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DESIGN OF A LATERAL AND VERTICAL SEMI-ACTIVE SUSPENSION SYSTEM FOR AN HIGHSPEED TRAIN

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Abstract: *Semi-active suspension systems have been widely applied to road vehicles especially for expensive applications like sport/luxury cars or military applications. Moreover, thanks to technological development of a new generation of magneto-rheological fluids in the late nineties, a new generation of more reliable and cost effective dampers has been available on the market. In this paper Authors have studied the application of this technology to the railway engineering field and in particular to high speed trains. In particular has been developed a tridimensional model using Mathworks Matlab-Simulink™. The multibody model have been developed using standard Simulink Simmechanics™ blocks; in addiction Authors have implemented a library of customized non linear elements able to model tridimensional wheel-rail contact, or the behavior of typical mechanical components like non-linear springs, dampers and endstops. At the end of the work a very accurate model has been developed and it has been used to simulate the performances of semi-active control system applied to secondary suspension stage and to lateral dampers. In this paper some preliminary results are reported.*

INTRODUCTION: MAIN CONCEPTS AND SPECIFICATIONS

Since the late seventies, with the first Italian prototype of Pendolino [1] mechatronic sub-systems, like active tilting, have been introduced in high speed train in order to improve comfort and to increase the maximum operative speed. On further developments of the design principles characterizing Pendolino is based the commercial success of tilting trains produced initially by Fiat Ferroviaria and then by Alstom.
More recently important railway suppliers, as for example Bombardier, have invested heavy resources in the development of mechatronic bogies with the innovative FLEXX tronic™[2,3,4,5] technology.

The aim of this work was to develop a general purpose semi-active suspension system that could be easily implemented with a reduced impact on a standard passive bogie from medium to high speed passenger service.

The main requirements of an ideal semi-active suspension control system implies it has to be less invasive as possible, simple, safe with an implicit fault tolerant robustness. So it preferably has to be designed as a easily customizable sub-system able to improve performances of existing bogies with affordable costs.

The layout of a typical trailer bogie is reproduced in Figure 1.

The simplest way to introduce a semi-active control system is to substitute the existing passive dampers with corresponding controlled ones; this is the solution that is proposed and investigated in this work too.

In particular, in this initial step of a wider research project, Authors have analyzed the possibility of replacing secondary suspension stage and lateral passive dampers with semi-active dampers.

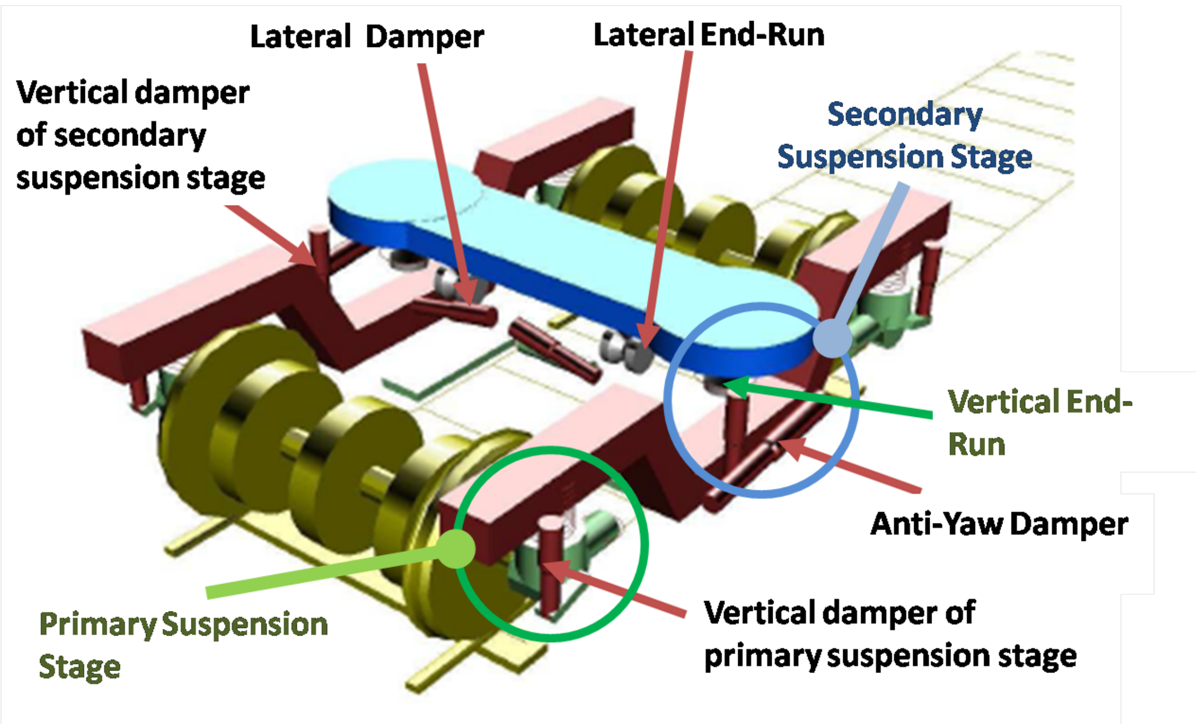


Figure 1: Typical trailer bogie layout

1 ACTUATION SYSTEM

A semi-active suspension solution is chosen. A damper with controllable/adjustable mechanical characteristic could be used to improve comfort, with the possibility of limited enhancements on vehicle dynamic stability.

Since the actuator is simply a semi active damper, it can only dissipate energy. At on hand it is a desirable consequence for system robustness since in case of control malfunction, the actuator is not able to inject mechanical energy in the system (no risk of instability). Nevertheless it unavoidably leads to low performances since the actuator is able to work only in two quadrants of damper force/speed characteristic plane, as shown in Figure 2, The dissipated power satisfies relation (1).

$$P = F \cdot V > 0 \qquad (1)$$
$$F = \textit{damper force}$$
$$V = \textit{damper deformation speed}$$
$$P = \textit{dissipated power}$$

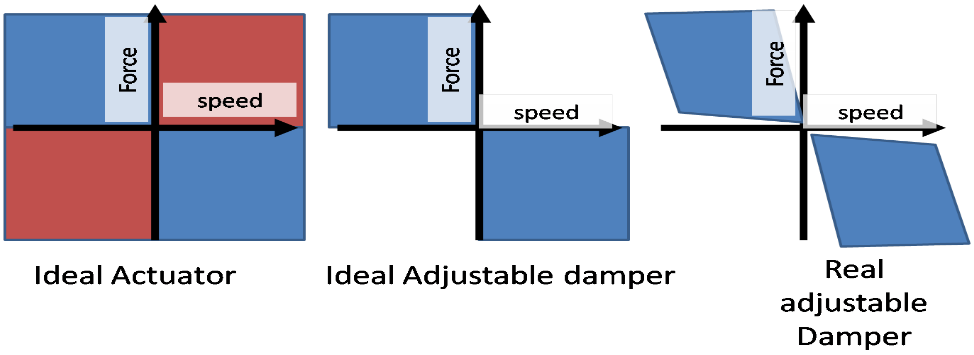


Figure 2: force-deformation speed diagrams of active and semi-active actuators

In this work Authors propose the use of magneto-rheological dampers as semi-active actuators.

The operating principle of this type of dampers is based on the particular behavior of magneto-rheological fluids. When a magnetic field is applied the disposition of ferromagnetic particles suspended in the fluid is affected. A limited or constrained position of this particles produced a “thickening effect” in the fluid increasing its apparent viscosity.

Usually this effect is modeled using a modified Bingham model [6,7] of the fluid described by equations (2) and (3).

$$\tau = \tau_y(H) + \eta\dot{\gamma} \qquad (2)$$
$$\tau = \textit{fluid stress}$$
$$\tau_y(H) = \textit{field dependent yeld stress}$$
$$H = \textit{magnetic field}$$
$$\dot{\gamma} = \textit{fluid shear rate}$$
$$\eta = \textit{plastic viscosity}\,(\textit{measured for } H = 0)$$
$$\textit{below the yeld stress}\,(\textit{strains of } 10^{-3})\,\textit{the fluid react viscolastically according}\,(3)$$
$$\tau = G\gamma, \tau < \tau_y \qquad (3)$$

The so called Bouc-Wen Model [8] is sometimes used to obtain a more accurate description of the hysteretic response below the yield strain. The approximated Carreu fluid model [10] is instead more suitable for a finite element modeling. The latter was used by Authors to design and to simulate the behavior of customized MR dampers suitable for this application. More detailed results of these activities will be surely the object of future publications.

