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# SEMI-ACTIVE SUSPENSION SYSTEM WITH AN MR ROTARY DAMPER

*SUMMARY*

*The paper is concerned with a laboratory semi-active suspension system (SAS), built to demonstrate and test a number of control algorithms. The heart of the system is the automotive engineering magnetorheological (MR) rotary damper used as an actuator. The MR rotary damper enables to control the SAS damping torque in a continuous way. A designed on-off controller automatically adjusts the damping coefficient to generate the torque required to reduce the amount of energy transmitted from the source of vibrations to the suspended equipment. The experimental results of laboratory investigations of the designed control algorithm in respect to the RD-2087-01 damper have been presented.*

***Keywords**: semiactive suspension, control algorithms, MR damper, fast prototyping*

*PÓŁAKTYWNY SYSTEM ZAWIESZENIA Z ROTACYJNYM TŁUMIKIEM MR*

*W artykule przedstawiono laboratoryjny system półaktywnego zawieszenia pojazdu, przeznaczony do analizy i testowania algorytmów sterujących. Jako element wykonawczy zastosowano obrotowy tłumik magnetoreologiczny (MR). Tłumik ten umożliwia sterowanie w sposób ciągły momentem tłumienia poprzez zmianę wartości prądu w cewce. Zaproponowany w pracy regulator on-off dostosowuje współczynnik tłumienia do wytworzenia momentu tłumiącego, wymaganego do zredukowania energii mechanicznej przekazywanej ze źródła wibracji do chronionej układem zawieszenia masy. Przedstawiono wyniki badań eksperymentalnych na zbudowanym stanowisku laboratoryjnym, przeprowadzonych z wykorzystaniem zaprojektowanego regulatora dla użytego tłumika RD-2087-01.*

***Słowa kluczowe**: system półaktywnego zawieszenia, algorytmy sterujące, tłumik magnetoreologiczny*

## 1. INTRODUCTION

The ability of MR fluids to rapidly change rheological properties upon exposure to an applied magnetic field is used in implementations of MR devices which provide an effective solution for the SAS control in a variety of applications.

The rotary MR damper incorporated in SAS acts as an interface between the electronic sensors (encoders), preprogrammed control algorithms and the mechanical structure of the suspension. The damping effectiveness is adjusted by varying the damper coil current (Sapiński and Bydoń 2004). The restoring torque in the MR damper depends on the rotational velocity of the damper's shaft and the magnetic field strength.

We use in our apparatus the Lord RD-2087-01 rotary damper manufactured by the Lord Corporation.

In the paper two experimental control polices are developed. The controllers operate in real-time. The performance of the vibration isolation system with the developed controllers is tested in a time and frequency domains that covers the SAS operating frequency range.

## 2. SYSTEM DESCRIPTION

The SAS is driven by a flat DC motor coupled to an eccentric small wheel (Fig. 1). The suspended car wheel rolls due to the small wheel rotation and oscillates up and down due to the small wheel eccentricity. A higher the angular DC motor velocity results in a higher frequency of the car wheel vertical oscillations. As far as the amplitude of oscillations is concerned one can notice that the maximal magnitude occurs at the resonance of car body. The goal of SAS is to reduce as much as possible the magnitude of oscillations in the operational range of applied excitation frequencies. We apply suspension control algorithms based on three measured signals. There are: two lever angles of the car wheel suspension (unsprung mass) and the car body suspension (sprung mass) and the small eccentric wheel rotational angle (kinematic excitation).

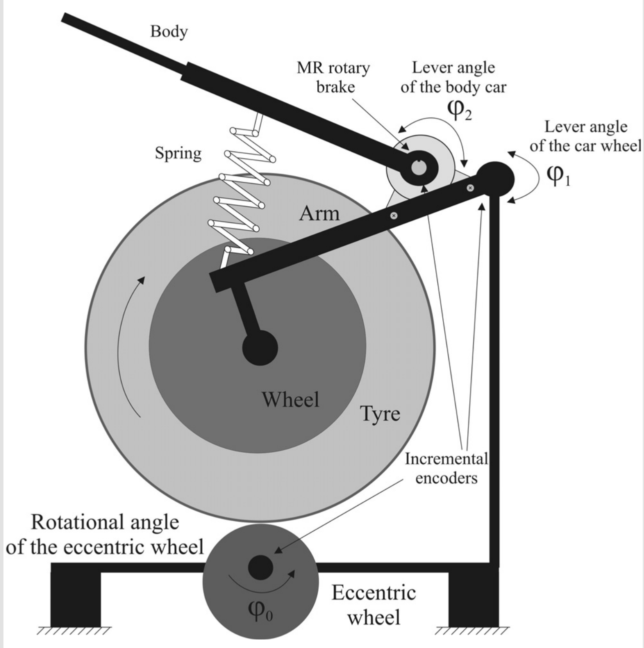


**Fig. 1.** Structure of SAS with the MR rotary damper

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The system consists of mechanical unit: rigid frame, DC high-torque flat motor, eccentric wheel, car wheel and suspension system (magnetorheological rotary damper and spring), position sensors: three incremental encoders, interface and power supply unit, RT-DAC I/O internal PCI or external USB board (PWM control and encoder logics are stored in a XILINX chip) (Gorczyca *et al*. 2009).

