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Vibration Reduction in a Washing Machine via Damping Control

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Abstract: The aim of this work is the analysis and design of a control system for the reduction of the mechanical vibration and the perceived acoustic noise in a washing machine. The control system is implemented via a semi-active magnetorheological damper located on the suspension that links the drum to the cabinet. The entire design procedure is outlined. The semi-active actuator is accurately described. An experimental protocol is proposed and tested on a sensored machine to highlight the system dynamical behavior. On this basis a simple adaptation control strategy is proposed, designed and tested. Finally some experiments are held in anechoic chamber: the reported results show the effectiveness of the proposed control system.

Keywords: mechatronics; appliances; semi-active control; vibration; acoustic noise.

1. INTRODUCTION

Among the many different types of controlled suspensions (see e.g. Ahmadiam and Song, 1999; Foo and Goodall, 2000; Campi *et. al.*, 2003; Fischer and Isermann, 2003; Savaresi *et* and Spelta, 2007; Savaresi *et al.* 2005a; Silani *et al.*, 2004; Spelta and Savaresi, 2007), semi-active suspensions have received a lot of attention since they seem to provide the best compromise between cost (energy-consumption and actuators/sensors hardware) and performance. The concept of semi-active suspensions can be applied over a wide range of application domains: suspensions in road, rail and agricultural vehicles, suspensions of appliances (e.g. washing machines), architectural suspensions (buildings, bridges, etc.), bio-mechanical structures (e.g. artificial legs) etc. This work focuses on the semi-active control of the suspension in a washing machine, namely the suspension system which links the drum to the machine cabinet. In this kind of appliance the aim of the suspension is to damp the drum movements and to reduce the vibrations transmitted to the chassis, which are strictly related to the perceived acoustic noise. The control objective is to reduce the vibration level measured on the cabinet panels, with no loss of performance in terms of damping.

The key idea is to replace the passive damping unit with a magnetorheological damper (MRD), capable of changing its damping characteristic according to a current command.

The problem of semi active control in appliances such as washing machine has been recently treated and discussed (see Chrzan and Carlson, 2001; Papadopoulos and Papadimitriou, 2001). Usually the focus is on the tub dynamics, in particular at low spin speed where drum resonance is induced; this work instead focuses on the entire dynamic range of the machine. The main contributions of this note are the following: an accurate comparison of different suspension mounting configurations; the design, implementation and testing of a control strategy able to ensure a reduction of the machine vibration level. Some test are held in an anechoic chamber with the aim to show the effectiveness of the proposed vibration-oriented control in term of acoustic-noise reduction.

The outline of this paper is as follows: in Section 2 the semi-active damper used in the washing machine suspension is described, and its main features are illustrated. Section 3 introduces the experimental set up. Section 4 is devoted to the characterization of the machine in different configurations. The control strategy is presented in Section 5, and then experimentally evaluated in Section 6. Section 7 ends the paper with some conclusive remarks.

Fig.1. The Ariston Aqualtis washing machine.

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2. CHARACTERIZATION OF THE MR DAMPER

The magnetorheological semi-active component used in this work is the MR Controllable Friction Damper RD-1097-01 (Fig.2) developed and distributed by Lord Ltd. This component can change continuously the damping characteristics and it is specifically designed for this kind of applications. The main feature of this device is that the damping force is obtained as friction between a foam covering the piston and the external case of the component (it is also named "sponge" damper). This foam is saturated with magnetorheological fluid. By applying a magnetic field to the foam it is possible to change its friction level. The magnetic field is generated by a current in a coil built in the damper piston. The friction force can be modulated continuously by a current imposed to the damper unit. Notice that this kind of component does not have fully integrated electronics, but it needs to be driven externally. Hence this cannot be considered a *smart* device (Savaresi, 2006).

Fig.2. The RD-1097-01 Friction Damper by Lord Ltd.

A concise control-oriented description of a semi-active damper is constituted by two main aspects: the description of *damping characteristic*, namely the capability of the device to deliver force with respect to the control signal (i.e. the current); the analysis of the *switching time,* namely the time necessary to the electric circuit to drive the damping force.

