

# Semi-Active Control of Cab Suspension in an Agricultural Tractor via Magneto-rheological actuator

Cristiano Spelta, Fabio Previdi, Sergio M. Savaresi, Diego Delvecchio, Simone Tremolada

**Abstract—** The topic of the paper is the analysis of the use of semi-active cabin suspension for agricultural vehicle. A magnetorheological damper is located in the cabin suspension with the aim of improving the comfort perceived by passengers. The entire control system design is outlined, with a focus on the experimental activity. An analysis of the suspension performances without control is proposed in order to highlight the potential trade-off of the system. The control of damping may overcome this trade-off as experimentally evaluated.

## I. INTRODUCTION

AMONG the many different types of controlled suspension systems (see e.g. [2, 8-12] semi-active suspensions have received a lot of attention since they 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: road vehicle suspensions, cabin suspensions in trucks or tractors, seat suspensions, suspensions in trains, suspensions of appliances (e.g. washing machines), architectural suspensions (buildings, bridges, etc.), bio-mechanical structures (e.g. artificial legs) etc ([1, 3, 6, 7, 13, 15]. This work focuses on the cabin suspension of agricultural vehicle, responsible of the passenger comfort.

The idea is to replace the current damper with a magnetorheological actuator (MR Damper) in order to perceive a comfort oriented control of the cabin vertical dynamics. In the literature, the problem of semi-active damping on agricultural vehicles has been recently treated and discussed [4, 15]. These works mainly focus on the comfort improvement obtained by controlling the seat damper. Recently the cabin control has been proposed [4]. It focuses on a completely different kind of vehicle layout and it

proposes a common control strategy based on “skyhook” algorithm [5], tested on road experiments.

The main contribution of this paper are:

- An analysis of the suspension dynamics and trade-off due to the limited stroke travel.
- A control strategy experimentally evaluated that overcomes the limits of a passive suspension

## II. EXPERIMENTAL SET-UP AND TEST-RIG.

In this section the experimental vehicle and the test-rig are presented.

The tractor tested in this work is a prototype vehicle, where the cabin of the tractor is characterized by a sophisticated set of suspensions (Figure .1):

- two rubber mounts at the cabin front;
- two full-fledged suspensions at the cabin rear.
- the available stroke of the suspension is 60mm. It is worth noticing that this represents a limit in terms of comfort, since an end-stops collision may occur.

Due to the comparatively high roll stiffness (caused by the small-stroke rubber mounts at the cabin front), the main movements of the cabin are the pitch rotation and the heave translation. This work mainly focus on the rear suspensions, since the cabin comfort is highly sensible to its configuration (the rear suspension is the most difficult to tune, but it is also responsible for the major vibration-attenuation benefits). Both left and right rear suspensions are equipped with the following sensors, so that the suspension dynamic is monitored (Figure .1 and Figure .2):

- a piezoelectric accelerometer cabin side, labelled as A1, and characterized by a  $\pm 4g$  range and by a 100Hz bandwidth;
- a piezoelectric accelerometer chassis side, labelled as A2 of the same characteristics of the accelerometer A1;
- An angular potentiometer labelled as S1 and characterized by a  $\pm 45^\circ$  range and by 50Hz bandwidth. Thanks to a leverage system, the measured angle is numerically converted to the suspension stroke measure;

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This work was partially supported by MIUR project *New methods for Identification and Adaptive Control for Industrial Systems*”, and by SAME Deutz Fahr SpA.

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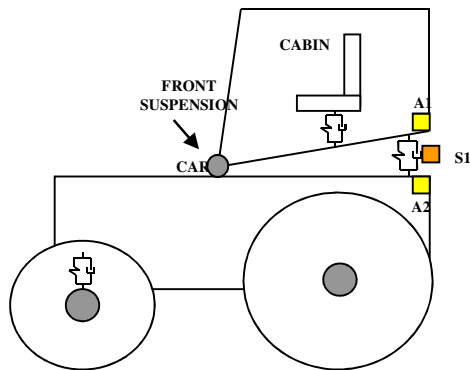


Figure .1. Graphical representation of the tractor and of the sensors layout.

The signals of the sensors have been measured with a 12-bit laptop-based portable *National Instruments* acquisition device, at the sampling frequency of 1KHz.

