MODEL OF SCALED TEST RIG FOR REAL TIME APPLICATIONS

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Abstract. The existing mechatronic models of scaled test rigs are usually built in multibody simulation programs or in the interactive simulation tool. In these cases, the simulation processes run so slowly that makes this approach not ideal and not realistic for test rig dynamics, especially when it is necessary to develop electronic controller systems. In the present work, the design approach for the a real-time simulation is proposed and based on a decomposition of the mechatronical system into groups with elements, which are modeled separately, and then all groups are integrated in the full system. The mechanical system can be split into two subsystems. The first subsystem model has been realized in Simpack multibody symbolic code generator and describes the behavior of mechanical components of the test rig. The second one, the wheel-roller contact model, has been developed in C program language and integrated in Simulink as S-function blocks. The controller and the electrical parts have also been simulated in Simulink. The numerical experiment for the mechatronical system was made in Simulink. The obtained data about the behavior of the mechanical model has been used for the verification of the real-time contact model inside of a dSpace Modular System, built on the DS1006 Processor board.
Keywords: test rig, wheel, roller, contact, real-time simulation.

1. INTRODUCTION

The analysis of rail vehicle mechatronic systems shows that at the development and design stages for the estimation of behavior of the proposed solutions of such systems three classical approaches can be applied.

The first approach (Iwnicki, 2006) (Iwnicki and Wickens, 1998) (Chudzikiewicz, 2007) (Eichberger, 2002) (Kotz, 2003) (Spiryagin *et al.*, 2008) is based on the use of multibody simulation programs or in the interactive simulation tools, such as Simulink in Matlab environment for the theoretical investigation. This approach does not allow modeling the real behavior of rail vehicles because of the great numbers of nonlinear characteristics inside the mechatronic model; moreover the time response of the rail vehicles on the control signals cannot be simulated in the proper way because the simulation is not in real time.

The second approach (Bosso *et al.*, 2004), (Bosso *et al.*, 2009) is based on experimental investigation and the proposed algorithms for the mechatronic system are analyzed on special test rig. Solutions obtained by using this approach are more close to real railway vehicles even if some characteristics cannot be taken properly into account, depending on the specific test rig.

The third approach refers to the real vehicle; theoretical and experimental investigation can be carried on. This approach is a very expensive one; however, it shows a real picture of the work of mechatronic systems.

Nowadays, it is possible to say that these approaches are not very effective because they require very high computational time. Therefore, it becomes more reasonable to apply a new approach based on the HILS (Hardware-in-the Loop Simulation) vehicle simulator, which can connect some design and development stages of the prototype and to avoid some problems with the compatibility between parts of complex mechatronic model. It means that it allows the possibility to obtain results immediately, like as it can be done during an experimental work with real equipments, and to make change of the design very quickly without expensive hardware tools. After the end of the design process, it is possible to change a simulated prototype of a physical system on a real test rig or a real vehicle without necessity to redesign input/output interfaces.

To summarize the written above, it is necessary to clarify the goals for the research activity presented in this paper:

1. Development of the HILS model of the physical system (test rig), which may apply for the verification of traction and braking algorithms.
2. The architecture of the HILS model should allow the easy change of included elements in order to support further improvements of physical system.

The HILS is a simulation technique that is based on the application of real-time models. In the case of the application for railway engineering, the real-time model can be divided into the following sub-models:

- mechanical subsystems;
- contact model between wheel and rail;
- electric and electronics parts;
- control algorithms.

Commercial tools allow adapting offline models and sub-models to real-time simulations. However, not all sub-models put on the list can be created with such software. One of these exceptions is the wheel/rail contact model. It is necessary to underline that existing commercial program codes for multibody simulation in the field of railway dynamics (Simpack, VI-Rail, Universal Mechanism and etc.) allow to simulate only classical contact between wheel and rails. It means that the contact model which is needed for the simulation of the test rig cannot be realized in the current version of the commercial software. Furthermore, the contact model used for the classical approach cannot be applied for rail vehicle real-time simulation. Therefore, this paper is focused on the adaptation of the contact model for the real-time application. The procedure of the adaptation, described below, includes a new approach which consists of the following steps:

- development of the algorithm of the contact model, which can be used for real-time simulation;
- co-simulation of the multibody model of the test rig with the proposed contact model and control system;
- real-time simulation suitable for test rig.

Results presented in this paper are related to the first step of the development of the real time test rig physical model. Results obtained from the simulation will be then compared with the results obtained during experiments, in order to validate and tune the simulation model.

2. MECHATRONIC SYSTEM OF TEST RIG

The approach for simulating mechatronics in railway vehicles was presented in (Kotz, 2003). It describes clearly how to create the toolkit for the automatic co-simulation processes of a bogie. However, it is necessary to improve the solution (Kotz, 2003) because a test rig also includes one more important and distinctive part called a roller. The roller, as a wheelset of a bogie, has its own control and electrical systems, which must be described inside the model in the proper way.

The design approach, shown in Fig. 1, is based on a decomposition of the mechatronic system into groups with elements, which are modeled separately, and then all groups are integrated in the full system. The basic mechatronic system includes four elements: mechanical system (also called a plant according to the terminology used in the control theory), controller, power converter and a feedback sensor. In this work, one more element, called "Creepage controlled module" must be added in order to evaluate the traction or braking forces due to a given longitudinal creepage value.

