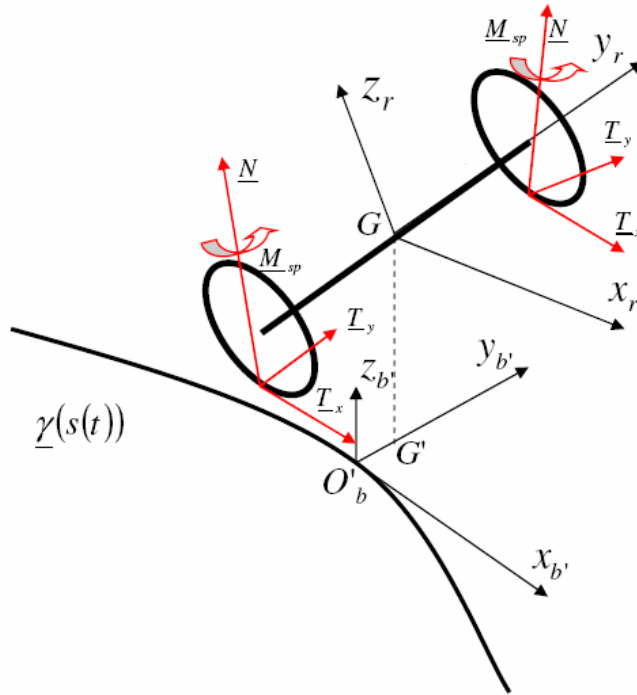

Figure 4: forces and torques acting on each wheelset

The forces acting on each wheelset (Figure 3) are the creep forces in the contact area, the forces due to the interaction with the boogie, the external applied braking or traction torque and the weight.
Then the dynamics of the wheelset can be described by the following differential equations:

$$
\begin{cases}
M\,\ddot{\underline{G}} = \underline{P} + \underset{j}{\Sigma}\,\underline{F}_j + \sum_{i=1}^{n}\left[\underline{N}_i + \underline{T}_{ix} + \underline{T}_{iy}\right] \\
\dot{\underline{K}}(\underline{G}) = \sigma(\underline{G})\,\dot{\underline{\omega}} + \underline{\omega}_s \wedge \underline{K}(\underline{G}) = \underset{j}{\Sigma}\,\underline{M}_j + \sum_{i=1}^{n}\left[\underline{M}_{iN} + \underline{M}_{iT_x} + \underline{M}_{iT_y} + \underline{M}_{isp}\right]
\end{cases}
\tag{1}
$$

3   CONTACT POINT BETWEEN WHEEL AND RAIL

The body shape of rail and wheel was is defined on the basis of real profiles, the wheel is modeled as a revolution surface, while the rail as an extrusion in the longitudinal direction. The proposed model allows to perform simulations on a generic track, that can be defined as input by the user in the pre-processing phase.
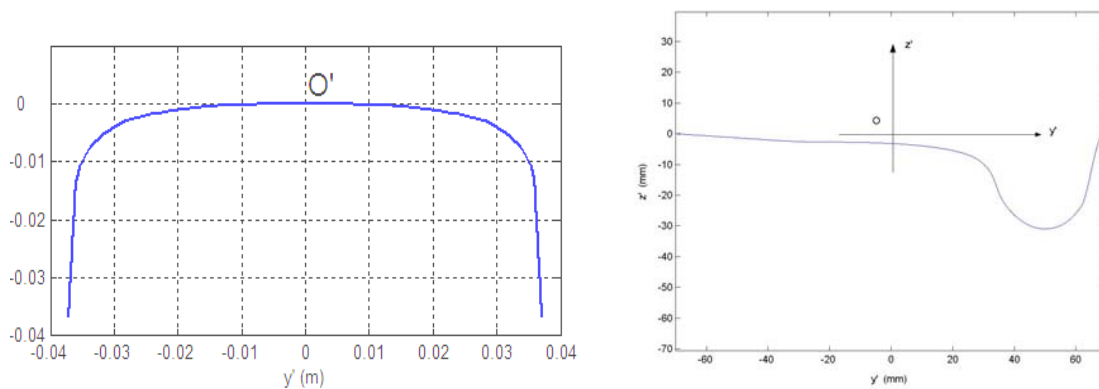Figure 3 shows an example of a rail (UIC 60) and a wheel (ORES1002) profile used for the simulations.

