Dynamical analysis of the HIL MDM scaled roller rig for the simulation of wheel-rail degraded adhesion condition.

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ABSTRACT

In the railway applications [2], scaled roller rigs play a great improvement with respect both to experimental tests and numerical simulations, since they combine the Hardware in the Loop benefits with a reduced economic investments [5]. In particular, the realization of a scaled roller rig able to reproduce degraded adhesion condition allows the development and analysis of safety on board system like odometry algorithms, wheel slide protection systems and traction controls. Referring to these developments, recently Trenitalia built in Florence an innovative fullscale roller rig to investigate these problems, implementing the control scheme proposed in [4]. According to this control scheme and the technical requirements of the Trenitalia fullscale roller-rig, the MDM Lab has designed a scaled roller rig to simulate the same dynamical effects and degraded adhesion conditions of the fullscale one [4]. Most of the problems in the development of the scaled roller rig are referred to the major complexity of the system, in terms of hardware parts, software parts and knowledge of the model (because of the necessity of simulating degraded adhesion conditions). Moreover, in order to study the control algorithms and the dynamics of the scaled roller rig, a complete multibody model of the HIL scaled roller rig is needed. Authors, in this paper, present an accurate model of the HIL scaled roller rig, with a particular focus on the interactions between the multibody model of the roller rig and the control system.

1 INTRODUCTION

In recent years the development and calibration of safety relevant on board subsystems like traction controls [1], odometry algorithms, WSP (Wheel Slide Protection Systems) [3] began to use HIL (Hardware In the Loop) system[5]. According to this, Trenitalia built in the research Center of Firenze Osmannoro an innovative fullscale roller-rig in which it is also possible to simulate degraded adhesion conditions in railway vehicle using a specific control scheme previously described in [4]. In [4] the control laws for the definition of the roller torques and speeds were developed by means of considerations based on a simplified model that considered the system dynamics only in the longitudinal plane. This control scheme performs the on line estimation of the vehicle motor torque (to reduce the time of the setting up phase) by means of an estimator procedure that uses both the tangential component of the contact forces (measured on the support) and the roller angular velocity. Thanks to the implementation of the Osmannoro facilities, a scaled roller rig was designed and realized, with the purpose of investigating the feasibility and the calibration of different control layouts that may be used also for the full scale version [4].

In this paper a sensitivity analysis with respect to several dynamic disturbances is performed. For this objective, authors have been used a numerical model of the scaled version of the roller rig (internally developed in [9]). The numerical model reproduces both the hardware (scaled bogie, rollers) and the software (virtual train model, virtual bogie roller rig) parts of the roller rig. The hardware parts are modeled like a three dimensional multibody model and, in particular, the contact between wheels and rollers have been modeled by adapting a previously existing algorithm (described in details in [6]). The software part of the roller rig and the control laws of the roller motors were derived substantially from those described in [4], even if

some improvements were necessary: first of all, since the roller rig described in [4] tests a whole vehicle, while the scaled version tests only one bogie, the controller has to include a virtual vehicle, in which one of the bogies is real and the other is simulated. Therefore, the proposed study may be seen as a Software in the Loop (SIL) calibration of the control algorithms that will be directly compiled and implemented in the scaled rig. In particular, in this paper the authors investigated the robustness of the proposed layout by means simulations where degraded adhesion condition were tested together with some typical disturbances like the hunting and dynamical imbalance of rotating parts. The numerical control performance were evaluated in terms of angular speed error (between the simulated train model and the roller velocity) and torque estimation error (between the real torque and the estimated torque).

2 ARCHITECTURE OF THE HIL SYSTEM

The architecture of the system reproduces the same architecture of the Osmannoro roller rig [4]. The objective of this bogie roller rig is to reproduce on the test rig the train behavior in degraded adhesion conditions. In order to reach this goal, Hardware In the Loop (HIL) approach is used: a virtual vehicle model with a simplified adhesion model simulates the desired vehicle dynamics; then the test rig rollers are controlled in order to approximate the dynamical behavior of the vehicle in terms of wheel angular velocity and vehicle motor torque. The simulated roller rig has two interesting characteristics: the reproduction of the same slidings between wheel and rail (to obtain a reliable simulation) and the implementation of a virtual scaled roller rig in a manner to simulate the whole vehicle (this strategy allows the evaluation of the dynamical behavior of the whole vehicle using a single scaled roller-rig and reducing the computational load). The HIL system can be logically divide in the following parts:

1. *Scaled Roller Rig*: including 3D roller rig multibody model, 3D contact model, 3D virtual roller rig and a simplified antiskid device.