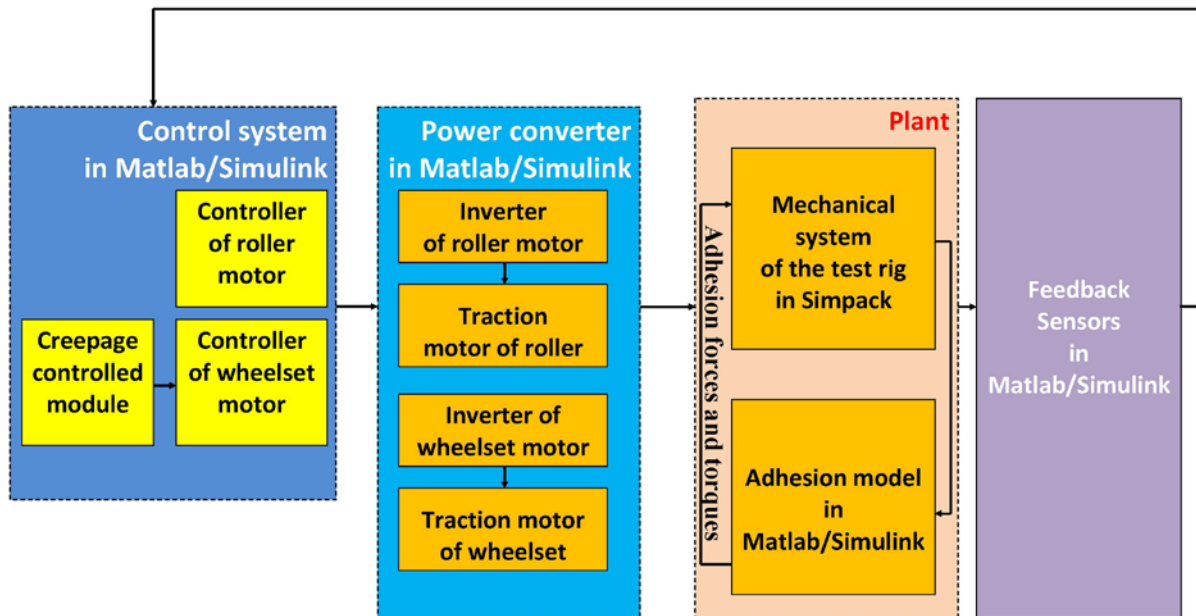In the following sections, the elements of the mechatronic system are described.

Figure 1: Block diagram of the model for a mechatronic system.

3. MULTIBODY MODEL OF THE MECHANICAL SYSTEM

An experimental device to carry on tests on scaled prototypes of railway vehicles has been developed at Politecnico di Torino. This device can be also used to perform experimental test to validate models of adhesion. It is then possible to obtain experimental measurements in controlled test condition, investigating a wide range of variation of the physical parameters of interest. The experimental system (scaled test rig) used in this work for the simulation in MBS (Multi Body System) packages includes a suspended wheelset and two rollers, which are pressed in the contact (Fig. 2).

The multibody model of the roller-rig (Fig. 3) has been developed using Simpack multibody code. The model represents the real device and is composed by two modules of the roller-rig which can support a single wheelset, a wheelset and the relative suspension system.

The roller rig has been modeled considering the two rollers as fixed together by a rigid joint, which is located in central position. The bearings which connect each roller to the relative support are also modeled using an ideal revolute joint (rigid).
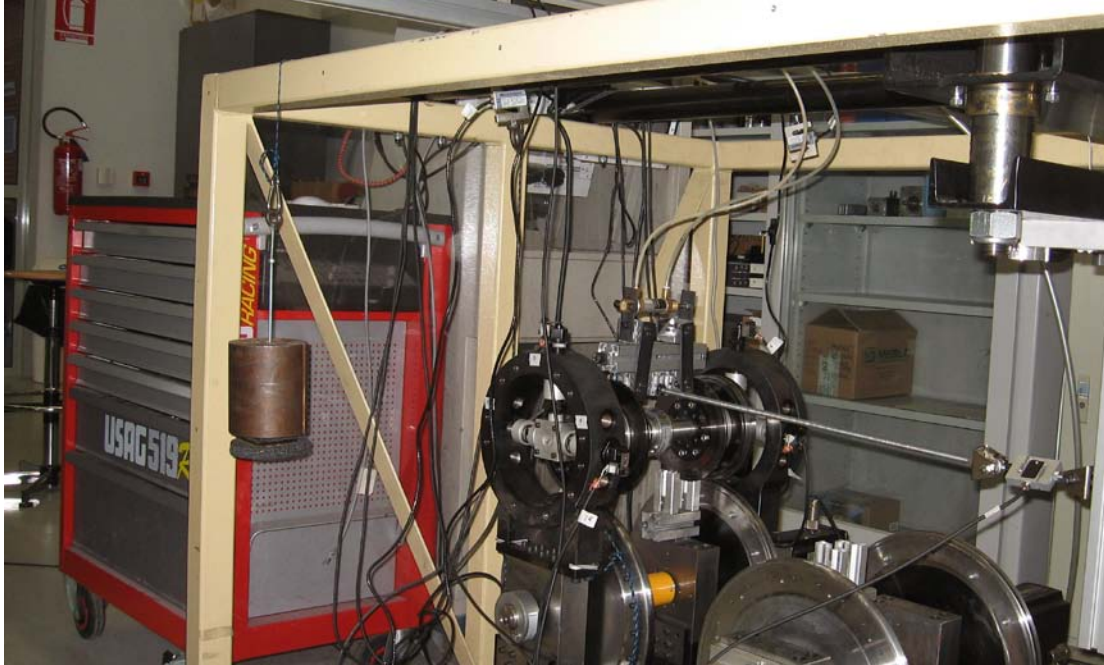