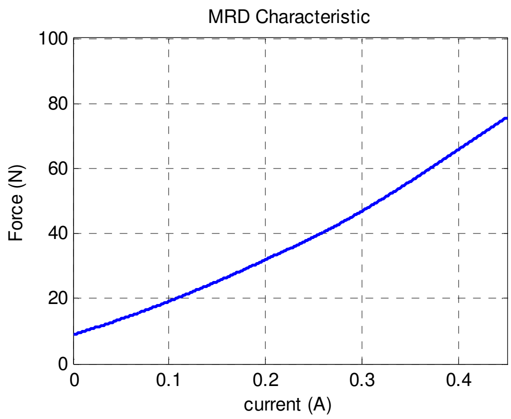

Fig.3. Characteristic of the MRD Sponge Damper.

The damper characteristic is pictured in Fig.3, where the friction force delivered by the damping unit is displayed with respect to the driving command. Some comments can be done:

- The delivered force does not depend on the velocity of the piston rod. Notice that this kind of device is regarded as a friction actuator . This behavior is typical of a low cost damper, and appropriate to this application. (Savaresi *et. al.* 2005b)

- According to the provided specifications, the current range is 0-450mA. Within this range the device can deliver a force of ±78N maximum. This level of force is consistent with the typical force measured for damping the drum movement in a washing machine.

- The semi-active actuator's forces range is very large. The ratio between minimum and maximum force level is 1:8. This ratio is an appealing feature for control design purposes.

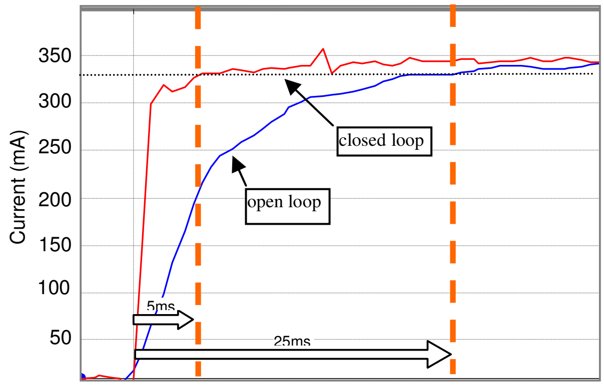

Fig.4. Time response to a step-like current command. Open and closed loop response.

In semi-active devices, the force range is not the only control-relevant feature. Another key characteristic is the electric dynamic of the embedded coil, which can be approximated as an impendence-inductance system. Notice that the resistance of the electrical system may drift significantly, due to heating (Joule effect) or aging. Furthermore the damping is current driven, but the system is obviously voltage driven. So a drift of the nominal resistance means a drift of the nominal damping. In order to preserve the nominal characteristics of the device, a current feedback control is necessary. In this work a proportional-integral controller has been implemented in the internal ECU (the design details are here omitted for the sake of conciseness).
The closed-loop electrical dynamic can be well approximated by the settling time of the current step response (the step is on the input voltage). The nominal "open-loop" response time of the device is about 25ms (see Fig.4). Notice that a closed loop control of the electrical dynamic has the benefit of reducing the settling time to 8ms.

3. EXPERIMENTAL SET UP AND TESTIGN PROTOCOL

The washing machine object of this work is an Ariston Aqualtis (Fig.1) produced and delivered by Indesit Group S.p.A. A schematic view of the machine components is reported in Fig.5. The drum, the clothes and water, the motor, and the transmission unit constitute the so called suspended mass. The suspended mass is linked to the top panel of the cabinet by two coil springs, and it is linked to the cabinet base by two damping units.

Vibrations in a washing machine are induced by the movements of the suspended mass during a spin cycle, in the case it is not balanced, i.e. the masses position is so that the centre of gravity does not coincide with the geometric centre of the drum (Mc Donald, 1998). These movements are transmitted to the cabinet through the suspension system. Notice that the spin cycle is commonly the most annoying working condition in term of acoustic noise.

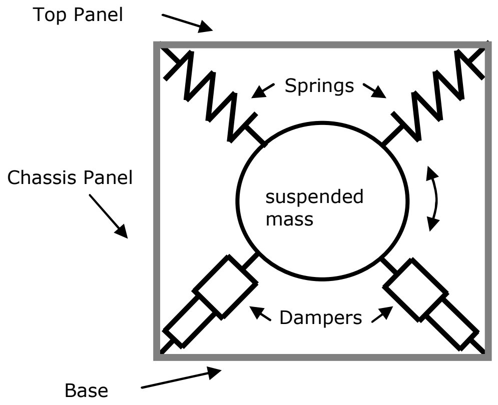

Fig.5. Scheme of the washing machine.