2. *Virtual train model*: including a simplified model of a whole railway vehicle and a simplified adhesion model.

3. *Controllers*: its task is to reproduce on the hardware roller-rig the same dynamical behavior of the virtual train model [4].

4. *Estimators/Filters*: systems used to reduce disturbances and to estimate the vehicle torque.

The general architecture of the test rig is schematically shown in the block diagram of Fig. 1. The sim-
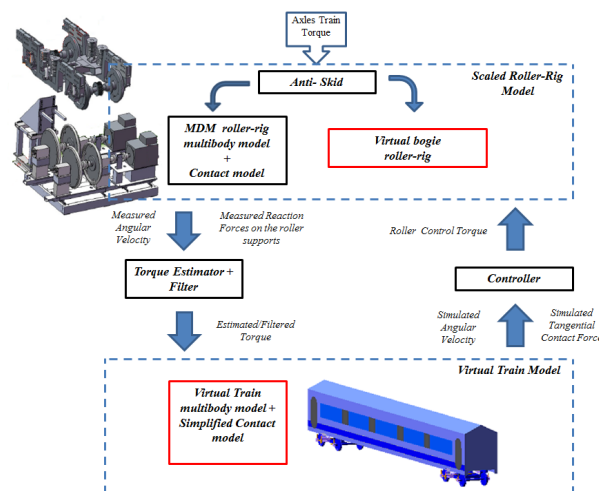

Figure 1. General architecture of the scaled roller rig

ulator of the HIL system includes the software part of the system (virtual train model, virtual bogie roller

rig) and a 3D multibody model of the hardware part (the scaled bogie and the rollers). The whole system simulator has been implemented in the MATLAB®-Simulink®environment; in particular the multibody model has been implemented in the MATLAB®toolbox SimMechanics. The use of the MATLAB®-Simulink®environment allows the use of many numerically efficient integration algorithms; moreover the structure of the Simmechanics is modular and parametric and therefore particularly suitable for modeling complex multibody systems.

2.1 Scaled Roller Rig

The multibody model of the scaled roller rig is used to simulate the dynamic behavior of the MDM test rig. Some previous works described the design of MDM roller-rig and the considered scaling hypothesis [3][5].
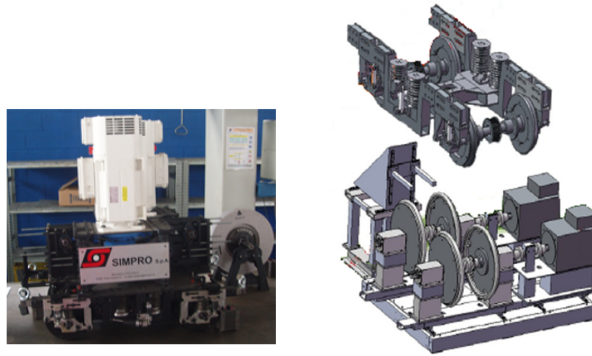

Figure 2. Comparison between MDM scaled roller rig (a) with the simulated model (b) referring to a bogie.

The 3D multibody model of the MDM roller-rig presented in this paper is more realistic than those described in the previous works [4] because it defines two different multibody model both for the scaled test rig and for the scaled vehicle. The scaled bogie physical and geometrical dimensions are referred to the Manchester wagon [10] by means of properly scaling factors. In order to simulate the behavior of the system in realistic conditions, in the multibody model some physical disturbances like geometrical errors and dynamical axle imbalance were introduced. In fact, some preliminary tests on the real roller-rig showed that this type of disturbance has a relevant effect on the control performances. The objective of this paper is to analyze the performance of the controller and the dynamical behavior of the MDM roller-rig when different kind of disturbances are coupled with degraded adhesion condition. The disturbances derive from the model uncertainties concerning the geometrical and inertial parameters. The simulated disturbances are:

- Longitudinal variation $\delta G_x(t_0)$ of the position of the center of mass of the bogie: this disturbance simulates the uncertainty on the bogie position in longitudinal direction when the bogie is positioned on the rollers.

- Lateral variation $\delta G_y(t_0)$ of the position of the center of mass of the bogie: the introduction of this alteration produces the bogie hunting on the MDM roller-rig.

- Dynamical imbalance of wheelsets and rollers (δ_M and a variation δ_G of the CG position to create an eccentricity effect for the mass imbalance and δ_J for the inertial imbalance): the dynamical imbalance of the MDM wheelsets/rollers have to be modeled because there are uncertainties in the exact knowledge of the inertial and mass parameters. These effects can disturb the controller performance and stability. In literature there is an European standard referring to the wheelset dynamical imbalance [7].