Figure 2: Roller Rig realized at Politecnico di Torino with a single suspended wheelset and braking device.
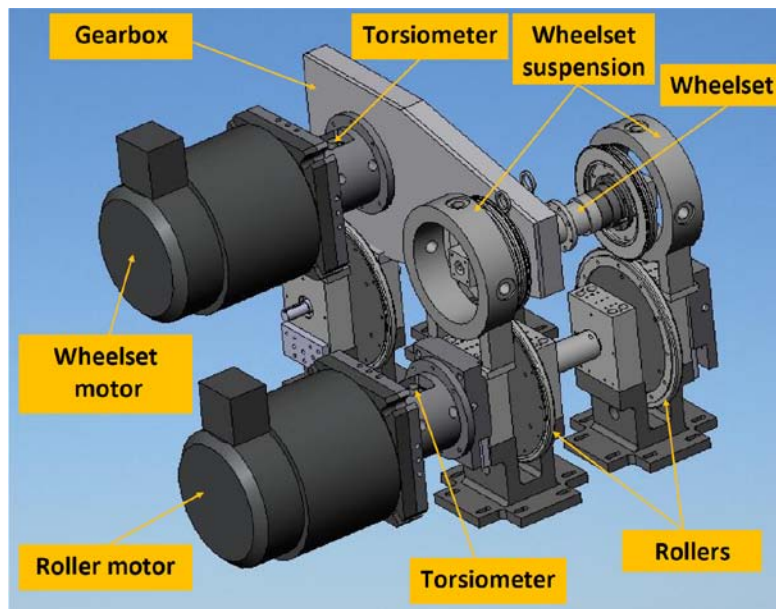

Figure 3: Multibody model in Simpack.

The left roller is connected to an electric motor simulated with a torque applied between the rotor and the stator, the stator has been fixed to the support part of the roller while the rotor has been fixed to the roller itself.

Rigid joints have been used both in the model and in the real device in order to achieve a better regulation of the motion, not influenced by the joint stiffness. Joint are simulated as rigid since their stiffness is higher than the stiffness of the shaft.

The wheelset is modeled as a rigid body; it is connected to the two axle-boxes with revolute joints which allow only the relative rotation along the wheelset axis. Each axle-box is then connected by means of a set of three springs to the suspension structures (Fig. 3), which are rigidly connected to the rollers supports.

The spring elements are modelled with a spring-damper parallel component force elements, which are supported by Simpack code export interface for the real time application.

In the model of the wheelset on roller-rig, the normal forces have been applied to the wheelset directly in the MBS model as user routines, which are also supported by Simpack code-export interface. The integration has been done because the constraint forces on roller rig must allow the wheelset be tangent to the roller profile during the longitudinal motion. For this reason two additional dummy bodies have been created, those bodies are rigidly constrained to the rollers with a revolute joint (each) allowing the rotation along the roller axis. On each dummy body two markers have been created, the first located at a distance equal to the sum of wheel and roller radius. This first marker has been used to create a force between wheelset and the dummy body acting as a stiffness on the longitudinal direction, this force has the effect to pull the dummy body during the wheelset longitudinal movement and does not affect the wheelset dynamic significantly since its mass is concentrated on the revolute joint and its rotational inertia is very low in comparison to the wheelset.

The second marker has been created on the dummy body at a distance equal to the roller radius, in this way this marker defines the contact point location when the wheelset is centered in lateral direction, also during yaw and longitudinal movements. This second marker has been used to create two contact reference markers on wheel and roller with a congruent position to the marker located on the dummy body (using Simpack "congruent" floating markers); the contact force (tangential and normal) has been created between the two contact reference marker on wheel and roller.

In case of the simulation on the roller, since a conical wheel has been used in the prototype, the equation defining the normal load can be written explicitly in the MBS model as:

$$F_z = k_{HZ} \cdot (\Delta z + y_W \cdot \tan(\alpha)) = 0.5 * (F_{SPRZ} + M \cdot g) \tag{1}$$

$$F_y = F_z \cdot \tan(\alpha) \tag{2}$$

where Δz is the displacement between wheel and roller measured along the Z axis of the contact marker placed on the dummy body; y_W is the lateral displacement of the wheelset; α is the contact angle; M is the wheelset mass, $F_{SPR,Z}$ in static condition is the preload of the vertical spring; k_{HZ} is the contact stiffness (assumed equal to 10^7 N/mm).

In case of real profiles, this approach cannot be used and it is necessary to decrease significantly the time step for the co-simulation process.

4. CONTACT MODEL

The proposed model described in this section is based on the work developed by the authors (Bosso *et al.*, 2001) and considers rigid single point contact. The contact model, that has been further developed and improved, can be divided into the following modules:
- geometrical module;
- Hertz module;
- normal forces calculation module;
- creepage module;
- creep forces calculation module.

4.1 Geometrical module

One of the most critical aspects in the wheel-rail contact problem is to develop an algorithm related to the contact point determination that can be easy translated in a numerical code.

The geometrical contact position is dependent on wheel and rail profiles. On the first stage of this investigation, the conical wheel and roller profiles has been used. The profiles are approximated using a left-side parabolic interpolation. The flow chart of the geometrical module is shown in Fig. 4.