The experimental facility used in this work is a state-of-the-art “four-poster” based plant provided by MTS™; this test-rig is located at the CRA-ING at Treviglio (Bergamo, Italy), and it is intended for full-scale vehicles up to 15 Tons. The test-rig is basically composed by an high pressure hydraulic system, a reinforced concrete seismic mass and an electronic control device. The maximum force of each actuator is 160 kN, and the allowed excursion is of 250 mm. The dynamic characteristics are 1.6 m/s speed and 30 m/s<sup>2</sup> acceleration with a range of vibration reproduction of 0.1-100Hz. The test-rig is consistent with this kind application and with the experimental protocol, described in the next section.

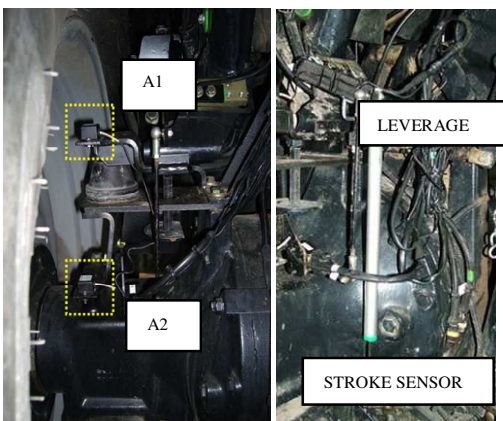


Figure .2. Sensors positioning on the rear cabin suspension.

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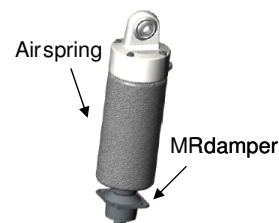


Figure .3. The Lord Ltd MRD Cab Suspension Unit.

### III. CHARACTERIZATION OF THE SEMIACTIVE DAMPER AND LOW LEVEL CONTROL

A magnetorheological damper can change the damping thanks to the physical property of the fluid flowing inside. Indeed the damping level is not varied by moving electro-actuated orifices, but by applying a magnetic field to the fluid [9]. Therefore the component can change continuously the damping due to an imposed current.

The magnetorheological semi-active device used in this work is a prototype developed by LORD Ltd., and it is specifically designed for this kind of application. As shown in Figure .3, it comprises a load-levelling air spring and a magnetorheological damper. The damping characteristics can be varied continuously and it is current driven. The electric characteristics are as follows:

- Minimum current: 0 Ampere, corresponding to the minimum damping level;
- Maximum current: 2.5 Ampere, corresponding to the maximum damping level.
- The current command is internally controlled by a PI servoloop.

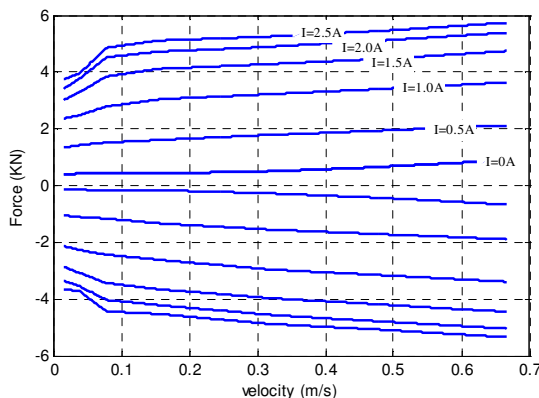


Figure .4. MR Damper characteristics for different values of current *I*.

In the design and testing of a semi-active systems an accurate description of the actuator is very important. A

good description of a semi-active damper is constituted by two parts:

- description of the damper characteristics;
- description of the dynamical behaviour of the current command. As obvious this is an important feature of this kind of component, since the damping is current-driven.

In Figure .4 the characteristics of the component are displayed in the Force-Velocity space, for several values of current. Some considerations are due:

- The explored speed range is  $\pm 6\text{cm/sec}$ ; the corresponding force range is  $\pm 5.5\text{KN}$ , for the curve of maximum damping; these speed and force ranges are consistent with the typical use of the cabin damper.
- The controllable damping range is very large; the ratio between the minimum ( $I = 0\text{A.}$ ) and maximum ( $I = 2.5\text{A}$ ) level is about 1:9. Notice that this is an important and appealing feature for control design purposes.
- The deliverable force at the maximum level is very high even at low speed. However this can be compensated by the control strategy which chooses the best current level at every control interval.
- The force delivered at each level is almost constant with respect to speed. In other words, the damping characteristics seem to be current depending only, so the magnetorheological damper can be viewed as a “friction actuator”.

