An investigation of a semi-active suspension for a fork lift truck

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Abstract

This paper investigates the feasibility of a semi-active suspension of the cabin of a fork lift truck as a way of reducing the effect of harmful vibrations on the health of truck drivers. A suspension based on MR fluid dampers has been designed and implemented on an actual vehicle, an heuristic control strategy has been developed, which preserves the filtering properties of the passive suspension when the vehicle moves in straight line with a constant velocity and suppresses the large amplitude pitch and roll motion during turns and braking. Field tests have demonstrated a substantial comfort improvement with respect to the passive suspension, during the braking and turning phases, without any noticeable detrimental effects.

1 Introduction

Fork lift trucks drivers are often exposed to high level of low-frequency vibrations due to the unevenness of the ground [1]. These harmful vibrations, aggravated by difficult working conditions (uncomfortable postures, inappropriate seats, frequent handling operations, etc...) lead frequently to spinal disorders [2]. Fork lift trucks are vehicles that do not have standard suspension systems such as those used in the automobile industry. On numerous models, the possibility of uncoupling the cab from the chassis allows the insertion of flexible mechanical components (springs and dampers) at each of the fixing points (Fig.1).

INRS has shown, through the development of two appropriate prototypes on two commercially available trucks, that a cabin suspension can reduce the vibration transmitted to the driver by 50 % [3]. As the mechanical components employed are relatively cheap, industrialisation of such a system will allow it to be fitted as standard to future models of fork lift trucks. To ensure the successful transfer of the concept of a suspended cab to constructors of industrial trucks, the problems that may result from the changes to the habits of the drivers must also be anticipated. The experience acquired by the development of suspension systems for other types of vehicles (agricultural tractors, trucks, etc.) shows that the angular movements (pitch and roll) produced by the insertion of flexible elements, although of low amplitude and therefore having no negative impact on health, can nevertheless bother the drivers. To reduce this discomfort, the passive suspension has been improved by introducing components whose mechanical characteristics can be

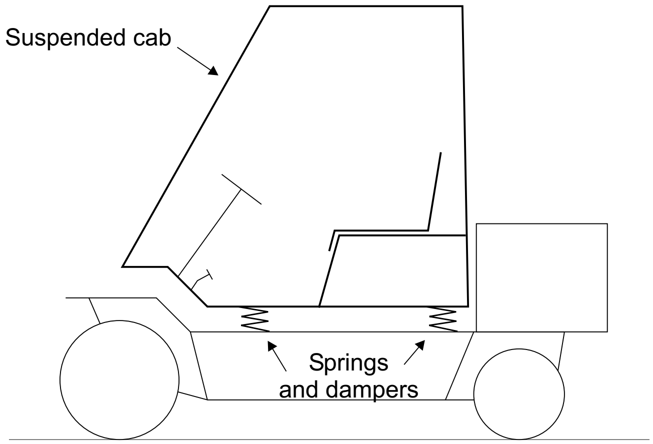

Figure 1: Cabin suspension. The seat is attached to the cabin while the steering wheel and the pedals are fixed to the truck

controlled. The aim of this study was therefore to design and develop a strategy to control these adjustable components (or semi-active actuators) in order to limit excessive angular movements of the cab while retaining the filtering qualities of the passive suspension. The technical feasibility of such a system has been assessed by developing a suspension prototype and testing it in realistic driving configurations.

The choice of the active elements was a key point in determining the suspension control strategy. In this feasibility study, we focussed on magneto-rheological (MR) fluid actuators, whose dissipation characteristics vary in relation to the electric current passing through it. MR dampers have a dynamic behaviour that is relatively well known and that has been described at length in the scientific literature [4, 5, 6]; the data available in the literature were verified and supplemented by systematic component testing prior to this study [7]. In particular, the response time in relation to both the intensity applied and the speed of piston movement of the MR damper was investigated. These tests allowed verification that this type of actuator is well suited to use in semi-active suspension systems in terms of frequency bandwidth.

