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FUZZY CONTROL FOR MR DAMPER IN A DRIVER'S SEAT SUSPENSION

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The paper concerns the application of fuzzy logic to the control of a magnetorheological fluid damper (MR damper) employed in a driver's seat support. The seat is modeled as a 1 DOF system to which a generalized model of an MR damper valid for fluctuating magnetic fields is attached. The performance of the feedback system with a fuzzy controller is tested in computer simulations and compared with the performance of an open loop system and the feedback system with an *on-off* controller. The obtained results are verified in laboratory experiments. For this purpose, the *on-off* and fuzzy real-time controllers for the MR damper are developed in integrated design and control environment of MATLAB/Simulink. The advantages of fuzzy control are proved in experimental investigations.

Key words: MR damper, driver's seat, vibration, fuzzy, control, algorithm

1. Introduction

MR devices provide modern and elegant solutions for semi-active control in a variety of applications, offering several advantages: simplicity of a structure, small number of mobile components, noise-free fast operation and low power demands. They act as interfaces between electronic control systems and mechanical systems. In recent years, several MR devices such as dampers and brakes have been commercialized and developed to the mass production stage. At present, MR dampers are successfully applied in machine engineering and construction industry.

In the automotive field, new commercially available devices, in which unique features of MR fluids are used, include: Delphi Automotive's MagneRide$^{\mathrm{TM}}$ shock absorbers, Carrera's MagneShock$^{\mathrm{TM}}$ automotive racing shocks and the Motion Master Ride Management System. MR dampers employed in those devices can be powered directly from common, low-voltage sources such as batteries, 12-volt automotive supplies or inexpensive AC/DC converters. Moreover, standard electrical connectors and wires can be reliably used, even in mechanically aggressive and dirty environment, without fear of dielectric break down.

Delphi shock absorbers and Carrera's automotive racing shocks provide the possibility for real-time control and optimization of suspension damping characteristics that improve ride comfort and maneuverability of a vehicle (all four corners of the vehicle are independently controlled adjusting the damping force in response to vehicle speed and road conditions).

The Motion Master system offers safety and comfort for drivers via automatic adaptation to both driver's body weight and continually changing levels of shock and road vibration. In this way, driver's responsiveness and control while reducing fatigue and risk of injury can be improved.

The driver's seat, whose suspension is equipped with a MR damper to be controlled, requires that displacement, velocity or acceleration signals be measured and processed in accordance with a specified control algorithm to produce an input signal for the MR damper. The main task of the MR damper is to quickly adjust the damping force to current operating conditions in accordance with the predetermined objective function. The damping force is controlled with the use of a controller programmed in accordance with an algorithm which adjusts the current driver connected to the MR damper coil. That endows us with means to minimize the gain of amplitude vibrations transmitted onto the driver's body at excitation frequencies near the resonance frequency, while the efficiency of vibro-isolation at higher frequencies should not deteriorate.

In many algorithms, velocity signals from suspension components are made use of. Generally, these algorithms fall in two categories: *on-off* and continuous algorithms (Ahmadian, 1999). The already patented *on-off* algorithm (U.S. Patent 5,712, 783, 1998) involves switching of the suspension system from the minimal (*on*) to maximal (*off*) positions which correspond to the minimal and maximal damping states for the MR damper. The continuous algorithm allows us to enhance the number of switching levels of damping for the suspension system such as a continuously variable damping coefficient may be achieved.

This paper is focused on fuzzy control for an MR seat damper which belongs to the category of continuous control algorithms. It is well known that no matter which system is considered, there are always three basic steps characteristic for all fuzzy controllers, i.e. fuzzyfication of controller inputs, execution of controller's rules and defuzzyfication of outputs to a crisp value to be implemented by the controller (Piegat, 1999). At any fuzzy controller output we might get any value from the interval bounded by the minimal and maximal damping states for the MR damper.

The paper reports a part of a research work (Sapiński, 2003) proceeding the development and performance testing of an autonomous control system for an MR seat damper of the RD-1005 type, engineered by Lord Corporation. This system was based on a microcontroller with a fuzzy set processing unit.