Using multiphisic FEM models and heuristic calculations taken from literature, Authors were able to design compact MR actuators suitable for this particular application, as shown in Figures 3/a/b.

The force exerted by the damper is approximately proportional to the applied magnetic field *H*. Since the field is produced by an electromagnet/solenoid, the force is also proportional to the applied current *I* and it's quite independent from damper deformation speed *V*. So the damping force exerted by the damper can be easily regulated without the need of complex models or force sensors. The electrical power required is quite low, about 10-20W. In case of system failure the fluid is "free" to flow inside the damper in function of its (low) "natural" viscosity; in these conditions the equivalent damping is minimum (but not zero). However, with a proper design and/or the use of a backup permanent magnet, it is possible to assure/adjust this minimum value to a desired optimal value.

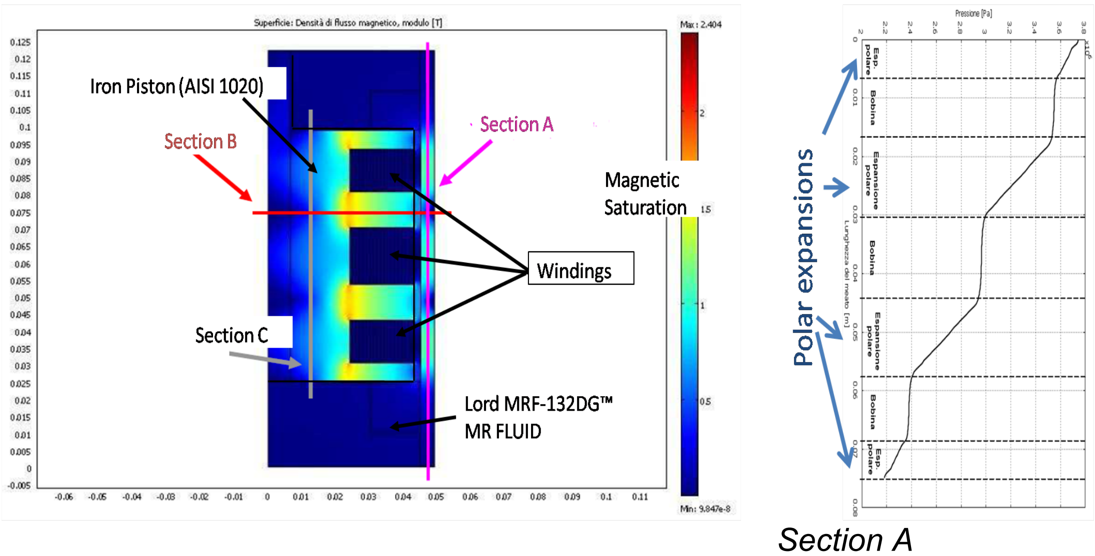


Figure 3/a: An example of Comsol™ FEM Model of MR damper, magnetic flux distribution (left), and corresponding pressure drop in the fluid flow on section A (on right)

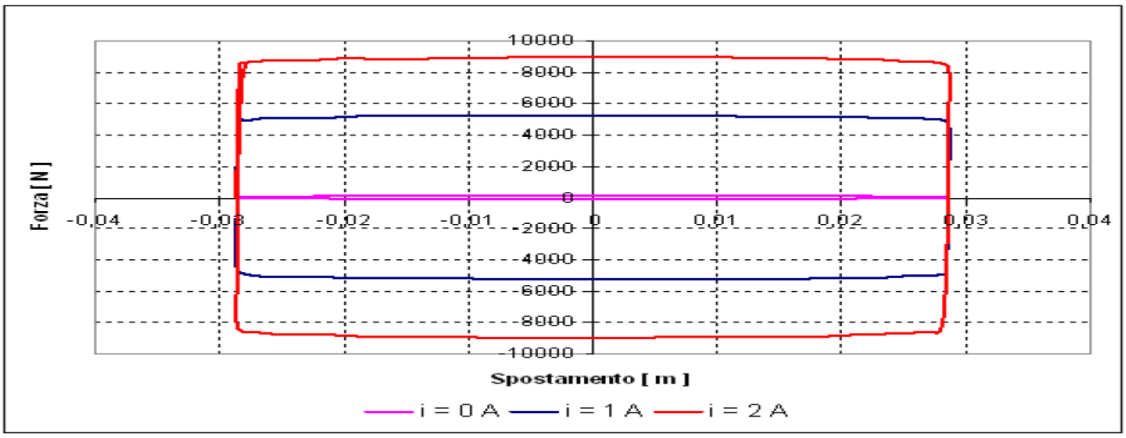


Figure 3/b: Example of force-displacement diagram for an MR damper with sinusoidal imposed motion(max speed about 0.1m/s), when different currents are applied on windings

2 CONTROL PRINCIPLES: THE SKYHOOK DAMPER TECHNIQUE

There is a huge bibliography in scientific literature concerning of semi-active suspension control systems [4, 5, 10, 11, 12, 13, 14, 15].

Most of these techniques could be treated as particular cases of the more general concept of mechanical impedance control [16].

In particular most of the proposed impedance controls try to emulate or to shape virtual damping in order to dissipate mechanical energy associated to vibrations or mechanical instability phenomena. This approach is often called/classified as Skyhook Control.

In Figure 4, a simplified scheme referred to a SDOF (Single Degree of Freedom) system is shown. This model is often called “quarter vehicle model” [15] when referred to vertical suspension system.

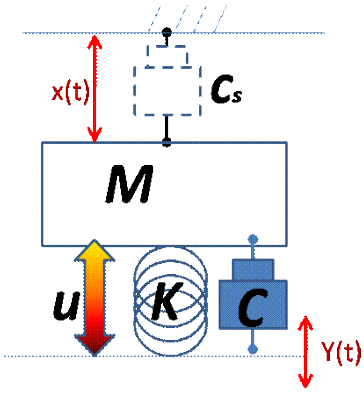


Figure 4: example of Skyhook Control applied to SDOF system

Equation (4) represent the dynamical behavior of the system.

$$m\ddot{x}+c(\dot{x}-\dot{y})+k(x-y)=u \tag{4}$$

A control force u is applied in order to emulate a desired dynamical response modeled by (5). In particular the force C_s generated by a virtual viscous damper (see Figure 4 for more details) is emulated.

$$m\ddot{x}+c(\dot{x}-\dot{y})+c_s\dot{x}+k(x-y)=0 \tag{5}$$

By subtracting equation (5) from (4), the requested control force is then calculated by (6):

$$u=-c_s\dot{x} \tag{6}$$

More generally, for a linear MIMO system whose dynamical behavior is described by eq. (7) through the classical State-Space formulation. By imposing the desired response (8), it's possible to calculate the corresponding vector of control actions *[u]*(9).

$$[\dot{x}]=[A][x]+B[u] \tag{7}$$

$$[\dot{x}]=[A_{des}][x] \tag{8}$$

$$[u]=[B]^{-1}[A_{des}-A][x] \tag{9}$$

In this way it's theoretically possible to emulate a the dynamics of every electro-mechanical component. For example, a suspension system characterized by a virtual spring with variable stiffness [4].

2.1 Limitations arising for the adoption of semi-active systems

Since vehicles, or more generally real mechanical systems are implicitly non-linear, linear eqs. (7), (8) and (9) have to be obtained as a local linearizationn of the system.

Moreover, if the system is semi-active, the actuators are adjustable dampers; this implies that their response is highly non linear since relations (1), (2) and (3) have to be satisfied.