In order to evaluate the washing machine performances and to control the dampers appropriately, the laundry is equipped with the sensors showed in Fig.6: two 3-axis accelerometers monitoring the top panel and the chassis panel, labelled as A1 and A2 respectively, and characterized by a ±2g range and a bandwidth of 130Hz; a 3-axis accelerometer monitoring the drum dynamic, labelled as A3, and characterized by a ±6g range and a bandwidth of 130Hz

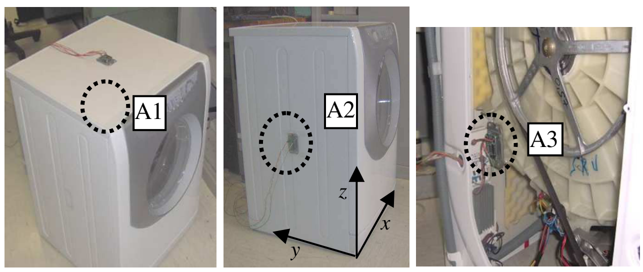

Fig.6. Sensor positioning. Top Panel (A1), Chassis Panel (A2), Drum (A3) accelerometers.

In order to fully characterize the machine, the following set of experiments is defined:

- The machine is tested for several spin velocities, from 100 to 1400rpm, with a step of 100rpm. Each spin cycle is 60sec long.
- Every set of spin cycles is repeated for several unbalanced weight, accurately positioned in the drum room. No details about the unbalanced mass are provided for confidentiality reasons.

4. CHARACTERIZATION OF THE "PASSIVE-LIKE" MACHINE

The current passive configuration includes two damping units. In designing a new suspension, it is possible to analyze the effect of different solutions. In this section the performances comparison of different kinds of suspension scheme is presented. Herein the magnetorheological device is used passive-like, namely with a constant current command with no modulation. So the following configurations are explored:

- Passive damper (current configuration);
- No dampers;
- 2 MR Dampers driven with constant current from 0mA to 450mA, with a 50mA step;
- 1 MR positioned motor side (labeled as DX);
- 1 MR positioned opposite to motor side (labeled as SX);

In order to evaluate the performance of the above configurations the following performance index is defined:

$$J_{vib}(spin) = \sum_{axis=x,y,z} \frac{\sum_{t=1}^{N}\left(a_{chassis}(t;axis)\right)^2}{N} + \sum_{axis=x,y,z} \frac{\sum_{t=1}^{N}\left(a_{top}(t;axis)\right)^2}{N} \quad (1)$$

Index (1) indicates the overall vibration level and it represents the total amount of energy of the accelerations measured over the tree axes on the panels. Notice that index (1) depends only on the vibrations of the cabinet (top and chassis panel); these vibrations are responsible of the acoustic noise perceived during a spin cycle.

The results of the experimental are condensed in Fig.7, where performance index (1) is reported with respect to the spin speed, for every proposed suspension configuration. The magnetorheological configurations (1 or 2 devices) are concisely represented in the case of their best damping condition. The results reported in Fig.7 are very clean, and the following remarks can be done:

- The spin velocity from 1200rpm to 1400rpm are the working conditions where the highest vibration level is measured. A high level of vibrations is measured also at 1100rpm for 1-MRD-SX configuration. However in the rest of the note we explore only the best mounting configuration, namely the 1-MRD-DX.
- The "no dampers" configuration presents good filtering performance in the critical spin velocities range. Unfortunately this configuration is not applicable since the drum movements are not damped in correspondence of the natural resonance of the suspended mass. This resonance is clearly visible in term of the cabinet vibrations at about 200rpm. Notice that in this working condition the drum hits repeatedly the top panel with a critical stress of the entire system. This represents the classical suspension trade-off: a low damping has a good filtering of high frequency vibration but it is not able to vanish the low frequency resonances. So every passive damper is a

compromise between best filtering and best damping.

- A single MR damper positioned at motor side represents the best applicable suspension configuration in terms of vibration level. Notice that no resonance appear around 200rpm, and furthermore this configuration outperforms the passive configuration for every working condition.

- There is no a unique current value driving the MRD which ensures the best vibration level for every spin velocity. In other words it is not possible to fix a current a priori so that the best performance is guaranteed.