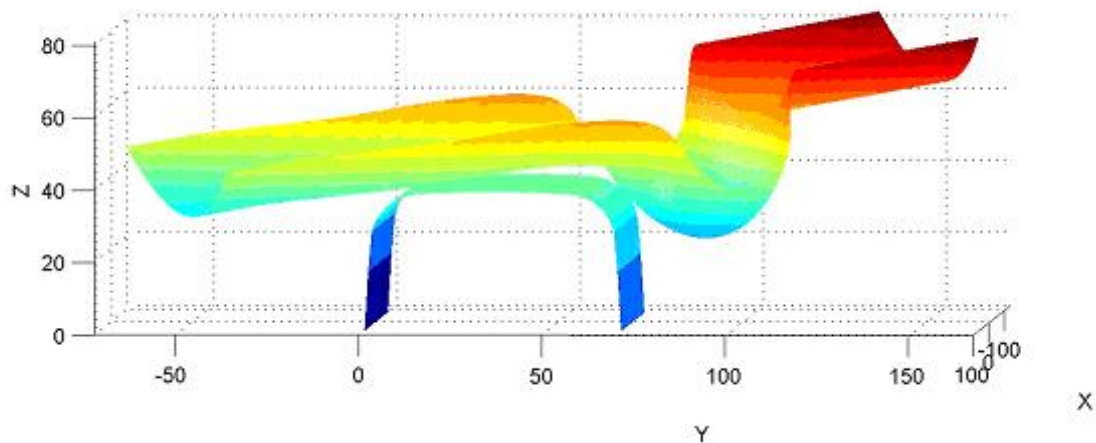Figure 5: UIC 60 rail and ORES1002 wheel profiles.

Figure 6: wheel and rail three-dimensional geometry, configuration in which a double contact point is present.

The contact point calculation is performed taking into account the complete wheelset three-dimensional motion, and it is based on a lookup table generated by an analytical procedure running offline.

In this procedure the contact points are calculated using the Simplex method (developed by Nelder e Mead in 1965) as the minimum values of the surface given by the difference between the rail and the wheel [4]. When the value of this difference minimum is positive (Figure 7 a)) the contact between the wheel and the rail is not present and no forces are exchanged in the contact point (in this case it is not defined). When the minimum difference is negative there is a penetration between the bodies (Figure 7 b)).

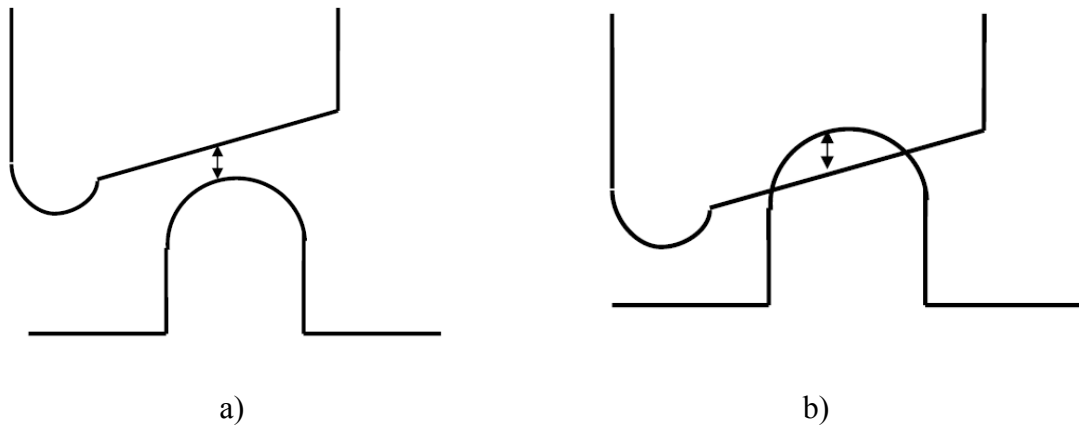<div align="center">a)   b)</div>

Figure 7: wheel and rail relative position, a) the minimum value of the difference surface is positive, no contact is present, b) the minimum of the difference surface is negative, the normal component of the contact force is positive.