The consequence is that the forces *[u]*calculated according to eq. (9) can be applied only if they are admissible according to damper behavior. This limitations also affect control on system frequency response. Considering the example in Figure 4, the calculated control action (6) has to be reproduced by an adjustable damper whose response can be roughly described by (10) or, in the particular case of a MR actuator, by (11).

$$u = c_d(\dot{x} - \dot{y}) \tag{10}$$

$$u = c_{mr}\, sign(\dot{x} - \dot{y}) \tag{11}$$

From (6), (10), (11), c_d or c_{mr} can be calculated obtaining (12) and (13).

$$c_d = \frac{-c_s \dot{x}}{(\dot{x} - \dot{y})} > 0 \tag{12}$$

$$c_{mr} = \frac{-c_s \dot{x}}{sign(\dot{x} - \dot{y})} > 0 \tag{13}$$

Since both coefficients have to be positive the difference between body speed $\dot{x}$ and disturbance $\dot{y}$ have to be negative and higher of a minimum value. At low frequencies $\dot{x}$ and $\dot{y}$ are almost equal so the correction of static or low frequencies disturbances may be difficult. Comparing (11) and (13) the behavior of MR damper seems to be better than conventional ones. Also considering also bandwidth and reliability characteristic of MR damper are quite superior to conventional valve-controlled adjustable dampers.

The use of these components has also some further consequences: the desired operating point can be reached and stabilized only if it is a stable equilibrium point corresponding to a minimum of system energy. This last physical consideration deeply influences the design of lateral suspension system where the lateral air spring, introduced for instance on the Sumitomo[17,18] solution, is used to introduce a centering force able to assure or to enforce the existence of a stable equilibrium point in the lateral direction. Some photos and schemes of Sumitomo semi-active lateral suspension system installed on Shinkansen high speed trains are shown in Figures 5/a/b/c

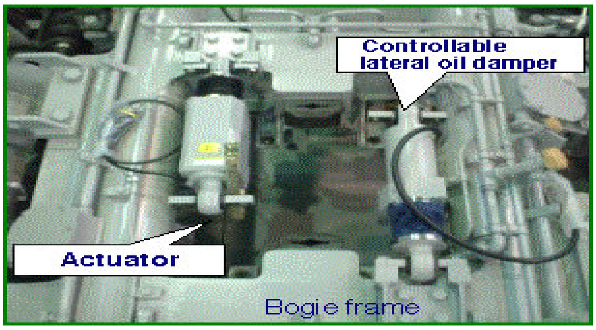


Figure 5/a: Semi-active lateral suspension system on Shinkansen E2-1000 and E3[17,18]

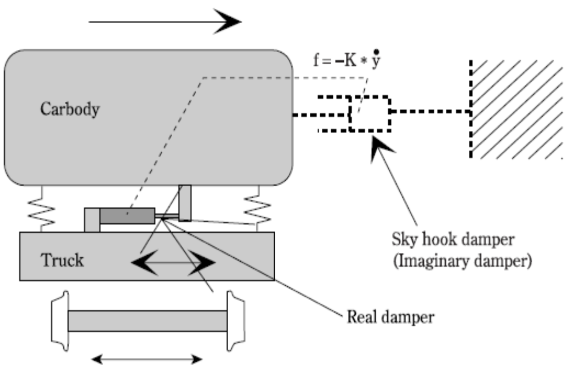


Figure 5/b: Principle of operation of semi-active lateral suspension system installed on Shinkansen E2-1000 and E3[17,18]

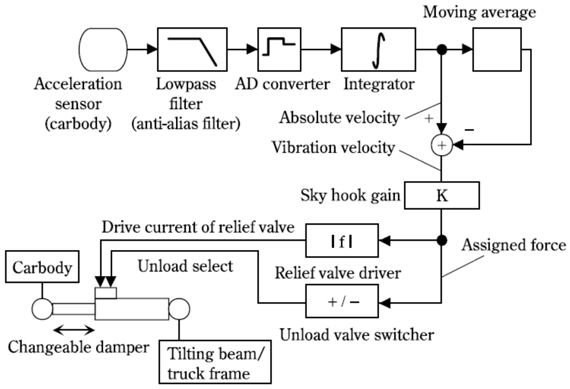


Figure 5/c: Layout of lateral semi-active suspension system according [17,18]

3 VEHICLE MODEL

3.1 Vehicle description

Authors have developed a complete multibody model of a coach of an high speed train in order to test performances of the proposed solution. This reference model is inspired to state of the art existing trains, and in particular to Ansaldo EMU V250.The vehicle is composed by a carbody with two bogies; the layout of the suspension system is shown in Figure 1 and is composed by:

•a double stage suspension system with lateral and anti-yaw dampers.

•a torsion bar to prevent an excessive souplesse of carbody

•a Watts linkage to assure the transmission of longitudinal efforts between bogies and the carbody

•bumpstops, on vertical and lateral direction to prevent excessive carbody souplesse on curves.

•Wheel profile is a standard ISO ORE S1002 while the rail is supposed to be a standard UIC 60 (1:20 of rail cant)

•Rigid wheel base is supposed to be 2.85m and the pivot pitch is 16.9m.

In Figure 6 axis alignment of the reference system is shown. Data concerning damping and stiffness of vertical suspension stages and axleboxes are shown in Table 1, inertial properties of wheelsets, axleboxes, bogies and carbody are listed inTables 2 and 3. Non linear responses of dampers and bumpstops are modulated as tabulated functions as visible in Figures 7/a/b/c/d.

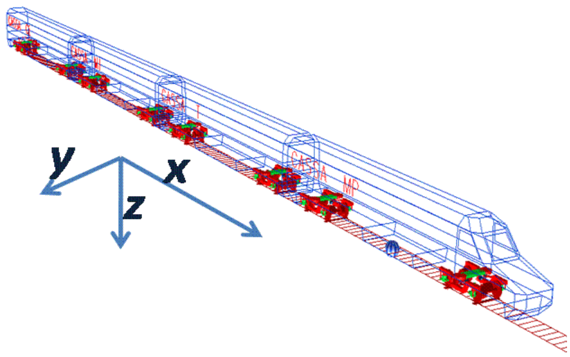


figure 5: axis orientation of the fixed/inertial reference frame

Axle Box		Primary Susp.Stage	
K_x [*N/m*]	*40000000*	**K_x** [*N/m*]	*844000*
K_y [*N/m*]	*6500000*	**K_y** [*N/m*]	*844000*
K_z [*N/m*]	*40000000*	**K_z** [*N/m*]	*791000*
K_{Al} [*Nm/rad*]	*45000*	**K_{Al}** [*Nm/rad*]	*10700*
K_{Be} [*Nm/rad*]	*9680*	**K_{Be}** [*Nm/rad*]	*10700*
K_{Ga} [*Nm/rad*]	*45000*	**K_{Ga}** [*Nm/rad*]	*0*
C_x [*Ns/m*]	*15000*	Secondary Sup. Stage	
C_y [*Ns/m*]	*10000*	**K_x** [*N/m*]	*124000*
C_z [*Ns/m*]	*0*	**K_y** [*N/m*]	*124000*
		K_z [*N/m*]	*340000*

Table 1: damping and stiffness of primary/secondary suspension stages and of axleboxes.

wheelset		**Axlebox**	
M [*kg*]	*1554.5*	**M** [*kg*]	*212,2*
I_{xx} [*kg*m^2*]	*800*	**I_{xx}** [*kg*m^2*]	*3*
I_{yy} [*kg*m^2*]	*160*	**I_{yy}** [*kg*m^2*]	*12*
I_{zz} [*kg*m^2*]	*800*		

Table 2: inertial properties of wheelsets and axleboxes

	Carbody	**Bogie**
M [*kg*]	*44700*	*2874*
I_{xx} [*kg*m^2*]	*112000*	*2400*
I_{yy} [*kg*m^2*]	*2190000*	*1900*
I_{zz} [*kg*m^2*]	*2150000*	*4000*
X_g [*m*]	*-0.174*	*0*
Y_g [*m*]	*0*	*0*
Z_g [*m*]	*-2.050*	*-540*