Both for the necessity to reproduce disturbances and to better the modeling, the 3D wheel/roller contact model considered in this work is an improvement of previous models developed by authors for the wheel-rail pair [6] and it is detailed in [9]. The contact model can be logically divided into two parts: the contact

point detection between two revolute surfaces (rollers and wheels) and the calculation of the normal and tangential contact forces.

The first problem is based on a semi-analytical procedures to find out the contact points; the implemented algorithm assures the following advantages:

- The contact detection algorithm between revolute surfaces is fully 3D and does not introduce simplifying assumptions on the problem geometry and kinematics

- Generic wheel-roller profiles

- Accurate management of the multiple contact points without limits on the point number

- High computational efficiency needed for the online implementation within multibody models

The second problem, the calculation of the normal and tangential forces, has been solved using the Hertz and the Kalker theories [11]. In order to improve the model of the Scaled Bogie mounted on the MDM roller-rig, an Anti-Skid device (ASD) is implemented [4]. The Anti-Skid is an electronic device that allows the reduction of the slidings between wheel and rail in degraded adhesion conditions. This component uses the informations about the adhesion state in the Virtual Train model to calculate the modulated torque that reduces the slidings. The Anti-Skid works only in traction phase. According to the architecture presented in Fig. 1, the scaled roller rig is composed by two parts: the MDM bogie roller rig and the virtual roller rig; the virtual bogie roller-rig model and the MDM bogie roller-rig model allowed the simulation of a full vehicle roller-rig. In order to reproduce the dynamics of the whole vehicle, this strategy permits to physically build only one roller-rig (that simulates the front bogie and the half carbody) and to simulate via software by means of a simplified model of the roller-rig the rest of the wagon. To improve the computational efficiency of the whole system, the virtual roller-rig is modeled by a simplified analytical model. The analytical model is simplified with respect to the MDM roller-rig multibody model because it is based on a 2D dynamical model of a wheelset-roller system. The input of these blocks are the rollers control torques u_i^{sc} and the modulated torque C_S^{sc} applied by the motors to the wheels. The outputs are the measured tangential contact force T_{mis}^{sc} and the measured roller angular velocity ω_{rm}.
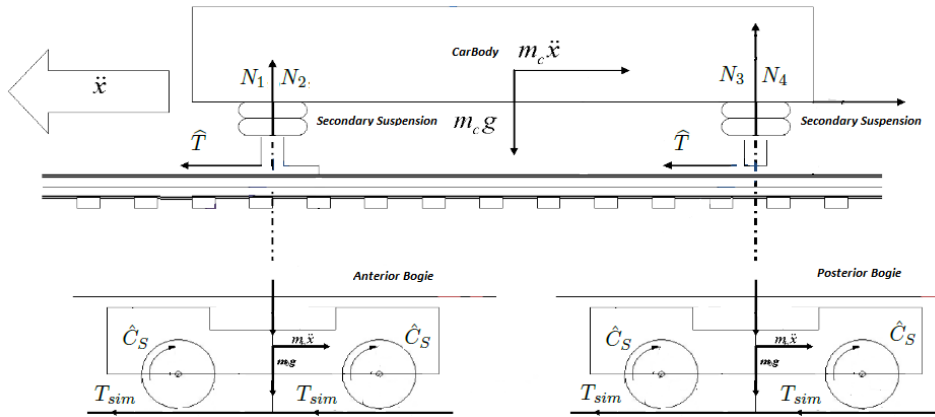
2.2 Virtual Train Model

Figure 3. Scheme of the virtual train model

The virtual train model allows the simulation of the dynamics of the fullscale railway vehicle. The goal of the whole system is to simulate the train dynamics in different scenarios: bad adhesion conditions, different physical/geometrical configurations, etc.. The multibody model is a simplified 2D model [1] of the longitudinal train dynamics, which using as inputs the estimated tangential forces T_{sim}^{sc} and the filtered torque $\tilde{C}^{sc}$, allows the calculation of the linear velocity $\dot{x}_i$, the linear acceleration $\ddot{x}_i$ and the wheelsets load

distribution N_i[9]. The virtual train model uses a simplified adhesion model in order to be implemented directly in real time software [4].

2.3 Controller

The controller reproduces on the roller rig the same values of angular velocity ω_{ws} and accelerations $\dot{\omega}_{ws}$ which have been calculated by the virtual train model [4]. In this way, the same dynamical condition simulated in the virtual train model are reproduced. The input of this block are the simulated tangential forces T_{sim}^{sc}, the simulated wheel angular velocities ω_{ws}, the estimated wheel angular velocities $\widehat{\omega}_w$ and the filtered motor torques $\hat{C}_S^{sc}$. The output are the four control roller torques u_i^{sc}. This architecture is studied to give the possibility to recreate different conditions on the four wheelsets. The complete algorithm used in the controller has been described in previous works [4].