2. MR damper

The concept of MR damper operation is based on the MR effect which involves quick changes in viscosity of an MR fluid under the action of magnetic field (Rabinow, 1948). In this study, we employ an MR damper operating in the flow mode, which means that the produced damping force is controlled by the flow resistance of the MR fluid portion contained in the gap (Jolly *et al.*, 1999). A schematic diagram of the MR damper is depicted in Fig. 1.

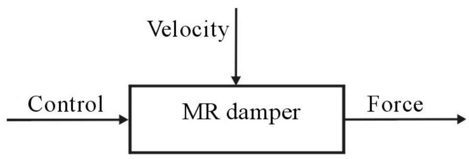

Fig. 1. Schematic diagram of an MR damper

Fig. 2. RD-1005 damper – general view

In this study, we used the RD-1005 damper that is recommended for driver's seats in trucks, buses and agriculture tractors. A general view of a model of the RD-1005 damper is shown in Fig. 2. Basic technical data of the RD-1005 damper are as follows (Lord Corporation, 2003): it has ± 25 mm stroke, 208 mm extended length and 155 mm compressed length. The main cylinder houses the piston head, magnetic circuit, accumulator and MR fluid. The input voltage is 12 V DC. The input current can be varied from 0 A to 2 A. The coil resistance is $5\,\Omega$ (at ambient temperature) and $7\,\Omega$ (at $71°$C). The damping forces (peak to peak) is 2224 N (velocity $50 \cdot 10^{-3}$ m/s, current 1 A) and 667 N (velocity $200 \cdot 10^{-3}$ m/s, current 0 A). The maximum extension force should be kept below 4448 N, the maximum operating temperature of the outer cylinder should not exceed $71°$C, and the maximum continuous current should not exceed 1 A. The durability of the RD-1005 service life damper is about 2 million cycles $(\pm 13 \cdot 10^{-3}$ m, 2 Hz with input current from 0 A to 0.8 A). Response time (dependent on the amplifier and power supply) is less than 25 ms (time to reach 90% of maximum level during 0 A to 1 A step input at velocity $51 \cdot 10^{-3}$ m/s).

3. *On-off* and fuzzy control algorithms for the MR damper

It was mentioned before that *on-off* algorithms and continuous algorithms are particularly recommended to the MR damper control. Principles of these algorithms are explained in Fig. 3.

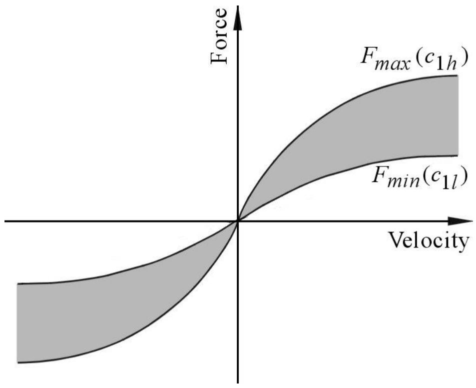

Fig. 3. Damping states in an MR damper

It appears that the benefit of the *on-off* algorithm lies in its simplicity. This is because the switching function is utilized where the damping force may assume the minimum F_{min} or maximum values F_{max} which correspond to damping states of minimum damping ratio c_{1l} or maximum damping coefficient c_{1h} (black bounded lines).

It is obvious that the benefit of the fuzzy algorithm is that the number of switching levels of damping is increased such as the controllable characteristic might become as effective as that of the continuously variable damping coefficient. This means that the damping force may assume any value from the interval bounded by F_{min} and F_{max} (c_{min} and c_{max}) (grey area). The above shows a potential application of fuzzy logic that can be used in an MR damper. Fuzzy logic works by executing rules that correlate controller inputs with desired outputs that may exist anywhere between the minimum and maximum damping states. These rules, in this case relating to the damping levels in the system, can be formulated intuitively or on the basis of an expert's knowledge about the system to be controlled. The switching criterion might be based on information from velocity sensors, however in practical applications it is usually based on information from either displacement or preferably acceleration sensors. This is so these quantities have to be controlled in considerations of comfort and loading gauge requirements.