The penetration value is used to calculate the normal component of the contact force, according to the elasto-viscous approach as sum of a term proportional to the normal penetration between the body, and a term proportional to the surface relative velocities in the contact point.

$$N = -k_1 \left| p \right|^{\gamma_1} + k_2 \left| \underline{v}^C \bullet \underline{n}_b(\underline{P}_b^C) \right|^{\gamma_2} \left| p \right|^{\gamma_3} \tag{2}$$

The constant values are calculated in accordance with the common models adopted in literature [5], [6]. The penetration value p is defined as the difference between the contact point on wheel and its vertical projection on rail.. This procedure can estimate both the main contact point on tread and the secondary contact point on flange for each wheel.

The magnitude of the tangential component of the contact forces is calculated on the basis of Kalker and Hertz theory: the Hertz theory is used to define the contact area dimensions and shape, that depend on the normal force magnitude, the material properties and the local profile geometry, while the Kalker linear theory results are used to define the components of the creep forces:

$$\begin{aligned} T_x &= -f_{11}\xi \\ T_y &= -f_{22}\eta - f_{23}\varphi \\ M_{sp} &= f_{23}\eta - f_{33}\varphi \end{aligned} \tag{3}$$

where $f_{11}, f_{22}, f_{33}, f_{23}$ are the linear creep coefficients, they are tabulated in function of contact ellipse semi-axis and combined modulus of rigidity, and the values of the creepage components ξ, η and φ for the right and the left wheel are defined as:

$$\xi = \frac{v_c \bullet \underline{i}}{\left| \dot{G} \right|}$$

$$\eta = \frac{v_c \bullet \underline{t}}{|\dot{G}|}$$

$$\eta = \frac{\omega \bullet \underline{n}}{|\dot{G}|}$$

The tangential Kalker values of creep forces are then saturated at the "adhesion limit".

4 RESULTS

The numerical model calculates the kinematic parameters of the body, and the forces on suspensions (between carbody and boogies and wheelset and boogie) and the whole contact information (creep forces and creepages) at the main contact points.

The "Manchester Passenger Wagon" (Figure 8) was chosen as benchmark for the validation of the model [8]. The numerical results were compared with those obtained with a model developed with Adams Rail. The validation process is necessary to evaluate the accuracy of the model and to verify the performance of the model in terms of computational burden.
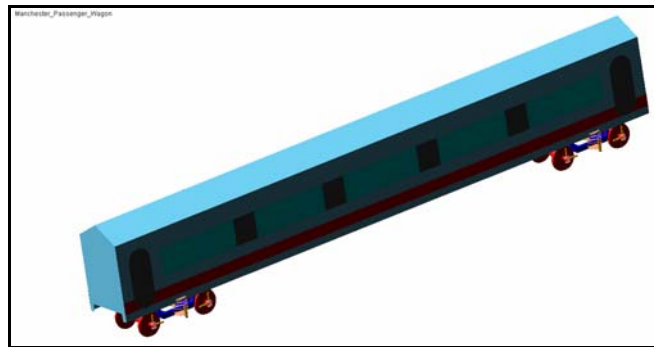

Figure 8: Manchester Passenger vehicle ADAMS model.

As example of results, in Table 1 and Figure 9 the lateral displacements from the center line of wheelset center of mass during a curve with radius 1200 m and cant 90 mm at several velocities are summarized.

Velocity	Matlab Simulink Model				ADAMS model			
	W_1 (mm)	W_2(mm)	W_3(mm)	W_4(mm)	W_1(mm)	W_2(mm)	W_3(mm)	W_4(mm)
45 m/s	6,1	3,1	5,8	3	5,9	3,2	5,8	3,1
30 m/s	6,1	4,5	6	4,7	6	4,7	5,8	4,8
15 m/s	6	6	6	6,3	6,1	6,2	6	6,3