2 Semi-active suspension

As far as comfort is concerned, the aim of an active suspension is to achieve a transmissibility with low overshoot at the sprung mass resonance and a high attenuation rate at higher frequencies (-40 dB/decade), which is typical of lightly damped isolation systems. This would amount to a passive suspension with frequency dependent damping properties leading to high damping up to the corner frequency, and then very low damping. The celebrated ”sky-hook” damper achieves just that with a feedback proportional to the absolute velocity of the sprung mass [8, 9, 10]. An alternative implementation with enhanced robustness has also been proposed [11]. Note that comfort is not the only concern when designing active suspensions; road holding and dynamic tyre forces may also be important aspects [12].

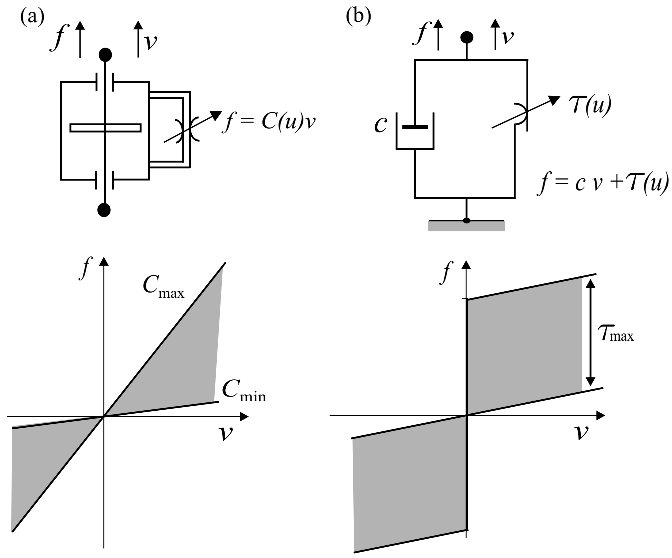

Figure 2: (a) Damper with variable damping coefficient (b) Bingham model of a MR fluid damper

Semi-active devices are essentially passive and cannot generate arbitrary control forces; they are restricted to control forces opposing the relative velocity which are exclusively dissipative ($fv \geq 0$) [13]. With this restriction, the control element allows one to vary the control force arbitrarily within the operating range (Fig.2).

Trying to emulate an active suspension with a low power device is a very tempting concept, because it eliminates the need for a costly power supply. Conceptually, the system works as follows (Fig.3) : Since the semi-active device is capable of generating essentially any force (in the operating range) such that

$$f_c \cdot (\dot{x} - \dot{x}_0) \geq 0 \tag{1}$$

the control strategy consists of setting the control signal u to the appropriate value whenever the inequality (1) is fulfilled, and setting u to achieve $f_c = 0$ whenever the requested force is violating inequality (1) [12, 13, 14].

Taking the damper with variable coefficient as an example (Fig.2.a), the sky-hook damper force, proportional to the body velocity, $-b\dot{x}$, can be best approximated with a damping coefficient

$$\begin{gathered} c = C_{min} + \mathbf{1}[\dot{x}(\dot{x} - \dot{x_0})] \left[\frac{|b\dot{x}|}{|\dot{x} - \dot{x_0}|} - C_{min} \right] \\ C_{min} \leq c \leq C_{max} \end{gathered} \tag{2}$$

where $\mathbf{1}(.)$ is the Heaviside step function taking care of the inequality (1). This strategy works indeed well for harmonic disturbances, because all the signals have the same frequency and it is sufficient to enhance the damping of the semi-active device at low frequency while making it as low as possible at high frequency. On the contrary, when the