4. Driver's seat suspension with a controllable damper – case study

Let us consider a simple passive suspension system equipped with a controllable damper represented by a 1 DOF model and compare the behaviour of the system complete with a fixed viscous damper upon applying a fuzzy algorithm and an *on-off* algorithm.

4.1. Fixed viscous damping

A schematic diagram of a simple passive suspension of a driver's seat with a fixed viscous damper is shown in Fig. 4. The following designations are used in Fig. 4: m_1 – seat mass, k_1 – spring stiffness, c_1 – damping factor of the damper to be controlled, $x_0(t)$ – displacement-input excitation (base displacement), $x_1(t)$ – seat frame displacement.

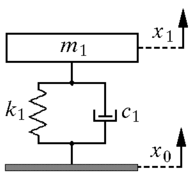

Fig. 4. Schematic diagram of a passive suspension

Let us assume the initial conditions $x_1(0) = 0$, $\dot{x}_1(0) = 0$ and the base harmonic motion $x_0(t) = X_0 \sin(2\pi f_0 t)$. The system response can be expressed as $x_1(t) = X_1 \sin(2\pi f_0 t - \varphi_1)$. It is apparent that the output displacement amplitude X_1 depends on X_0, f_0 and c_1. If $m_1 = 100$ kg and $k_1 = 36861$ Nm, the undamped natural frequency of the suspension is $f_{0s} = 3.06$ Hz and cross-over frequency is $f_{0c} = \sqrt{2}\, f_{0s} = 4.31$ Hz (f_{0c} is the frequency value such that the transmissibility $X_1/X_0 = 1$). Note that the fixed "compromise" damping coefficient $c_{1m} = 1000$ Ns/m will provide rapid damping of free vibrations.

Displacement transmissibility characteristics obtained for a suspension with the fixed viscous damper $c_{1m} = 1000$ Ns/m and with controllable (switched damper) for two values of the damping coefficient ($c_1 = 4c_{1m} = 4000$ Ns/m and $c_1 = c_{1m}/4 = 250$ Ns/m), showing the minimum attained displacement transmissibility for a fixed spring ($k_1 = 36861$ Ns/m) are provided in Fig. 5. It is seen that the transmitted displacement amplitude is significantly reduced as long as:

– the damping coefficient is increased to $c_1 = 4c_{1m}$ at a low frequency,
– the damping coefficient is decreased to $c_1 = c_{1m}/4$ at high frequency.

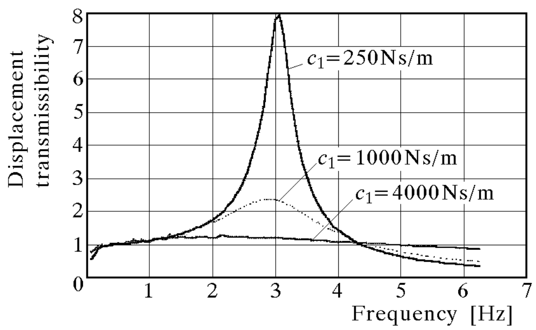

Fig. 5. Displacement transmissibility for the suspension with various damping levels

It implies also a certain trade-off between the resonance control and high-frequency isolation associated with the passive suspension system; as the damping increases, the resonance peak is attenuated but the isolation is lost at higher frequencies. However, as the value of the damping coefficient is increased, the location of the peak shifts towards smaller frequencies.

4.2. *On-off* controllable damping

The *on-off* control algorithm, applied to the MR damper involves switching of the magnetic control field between two constant levels: the upper boundary – maximum field excitation or the lower boundary – no field (or "fail-safe" condition). This produces a high or low level of damping ratio depending on whether the acting disturbances are below or above the cross-over frequency of the suspension transmissibility characteristic.

Let us consider the following input – displacement $x_0(t)$ applied to the base (Fig. 6):

– step of $10 \cdot 10^{-3}$ m, amplitude $\pm 10 \cdot 10^{-3}$ m at frequency 2.8 Hz, time 0-5 s,

– step of $-10 \cdot 10^{-3}$ m, an amplitude $\pm 10 \cdot 10^{-3}$ m at frequency 3.06 Hz, time 10-15 s,

– step of $10 \cdot 10^{-3}$ m, an amplitude $\pm 10 \cdot 10^{-3}$ m at frequency 4.3 Hz, time 20-25 s,

– 0 m in between.