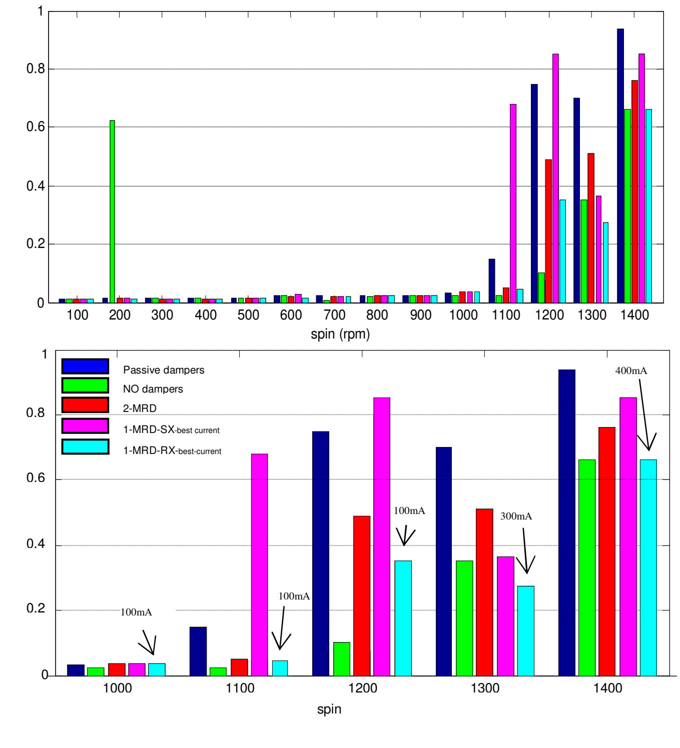

Fig.7. Performance index J_{vib} for different mounting configuration

5. ADAPTIVE CONTROL STRATEGY

In this section the adaptive control of the MR damper located on the DX side of the laundry is discussed. As already pointed out at the end of the previous section, the first control objective is to find the best current (fixed) value in a constant-spin working condition. It has been shown that the best current value can significantly vary; a run-time optimization hence is required.

The adaptation technique proposed herein is very simple, but it is consistent with the specific features of this application. It can be summarized as follows:

• when a constant-spin condition starts, the training phase is activate: the algorithm explores the whole current range with a comparatively slow ramp signal, possibly repeated several times;

• during this training phase, the instantaneous vibration level is measured, and a current→vibration map is built;

• at the end of the training phase, the current value corresponding to the minimum vibration level is computed, using a smoothed version of the current→vibration map; this "optimal" current is then applied for the remaining phase of the constant-spin condition.

In order to monitor the vibration level an on-line performance index is defined:

$$J(t) = \sum_{axis=x,y,z} \left(a_{chassis}(t;axis)\right)^2 + \sum_{axis=x,y,z} \left(a_{top}(t;axis)\right)^2 \qquad (2)$$

Index (2) stands for the instantaneous vibration level and it represents the on-line amount of energy of the accelerations measured over the tree axes on the panels, and transmitted by the drum movements. Notice that index (2) is the *instantaneous* version of the performance index (1).

Finally, during a spin cycle the best current is automatically chosen:

$$\bar{I} = \arg\min_{I} J \qquad (3)$$

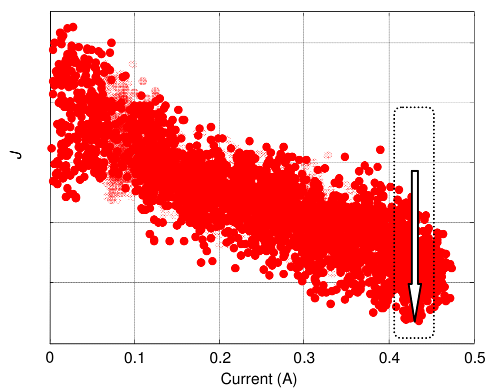

Fig.8. Performance index J with respect to the driving current. (1400 rpm)

The literature of adaptive control is vast, and a lot of methods and techniques have been proposed (see e.g. Goodwin and Sin, 1984, and references cited therein). The approach described above, however, is not a genuine adaptive-control method; it can be better classified as a batch optimization procedure performed on-line, when required. The choice of this simple approach is motivated by some peculiar features of this application:

• the controlled signal is not a single one, but a non-trivial combination of six acceleration signals, as stated by (2);

• the relationship between the MRD current and the vibration index is non linear and not monotonically increasing-decreasing (see Fig 7);

• the system is very noisy and the optimization-adaptation procedure must be terminated in a short time (only a few seconds are allowed to complete this optimization routine).