A number of control algorithms written in the MATLAB/Simulink are shown. The RTW, RTWT toolboxes are included and they are necessary tools to apply real-time controls in the real SAS system. The very important issue is that certain control ideas based on modeling can be verified by a real-time experiments. Therefore, the constructed laboratory apparatus is an example of a mechatronic system suitable for analysis and synthesis of control algorithms. It creates a fast prototyping environment for academic and also industrial research.

## 3. EXPERIMENTS

Experiments were conducted at the laboratory rig shown in Figure 2. The block diagram of the rig comprises the SAS, power interface, PC working under Windows XP equipped with RT-DAC/PCI board.

The vibration excitation circuit is based on the eccentric wheel directly connected with the controlled DC motor. Two lever angles of the car wheel and the car body and rotational angle of the eccentric wheel are measured with the incremental encoder sensors HEDM-5055 (4096 counts per revolution). These signals are utilized by the measurement and control algorithms running on PC to generate a control signals for the RD-2087-01 rotary damper (Lord Corporation 2006) and DC motor. The control signals from the digital output interface are the PWM duty cycle in the range (0÷1). These signals are amplified by the power interface and supplied to the damper and DC motor coils.

Two control algorithms are applied. The first one is an *on-off* utilizing two measured signals. There are two lever angles of the car wheel suspension (the unsprung mass: $\varphi_1$) and the car body suspension (the sprung mass: $\varphi_2$). The goal of the *on-off* controller is to reduce as much as possible the magnitude of the body oscillations. In the *on-off* control algorithm the simple switching function (the rotational velocities product $\dot{\varphi}_2(\dot{\varphi}_2+\dot{\varphi}_1)$) is utilized to generate the current $I_{MR}$ in the MR damper coil (1). It uses the car body angular velocity $(\dot{\varphi}_2+\dot{\varphi}_1)$ and the rotational velocity of damper's shaft $\dot{\varphi}_2$ as the input signals. The output signal is current in the coil (Lin *et al*. 2002).

$$I_{MR}=\begin{cases} I_{On},\ \dot{\varphi}_2\left(\dot{\varphi}_2+\dot{\varphi}_1\right)\geq 0 \\ I_{Off},\ \dot{\varphi}_2\left(\dot{\varphi}_2+\dot{\varphi}_1\right)<0 \end{cases} \tag{1}$$

The constants $I_{On}$ and $I_{Off}$ relate to the maximal and minimal assumed current values which are compatible with the MR rotary damper current vs. velocity characteristics (Bydoń *et al*. 2005). In fact, we have applied: $I_{Off}$ equal to 0 A and $I_{On}$ equal to 0.5 A or 1.0 A.

The second PID control algorithm is applied to stabilize the rotational velocity of the small eccentric wheel. The PID controller utilizes the rotational velocity signal $\dot{\varphi}_0$ and generates the PWM excitation control signal (current) to the DC motor.

The primary experiment is to observe a free motion of the car body for the constant value of the MR coil current. Value of the coil current is changed in the range (0÷1) A. This and further experiments are performed for a simplified structure of the SAS presented schematically in Figure 3.