The control performances are evaluated by means of three parameters:

- *Speed Error e_ω*: error between the simulated wheel angular velocity ω_{ws} and the estimated wheel angular velocity $\widehat{\omega}_w$

- *Torque estimation Error e_c*: error between the real torque C_S and the filtered torque $\hat{C}_S^{sc} * \phi_C$

- *Control Torque u_i^{sc}*: the torques are defined in order to minimize the speed angular error and the torque error between the MDM roller-rig and the Virtual roller-rig.

2.4 Estimator and Filtering

According to the necessity to reduct the number of sensors installed on the bogie and on the architecture of the HIL system (see Fig. 1), the torque applied by the wheel motor to the roller cannot be directly measured because it is a function of the behavior of the virtual train model. The only outputs of the scaled roller rig that can be measured by the sensors are the roller angular velocity ω_{rm} and longitudinal component T_{mis}^{sc} of the reaction force evaluated on the roller support. The goal of the estimator block is to estimate the wheel angular velocity $\widehat{\omega}_w$, the wheel angular acceleration $\widehat{\dot{\omega}}_w$ and the estimated torque of the wheel motor $\tilde{C}_S$. In the MDM roller-rig the adhesion condition between wheel and roller is fully adhesion (the adhesion coefficient is $\mu = 0.3$). In this adhesion state, the slidings between wheel and roller can be neglected. With this hypothesis (supposing that the roller radius is equal to the wheel radius $r_s = R$) the wheel angular velocity, the tangential contact force and the wheel angular acceleration can be estimated as follows:

$$\widehat{\omega}_w = -\omega_{rm} \qquad \widehat{T}^{sc} = T_{mis}^{sc} \qquad \widehat{\dot{\omega}}_w = -\tfrac{d}{dt}\omega_{rm} \qquad \tilde{C}_S^{sc} = \widehat{T}^{sc} r_s + J_B \widehat{\dot{\omega}}_w; \qquad (1)$$

where $\widehat{\omega}_w$, $\widehat{T}^{sc}$, $\widehat{\dot{\omega}}_w$ are the estimations of the considered quantities and r_s is the wheel radius, J_B is the total momentum of inertia of the axle/roller system. Considering the complex architecture of the system and the interactions between hardware and software parts, a reliable value of the wheel motor torque is fundamental for the control system performance. The torque estimator equation is a function of the wheel estimated angular acceleration $\widehat{\dot{\omega}}_w$ and the estimated tangential contact force $\widehat{T}^{sc}$. However, the measured values of ω_{rm} and T_{mis}^{sc} (and thus $\widehat{\omega}_w$, $\widehat{\dot{\omega}}_w$ and $\widehat{T}^{sc}$) can be affected by noise/disturbances that may lead to instability problems and to reduce the controller performances. The dynamical imbalance, simulated as an alteration of mass δ_M, inertia tensor δ_J and defining a wheelsets eccentricity δ_{CG}, generates on the roller an approximatively sinusoidal force (at the wheel rotation frequency) that produces a disturbance on the torque estimation. The force disturbance modifies the estimated torque $\tilde{C}_S^{sc}$ because it alters the estimation of the tangential component of the contact force $\widehat{T}^{sc}$ and the wheels angular acceleration $\dot{\omega}_{rm}$. The hypothesis made in this work is that the torque disturbance term could be modeled as a nearly sinusoidal analytical contribution:

$$C_D^{sc}(t) = A(t)\sin(\hat{\omega}_w(t)t + \phi) \qquad (2)$$

where $A(t)$ represents the amplitude approximatively proportional to the centrifugal term, $\hat{\omega}_w(t)$ represents the estimated wheel angular velocity (depending on the work operative condition) and ϕ is the phase and it is approximatively constant in time and function of the initial conditions of the system. The disturbance $C_D^{sc}(t)$ is defined in the whole work range (in terms of frequency) but its amplitude is variable with the velocity: at low angular speed the error between the real torque and the filtered torque is acceptable while