An example of the current-optimization cycle is illustrated in Fig.8, for the 1400rpm condition. Note that the current range 0A-0.45A is explored. It is clear the dependence of the vibration index on the MRD current. The best choice be clearly distinguished.

6. EXPERIMENTAL RESULTS OF CONTROL

In this section the experimental results of the proposed control strategy are reported. Two kinds of experiments are presented: a first test is focus on the measure and optimization of cabinet accelerations, according to the proposed strategy. A second test has been held in an anechoic chamber. The experiment goal is to show the relation between vibrations and acoustic noise, and then validate the proposed strategy in terms sound level reduction.

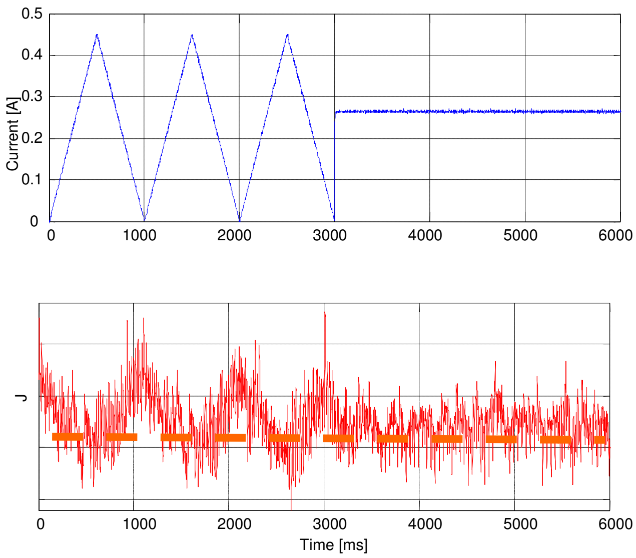

Fig.9. Adaptation strategy of the current in the time domain (1400rpm). Control signal (top) and performance index J (bottom - the scale is omitted for confidentiality reasons).

The experiment presented in Fig.9 concerns a 1400rpm spin test. Both the current and the performance index (2) are displayed in time domain: Fig.9 confirms the effectiveness of the control strategy: the performance index clearly varies with respect to the current, so that it is possible to find out the best friction force associated to the lowest vibration level. The strategy is able to compute and select the best current command in 3 seconds. The vibration level corresponding to the best chosen current is maintained during the rest of the spin cycle

The acoustic test has been made at the anechoic chamber of the Departimento di Energetica, Politecnico di Milano.

The acoustic noise is measured as a pressure variation of the air by a microphone at sampling time $\Delta T_f = 1\mu s$. The measured signal is then processed by a real time sound level meter (SLM 2900) developed by Larson Davis. The analyzer provides a sound index computed every $\Delta t = 125ms$ and defined as follows:

$$A_i(k\Delta t) = \frac{\Delta t}{\Delta t_f} \sum_{i=1}^{\Delta t/\Delta t_f} 20\log_{10} \frac{p\left((k-1)\Delta t + i\Delta t_f\right)}{p_0} \qquad (4)$$

Where $p(t)$ is the acoustic pressure measured by the microphone, and $p_0 = 2mPa$ is a standard reference for sound measurements, and it represents the sensitivity limit of the human ear. Equation (4) provides a standard overall index, and it stands for the averaged measure of acoustic noise, sampled every 125ms. Finally it is commonly expressed in Decibel as the human ear is roughly *logarithmic*.

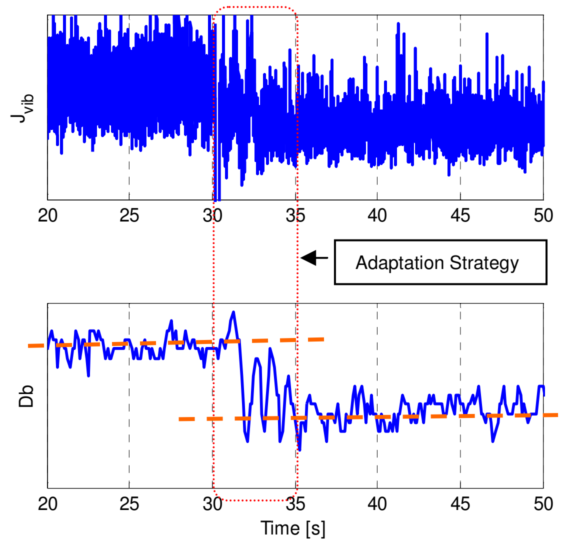

Fig.10. Experiment in anechoic chamber. Performance index of vibration level (top) and acoustic noise (bottom) during a 1400rpm adaptive control test. Both the scales are omitted for confidentiality reasons.