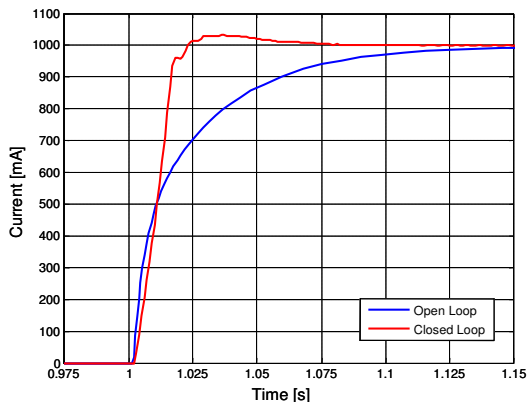


Figure .5. Current response of the MR Damper: open loop and closed loop performance

Both in control of the cabin suspension and in emulating the linear damper characteristics a fast transient of the damper current dynamic is necessary. In Figure .5 the current response of both open and closed loop MRD electrical dynamics are pictured. The feedback control of the current is actuated with a PI regulator, which is embedded in the vehicle control unit and runs at a control frequency of 1KHz. The controller design is here omitted for sake of conciseness. In Figure .5 it is shown that the closed loop response is about

12ms, compared to an open loop time response of 50ms. The switching time is fast enough to guarantee a good result in semi-active control. As a matter of fact the cabin vertical dynamic presents a main resonance at about 3 Hz (see next section). This means that a switching time of 12ms corresponds to about 1/28 of the oscillating period of the main vehicle dynamic. This ratio is acceptable for a good semi-active control of the vehicle dynamics

The friction behaviour makes this kind of component far from ideality. Indeed an ideal damper delivers a force proportional to the stroke speed. An ideal semi-active device regulates the proportional ratio between force and velocity. Moreover most of known semi-active strategies are designed under the assumption of an ideal damper [5, 13]. Yet the non-ideal behaviour is easy to overcome. From previous considerations, it is possible to assess that the delivered force  $F$  is approximately proportional to the current  $I$ ,  $F \propto I$ . Making the current proportional to the damper velocity  $I = Kv$ , the force becomes approximately linear with respect to the stroke speed,  $F \propto Kv$ . In this way  $K$  becomes the real control variable, which indicates the damping level of a virtual ideal damper obtained from the magnetorheological device.

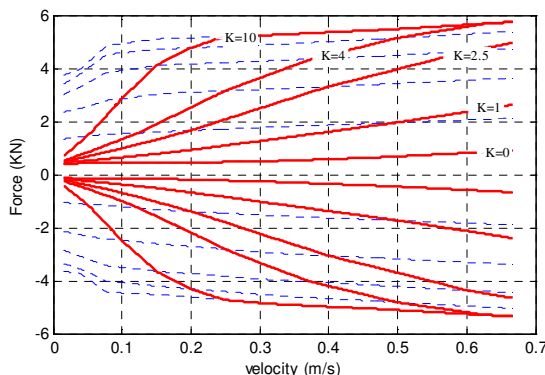


Figure .6. Linearized MR Damper characteristics

The “linearized” damping characteristics are depicted in Figure .6 and some considerations are due.

- The plotted characteristics are really similar to the ones of an ideal damper (see e.g [13]).
- The speed and force range are obviously still consistent with a device for cabin application, and the emulated ideal semiactive devices can be modulated continuously.
- A *linear* damping level is ensured if the maximum requested force is included in the force range of device (small value of  $K$ )
- In case of large value of variable  $K$ , the linearized damping level saturates to the non-linearized characteristic, at high velocity.

#### IV. EXPERIMENTAL ANALYSIS OF THE SUSPENSION DYNAMICS.

The objective of this section is to present an analysis of the suspension dynamics with a “passive-like” setting, namely

when the MR damper is controlled at fixed level of damping level  $K$ .

Since the use of cabin semi-active suspension is comfort oriented, the analysis of the performance is mainly concentrated on the vertical acceleration measured on cabin side. Indeed it is well known that the human body is sensitive to vertical accelerations. For evaluation the following index may be adopted [8, 15].

$$J = \frac{\sum_{t=1}^N [\ddot{z}_{cabin}(t)]^2}{\sum_{t=1}^N [\ddot{z}_{chassis}(t)]^2} \quad (1)$$

Where  $\ddot{z}_{cabin}$  represents the measure of the vertical cabin accelerations provided by the sensor A1;  $\ddot{z}_{chassis}$  represents the measure of the vertical cabin accelerations provided by the sensor A2.  $N$  is the number of samples in the considered time-history.