Table 3: inertial properties of carbody and bogies

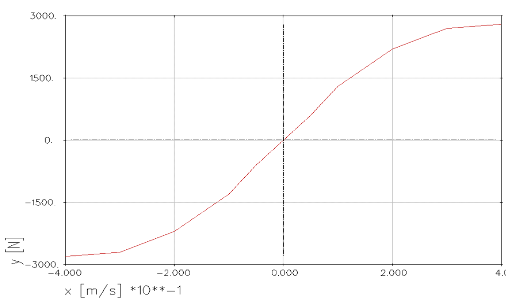


Figure 7/a: force/velocity mechanical characteristic for the primary suspension stage

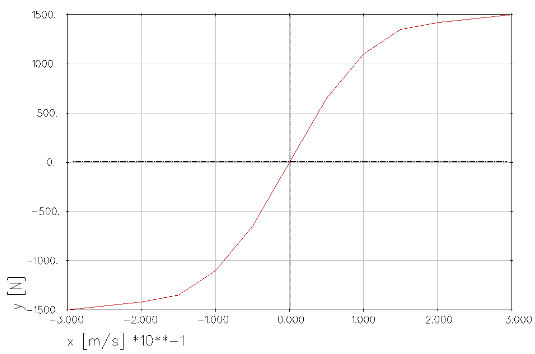
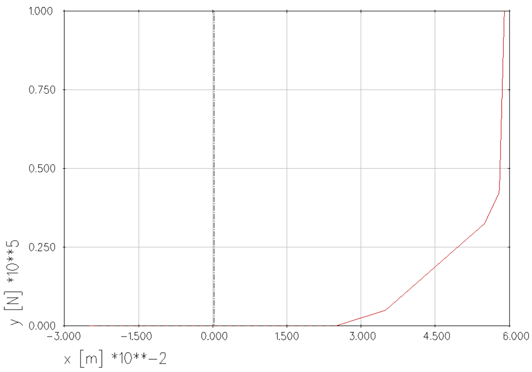


Figure 7/b: force/velocity mechanical characteristic for the secondary suspension stage



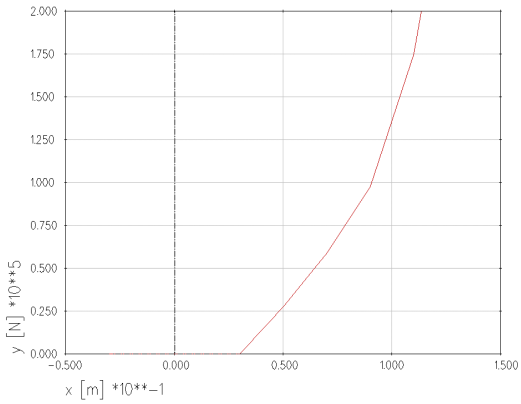


Figure 7/c: force/deformation mechanical characteristic for lateral end run

Figure 7/d: force/deformation mechanical characteristic for vertical end run

3.2 Model Implementation

For simulation and fast prototyping of mechatronic on-board subsystems, an approach based on the co-simulation is often very useful to fully exploiting the possibilities of the large number of commercially available specialized simulation software. Typical is the association of a multibody model of the mechanical enviroment, developed with specialized commercial software like MSC Adams™ or Intec Simpack™, and the models of on-board electro-mechanical (or pneumatic, or idraulic, …) subsystems developed with more flexible and suitable environments like Matworks Matlab-Simulink™ or Lms Amesim™.

For this work Authors have preferred to develop the whole dynamic model using only Matlab and Simulink™.

In particular, for the development of the multi-body model of the vehicle the Simulink-Simscape-SimMechanics™ library was used. Since non-linear dampers and bumpstops were not available in this software, Authors have developed some customized SimMechanics blocks by modifying the existing "*Body Spring and Damper*" block.

Authors have used a Simulink model developed during previous research activities [19, 20] in order to model the interaction between rail and wheels. The wheel-rail contact model is tri-dimensional and it considers all the relative motions between rail and wheel. Multi-contact condition is simulated and a maximum of four separated hertzian contact points can be simulated for every wheel (corresponding to a maximum of eight points for every axle).

The Simulink model has been developed with a modular architecture reproducing the layout and functionalities of single vehicle sub-components and assemblies (wheels, bogies, carbody etc.) as shown in Figure 8. A Matlab Simulink™ pre-processor is used to automatically design and generate the virtual track by considering curves, clothoid transitions, cant and track irregularities.

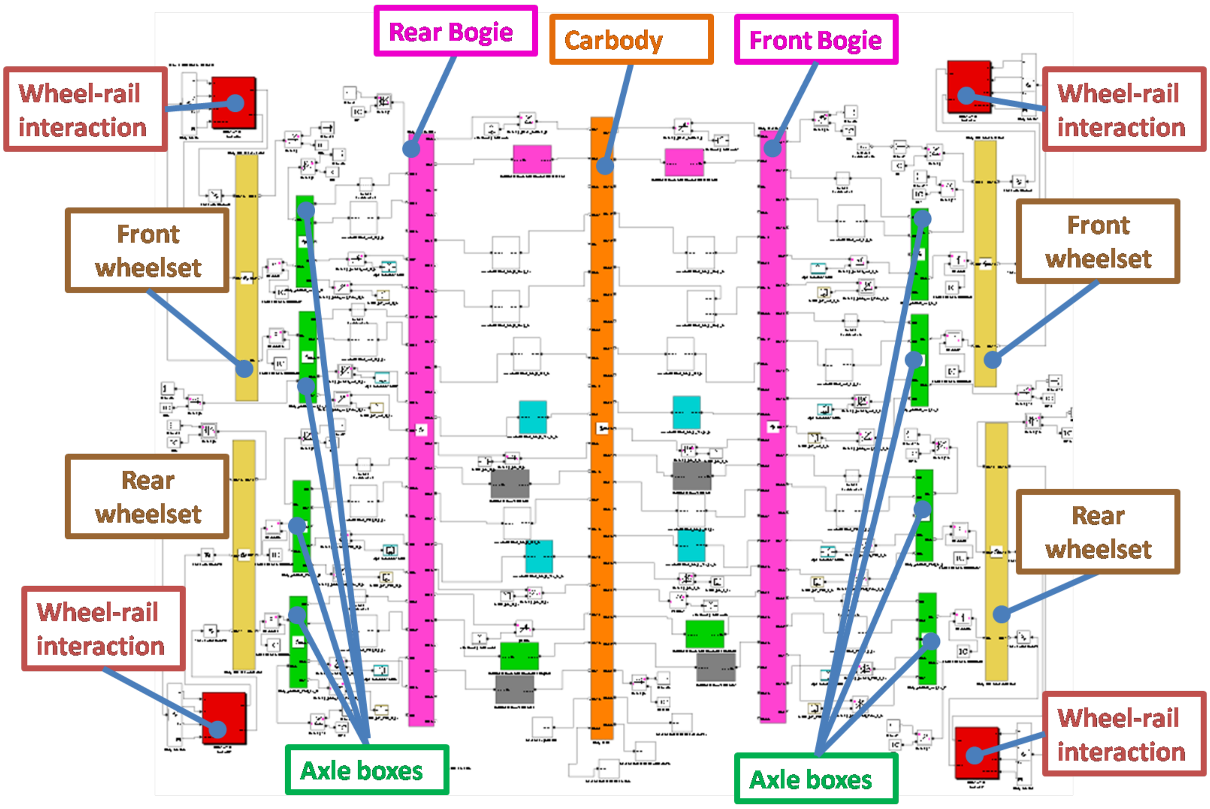


Figure 8: Mathworks Matlab-Simulink™ multibody vehicle model

4 DEFINITION OF REFERENCE PERFORMANCES

Performances of the proposed solutions are compared (in simulations) to those obtained by a vehicle equipped with passive/standard dampers. The comparison involves both stability and comfort considerations, so two different simulation test run are defined:

1. *Comfort Test*: vertical, lateral and track irregularities are applied on a straight track as shown in Figure 9/a. Accelerations are measured in three different position of the carbody, as shown in Figure 9/b. Measured accelerations are used to calculate a_{zp95} and N_{mv} comfort index, according to the prescriptions of regulations in force [21]. Measured accelerations are filtered using the weighting curves shown in Figure 9/c.
2. *Lateral Ride Test*: The irregularities described in Figure 9/d are applied on a straight track (700 m long). A sharp, discrete lateral and gauge irregularity applied at about an half of the track length is superimposed to a noisy continuous disturbance.. Smoothing and decaying of lateral accelerations is observed in order to evaluate how the proposed solution affect lateral ride quality.