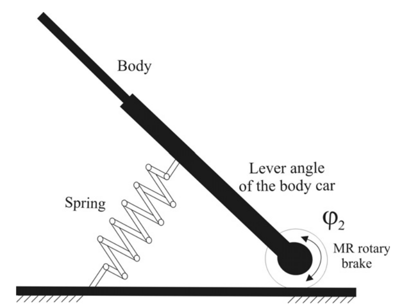


**Fig. 3.** Structure of the body suspension

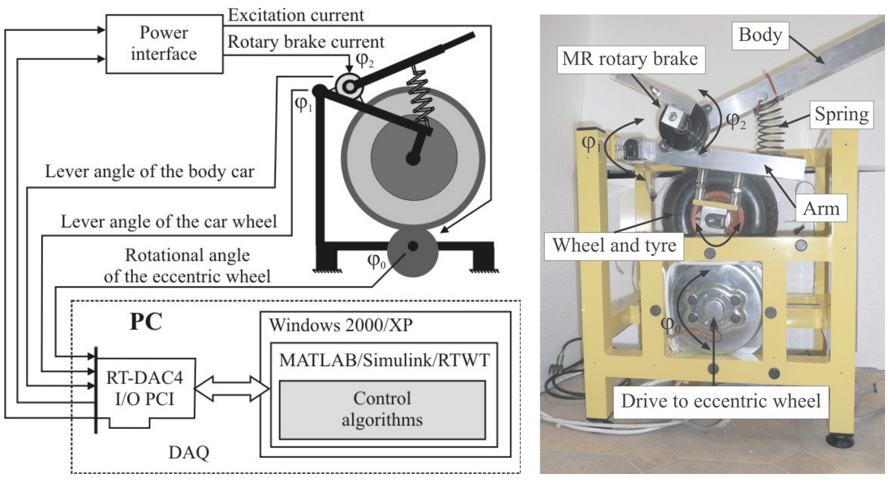


**Fig. 2.** Block diagram and a general view of the rig

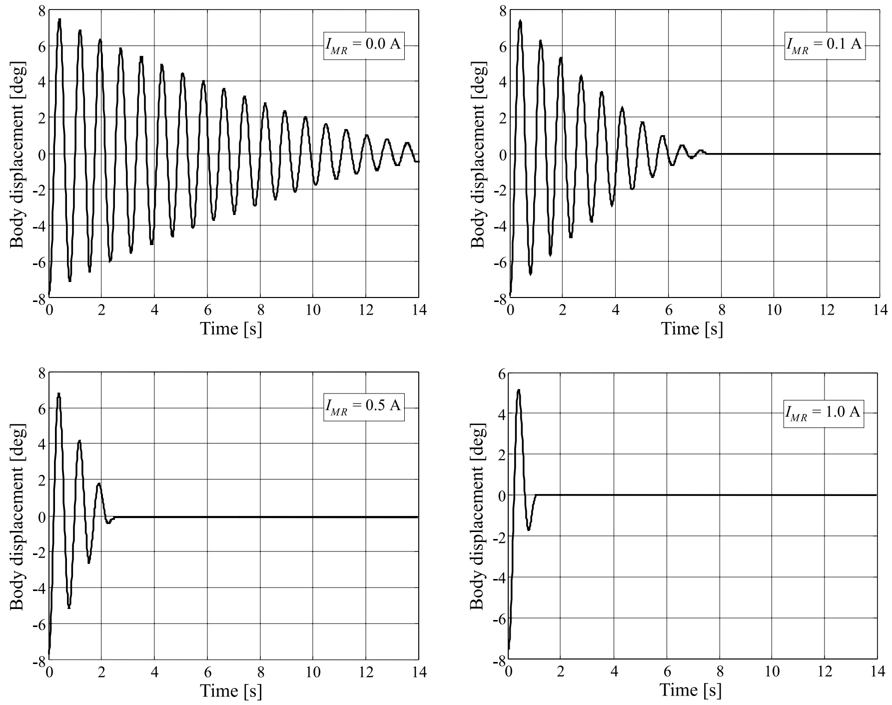


**Fig. 4.** Free motion of the car body

The results are illustrated in Figure 4. The car body is disturb manually to abandon the steady-state position. Then it begins to vibrate. The frequency of oscillations which in fact is the resonant frequency of the system is equal to 1.28 Hz. As can be noted the number of vibration cycles decreases when the current value increases.

The parameters of the MR rotary damper and the spring have been identified assuming that the system dynamics is described by the second order linear equation (2).

$$J_b \frac{d^2\varphi_2}{dt^2} + K_T \frac{d\varphi_2}{dt} + K_S\varphi_2 = 0,\ \varphi_2(0) = \varphi_{20} \qquad (2)$$

where:

- $J_b$ – the body moment of inertia [kgm$^2$],
- $K_T$ – damping coefficient [Nms],
- $K_S$ – spring stiffness [Nm],
- $\varphi_2$ – angle position of the body [deg].

The optimization procedure is based on the *fminsearch* function of the MATLAB Optimization Toolbox. It confirms that the damping coefficient $K_T$ of the MR rotary damper is the function of the magnetic coil current. The relation between the damping coefficient and the current is shown in Figure 5. As can be noted the spring stiffness $K_S$ vary around the mean value 73.54 (Fig. 6) but it can be admit as constant in future consideration (it does not depend on the coil current).