The results of the test are reported in Fig.10. The test refers to a 1400rpm spin cycle and both the performance index (2) and the acoustic level (4) are displayed in time domain. The results are clean and the following comments can be drawn

- The searching procedure is clearly visible and it can *measured* by the microphone.

- The improvements in term of vibration level corresponds to an improvement of perceived noise. This confirm both the effectiveness of the adaptation strategy and the relationship between panels vibrations and acoustic noise.

7. CONCLUSIONS

In this work the complete development and analysis of a damping control system for a washing machine suspension has been presented. The experimental analysis clearly shows that the proposed damping control ensures a low level of vibration for every working condition of the machine. Finally the experiment in anechoic chamber shows the effectiveness of such a control in terms of acoustic noise.

REFERENCES

Ahmadian M., X. Song (1999). A non-parametric model for magneto-Rheological dampers. Proceedings of 1999 ASME Design Engineering Technical Conference, Las Vegas, Nevada.

Campi M.C., Lecchini A., Savaresi S.M. (2003). An application of the Virtual Reference Feedback Tuning (VRFT) method to a benchmark active suspension system. European Journal of Control, vol.9, pp.66-76.

Chrzan M.J., J. David Carlson (2001) MR Fluid Sponge Devices and Their Use in Vibration Control of Washing Machines. Proceedings of the 8th Annual Symposium on Smart Structures and Materials. Newport Beach, CA, USA.

Fischer D., R. Isermann (2003). Mechatronic semi-active and active vehicle suspensions. Control Engineering Practice, vol.12, n.11, pp.1353-1367.

Foo E., R.M. Goodall (2000). Active suspension control of flexible-bodied railway vehicles using electro-hydraulic and electro-magnetic actuators. Control Engineering Practice, vol. 8, pp.507-518.

Goodwin G. C., K. S. Sin (1984). Adaptive Filtering Prediction and Control. Englewood Cliffs, NJ: Prentice-Hall.

McDonald K.T. (1998) Physics in the Laundromat. American Journal of Physics. Vol. 66, Issue 3, pp. 209-211.

Papadopoulos E. and I. Papadimitriou (2001): Modeling, Design and Control of a Portable Washing Machine during the Spinning Cycle Proceedings of the 2001 IEEE/ASME International Conference on Advanced Intelligent Mechatronics Systems Vol. 2, pp:899 – 904.

Savaresi S.M., C. Spelta (2007). Mixed Sky-Hook and ADD: Approaching the Filtering Limits of a Semi-Active Suspension. ASME Transactions: Journal of Dynamic Systems, Measurement and Control. Vol. 129, Issue 4, pp. 382-392.

Savaresi S.M. (2006). The role of real-time communication for distributed or centralized architectures in vehicle dynamics control systems. 6th IEEE International Workshop on Factory Communication Systems. Torino, Italy (plenary presentation), pp.1-6.

Savaresi S.M., E. Silani, S. Bittanti (2005a). Acceleration-driven-damper (ADD): an optimal control algorithm for comfort-oriented semi-active suspensions. ASME Transactions: Journal of Dynamic Systems, Measurement and Control, vol.127, n.2, pp.218-229.

Savaresi S.M., S. Bittanti, M. Montiglio (2005b). Identification of semi-physical and black-box non-linear models: the case of MR-dampers for vehicles control. Automatica, vol.41, pp.113-117.

Silani E., Savaresi S.M., Bittanti S., Fischer D., Isermann R. (2004). Managing Information redundancy for the design of fault-tolerant slow-active controlled suspension. Tire Technology International, vol.2004, pp. 128-133.

Spelta C., S.M. Savaresi. (2007) Mix-1-Sensor: a control strategy for semi-active suspension. Proceedings of the IFAC Advances in Automotive Control, Monterey, CA, USA.