The evaluation of the position for one contact point is based on a discrete minimization of the interference between points for wheel and rail profiles.

The wheelset is rototraslated by the algorithm in order to reduce the interference; the process continues until the compenetration exceeds the allowed value (in this case 10^{-9} m).
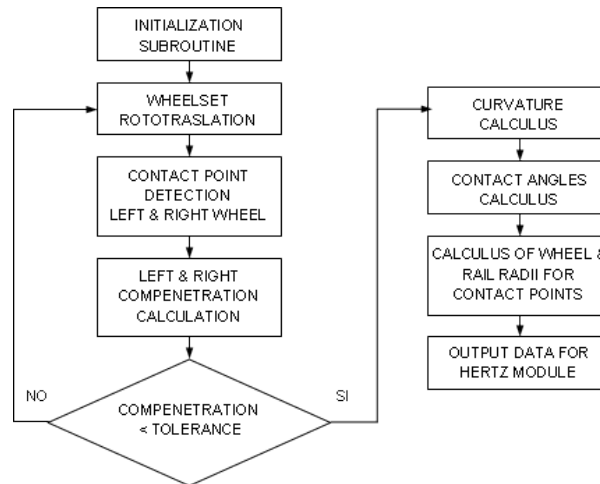
Figure 4: Flow chart for the geometrical module.

4.2 Hertz module

We assume that an elliptical contact surface is generated when wheel and rail are pressed together. The task of this module is to evaluate the ellipse's semi-axis at each time step. To obtain these results it is necessary to use the profiles curvatures in the contact point and the normal load to solve the Hertzian problem. Pre-determined tables are used for the evaluation of the elliptical integrals.

4.3 Normal forces calculation

Normal force calculation can be made directly solving the compenetration problem at the contact point (Eq. 1), anyway those forces have strong influence on wheelset dynamic.

Since wheelset dynamic in this work has been solved using Simpack, therefore the normal forces have been recalculated from the wheelset dynamic (acceleration, displacement) according to Eq. 3 and Eq. 4 the same forces have been recalculated using Eq. 1 to verify the congruence of the elastic model. This process has been used because in a co-simulation environment with variable transferred from two models in discrete time, it is not possible to ensure the stability of the solver with big time steps on the normal problem solution.

The vertical forces for the left, F_{zL} , and the right wheel, F_{zR} , can be defined as:

$$F_{zL} = \frac{1}{2}\left[m(\ddot{z}_w - g) - \frac{I_w}{s}\ddot{\theta} - m\ddot{y}_w(\theta + r_o / s) - V + \frac{M_w^x}{s} \right] \tag{3}$$

$$F_{zR} = \frac{1}{2}\left[m(\ddot{z}_w - g) + \frac{I_w}{s}\ddot{\theta} - m\ddot{y}_w(\theta - r_o / s) - V - \frac{M_w^x}{s} \right] \tag{4}$$

where m is wheelset mass; $\ddot{z}_w$ and $\ddot{y}_w$ are accelerations of wheelset in vertical and lateral directions; I_w is the wheelset moment of inertia; s is a half of current gauge; r_o is the nominal radius of a wheel in the central position of a wheelset; V is primary suspension vertical forces; M_w^x is roll bogie frame/wheelset suspension torque.

For the calculation of nominal forces, the rotation of the wheelset ahead for the test rig must be taken into account. Therefore, nominal forces for the left, N_L, and the right wheel, N_R, can be written as:

$$N_L = F_{zL} \sec \alpha_L - F_{zL} \sec \beta - F_{yL} \tan \alpha_L \tag{5}$$

$$N_R = F_{zR} \sec \alpha_R - F_{zR} \sec \beta + F_{yR} \tan \alpha_R \tag{6}$$

where α_L and α_R are the contact angles for left and right wheels, rad; β is the angle of rotation of the wheelset around the roller axis (due to longitudinal displacement), rad; F_{yL} and F_{yR} are lateral creep forces in the contacts for left and right wheels, N.

The effect of the lateral creep forces can be observed also on the normal forces (which act on a plan that is normal to the plane where actually friction take places) as a consequences, because friction forces can increase or decrease the vertical forces.

4.4 Creepage module

The following equations are defined for longitudinal, ξ_x, lateral, ζ_y, and spin, φ_S, creepages for the wheel-roller contact.

$$\xi_{xj} = \frac{(r_o - r_j)(r_o + r_r)}{r_o r} + \frac{\dot{u}}{V_n} + \frac{\dot{\chi} r_o}{V_n} \mp \frac{a_j \dot{\psi}}{V_n} \tag{7}$$

$$\zeta_{yj} = -\dot{\psi} + \frac{1 + A_0 \left(r_j + \alpha_j a_j \right)}{V_n} \dot{v} \tag{8}$$

$$\phi_{sj} = \mp \alpha_j \left(\frac{1}{r_o} + \frac{1}{r_r} \right) \tag{9}$$

where $j=1$ for the right wheel and $j=2$ for the left wheel; r_j is the actual rolling radius; $\dot{u}$ is the longitudinal translation velocity; V_n is the nominal velocity in the longitudinal direction; $\dot{\chi}$ is the wheelset angular velocity; a_j is the actual distance from the central line of the track to the wheel contact point in the lateral direction; ψ is the yaw angle of the wheelset; α_j is the contact angle; A_0 is a geometrical parameter (Bosso *et al.*, 2004); $\dot{v}$ is the velocity of a wheelset in the lateral direction, r_r is the curvature radius of the roller. A special procedure has been adopted for the case when $V_n = 0$.