Note that index (1) is a concise time-domain representation of the comfort perceived by the driver. A frequency-domain representation of the comfort is provided by the “approximate frequency response” from the car acceleration to the cabin acceleration. Note that for non-linear systems the concept of “frequency response” is not well-defined. However for this kind of systems the “describing function” and the “approximate transfer function” represent a useful tool of analysis and comparison (see for further details [13]). To have a comprehensive analysis the suspension signals are also observed in the time domain. To this purpose the cabin acceleration signal and the stroke signal (measured by sensor S1) are reported.

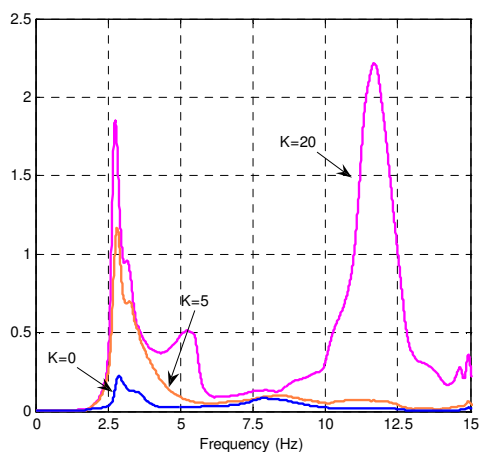


Figure 7. Spectra of the cabin acceleration for different values of  $K$ . (rear sweep excitation).

For analysis purposes a set of experiments on the test-rig is defined as follows:

- a sweep signal actuated on the front wheels of the vehicle, characterized by an amplitude of  $\pm 5\text{mm}$  and by a frequency that varies linearly from 0 to 15 Hz;

- a sweep signal actuated on the rear wheels of the vehicle, characterized by an amplitude of  $\pm 5\text{mm}$  and by a frequency that varies linearly from 0 to 15Hz;
- a random walk actuated on the four wheels simultaneously and corresponding to a real measurement on-field.
- a 10cm high “pothole” encountered by the rear wheel and by the front wheel separately. The amplitude is consistent with the common working profile of the suspension.

The test-rig protocol is complemented with a on-road experiment that excites strongly the vertical dynamic: a braking manoeuvres encountered at the speed of 30Km/h.

For the sake of conciseness, throughout this paper the results of rear sweep and rear step excitation are presented, as they provide a complete picture of the vertical vehicle dynamic of the vehicle.

The tests protocol has been proposed for different value of linear damping level  $K$ . The results about the sweep experiment on the rear wheels are described in Figure .7, where the power spectrum of the cabin-side acceleration is depicted. The following remarks are due:

- Due to the cabin mass and suspension stiffness, there exists a structural resonance at 1Hz that is over damped for every value of  $K$ .
- A clearly visible resonance appears at about 2.8Hz due to the vertical elasticity of the rear tires. A low damped suspension system is able to filter it.
- There is a resonance at about 5.5Hz induced by the front rubber mounts. However, it is well filtered for every value of  $K$ .
- A low damped suspension ( $K = 0$ ) ensures an optimal filtering at every frequency.

From this first analysis it appears that a low damped suspension may ensure the best comfort performance overall. This may suggest that a semi-active control is not strictly necessary, since there is no compromise among the use of “passive-like” settings. However this is true for excitations relatively low.

Notice that the sweep experiment has been designed so that the suspension works within its stroke range (linearity constraint). The “pothole” experiment represents a complementary analysis of the vehicle dynamics in order to show how the end-stop event can influence the perceived comfort.

In Figure .8 the results of the rear “pothole” experiment is pictured for two level of damping:  $K=0$  (minimum) and  $K=10$  (maximum), respectively. Both the suspension stroke and the related acceleration are depicted. The following consideration about the pothole experiment can be done:

- with a low damped suspension ( $K = 0$ ), the end-stop event occurs. The corresponding cabin-side accelerations present several peaks with a consequent discomfort;
- a high damped suspension ( $K = 10$ ) guarantees a response that avoids the end-stop. The event is efficiently filtered

with an acceleration level lower than the one in measured in case of low damped suspension.