The line is supposed to be perfectly straight and rail profile a standard UIC60 1/20 one in both test.

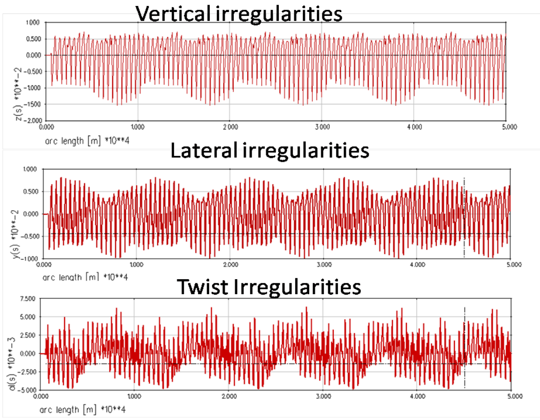


Figure 9/a: irregularities applied on track for *Comfort Test*

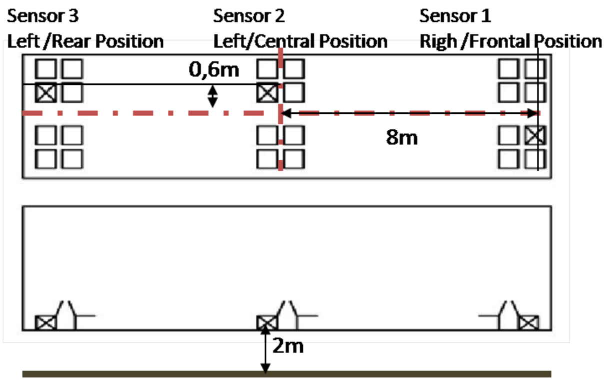


Figure 9/b: position of acceleration measurements on vehicle carbody

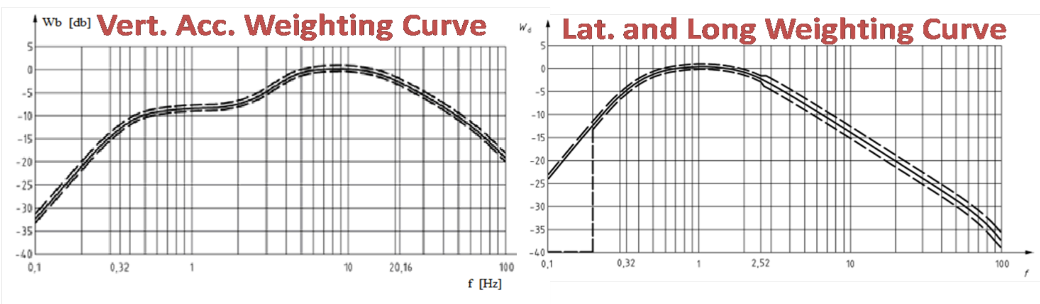


Figure 9/c: frequency weighting curve applied to acceleration measurements according regulations in force [21]

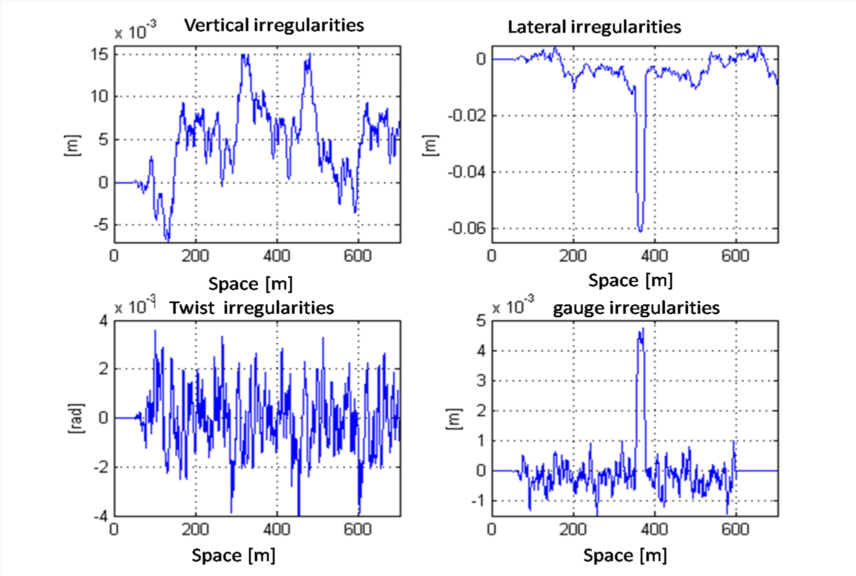


Figure 9/d: irregularities applied for *lateral ride test* [21]

5 IMPLEMENTATION OF THE PROPOSED SEMI-ACTIVE SUSPENSION SYSTEM

As previously mentioned, in this first study the possibility of substituting the lateral and the vertical passive dampers of the secondary suspension stage with semi-active Magneto-Rheological Actuators is investigated.

5.1 MR Actuator implementation

Using the FEM model described in section 1 of this document, Authors were able to properly design and dynamically model MR actuators with the same encumbrances of the corresponding passive dampers. Dynamic performances of actuators have been calculated through FEM models simulation results. FEM models were also very useful to calibrate an approximated model of actuators to be implemented on the Matlab-Simulink model.

In particular the MR damper is modeled using approximated relation (14) where the damper force u is calculated as the sum of a Coulombian-like contribution which value is proportional to the input current I (c_{imr} is the proportionality factor) and a residual viscous term depending on c_d.The variables $\dot{x}$ and $\dot{y}$ respectively represent the velocities of the damper piston and the cylinder, so the term $\dot{x}-\dot{y}$ is the deformation speed.

$$u=c_{imr}\, I\, sign(\dot{x}-\dot{y})+c_d(\dot{x}-\dot{y}) \tag{14}$$

In order to simulate the magnetic field saturation the maximum input current I has a maximum value I_{max} and it implies a corresponding upper limit to the maximum force that the actuator is able to exert. A first older filter is also applied to the input current I in order to simulate delays corresponding by several different physical phenomena:

1. since the adjustable magnetic field is produced by a solenoid, this inductive load unavoidably introduces a delay;
2. the changes in MR properties due to the variation of the magnetic field are very fast, but a delay of about 10^{-3} s (minimum) is suggested by scientific literature;
3. Elastic and compliance effects of both fluid and damper mechanical structure introduce a further cut-off in the damper frequency response.

The corresponding Simulink implementation is shown in Figure 10

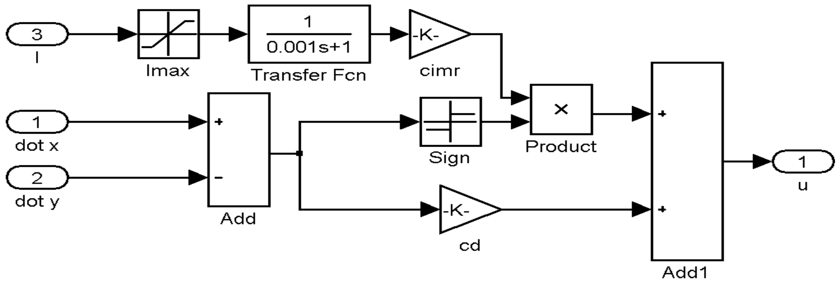


Figure 10: Simulink implementation of an MR Damper

5.2 Control implementation: SISO Skyhook Control

Several works are present in scientific literature concerning control and stabilization techniques applied to railway vehicles [4, 11, 13, 22, 23, 24, 25]. In this paper Authors have approached the problem by testing a quite simple solution: every actuator has its own independent control loop. Every actuator then works independently from the others with a very simple and robust logic shown in Figures 11/a/b:

1. carbody and bogie frame accelerations are measured using inertial sensors placed in correspondence of dampers junctions. Corresponding velocities are calculated by integrating acceleration measurements;
2. calculated velocities are used to implement a Skyhook control system according the model described by eqs. (4), (5) and (6);
3. since the actuator is a "real" MR damper, the control force u have to be compatible with the non-linear behavior of the actuator as described by equation (14). In particular it is possible to apply the control force u only if it dissipates mechanical energy. If this condition is satisfied, the corresponding current I is applied to the actuator producing the desired force. Otherwise no current is provided to the actuator, that is consequently free to move with the minimal residual viscous damping c_d.