The creep forces theory by Kalker (Kalker, 1982) have been applied to solve the wheel/rail contact problem. The simplified theory FASTSIM for the calculation of forces has been realized as S-function written in C language (Bosso *et al.*, 2009a), (Bosso *et al.*, 2009b).

5. MODELLING OF POWER AND CONTROL SYSTEMS

A digital speed controller with proportional and integral action is shown in Fig. 5, where ω_{ref} is reference angular velocity, G(z) is pulse transfer function of the digital speed controller, Tref is reference torque, T*$_{ref}$ is discrete-time reference torque, T_{in} is input torque to the mechanical system, T*$_{in}$ is input torque to the mechanical system after limiter, constraining the driving torque to the motor characteristics, and θ is angular position of the motor.

The control law for both motors is based on a simple proportional integral control and can be represented by the following equation, in a discrete-time form, with j varying from 0 to n:

$$T_{ref} = K_P(\omega_{ref(n)} - \omega_{m(n)}) + K_I \sum_{j=0}^{j=n} (\omega_{ref(j)} - \omega_{m(j)}) \tag{10}$$

where K_P and K_I are proportional and integral gains, which should be adapted to the applied load, and ω_m is output angular velocity of the motor.
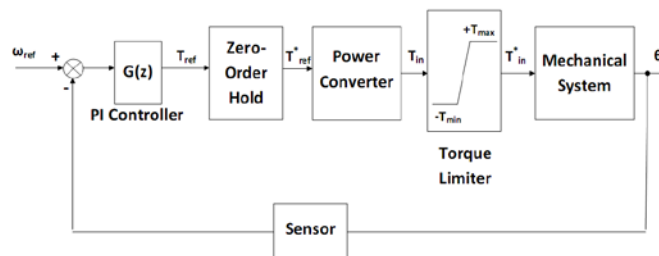

Figure 5: Block diagram of a discrete-time speed controller.

Based on Eq. (10), with the introduction of the transfer function $(1-z^{-1})^{-1}$ of the discrete time integrator, the pulse transfer function of the speed controller can be written as

$$G(z) = \frac{T_{ref}(z)}{\Delta\omega(z)} = K_P + K_I \frac{1}{1 - z^{-1}} \tag{11}$$

where $\Delta\omega$ is the difference between the reference angular velocity and the output angular velocity of the motor.

As well known from the design of mechatronic system, it is impossible to apply the obtained value of the reference torque directly to the mechanical subsystem because a time delay of electrical subsystem always occurs. Consequently, it has been added a low-pass filter to describe the torque dynamics of the inverter and the traction drive, which are parts of the power system.

5.1 Sensors

Sensors provide with the information of physical states for mechanical system and required to close control loops. Feedback sensors measure signals imperfectly and errors of such sensors can result in a poor performance of the control system (Vukosavić, 2007). A practical sensor supplies information on the shaft position. In this case, in the z-domain, the angular velocity as output from an encoder can be evaluated by using the transfer function defined as (Vukosavić, 2007), (Ellis, 2002) (Fig. 6):

$$S(z) = \frac{z-1}{Tz} \tag{12}$$

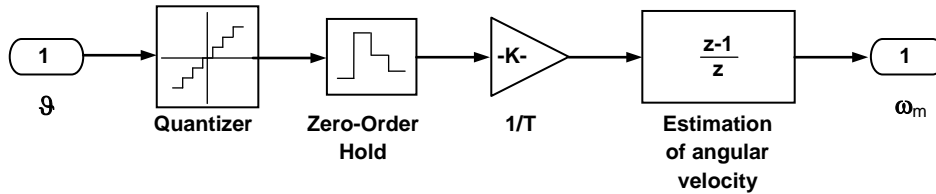where T is the sampling time interval for a sensor.

Figure 6: Simulink subsystem model of the speed encoder.

5.2 Creepage control module

In order to reach the required value of the longitudinal creepage, the angular velocity of the wheelset is modified based on the steepest gradient method. This method uses the longitudinal creepage difference, $\Delta\xi$, between the reference longitudinal creepage, ξ_{ref}, and the estimated longitudinal creepage, ξ_{est}, as a reference value for the velocity adjustment. In this case, the equations for the proposed method can be written as:

$$\text{if} \quad \Delta\xi \geq 0 \quad \text{then} \quad \omega_{ref}^{wheelset}(t+1) = \omega_{ref}^{wheelset}(t) + \alpha \cdot \Delta\xi \tag{13}$$

$$\text{if} \quad \Delta\xi < 0 \quad \text{then} \quad \omega_{ref}^{wheelset}(t+1) = \omega_{ref}^{wheelset}(t) - \beta \tag{14}$$

with t, time, α and β, constants, and $\alpha \cdot \Delta\xi$ is a small value corrector in order to reach the desired value of the angular velocity of the wheelset to provide the reference longitudinal creepage. A value of $\alpha = 0.01$ has been used in the simulations. If the value of the estimated longitudinal creepage is higher than a reference value, it is necessary to enable a fast recover to the reference value of the longitudinal creepage by decreasing the angular velocity of wheelset with a constant step length β. A value of $\beta = 0.002$ has been chosen in the simulations.

The reference angular velocity for the wheelset can be evaluated by using the following expression:

$$\omega_{ref}^{wheelset} = K \frac{r_r}{r_w} \omega_m^{roller} \tag{15}$$

where K is a coefficient ($0 \leq K \leq 1$), which is to be adapted to the desired value of the longitudinal creepage; r_r is the curvature radius of the roller; r_w is the curvature radius of the wheel.