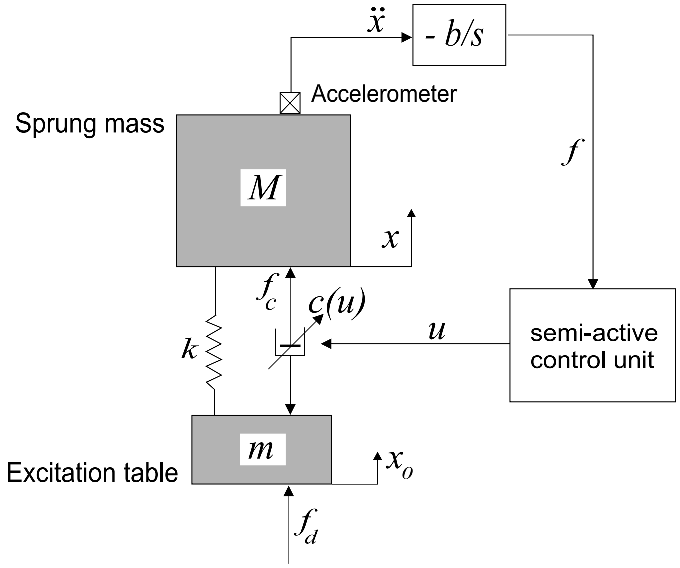

Figure 3: Principle of a sky-hook semi-active suspension

excitation is broad band, the semi-active control strategy faces the difficulty that the relative velocity between the two bodies $(\dot{x} - \dot{x}_0)$ on which the semi-active controller operates contains higher frequency components than $\dot{x}$ which one tries to emulate (a discussion of the frequency content of the two signals can be found in [15]). Figure 4 illustrates this difficulty of trying to emulate a low frequency signal from segments of a signal of higher frequency : when $(\dot{x} - \dot{x}_0)$ and $(\dot{x})$ have the same sign, it is, in principle, possible to follow the requested force (with an appropriate strategy); on the contrary, when they have opposite signs, the best thing the active controller can do is to set the dissipative properties to the minimum, to minimize the amplitude of the control force, which is in fact of opposite sign to that actually wanted. This strategy produces step variations and introduces higher harmonics in the actual control force, which is a parasitic excitation to the system.

This problem has not been widely recognized in the literature; it has been verified with a single axis semi-active suspension at INRS; the experimental set-up is shown in Fig.5; the semi-active device is the Lord RD-1005 damper. The semi-active control is done via a proportional controller comparing the desired and the actual control force [16] (Fig.6). The experimental transmissibility curves are compared in Fig.7; the four curves represent respectively (i) the transmissibility without control, (ii) with the semi-active sky-hook implemented according to Fig.6 for broad band excitation, (iii) with semi-active control (same) for harmonic disturbance and (iv) with constant current applied to the MR damper. One sees that the semi-active control behaves significantly better at high frequency for narrow band disturbance (this situation is representative of, for example, a washing machine [17]) than for broad band disturbance, and that the behaviour in broad band of the semi-active control is not significantly better than that of a MR device excited with a constant current of 0.25 A. This important observation is not related specifically to the sky-hook damper control law that one tries to emulate, but rather to the frequency content of the signals, the control signal that one tries to generate on one hand, and the one on which the semi-active device operates, on the other hand; this has motivated the

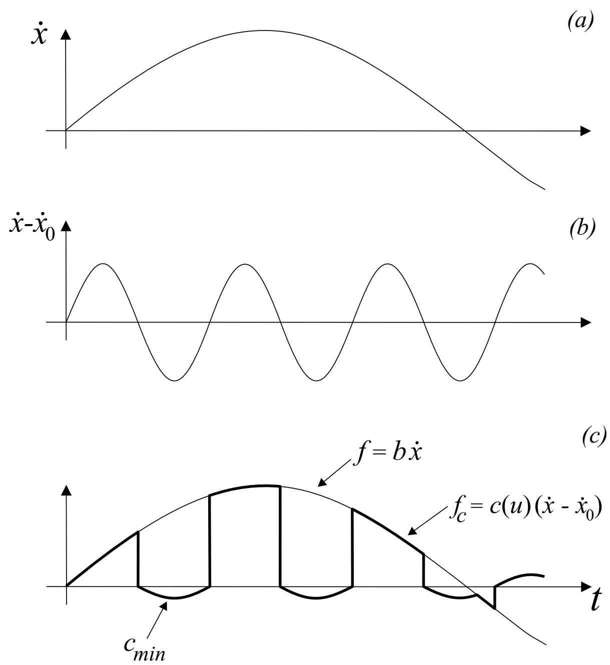

Figure 4: Comparison of the requested force and the actual force when trying to emulate a low frequency signal $b\dot{x}$ with segments of $c(u)(\dot{x} - \dot{x}_0)$