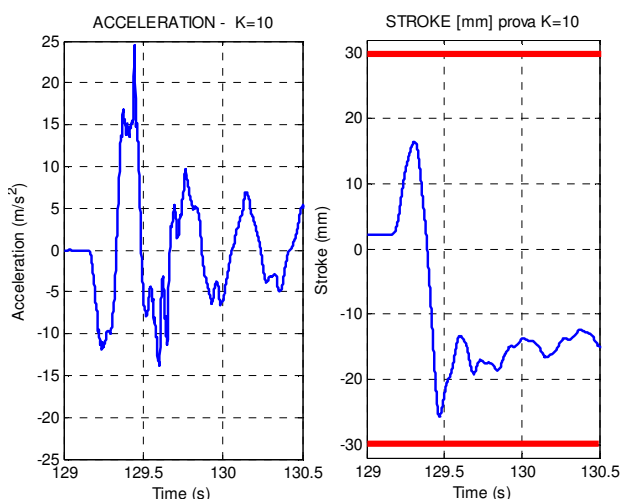
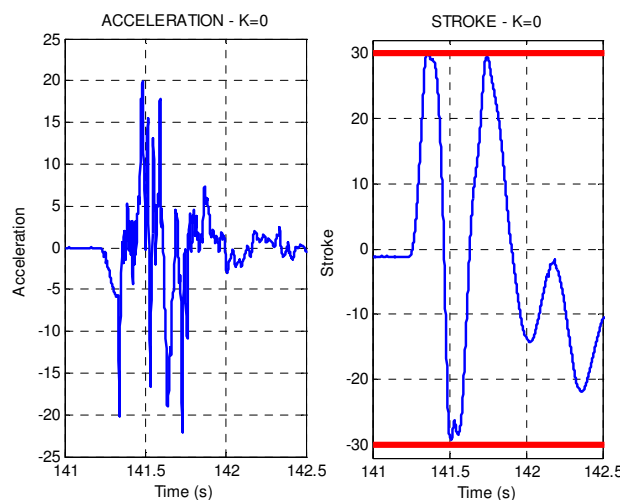


Figure 8. Response to the pothole excitation: acceleration (left) and stroke suspension (right), for low damping level  $K=0$  (top) and high damping level  $K=10$  (bottom)

The analysis puts in evidence the clear trade-off of the suspension system. In the case of a “linear” use (on-road at constant velocity, on-field use) a low damped suspension guarantees an optimal filtering and a corresponding good level of comfort. In the case of a “non-linear” use (encountered potholes and steps, braking manoeuvres, etc.) a medium and high damped suspension outperforms a low damped one. Thus any passive suspension system represents a compromise of this trade-off, which can be overcome by the use of a semi-active damper.

Furthermore, this kind of trade-off is commonly observed in non-linear systems. In fact this is mainly due to the amplitude of the input and not to a specific parameter of the system.

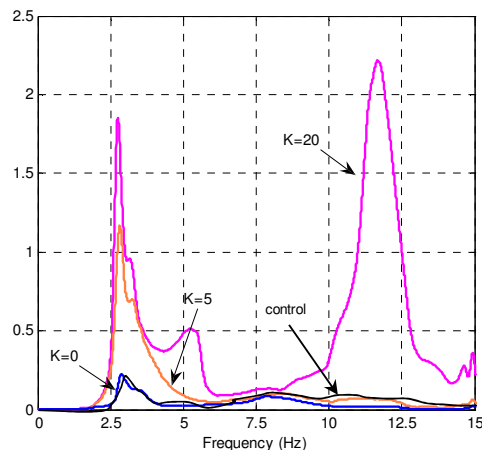


Figure 9. Spectra of the cabin acceleration for different values of  $K$ , compared to the control algorithm performance (rear sweep excitation).