The main advantage of this algorithm is the fast and simple calibration and the robustness. Since every actuator has an independent control loop, the failure of a single component in most cases will produce a limited performance de-rating. The whole control system emulates an array of 6-8 Skyhook dampers. It is clearly redundant with respect to the effective number of controlled degree of freedom (only 5, as shown in Figure 11/b):vertical and lateral translations, roll, pitch and yaw rotations. It is also quite simple to take count of the non-linear behavior of the actuators; this would make much more difficult the design of a centralized control system.

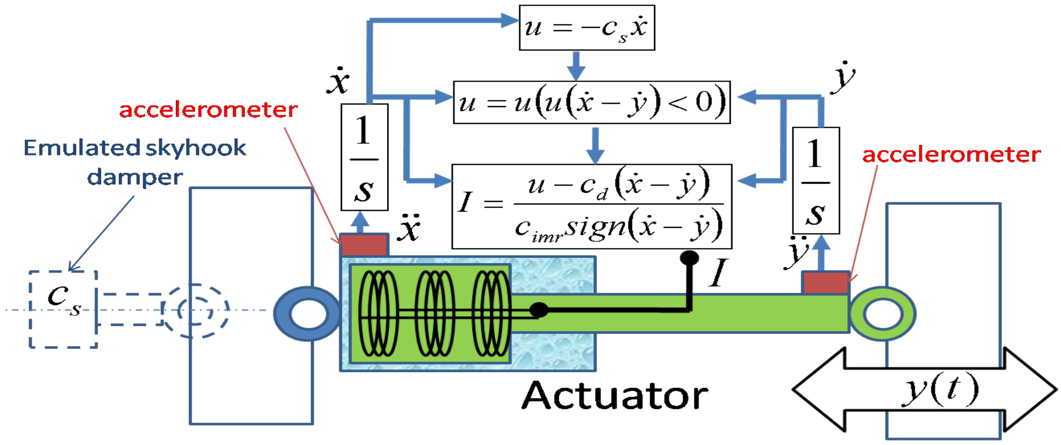


Figure 11/a: independent SISO control loop applied to the MR Actuator

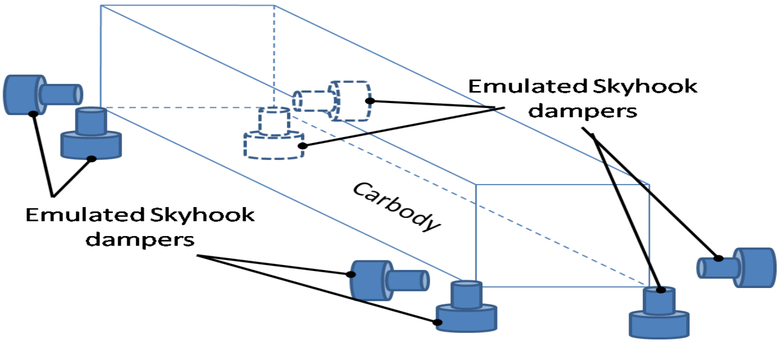


Figure 11/b: semi-active suspension system emulate an array of 6-8 skyhook dampers.

6 SIMULATION RESULTS

The control system described in section 5 was optimized iteratively using the testing procedure described in section 4. The emulated damping factors of the sky-hook dampers were the only parameters. In particular for lateral and vertical suspension dampers two different values of emulated damping, C_{sky_vert} and C_{sky_lat} were defined. Performances have been evaluated in terms of acceleration a_{zp95} and comfort index N_{mv} defined as recommended by regulations in force[21], especially when analyzing the results of "*comfort test*" simulations. Authors didn't define a particular index for analysis of *lateral ride* simulations: the chosen calibration set which corresponds to a smooth, damped, lateral response of the system in terms of lateral and vertical acceleration.

This heuristic approach has lead to near to optimal set of parameters described in Table 4.

C_{sky_vert}	*200000 Ns/m*
C_{sky_lat}	*50000 Ns/m*

Table 4: near to optimal calibration set

Applying this sub-optimal calibration set, relative improvements a_r of acceleration a_{zp95} and N_r of comfort index N_{mv} of about 20-40% are recorded as visible in figures 12/a/b.

a_r and N_r are defined as stated by (15) and (16).

As shown in Figures 11/a/b, significant improvements a_r of acceleration a_{zp95} and N_r of comfort index N_{mv} (about 20-40%) were reached by applying this sub-optimal calibration set.

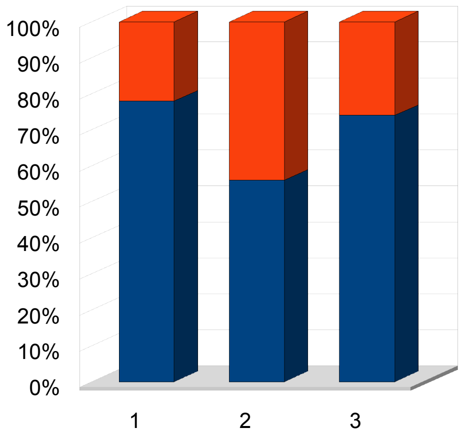
As stated, indexes a_r and N_r are defined by (15) and (16).

$$a_r = 100 \frac{a^a_{zp95} - a^p_{zp95}}{a^p_{zp95}} \tag{15}$$

$$N_r = 100 \frac{N^a_{mv} - N^p_{mv}}{N^p_{mv}} \tag{16}$$

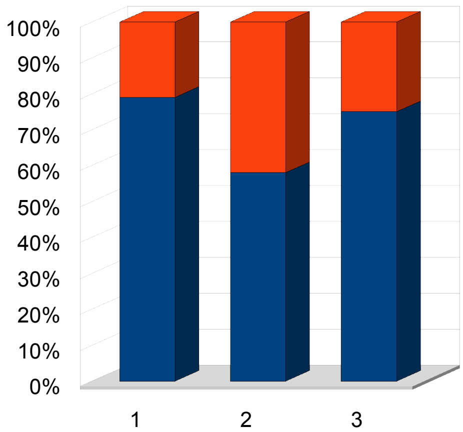
where a^a_{zp95} , N^a_{mv} *are measured when semiactive suspension system is working* a^p_{zp95} , N^p_{mv} *are measured with standard passive suspension system*

More in details, best results are obtained for the measurement point *2*, corresponding to the center of the carbody, as shown in Figure 9/b. The differences in comfort index values among the various monitored points are mainly due to significant carbody rotations, whose kinematic center is quite near to the carbody midpoint.



Sensor	a_r
1	78%
2	56%
3	74%

Figure 12/a: relative improvement a_r of a_{zp95} for the three measurement points defined as in figure 9/b



Sensor	N_r
1	79%
2	58%
3	75%

Figure 12/b: relative improvement N_r of N_{mv} for the three measurement points defined as in figure 9/b

The results obtained analyzing the *Lateral Ride Tests* are More interesting. In particular, in Figures 13/a/b/c the vertical and lateral accelerations measured on different carbody positions are shown. By comparing the obtained performances with those of the passive suspensions ones, the following considerations can be done:

1. the maximum values of accelerations due to an impulsive lateral irregularity are quite similar;
2. thanks to the actions of semi-active dampers, both vertical and lateral oscillation decay with a smooth behavior.