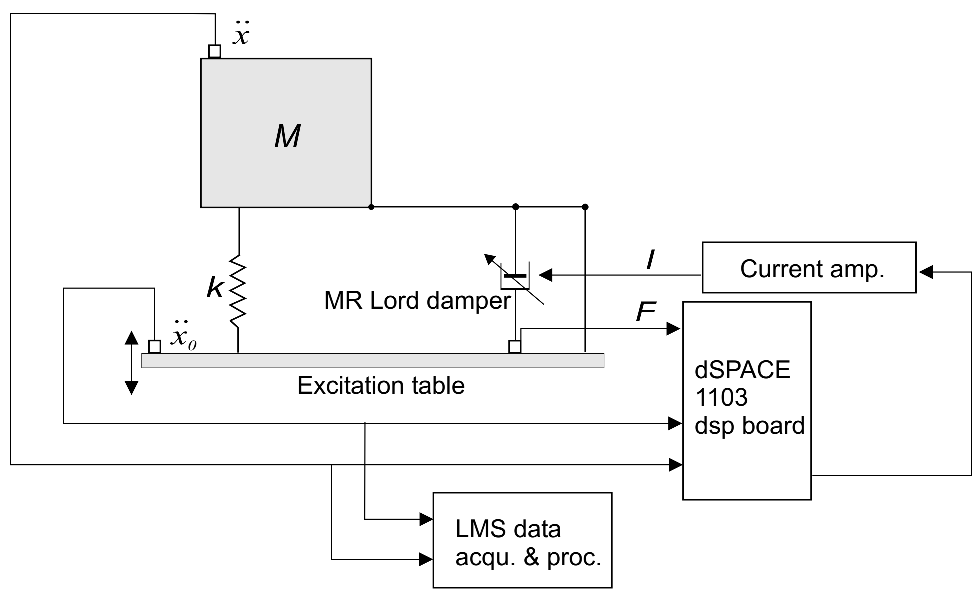

Figure 5: Semi-active sky-hook test setup on the shaking table of INRS

heuristic control strategy that follows.

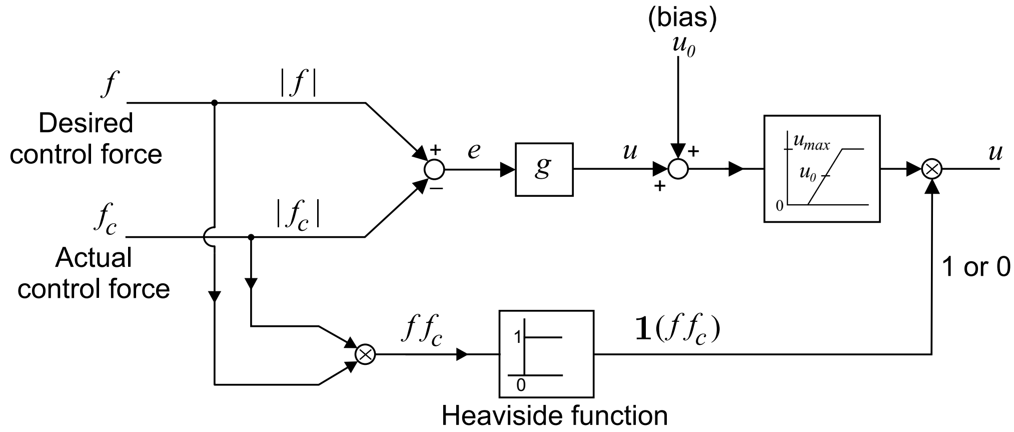

Figure 6: Semi-active proportional controller [16]

3 Control strategy

The choice of the control strategy has been motivated by two things:

1. The passive suspension is perfectly adequate and quite effective when the fork lift truck moves in straight line and at constant speed, but it has been found to generate some discomfort when the driver is subjected to an acceleration, during turns or decelerations (accelerations are not large enough to generate substantial pitch).
2. The ground disturbances involved here being broad band, a semi-active implementation of the sky-hook damper, or anything of this kind, is likely to degrade the high frequency filtering properties of the passive suspension, which is not advisable.

Accordingly, the control strategy has been based on detecting the lateral acceleration and the brake-pedal depression. The control logic is as follows:

- Maximum electrical current supply to the MR dampers when brake depression is detected (thus producing a soft blocking of the cabin).
- When the lateral acceleration exceeds some threshold, an acceleration dependent current is delivered to the MR dampers. After extensive simulations and testing, the current/acceleration law adopted is shown in Fig.8. An alternative control logic based on the detection of the steering wheel angle was also tested but not successfully.