Table 1 comparison between Matlab Simulink and Adams model results, wheelset lateral displacement values.
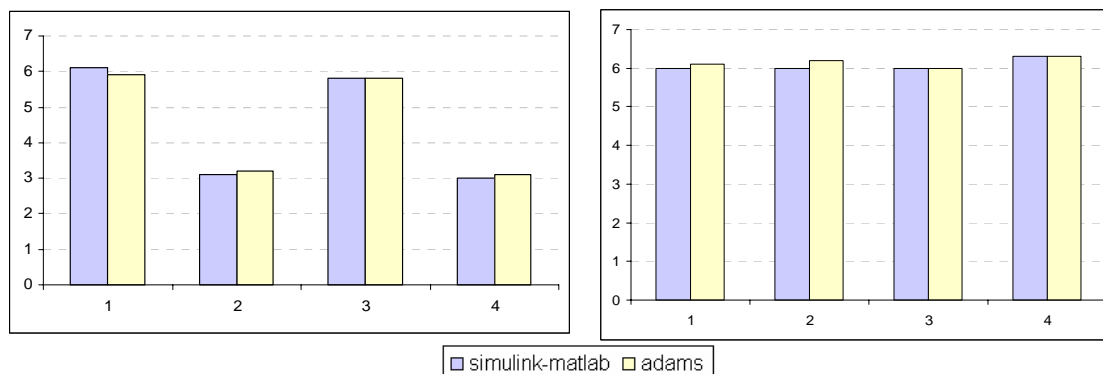
Figure 9: Lateral displacement: comparison between Simulink-Matlab and Adams model results corresponding to 45 m/s and 15 m/s.

Then the vehicle track disturbance response was analyzed. Vertical and lateral track irregularities was realized in order to estimate the carbody center of mass acceleration and the contact force acting on a wheel. The multibody model results are compared with those obtained from Adams model. As it can be seen from the diagrams shown in Figure 9 and 10, there is a good agreement between the results obtained from the different models.
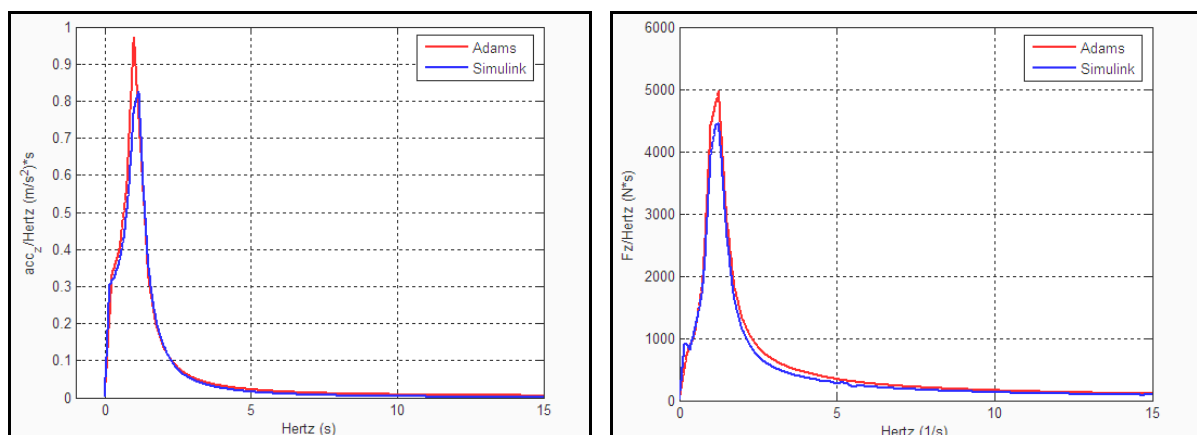

Figure 10: Vertical carbody acceleration and vertical contact force for the vertical track disturbance.

5 CONCLUSIONS AND FUTURE IMPROVEMENTS

The paper summarizes the features of a railway vehicle multibody model realized in the Matlab-Simulink environment. The model is used to simulate the dynamical behavior of a benchmark railway vehicle, the results of the model are compared with those obtained with the Adams Rail multibody model of the same vehicle. The comparison shows a good agreement between the models and the relative errors are acceptable (it has to be highlighted that the models evaluate the deformation of the contact bodies by means of two completely different approaches). The multibody model real time implementation is currently being investigated. The research activity is furthermore focused in the definition of more efficient procedures for the calculation of the contact points.

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