Actuators response have been modeled using accurate data obtained from a multi-physic FEM model. So simulated actuators have maximum force limitations that reflect a very realistic behavior. This maximum force limitations have been chosen considering maximum forces exerted by passive dampers and some other considerations about stiffness, resistance and modal response of the attached structures of carbody and bogie frames. As a consequence, the maximum effort is limited to reasonable values.

In Figure 14, the effort exerted by two of the four vertical semi-active dampers is shown. Here it's clearly visible that the response of the controlled system heavily suffers of the limitations arising from the force saturation, especially for the first two-three carbody oscillation periods. As a consequence, carbody roll and yaw rotations are insufficiently damped and this is one of the causes partially justifying the different performances achieved on different carbody points. It is realistic to retain that designing actuators in a different way, better performances could be achieved. However the more important result of these preliminary tests is that the control system architecture was demonstrated to be sufficiently robust, even if the applied external excitation it's much larger than the expected one and all the non-linearities of the system are excited.Further test have been repeated by simulating a malfunction on one of the vertical semi-active dampers. The results were quite good: the system have demonstrated to remain stable, low performance reductions were registered, especially those affecting roll and pitch rotations of carbody. From the stability point of view, an improvement of about 5-10% of the bogie critical speed has been demonstrated. It's a light, but noticeable, improvement: it's feasible that in real operative conditions the application of this solution will have no negative consequences on vehicle stability even considering a very pessimistic scenarios

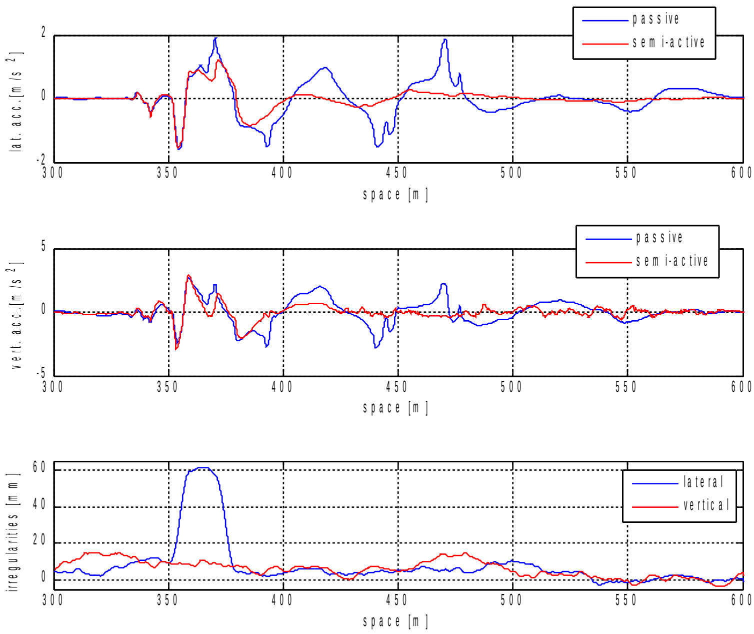


Figure 13/a: vertical and lateral carbody accelerations measured on point *2* as a function of position and applied irregularities

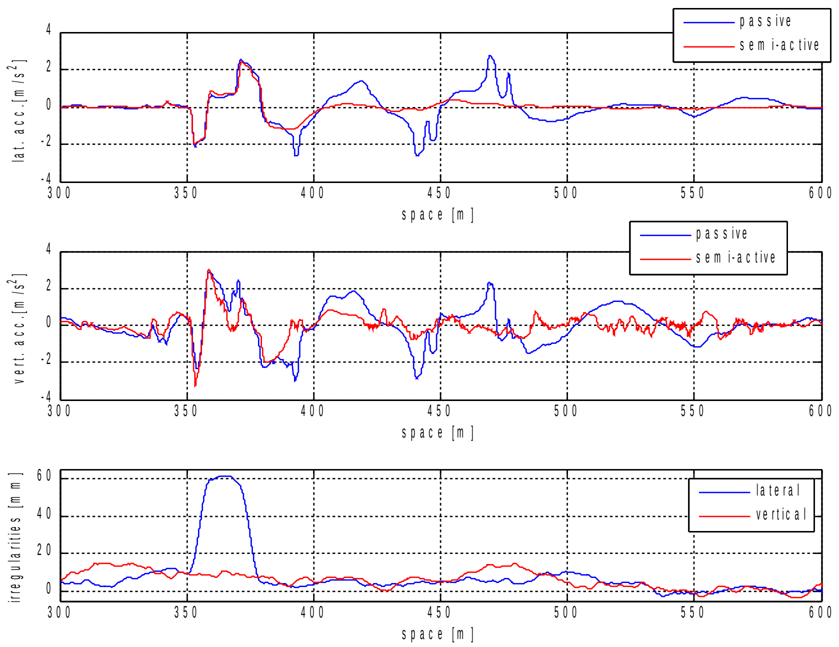


Figure 13/b: vertical and lateral carbody accelerations measured on point 3 as a function of position and applied irregularities

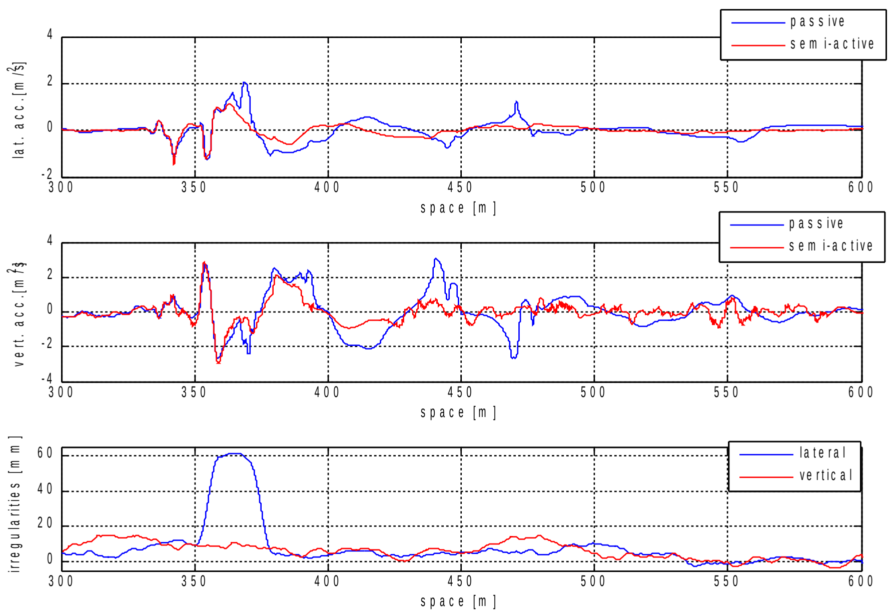


Figure 13/c: vertical and lateral carbody accelerations measured on point 1 as a function of position and applied irregularities

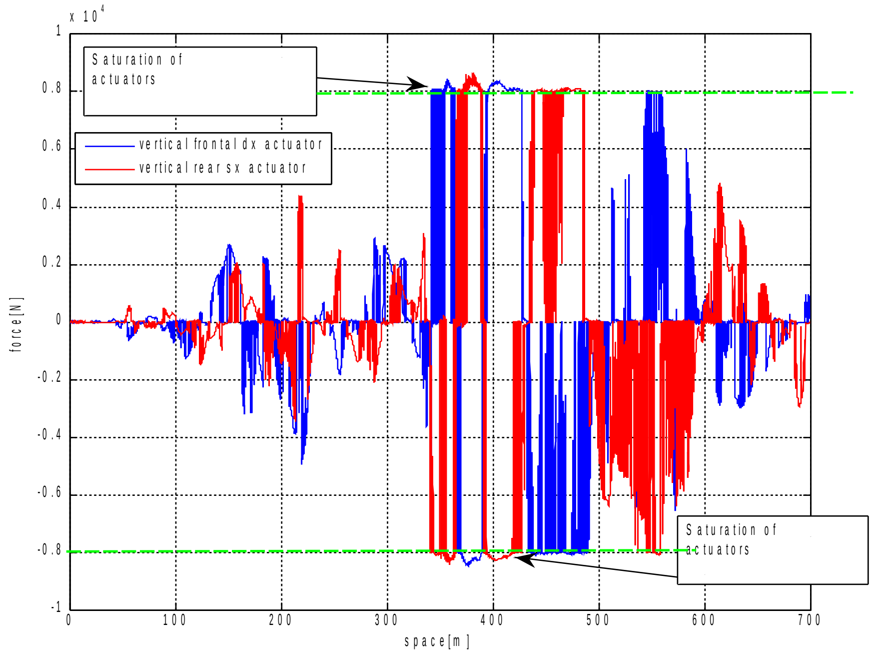


Figure 14: force exerted by two vertical semi-active dampers the left one on the rear bogie and the right one on the frontal bogie