4 Hardware

The fork lift truck used in this study is a KOMATSU FD 20 with a load lifting capability of 2000 kg; the experimental set-up is shown in Fig.9; the passive suspension was

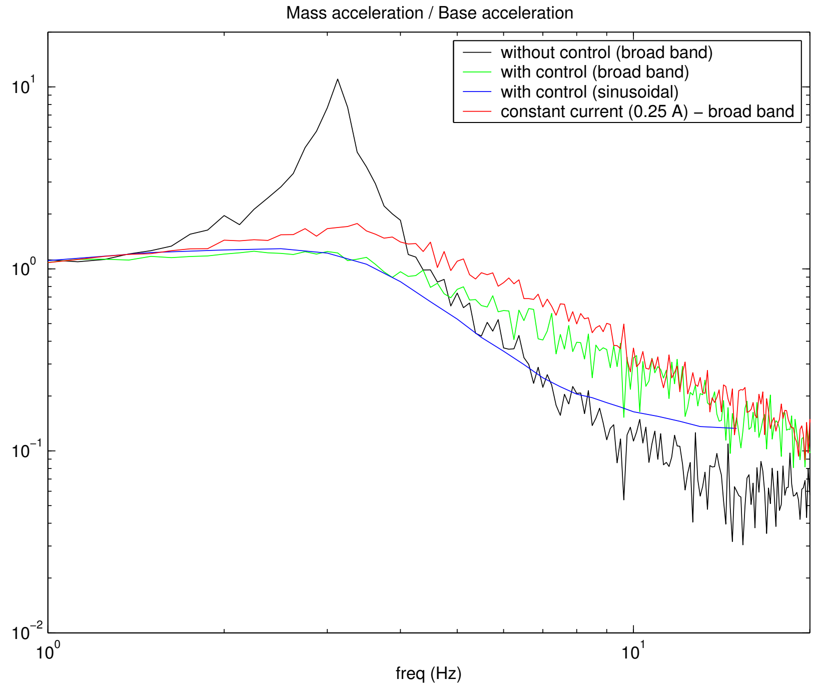

Figure 7: Transmissibility curves : without control broad-band disturbance (black) - semi-active sky-hook broad-band disturbance (green) - semi-active sky-hook harmonic disturbance (blue) - constant current broad-band disturbance (red)

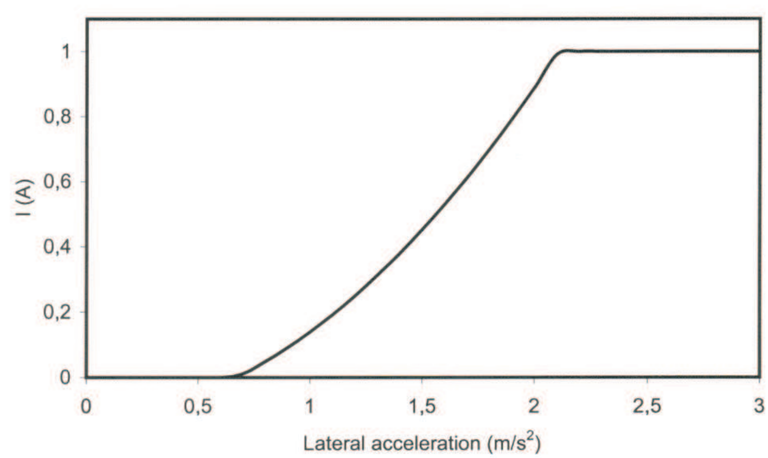

Figure 8: Law linking the lateral acceleration of the truck with the MR damper command

designed (at INRS) by replacing the original elastomer studs at each of the four fixing points of the cabin by suspension spring assemblies occupying the same space; these allow vertical motion with higher flexibility (spring stiffness of 20000 $\frac{N}{m}$) as well as roll and pitch, while preventing motion in the horizontal plane [3]; end stops limit the vertical stroke of the suspension. Four MR dampers (Lord 1005) are mounted with appropriate lever arms to scale the MR dampers to the stroke/force requirements of this application. The scaling system was made necessary by the large passive damping of the actuator which would otherwise deteriorate the passive properties of the suspension (high frequency roll-off); a reduction ratio of two was finally adopted.