As it was mentioned before the SAS system is excited by the eccentric disk driven by the DC drive. The shape of the eccentric disk is designed in such a way that the excitation is semi-sinusoidal if the rotating velocity of the disk is constant. The force required to maintain the constant velocity changes and the dedicated PID controller is applied to stabilize the velocity. The velocity is calculated internally by the FPGA chip based on the rotational angle signal $\varphi_0$.

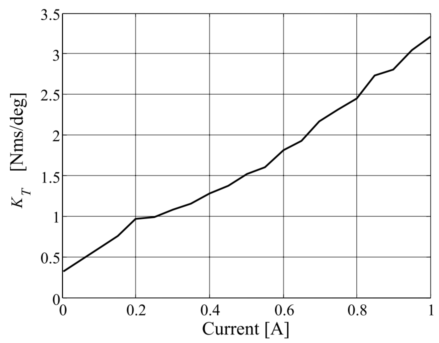


**Fig. 5.** Damping coefficient vs. current

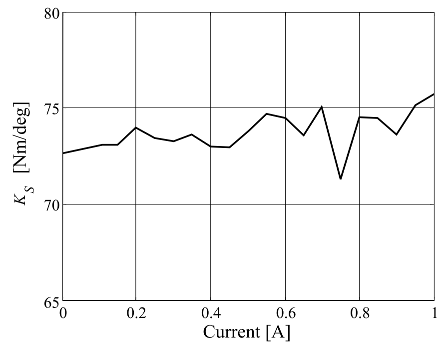


**Fig. 6.** Spring stiffness vs. current

To observe the behaviour of the SAS model in a reasonable frequency range a ramp signal is applied (desired curve in Fig. 7), which is the source of the reference frequency of the velocity of the eccentric driving wheel. The chirp signal in the range (0÷6) Hz is generated to measure the frequency characteristics of the system. The experimental data curve in Figure 7 presents the measured frequency controlled by the PID. One can observe that the maximum frequency stabilisation error is equal to 1.4 Hz (see Fig. 8). The zoomed part of PWM duty cycle control signal is shown in Figure 9. One can notice that the control signal can be only positive.