CONCLUSIONS AND FUTURE DEVELOPMENTS

In this paper has been presented a complete tri-dimensional model of a railway vehicle developed in the Matlab-Simulink environment. The use of this particular environment makes easier further applications concerning the design and the simulation of on board mechatronic sub-systems or even a real time implementation for Hardware in the Loop testing of existing components. In this paper an application a semi-active suspension system based on Magneto-Rheological dampers and sky-hook control technique is proposed.

Some preliminary simulations done with quite realistic testing conditions seems to confirm feasibility of the studied solutions.

As further development authors will investigate the following matters:

1. How different actuators design and layout can affect system performances. For example also anti-yaw dampers can be semi-active.
2. A different control algorithm may be used to improve system performances or to use in a more efficient way the actuators. Probably a single MIMO (multi input-multi output) regulator can reach higher performances than the proposed solution based on an array of SISO regulators. Since the used actuators have an high non linear behavior. Authors are working on optimization algorithms based on Gradient Projection Method [26,27,29] that have been successfully used to solve model problems with non linear or mono-lateral constraints. Further improvements may be introduced in order to introduce self-calibration properties to the regulator through the introduction of adaptive controllers,[15,29] or modal considerations [30]. Also other mechanical layout, for example the application of MR actuators to anti-yaw dampers, should be investigated.

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REFERENCES

[1] Paolino Camposano *Italy's Pendolino a succesful experiment International Railway Journal* June 1979.

[2] Bombardier Transportation free technical documentation downlodable at http://www.bombardier.com/en/transportation/sustainability/technology/flexx-tronic-technology?docID=0901260d800486e2

[3] Ralph Streiter, Marco Boller,Dr. Britta Riege,Richard Schneider,Günther Himmelstein, *Active Lateral Suspension for High Speed Trains tech documentation* free download at www.**streiter**.com/Veroeffentlichungen/230.pdf

[4] R. Goodall*, Active Railway Suspensions: Implementation Status and Technological Trends,*Vehicle System Dynamics, Volume 28, Issue 2 & 3 August 1997 , pages 87 – 117

[5] S. Iwnicky, R.Goodll, T.X.Mei, *Handbook of railway vehicle Dynamics, Cap. 11 Active Suspensions,* CRC PRESS Taylor and Francis

[6] Lord Corporation-Material Division , *Engineering Note Designing with MR Fluids,* free technical documentation available at site www.lord.com

[7] Andrzej Milecki, *Investigation and control of magneto–rheological fluid*, International Journal of Machine Tools & Manufacture 41 (2001) 379–391

[8] Y.K. Lau e W.H. Liao "*Design and analysis of magneto-rheological dampers for train suspension*", Journal of Rail and Rapid Transit 219 (2005), pag. 261-276

[9] M. Benedetti e E. Dragoni "*Nonlinear magnetic analysis of multi-plate magnetorheological brakes and clutches*" COMSOL Users conference Milano 2006

[10] Delphi Magneride™ Technical Documentation site of the manufacturer www.delphi.com

[11] Peel, D.J., Stanway, R., Bullough, W.A., "Design Optimisation of a Controllable Vibration Damper for Vehicle Suspension Applications", Active Control of Vibration and Noise, American Society of Mechanical Engineers, Design Engineering Division Publication, DE-Volume 93, 1996, pp. 205-214.

[12] Ahmadian, M., Poynor, J.C., Gooch, J.M. "Application of Magneto Rheological Dampers for Controlling Shock Loading", American Society of Mechanical Engineers, Dynamic Systems & Control Division (Publication) DSC-Volume 67 1999. pp. 731-735.

[13] K.Sasaki A lateral semi-active suspension of tilting train, Q.Report of RTRI Vol. 41 (2000) , No. 1 pp.11-15. W. H. Liao, D. H. Wang, *Semiactive Vibration Control of Train Suspension Systems via Magnetorheological Dampers* ,Journal of Intelligent Material Systems and Structures, Vol. 14, No. 3, 161-172 (2003)

[14] A Alleyne, JK Hedrick , *Non linear control of a quarter car active suspension*, proc. Of IEEE Control Conference, American, 1982

[15] Dean Karnopp; Donald Margolis; *Adaptive Suspension Concepts for Road Vehicles,* Vehicle System Dynamics, Volume 13, Issue 3 November 1984 , pages 145 – 160

[16] L. Sciavicco and B. Siciliano, *Modelling and Control of Robot Manipulators*. New York: McGraw-Hill, 1996

[17] K.Sasaki *A lateral semi-active suspension of tilting train,* Quarterly Report of RTRI Vol. 41 (2000) , No. 1 pp.11-15

[18] Sumitomo Metals, *Technical Documentation,*

[19] Monica Malvezzi, Enrico Meli, Jury Auciello, *Evaluation of wheel-rail contact points with semi-analytic methods*, Vehicle System Dynamics Vol 00, No. 00, January 2008 1-30

[20] Monica Malvezzi · Enrico Meli · Stefano Falomi · Andrea Rindi, *Determination of wheel–rail contact points with semianalytic methods*, Multibody Syst Dyn (2008) 20: 327–358

[21] pr EN 12299 *Railway Applications-Ride comfort for Passengers- Measurement and evaluation* Version of July 2006

[22] S.Shen, T.X.Mei, R.M.Goodall, J.Pearson and G.Himmelstein: A Study of Active Steering Strategies for a Railway Bogie. IAVSD03, Kanagawa Japan, 2003.

[23] Goodall, R. and Li, H., *"Solid axle and independently-rotating wheelsets-a control engineering assessment",* Journal of Vehicle System Dynamics, 1999

[24] Mei, T.X. and Goodall, R.M. *"Wheelset Control Strategies for a 2-Axle Railway Vehicle"*, Journal of Vehicle System Dynamics. Vol. 33 Supplement, 2000, pp.653-664

[25] M Gretzschel, TX Mei and O Vaculin: *Simulation of an Integrated Mechatronic Train*, IAVSD 2001

[26] Luemberger, D. G., 1984, *Linear and Nonlinear Programming,* Addison-Wesley Publishing Company, Reading, Massachusetts.

[27] Benedetto Allotta,Valentina Colla,Glanluca Bioli, *Kinematic Control of Robots With Joint Constraints*,Journal of Dynamic Systems, Measurement, and Control SEPTEMBER 1999, Vol. 121 / 435

[28] Charles A. Klein,Sakon Kittivatcharapong, *"Optimal Force Distribution for the Legs of a Walking Machine with Friction Cone Constraints"* IEEE TRANSACTIONS ON ROBOTICS AND AUTOMATION, VOL. 6, NO. I, FEBRUARY 1990

[29] Sunwoo, Myoungho; Cheok, Ka C. *Investigation of Adaptive Control Approaches for Vehicle Active Suspension Systems,* proceeding of American Control Conference. 28-29 june 1991 Pages 1542-1547

[30] Braghin, F.; Resta, F.; Sabbioni, E.; *A modal control for active/semi-active suspension systems* Proceedings of AIM 2007, 4-7 September 2007 Pages 1-6