A modular control system was developed (by Micromega Dynamics); it consists of three subassemblies (Fig.10) : an industrial microbox PC serving as the support for a DSP board on which the control law is programmed; an interface module containing four power amplifiers (voltage controlled current amplifier) ensuring the connection between the different actuators and sensors on the one hand, and the DSP board on the other; a DC/DC module allowing the above-described system to be used from the 12 V battery of the truck. Note that the mechanical as well as electronic hardware could be simplified drastically for future industrial applications.

The braking action was detected by a contact switch fitted to the brake pedal producing a voltage step of 1.5V; the lateral acceleration during turning was detected thanks to an accelerometer located at the vertical of the steering wheels at the rear of the truck whose sensitive axis is oriented horizontally in the left-right direction.

Additional instrumentation for performance assessment purposes included an optical encoder fitted to a disk in contact with the front right wheel, an accelerometer measuring the vertical acceleration of the driver's seat, and four sensors measuring the relative displacement of the cab with respect to the chassis (LVDT sensors at the rear and $5k\Omega$ potentiometric sensors at the front); all these data were transmitted to a data acquisition system via telemetry. In addition, a video camera was installed in front of the front right mechanical stop of the cabin, and another one took fixed-station views.

5 Experiments

The passive properties of the suspension have been examined in earlier studies [3]; field tests have demonstrated that the rms amplitude of the vertical acceleration at the driver's seat is divided by two.

The following field tests were done:

- *Test with obstacles* : These tests intended to verify that the semi-active system did not degrade the filtering properties of the passive suspension. The fork lift truck, moving in a straight line at different speeds, was firstly required to cross over a 1cm x 15cm obstacle (single obstacle) with both the left and right wheels simultaneously. The tests were repeated at speeds ranging from 6 to 14 km/h with 2 km/h increments. Each test was carried out with and without control and was systematically conducted twice. The tests were then repeated with a series of three 1 x 15 cm obstacles laid out alternately under the left and right wheels (multiple obstacles).

- *Braking test*: The fork lift truck was driven in a straight line at speeds varying from 6 to 14 km/h with 2 km/h increments and the brakes were then applied suddenly. Each test was carried out with and without control and was conducted twice.

- *Turning test* : The fork lift truck was required to drive in a straight line at a speed of 14 km/h then negotiate a turn with a constant radius of curvature. The tests were carried out for left and right turns, with and without control, and were conducted twice.

- *Slalom test* : The fork lift truck, while trying to maintain a globally rectilinear path, was required to steer slightly left and right successively at a speed of about 10 km/h then brake suddenly. Each test was carried out with and without control and was conducted twice.

The main results are as follows :

Stability of the cabin : For the tests involving tilting of the cabin (braking, turning, and slalom), the assessment of the control strategy was based on the extreme values of the cabin displacement with respect to the truck at the four measurement points (including possible contact with the end-stops) as well as the relative velocity. For the turning tests, the travel of the fixing points was globally reduced by 40 % to 60 % by the control, and the relative velocity was reduced even more drastically (Fig.11). The fixing points were systematically prevented from reaching their end-stops. The forces generated by the MR damper at low speeds were not sufficient to block the cab. The objective of alleviating severe tilting of the cabin was therefore achieved. For the slalom tests the amplitudes of the lateral tilting of the cab were reduced by 60 % to 80 %. The cab no longer reached the end-stops. For the braking tests, gradual blocking of the cabin was effective and the end-stops were no longer reached (Fig.12).

Vibration isolation : The assessment of the vibration isolation performance of the semi-active suspension was based on the retention of the properties of the passive suspension for the test with single and alternate obstacles. The assessment was based on the peak and RMS values of the vertical acceleration of the seat. These values were more or less constant, increasing slightly with the speed of the vehicle. For the same speed, they were sometimes higher during the test with control, and sometimes higher during the test without control; these slight variations are attributed to the statistical variability of the test conditions. The control activation threshold with respect to the lateral acceleration of the rear of the truck was determined correctly to prevent action of the MR device when crossing over an obstacle, which would reduce the isolation performance.

6 Conclusion

The aim of this feasibility study was to show that a semi-active control strategy could respond to the comfort criteria not satisfied by a passive suspension of the cabin of a fork lift truck, namely limiting the low-frequency tilting of the cab on turning and braking without degrading the initial vibration filtering performances of the passive suspension.