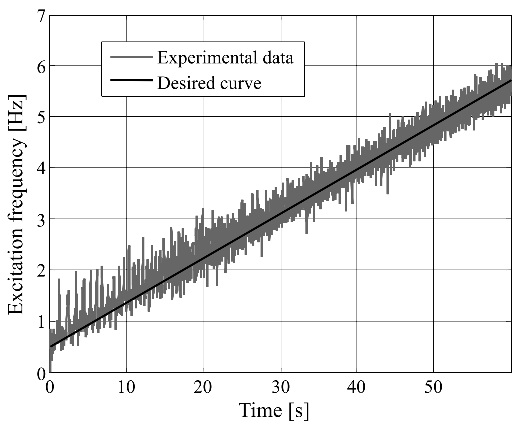


**Fig. 7.** Generation of the chirp excitation signal

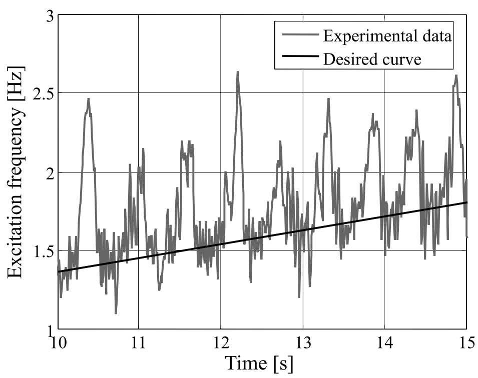


**Fig. 8.** Generation of the chirp excitation signal (zoomed)

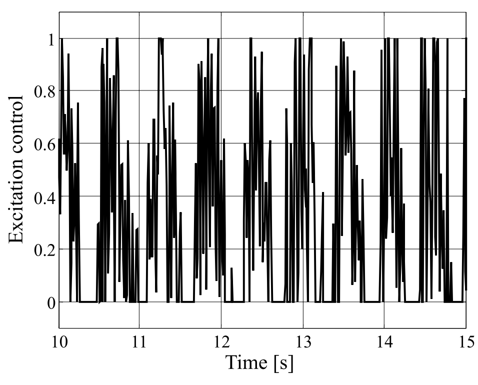


**Fig. 9.** Excitation control signal (zoomed)

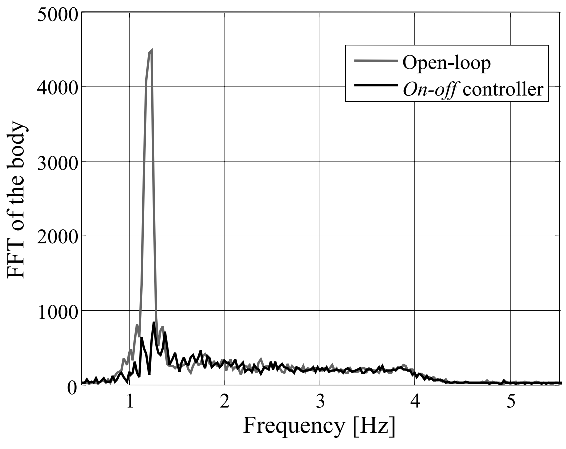


**Fig. 10.** FFT of the body

The results of the open-loop control are presented in frequency domain (as *Fast Fourier Transform* – FFT) in Figures 10 and 11. The open-loop response is compared to the response of the system controlled by *on-off* controller. In fact, in the open-loop the MR rotary damper operates as a passive one (Sapiński 2004). One can notice that the controllers significantly decrease the peak resonant response of the car body and the wheel.

The first peak at 1.28 Hz frequency shown in Figures 10 and 11 corresponds to the na-tural frequency of the controlled system for $I_{On}$ = 0.5 A. The most important is that the natural frequency of the investigated SAS structure does not significantly change with the coil current.

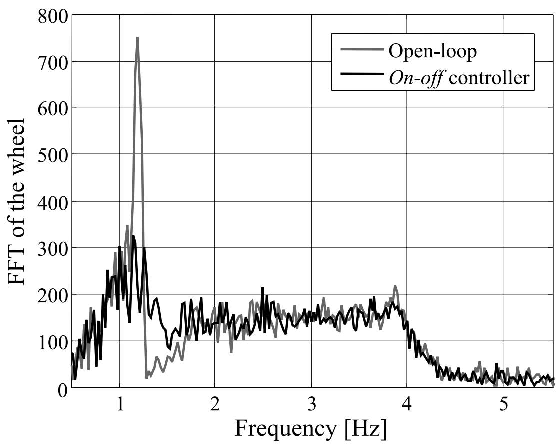


**Fig. 11.** FFT of the wheel

Figure 12 illustrates the behaviour of the SAS model in the time domain. The car body amplitude of the open-loop system is approximately four times smaller than the one of the closed loop system controlled by the *on-off* algorithm. In this way the car safety and comfort are significantly improved due to the reduced oscillations.

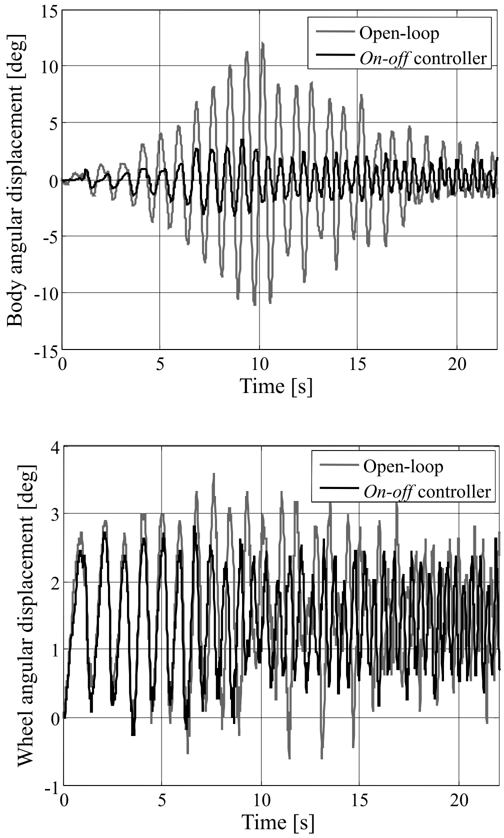


**Fig. 12.** The time responses of the open-loop and closed loop systems

## 4. CONCLUSIONS

The main goal of the SAS apparatus design is to create a laboratory rig to conduct research focused on the vibration isolation while applying RD-2087-01 rotary damper. A designed controller automatically adjusts the varying damping coefficient to generate the torque required to reduce the amount of energy transmitted from the source of vibrations to the suspended equipment. A number of new control algorithms, tested in a wide frequency domain, operating in real-time and examined by a fast prototyping manner can be developed. It is obvious that the semi-active rotary damper provide remarkable improvements over passive suspensions. The magnitude of vibrations, especially in the neighbourhood of the resonance frequency is significantly reduced. The SAS is a portable device operating in the MATLAB/Simulink environment. Therefore control results corresponding to vibration damping effects can be experimentally proved easily with this test rig. A future research will focus on the modelling of the SAS with MR rotary damper structure and developing a more effective control algorithms.

### Acknowledgement

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