at high angular speed produces vibrations and modifies the dynamical behavior of the Virtual Train model. As demonstrated in practice, it is possible to make a simplifying hypothesis: the *Uncoupled Spectrum Hypothesis*. The spectrum of the torque signal $\tilde{C}_S^{sc}(t) = \hat{C}_S^{sc}(t) + C_D^{sc}(t) + N(t)$ consists of three different contributes: the real torque signal $\hat{C}_S^{sc}(t)$ (its frequency spectrum is on the left part of Fig. 4), the high frequency noise $N(t)$ (its frequency spectrum is on the right part of Fig. 4) and the torque disturbance $C_D^{sc}(t)$ (its frequency spectrum is on the middle of Fig. 4 and it is variable with respect to the wheel angular velocity). If the filter works well the filtered torque $\tilde{C}'^{sc}_S$ coincides with the real torque $\hat{C}'^{sc}_S$. In conclusion
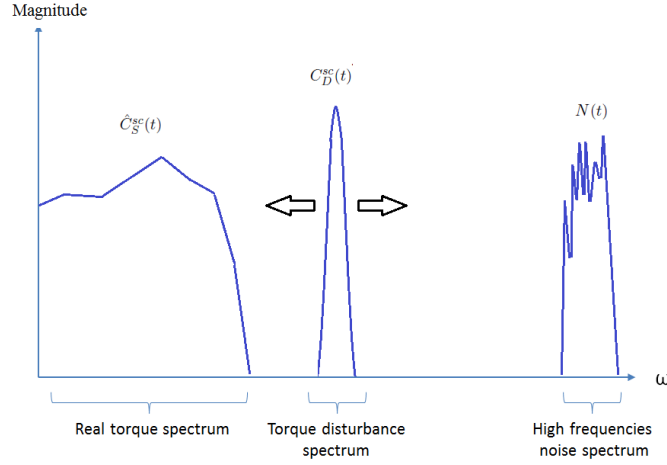

Figure 4. Spectrum of the torque signal

the situation can be summarized as follows: spectrum of the real torque $\hat{C}_S^{sc}(t)$ in the range $[0 - 10]$ Hz, spectrum of the torque disturbance $C_D^{sc}(t)$ in the range $[10 - 32]$ Hz and spectrum of the high frequency noise $N(t)$ range over 50 Hz.

A filter is generally designed to reduce the noise and some particular disturbance. The filter developed by the authors is studied to eliminate the particular disturbance $C_D^{sc}(t)$ and the high frequency noise $N(t)$, extracting from the estimated torque $\tilde{C}_S^{sc}$ the real torque $\widehat{C}_S^{sc}$. The filter performance is evaluated by means of the torque estimation error (that is defined by the difference between the original torque C_S and the real torque $\widehat{C}_S^{sc} * \phi_C$). The filter strategy, proposed by the authors, is based on the spectrum hypothesis; the filtering procedure is schematically sketched in Fig. 5. Initially, the estimated torque $\tilde{C}_S^{sc}$ is filtered by a lowpass filter to eliminate the noise $N(t)$. The ideal behavior of the filter block would be the extraction of the disturbance $C_D^{scf}(t)$ without modifying its amplitude A_{CD} and its phase ϕ_{CD} (with respect to $C_D^{sc}(t)$). The condition $C_D^{s}cf(t) = C_D^{sc}(t)$ (in terms of amplitude A_{CD} and phase ϕ_{CD}) represents one of the main requirement to employ the filter strategy. In this way the disturbance is eliminated in real-time, obtaining the filtered torque $\hat{C}_S^{sc}(t)$.
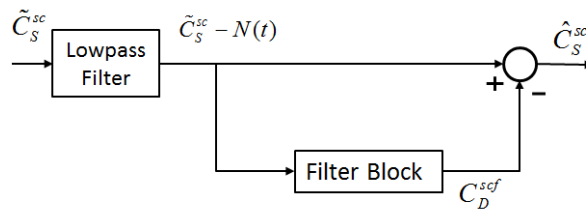

Figure 5. Filter scheme

The main problems in the proposed filter design were:

- Defining a filter that extracts $C_D^{sc}(t)$ without modifying amplitude and phase,

- Making the filter time variant in the whole frequency range $[10 - 32]$ Hz.