The semi-active devices chosen in this study are commercially available MR actuators that were mounted in parallel with suspension springs at the interface between the cabin and the chassis of an actual machine. A heuristic control strategy has been developed, based on sensing the braking action through a contact switch fitted to the brake pedal, and the lateral acceleration with an accelerometer.

Field tests including realistic obstacles, turning, braking and slalom have demonstrated a substantial improvement of the cabin stability with respect to tilting, while retaining the filtering properties of the passive suspension.

The system tested here is not intended for industrial applications; however, it appears that both the mechanical and electronic hardware could be simplified drastically to achieve a low-cost, efficient and reliable product.

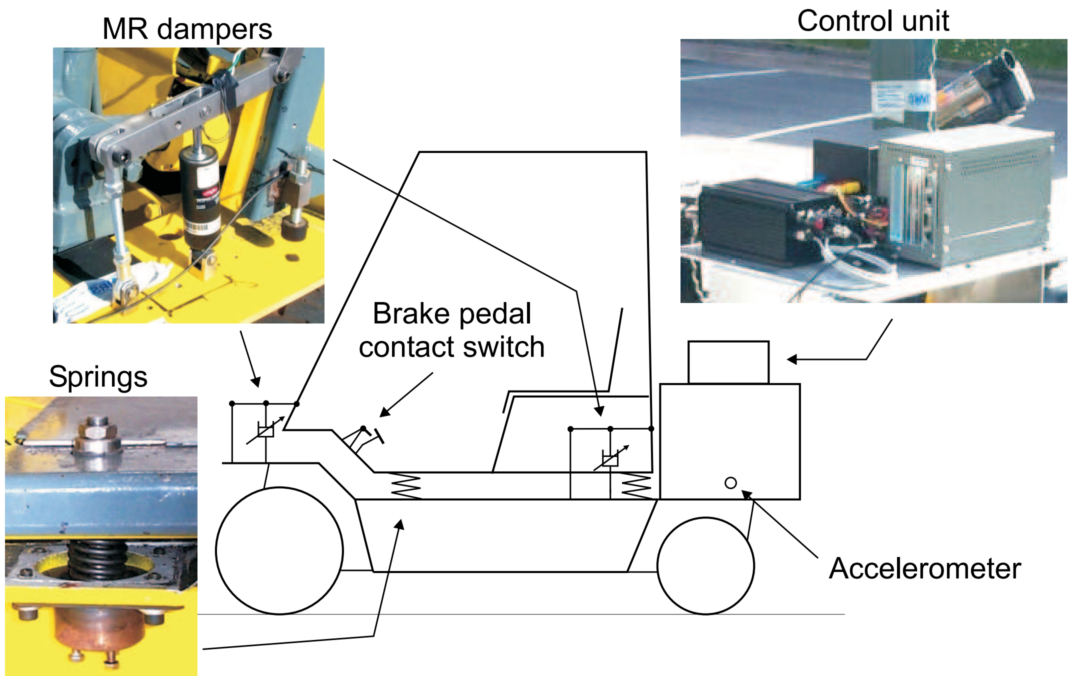

Figure 9: Main components of the semi-active suspension

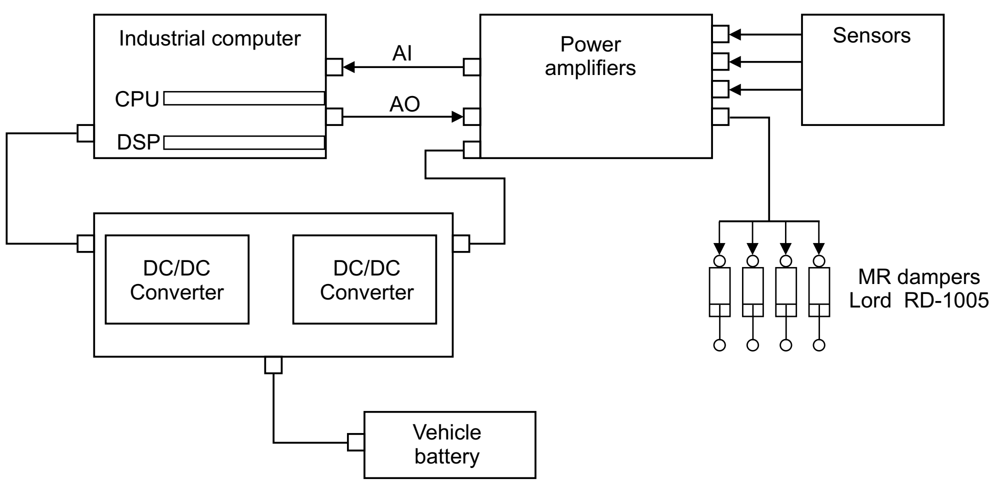

Figure 10: Modular control system

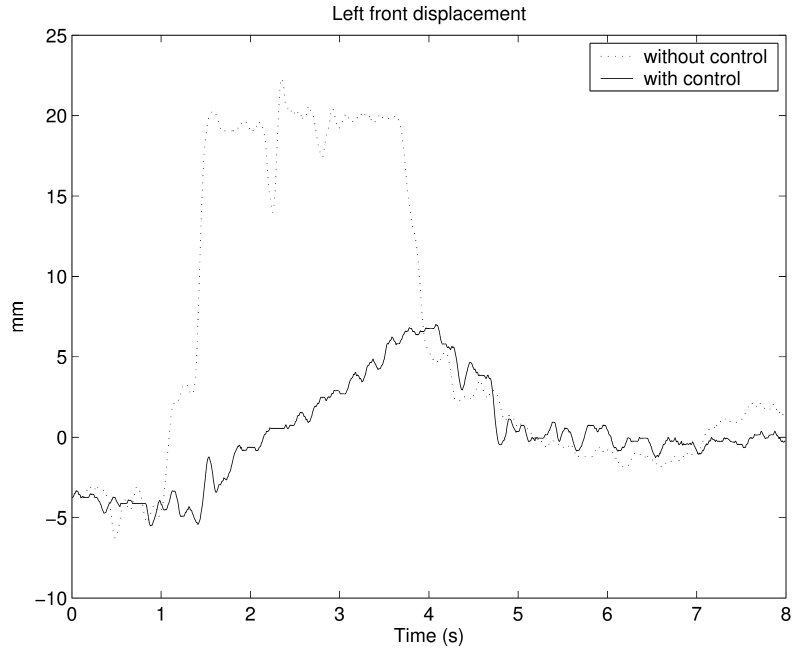

Figure 11: Travel of the front left fixing point when turning

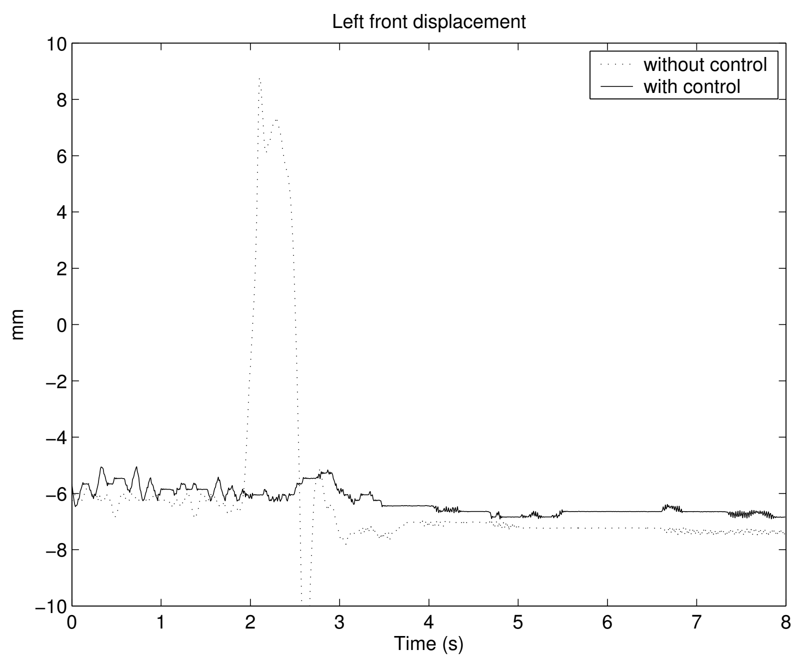

Figure 12: Travel of the front left fixing point during braking

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