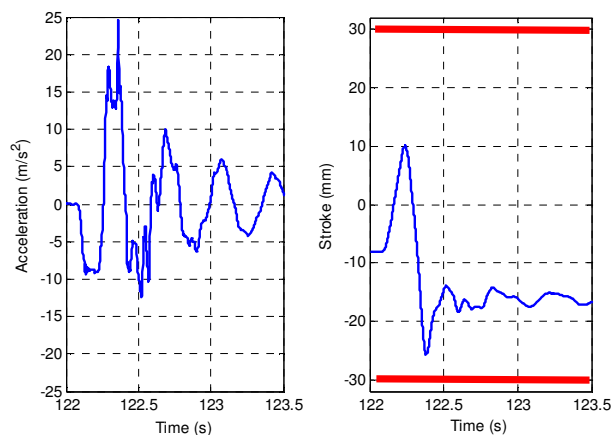


Figure 10. Response to the pothole excitation: acceleration (left) and stroke suspension (right), with semi-active control.

## V. SEMI-ACTIVE CONTROL AND EXPERIMENTAL EVALUATION.

Most of the common semi-active control strategies presented in the literature are based on basic assumptions [2, 5, 13]: the suspension is characterized by an unlimited stroke travel and the suspended mass (e.g. the wheel) is always connected to the ground by the tire (no contact loss assumption). The skyhook based algorithms belong to this family (see e.g. [5, 11, 12, 14]).

When the suspension non linear limits affect the system behaviour (e.g. the suspension travel is limited different kind of strategies are designed, so that the limitation is explicitly taken into account (see e.g. [14]).

The proposed semi-active strategy is represented by the following expression:

$$\begin{cases}
 v(t) = d(\text{stroke}(t)) / dt \\
 \text{thresh}(t) = S_{\max} - \text{sat}_{[0,1]} \left[ \frac{|v(t)|}{V_{\max}} \right] (S_{\max} - S_{\min}) \\
 \text{if } |\text{stroke}(t)| \leq \text{thresh}(t) \quad K(t) = K_{\min} + \text{sat}_{[0,1]} \left[ \frac{|\text{stroke}(t)|}{\text{thresh}(t)} \right] (K_{\max} - K_{\min}) \\
 \text{if } |\text{stroke}(t)| > \text{thresh}(t) \quad K(t) = K_{\max}
 \end{cases}
 \quad (2)$$

Where *stroke* is the suspension stroke measured at every sampling interval, by the sensor S1; *K* represents the actuated damping level;  $K_{\max}$  and  $K_{\min}$  stand for the maximum and minimum level of the available damping; *v* represents the stroke velocity; *thresh* stands for a threshold of the stroke signal;  $V_{\max}$ ,  $S_{\max}$ ,  $S_{\min}$  are parameters of the control strategy that can be tuned.

Practically speaking, at every sampling interval the strategy (2) actuates a damping level according to the stroke signal. If the stroke is within the limit *thresh*, the damping level is set at minimum plus a level depending on the suspension behaviour (e.g. low stroke velocity and far from the end-stops); otherwise the damping level is set at maximum to prevent the end-stops collision.

The control strategy (2) is rooted in the Skyhook algorithm, but the stroke value is here taken into account explicitly. Not that the control (2) has much in common to

The parameters of the strategy have been optimized during tests, according to the comfort index (1).

The results of the rear sweep excitation are reported in Fig. 9, where the spectrum of the cabin accelerations with damping control active is depicted and compared to the passive-like settings.

The results in the frequency-domain are complemented by those in the time-domain, as represented by Fig. 10. Figure 10 reports the time history of the cabin acceleration and of the stroke suspension with the damping control active, during a pothole test.

From the results in Fig. 9-10 some considerations can be drawn:

- From the frequency domain analysis, the semi-active control is able to ensure a performance comparable to the one obtained with a low damped suspension. A little degradation of the filtering is observed at high frequency, beyond 10Hz.
- The time-domain analysis shows that the control strategy is able to prevent the end-stop during a pot-hole test.
- The use of an appropriate damping control is able to overcome the trade-off of a passive suspension. Good filtering for relatively low excitation and prevention of the end-stop collision for highly exciting road inputs. This kind of result cannot be achieved by any passive suspension.

## VI. CONCLUSIONS

In this work the idea of using a magnetorheological semiactive damper in A cabin suspension for agricultural vehicle has been explored. A prototype device has been tested in a vehicle. Particular attention has been paid on the analysis of the suspension trade-off. For this purposes a testing protocol has been proposed and the related results presented. This paper proposes a control strategy to overcome such a trade-off. The effectiveness of the strategy is experimentally evaluated.

## ACKNOWLEDGMENTS

This work has been partially supported by MIUR Project “*New methods for Identification and Adaptive Control for Industrial Systems*” and by Same Deutz Fahr.

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