The estimated longitudinal creepage is defined as

$$\xi_{est} = \frac{\omega_m^{wheelset} r_w - \omega_m^{roller} r_r}{\omega_m^{roller} r_r} \quad \text{with} \quad \omega_m^{roller} \neq 0 \tag{16}$$

6. SIMULATION RESULTS

This section describes the strategy for two simulations. First simulation has been done in Matlab/Simulink. The test rig part of the mechatronic model has been used for simulation purpose only without the further application for the real-time mode because it is based on the standard solution, realized by Simpack and allowed to create a real-time multibody model limited by some requirements of the Simpack code-export interface. The ode23tb solver with a variable step size in SIMULINK has been used for the simulations. Main goal of the simulation is to collect the data needed for further verification of the proposed real-time contact model.

For the adjustment of PI controllers of the mechatronical systems, it is necessary to choose correct values of proportional and integral coefficients. Therefore, based on the requirement of the proposed system and obtained simulation results, $K_P=300$ and $K_I=2$ in the case of the roller and $K_P=100$ and $K_I=2$ in the case of the wheelset were chosen for the control systems of motors. The calculation of contact forces was made with Fastsim algorithm (Kalker, 1982).

The simulation for verifying the validity in the case of different input parameters for the proposed mechatronic system was performed. To investigate characteristics of the system, it is necessary to make the following assumptions:
1. the linear velocity of the roller has a constant value;
2. the longitudinal creep coefficient is adjusted regarding to the reference values by controlling the velocity of wheelset.

The simulation was performed with the linear velocity of the roller of 5 m/s. The static adhesion coefficient required for the calculation of dynamic friction coefficient was taken equal to 0.4. The velocity of the wheelset has been modified in order to reach the desired values of the longitudinal creepage (0.04, 0.015, 0.02 and 0.03) between wheel and roller. Results for the first set of simulations are presented in Fig. 7. The braking torque was applied to the wheelset and controlled by using the creepage controlled module and a PI controller until the end of the simulation. It allows reaching and adjusting the desired values of longitudinal creepage. The results, presented in Fig. 7 and Fig. 8, show a good performance of the proposed control system and a stable behavior of the mechatronic system.
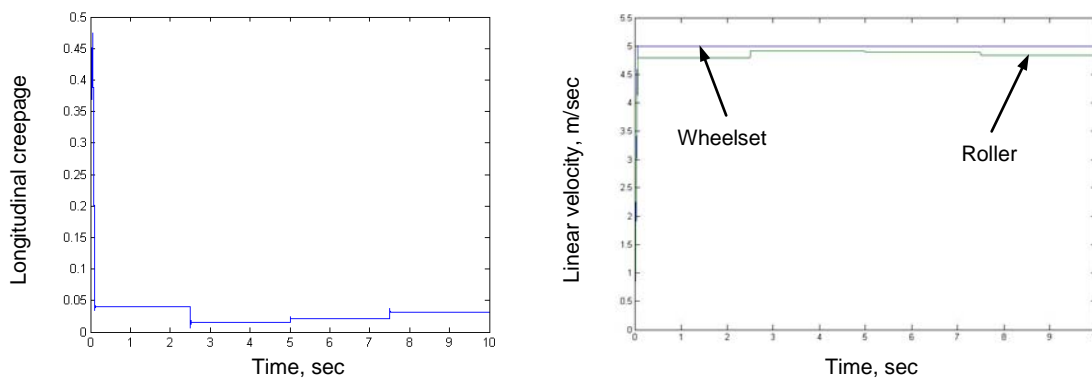

Figure 7: The creepage and velocity results obtained for a test rig during simulation process in Simulink (static friction coefficients 0.4, vertical spring load 900.85 N).