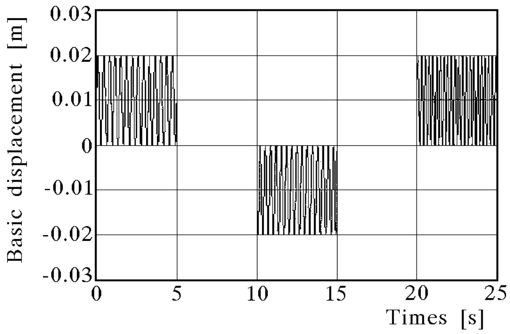

Fig. 6. Displacement – input excitation

Now, let us assume the switching criterion applied to the damper that with relation to the sprung mass acceleration $\ddot{x}_1$:

– if $|\ddot{x}_1| \geqslant 3\,\mathrm{m/s^2}$, then $c_1 = 4000\,\mathrm{Ns/m}$,

– if $|\ddot{x}_1| < 3\,\mathrm{m/s^2}$, then $c_1 = 250\,\mathrm{Ns/m}$.

Time patterns of displacement and acceleration responses obtained for the suspension with fixed viscous damping ($c_1 = 1000\,\mathrm{Ns/m}$) and with *on-off* damping (switched between $c_1 = 250\,\mathrm{Ns/m}$ and $c_1 = 4000\,\mathrm{Ns/m}$) are provided in Fig. 7 and Fig. 8. When analyzing time patterns of displacements in Fig. 7, the initial length of the spring with the stiffness k_1 was assumed to be 0.4 m.

The obtained results lead us to the following statements:

– acceleration response for individual components is reduced,

– acceleration response to simultaneously applied frequency components is reduced, too,

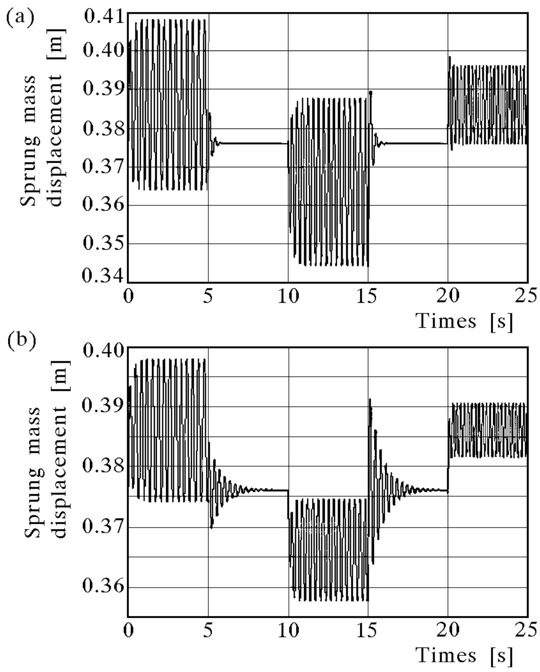

Fig. 7. Displacement responses for a suspension with: (a) fixed viscous damping, (b) *on-off* controllable damping

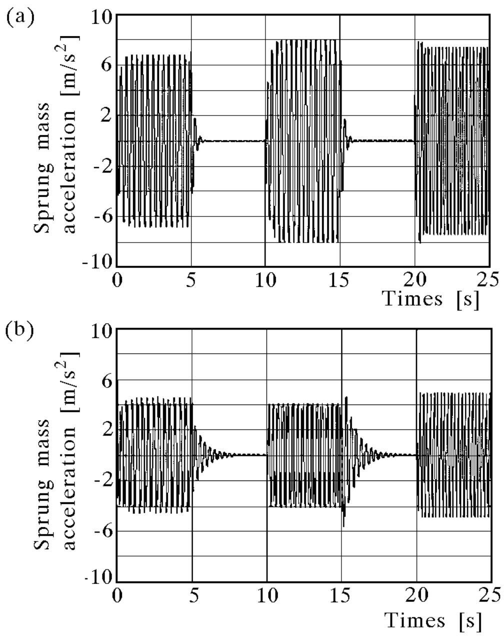

Fig. 8. Acceleration responses for a suspension with: (a) fixed viscous damping, (b) *on-off* controllable damping

– displacement response to individual frequency components is reduced,
– displacement response to simultaneously applied frequency components might increase – however, the higher frequency component is effectively eliminated, so that this represents a kind of trade-off in eliminating unwanted transmission of large high frequency accelerations.