The first part of the the problem can be solved using a passband filter implemented by means of sixth order elliptic filter (since with this filter the performances are reached with medium-low order). As can be see in Fig. (6), in the range of interest $\omega_C \pm PhaseLimit$, both the no ripple condition $A_{CD} \cong 1$ and the phase condition $\phi_{CD} \cong 0$ can be observed. Moreover, as can be seen from the red area in Fig. (6), the conditions
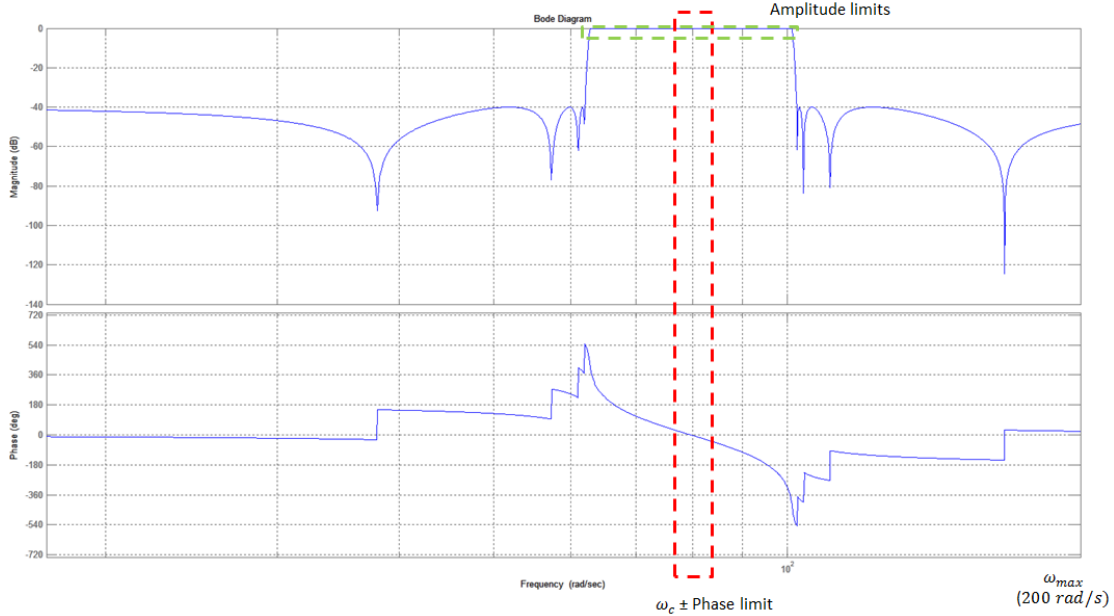

Figure 6. Bode diagram of the i-th Elliptic Passband filter: Magnitude-Phase

required are respected with the selected filter in the considered range but the problem, as previously said, is the narrow work range where these conditions are respected.

The problem of the small range (where the phase and the amplitude conditions are respected) can be addressed by discretizing the whole frequency range [10-32] Hz in intervals and using a multi filter approach. In this way, it is possible to design each single filter to work in the interval where the phase and the amplitude conditions of the considered filter are verified [9]. The proposed multi-filter approach allows the merging of the solutions of the two problems.

3 NUMERICAL SIMULATIONS

The goal of the numerical simulations is to analyze the behavior of the MDM roller-rig when degraded adhesion is coupled with a particular disturbance. The numerical simulations have been made combining bad adhesion condition with longitudinal variation $\delta G_x(t_0)$, lateral variation $\delta G_y(t_0)$ and dynamical imbalance of the wheelsets and rollers ($\delta_M, \delta_G, \delta_J$). In particular, the numerical simulation analyzed in this paper is summarized in Tab. 1. The Scenario purpose is to simulate the dynamical behavior of the MDM

Table 1. Simulated Scenario

	Disturbances	Initial Conditions	Control Parameters
Scenario	Wheelsets imbalance $\delta_M, \delta_J, \delta_G$ Rollers imbalance $\delta_M, \delta_J, \delta_G$ Low adhesion condition $\mu = 0.05$	$\omega_{init} = 150$ rad/s Traction Phase Saturated ramp	e_c front/rear e_ω front/rear

roller-rig and the control performances when degraded adhesion condition ($\mu = 0.05$) is coupled with

wheelsets/rollers dynamical imbalance [7] δ_M, δ_G and δ_J. In this scenario two different cases are tested: the MDM roller-rig model without Multi-filter and the MDM roller-rig model with Multi-Filter. In the first case, the effects of the dynamical imbalance on the whole system are showed, while the second one permits to describe the effects of the Multi-Filter block.