For the second set of simulations, the dSpace hardware system, built on the DS1006 Processor board, has been applied. The database, created during first set of simulations, contains information about the behavior of the mechanical system; it has been used as input parameters for the proposed real-time contact model. The model was created in Simulink based on S-functions for each module written in C language regarding to the rules of the standard "MISRA C". Based on TargetLink from dSpace, the embedded code for the real-time computer has been generated.
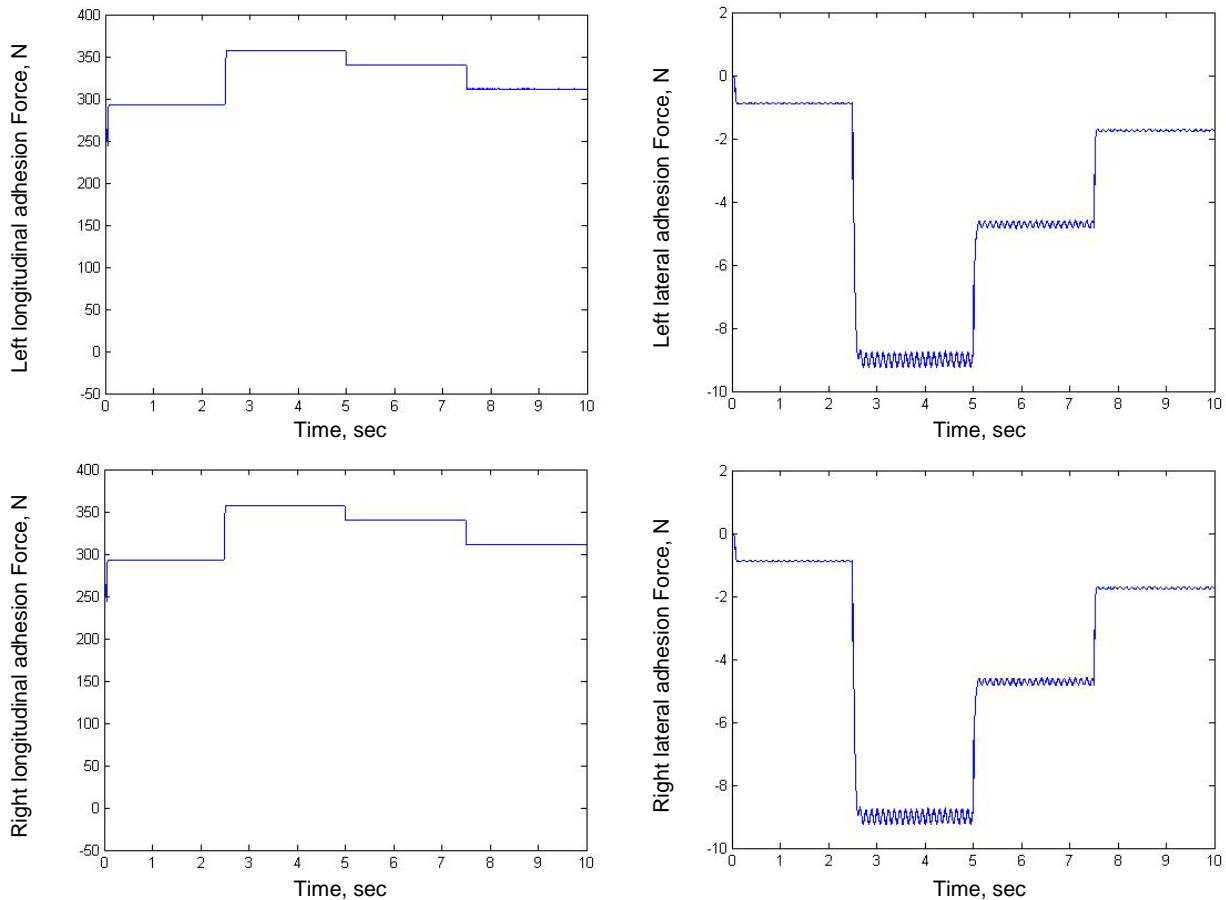
Figure 8: The adhesion force results obtained for a test rig during simulation process in Simulink (static friction coefficient 0.4, linear velocity of the roller 5 m/s, vertical spring load 900.85 N).

One of the problems encountered during implementation of the code into the dSpace system was to find a minimal value of the discrete time step to provide the possibility to work in the real-time environment. It has been found that the proposed model works with the minimal time step equal to 0.004 sec, allowing 250 calculations of adhesion forces per second.

Results are shown in Fig. 9; the real-time contact model (TargetLink compiler and dSpace platform) gives same results as the model simulated in Simulink (Fig. 8).

7. DISCUSSION AND FURTHER DEVELOPMENT

Two-dimensional contact model has been adopted for the real time application, based on the investigation made by other researchers (Piotrowski and Chollet, 2005) and our own experience (Bosso *et al.*, 2001), (Bosso *et al.*, 2009).

In the real time simulation the geometrical module is the most time consuming task, it requires about 70% of the computational time. It is then important to improve the interpolation algorithms for the geometrical module.

Future work will focus on the development of the full real-time mechatronic model in order to reach optimal balance between accuracy and calculation time. This should give new trends for further model-in-the loop (MIL), software-in-the loop (SIL) and hardware-in-the loop (HIL) simulations.
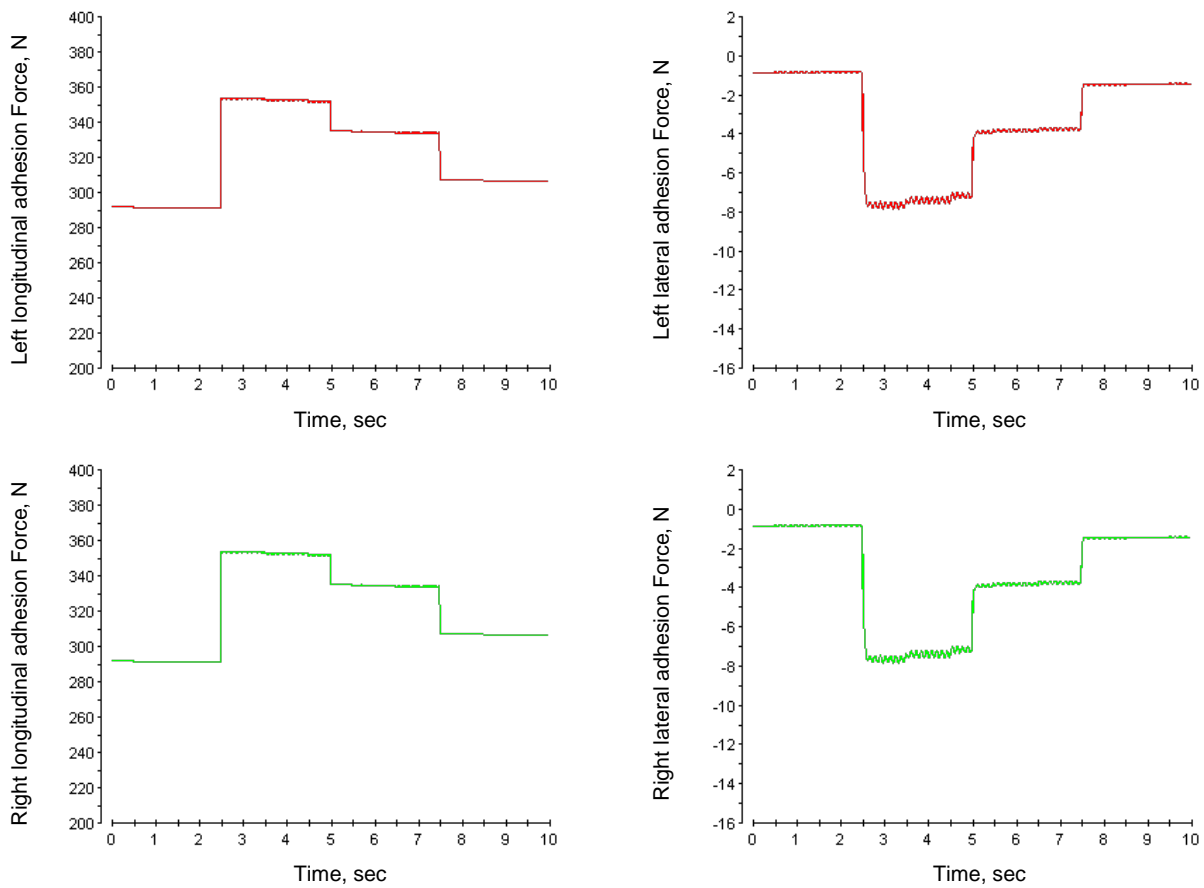
Figure 9: The results obtained for the contact model from dSpace ControlDesk software (static friction coefficient 0.5, linear velocity of the roller 5 m/s, vertical spring load 900.85 N).