4.3. Fuzzy control damping

Let us assume the switching criterion applied to the damper with respect to the sprung mass acceleration $\ddot{x}_1$:

– if $|\ddot{x}_1| \geqslant 2\ \mathrm{m/s^2}$, then $c_1 = 4000\,\mathrm{Ns/m}$,
– if $2\,\mathrm{m/s^2} > |\ddot{x}_1| \geqslant 1.5\,\mathrm{m/s^2}$, then $c_1 = 2000\,\mathrm{Ns/m}$,
– if $1.5\,\mathrm{m/s^2} > |\ddot{x}_1| \geqslant 1\,\mathrm{m/s^2}$, then $c_1 = 1000\,\mathrm{Ns/m}$,
– if $1\,\mathrm{m/s^2} > |\ddot{x}_1| \geqslant 0.5\,\mathrm{m/s^2}$, then $c_1 = 500\,\mathrm{Ns/m}$,
– if $|\ddot{x}_1| < 0.5\,\mathrm{m/s^2}$, then $c_1 = 250\,\mathrm{Ns/m}$.

Time patterns of displacement and acceleration responses for the predetermined damping levels are provided in Fig. 9 and Fig. 10.

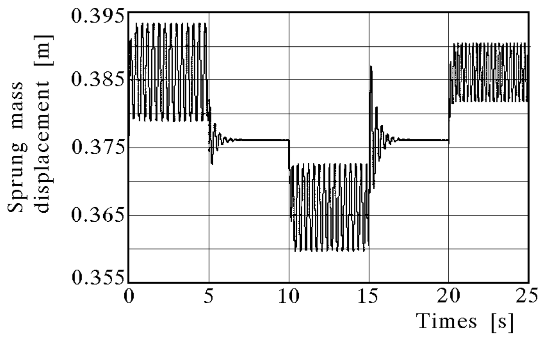

Fig. 9. Displacement response for a suspension with 5 levels of controllable damping

Apparently, in the case of fuzzy control damping more significant reduction of acceleration and displacement responses for individual components and simultaneously applied frequency components can be achieved than it is in the case of *on-off* control. The reduction in the acceleration response to the higher frequency displacement input is appreciable, while no significant transients are observed at switching of the damper rate. The increase of the acceleration response at each switching point occurs before switching and is associated

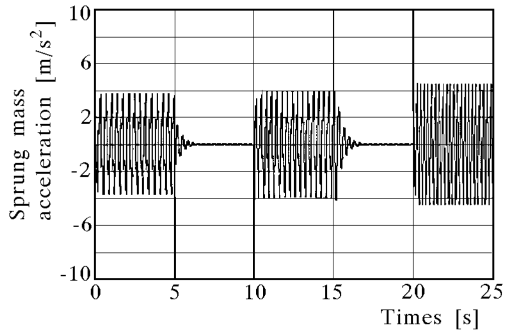

Fig. 10. Acceleration response for a suspension with 5 levels of controllable damping

with the damping level during the delay period. Nevertheless, the acceleration response for the switched damper remains within the overall limits of the responses for both the fixed damper and *on-off* control.

5. Simulation of a driver's seat suspension with the MR damper

In computer simulations we used the generalized model of an MR damper valid for fluctuating magnetic fields (Sapiński and Piłat, 2003). It captures real behaviour of the MRD over a wide range of operating conditions and proves adequate in control applications. This model was developed as a modification of the Spencer model (Spencer *et al.*, 1996).

5.1. Open loop system

We investigate responses of open loop systems (suspension with a fixed viscous damper and suspension with an MR damper for constant levels of the applied current) under sine displacement-input excitations. Simulation results are shown in Fig. 11.