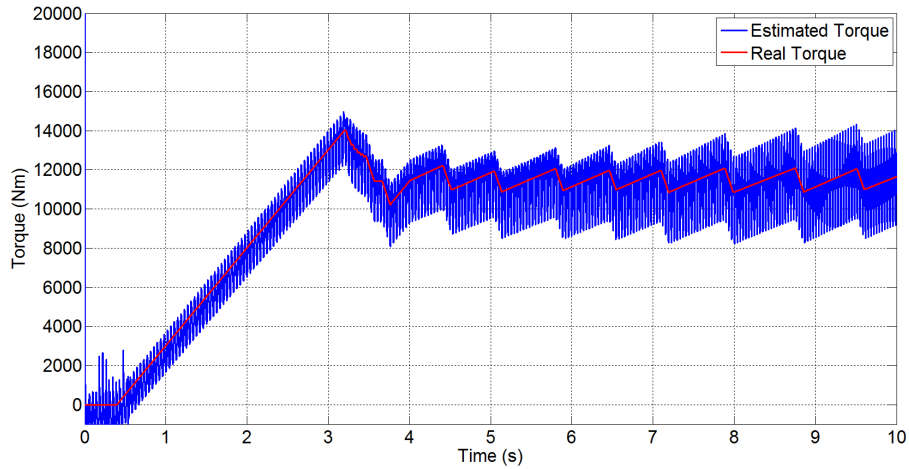## 3.1 Dynamical Imbalance Disturbance Without Multi-filter

Figure 7. Comparison between the estimated torque $\tilde{C}_S^{sc} * \phi_C$ and the real torque C_S on MDM the front wheelset
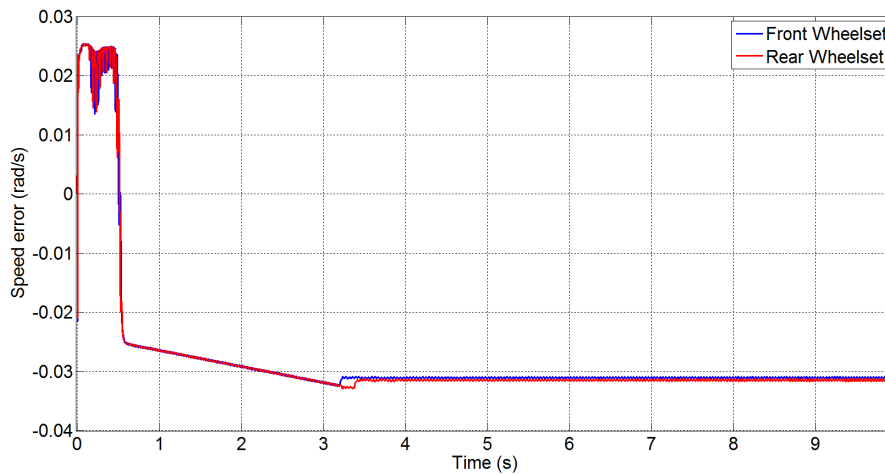

Figure 8. MDM roller-rig: Front/Rear speed error e_ω

As can be seen from the graphics, the presence of the dynamical imbalance produces a sinusoidal disturbance on the estimated signal. In Fig. 7 the torque behavior is shown: the original torque C_S follows the Anti-Skid algorithm while in the estimated torque $\tilde{C}_S^{sc} * \phi_C$ the effect of the disturbance $C_D^{sc}(t)$ is present. The controller is *not robust* in term of torque estimation error.

The speed error e_ω (see Fig. 8) after a transient is stabilized. This result confirms that the controller is *robust* in term of speed error.

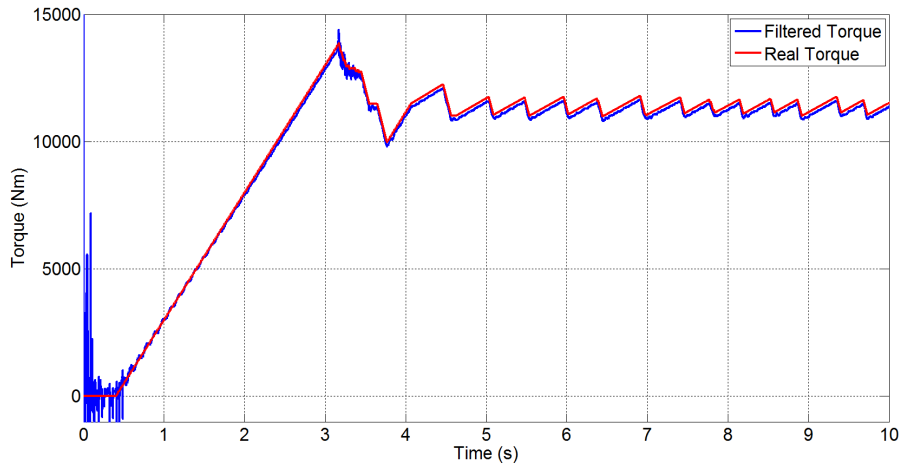## 3.2 Dynamical Imbalance Disturbance using Multi-filter