8. CONCLUSIONS

The design of complex mechatronic system, based on the integration of different software packages for the simulation of the test rig, has been proposed for the development of real-time models. All subsystems of the mechatronic system have been integrated in Matlab/Simulink and then the simulation results were used for the creation of the database, which contains information required for the verification of real-time contact model. The model for the real-time application was created in an automatic code generation called TargentLink, which was integrated in Simulink and integrated in the dSpace Modular System, built on the DS1006 Processor board.

The real time simulation on the contact model has been compared with the co-simulation model. It allows the possibility to apply the proposed solution for a real-time simulation of any existing rail vehicles and test rigs under traction and braking condition.

It is necessary to note that such approach can also be useful for further development of traction and braking control systems with software-in-the loop and hardware-in-the loop simulations for rail vehicles.

9. REFERENCES

Bosso, N., Gugliotta, A., Somà., A., 2001. "Introduction of a wheel-rail and wheel-roller contact model for independent wheels in a Multibody code", Proceedings of the ASME/IEEE Joint Rail Conference , Washington, DC.

Bosso, N., Gugliotta, A., Somà., A., 2004 . "Dynamic behavior of a Railway Wheelset on a Roller Rig versus Tangent Track", Shock and Vibration, 11, 467-492.

Bosso, N., Gugliotta, A., Somà., A., Spiryagin, M., 2009. "Adhesion force estimation on 1/5 test rig", Multibody Dynamics 2009, Eccomas Thematic Conference, Warshaw, Poland.

Bosso, N., Gugliotta, A., Somà., A., Spiryagin, M., 2009. "Improvement of adhesion force estimation on 1/5 scaled test rig and optimization of the control law of tractive and braking motors", XXXVIII Convegno Nazionale, Associazione Italiana per l'Analisi delle Sollecitazioni, Torino, Italy.

Chudzikiewicz, A., 2007. "Simulation of rail vehicle dynamics in MATLAB environment", Vehicle System Dynamics, 33, 107-119.

Eichberger, A., 2002. "Generating Multibody Real-Time Models for Hardware-in-the-Loop Applications", Proceedings of AVEC 2002, Hiroshima.

Ellis, G., 2002, "Observers in control systems", Elsevier Academic Press, New York.

Iwnicki, S., 2006. "Handbook of railway vehicle dynamics", CRC Press, UK.

Iwnicki, S. and Wickens, A.H., 1998. "Validation of Matlab vehicle simulation using a scaled test rig", Vehicle System Dynamics , 30, 257-270.

Kalker., J.J., 1982. "A fast algorithm for the simplified theory of rolling contact " (FASTSIM). Vehicle Systems Dynamics, 11, 1-13.

Kotz, H.P., 2003. "Simulation of effects based on the interaction of mechanics and electronics in railway vehicles", Simpack User Meeting 2003 - Freiburg, Germany.

Piotrowski, J., Chollet, H., 2005 "Wheel–rail contact models for vehicle system dynamics including multi-point contact", Vehicle System Dynamics, 43 (6-7), 455–483.

Spiryagin, M. , Lee, K.S., Yoo., H.H.,2008. "Control system for maximum use of adhesive forces of a railway vehicle in a tractive mode", Mechanical Systems and Signal Processing, 22, 709-720.

Shu, X., Wilson, N., Sasaoka, C., Alkins, J., 2006. "Development of a real-time wheel/rail contact model in NUCARS® and application to diamond crossing and turnout design simulations", Supplements of Vehicle System Dynamics, 44, 251-260.

Vukosavić, S.N., 2007. "Digital control of electrical drives", Springer, New York.

10. RESPONSIBILITY NOTICE

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