On that basis, the variability range of the applied current for the MR damper can be found, which would guarantee such a level of seat damping as would be achieved when viscous dampers were employed.

5.2. *On-off* and fuzzy controllers

Let us consider a feedback system in which the MR damper in the suspension is adjusted by *on-off* and fuzzy controllers. The structure of an *on-off*

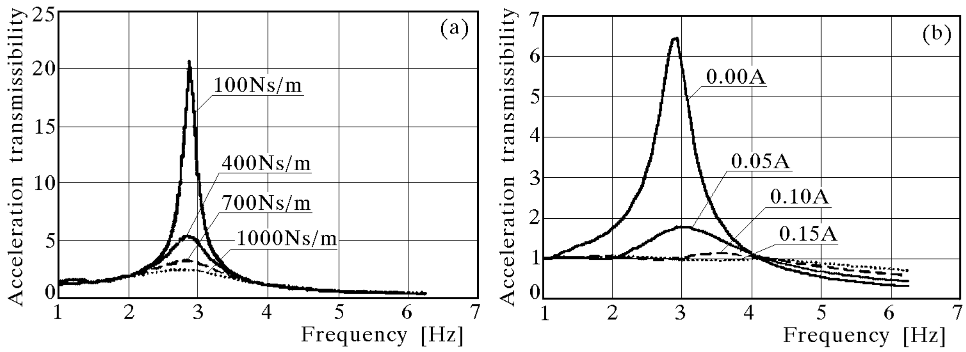

Fig. 11. Acceleration transmissibility for the suspension with fixed viscous dampers (a) and the MR damper (b)

controller is shown in Fig. 12, where the input signals are: $\dot{x}_0$ – base velocity, $(\dot{x}_1 - \dot{x}_0)$ – relative velocity between the seat frame and the base. The output signal (control signal) is: I – current in the MR damper coil.

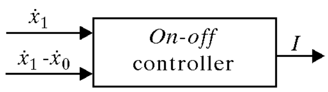

Fig. 12. Structure of an *on-off* controller

The *on-off* controller is governed by the formula

$$I = \begin{cases} c & \text{for} \quad \dot{x}_1(\dot{x}_1 - \dot{x}_0) \geqslant 0 \\ 0 & \text{for} \quad \dot{x}_1(\dot{x}_1 - \dot{x}_0) < 0 \end{cases} \tag{5.1}$$

A block diagram of fuzzy controller processing with three distinct stages (fuzzyfication, inference, defuzzyfication) is provided in Fig. 13. The considered fuzzy controller uses Mamdami's inference system.

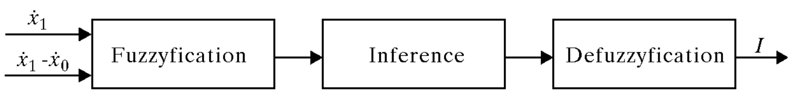

Fig. 13. Block diagram of fuzzy controller processing

The *on-off* and fuzzy controllers were designed on the basis of experiments on an open loop system ($m_1 = 100\,\text{kg}$, $k_1 = 36861\,\text{Nm}$) over the frequency range 1-12 Hz. The bounded values of output for the *on-off* controller were: 0.0 A and 0.15 A. The fuzzy controller has 5 rules (Table 1). The input membership functions for seat frame velocity and relative velocity are triangular-shaped (Fig. 14), while the output membership functions are of the singleton type (Fig. 15). The controller uses three linguistic variables for each input and three linguistic variables for the output: *Minimum*, *Medium*, *Maximum*. The linguistic variables for the output correspond to the values of the applied current: *Minimum* – 0.000 A, *Medium* – 0.075 A, *Maximum* – 0.150 A.