Figure 9. Comparison between the filtered torque $\hat{C}^{sc} * \phi_C$ and the real torque C on the MDM front wheelset
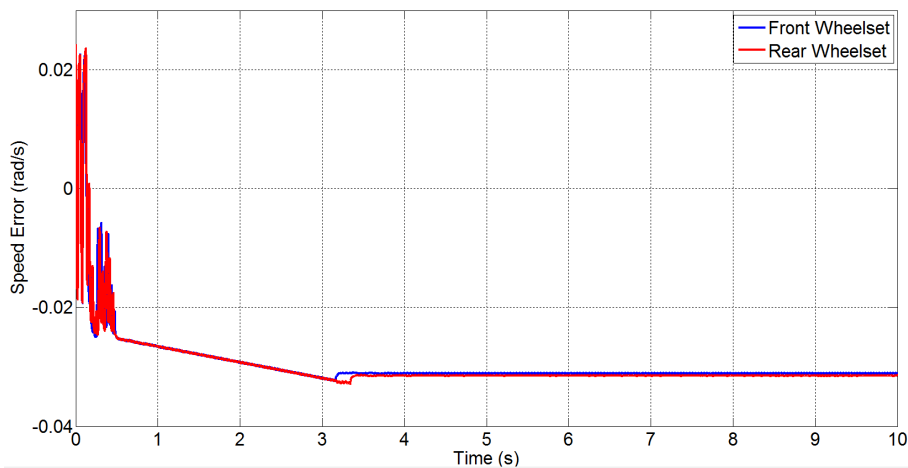

Figure 10. MDM roller-rig: Front/Rear speed error e_ω

In this simulation, the multi-filter approach has been used. The first figure (see Fig. 9) shows a comparison between filtered $\hat{C}_S^{sc} * \phi_C$ and the original torque C_S on the wheelsets and the improvement with respect to the Fig. 7 appears clearly: the disturbance $C_D^{sc}(t)$ is almost filtered. The torque error e_c, after the transient period, is ± 300 Nm and this result is satisfying (nearly 2.5% of the mean torque value 12000 Nm) as regards the system requirements. The speed error e_ω is included in the range limits (see Fig. 10).
In conclusion:

- the multi-filter approach and the controller result *robust in terms of torque estimation error e_c and speed error e_ω,*

- the implementation of the multi-filter approach in the architecture does not modify the stability of the system.

- the controller is robust and feasible when degraded adhesion condition and dynamical roller/wheel imbalance are coupled,

4 CONCLUSIONS AND FURTHER DEVELOPMENTS

In this paper authors presented a realistic multibody model of the MDM roller-rig implemented within an Hardware in the Loop (HIL) architecture in order to simulate a scaled railway vehicle test-rig. The main purpose of this work was to study and to analyze the performance of the controller and the dynamic behavior of the roller-rig when different types of disturbances are coupled with degraded adhesion condition, since the effects of the disturbances on the HIL system usually degrade the reliability of the simulations. The simulated disturbances were:

- uncertainty δG_x on the wheel center of mass exact position in longitudinal direction, when the bogie is set on the MDM roller-rig,

- alteration of the wheel center of mass initial position in lateral direction δG_y that produces bogie hunting on the MDM roller-rig,

- dynamical imbalance of the wheelsets and the rollers caused by the uncertainties in the exact knowledge of the inertial and mass parameters δ_J, δ_M and δ_G (considering to the European standard referring to the wheelsets dynamical imbalance [7]).

In particular, in this paper a stability test referring to the dynamical imbalance disturbance is presented. The analyzed scenario studies the system when degraded adhesion condition is coupled with dynamical imbalance. The results showed that the system is greatly influenced by the imbalance disturbance, in particular as regards the torque estimation error. The authors proposed a new torque estimation strategy that involves a multi-state filter especially designed to eliminate this particular disturbance. The new approach utilizes the same estimation equation presented in Chap. 2.4 but introduces a multi-state filter that allows the extraction of the disturbance $C_D^{sc}(t)$ in a real-time process. In the numerical simulations two different cases were presented: the case that used the previous torque estimation procedure and the case with the new torque estimation strategy. In the first case the results showed the disturbance effects on the system stability and on the torque estimation error. The system remained robust only in terms of speed error. At the contrary, the results of the second case demonstrated the efficiency of the new torque estimation strategy. The graphics showed that the system is robust in terms of speed error and torque estimation error. Moreover, the presence of the multi-state filter and the controller does not influence the system stability.

As regards the further developments, the comparison between the experimental data coming from physical model of the MDM roller-rig and the numerical results is scheduled for the future in order to validate the numerical model.

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