Table 1. Base of rules for the fuzzy controller

No.	Rule
1	IF $(\dot{x}_1 - \dot{x}_0)$ *is Medium*)) AND ($\dot{x}_1$ *is Medium*) THEN (*current is Minimum*)
2	IF $(\dot{x}_1 - \dot{x}_0)$ *is Minimum*)) AND ($\dot{x}_1$ *is Minimum*) THEN (*current is Maximum*)
3	IF $(\dot{x}_1 - \dot{x}_0)$ *is Maximum*)) AND ($\dot{x}_1$ *is Maximum*) THEN (*current is Maximum*)
4	IF $(\dot{x}_1 - \dot{x}_0)$ *is Minimum*)) AND ($\dot{x}_1$ *is Maximum*) THEN (*current is Minimum*)
5	IF $(\dot{x}_1 - \dot{x}_0)$ *is Maximum*)) AND ($\dot{x}_1$ *is Minimum*) THEN (*current is Minimum*)

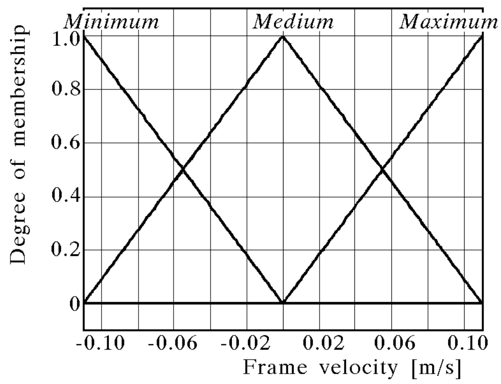

Fig. 14. Triangular-shaped input membership function

The input-output graph for the fuzzy controller with 5 rules is shown in Fig. 16a. As the number of rules goes from 5 to 9, the input-output graph is slightly changed (see Fig. 16b).

Whilst comparing Fig. 16a and Fig. 16b we see that when the relative and seat frame velocity have opposite signs, the current in the MR damper coil controlled by the fuzzy controller with 9 rules will be zero, which will not happen in the controller with 5 rules.

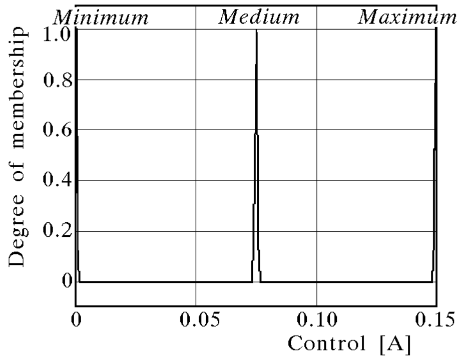

Fig. 15. Output membership function for the control signal

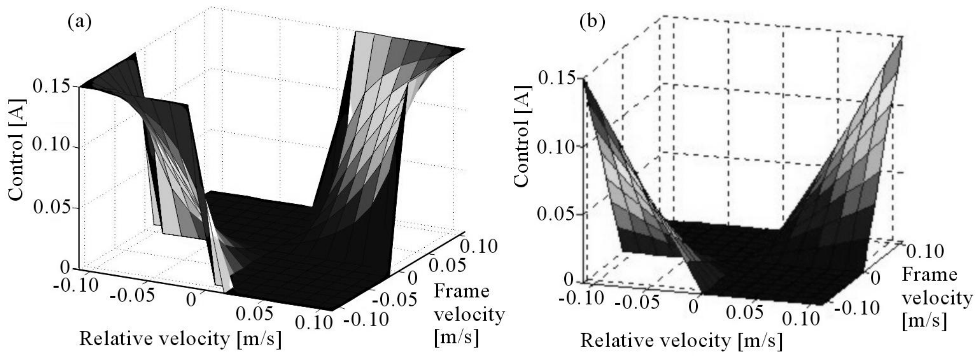

Fig. 16. Input-output graph for the fuzzy controller: (a) 5 rules, (b) 9 rules

5.3. Feedback system

The results obtained in computer simulations for the seat under sine displacement-input excitations are presented in the frequency domain, and in the time domain in Fig. 17 - Fig. 19. In Fig. 17, acceleration transmissibility characteristics for the open loop system and feedback systems with *on-off* and fuzzy controllers (5 rules) are provided. In Fig. 18 and Fig. 19, time patterns of the current in the coil and relevant control variables for feedback systems with *on-off* and fuzzy controllers are shown.

It is seen in Fig. 17 that the damping system with the MR damper and *on-off* controller performs well at frequencies exciding 3 Hz, and the system with the fuzzy controller – above 4 Hz. Time patterns in Fig. 18 prove correct performance of the implemented *on-off* controller that begins to function when the product $\dot{x}_1(\dot{x}_1 - \dot{x}_0) > 0$ assumes a value above zero. Time patterns in Fig. 19 show that the fuzzy controller is responsible for an increase in the

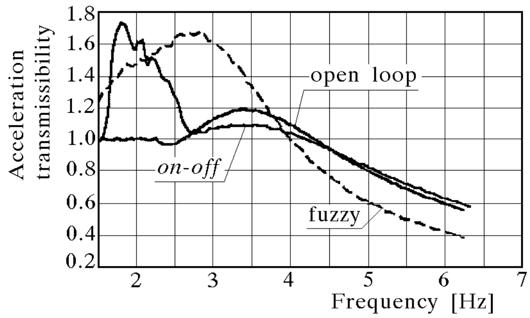

Fig. 17. Acceleration transmissibility in open loop and feedback systems with *on-off* and fuzzy controllers

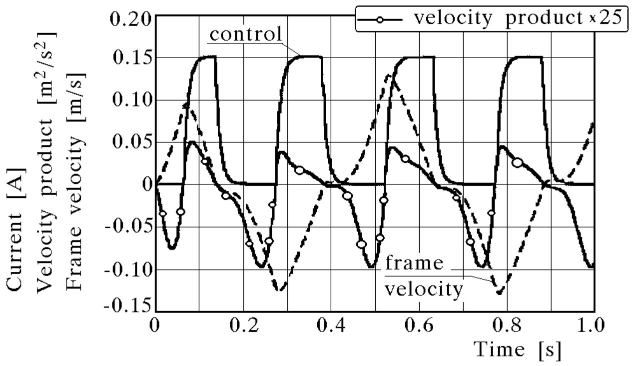

Fig. 18. Time patterns of control, velocity product and frame velocity in the feedback system with the *on-off* controller

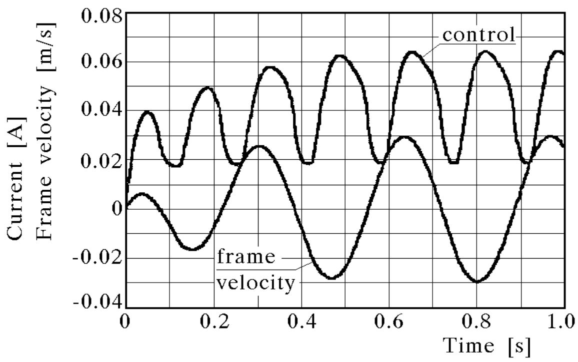

Fig. 19. Time patterns of control and frame velocity in the feedback system with the fuzzy controller

current in the coil when the seat velocity increases; at the same time, the amplitude of the current is still below 0.150 A.

Comparing the simulations results in the feedback system with fuzzy controllers containing 5 and 9 rules, we see that no significant improvement of system performance was achieved.

6. Laboratory testing of a driver's seat suspension with an MR damper

In this stage we investigate a feedback system (driver's seat suspension with an MR damper) with a fuzzy controller and compare its performance with an open loop and feedback system with an *on-off* controller. For this purpose, *on-off* and fuzzy real-time controllers were developed in an integrated design and control environment utilizing a PC computer.

6.1. Experimental setup

A schematic diagram of the experimental setup for a driver's seat is provided in Fig. 20. The power supply circuit consists of an electro-hydraulic shaker (EHS) with a hydraulic pump (P) and a control cubicle (CB). Input-output data was acquired using a data acquisition and control system based on a PC with a multipurpose I/O board (RT-DAC4), operating in the software environment of Windows 2000, MATLAB/Simulink and Real Time Windows Target (RTWT).

The displacements x_0 and x_1 were measured with linear displacement transducers (LVDT0 and LVDT1 with measuring range ± 0.01 m). The output signal from the controllers passes to the current driver and then to the MR damper coil.

A general view of the driver's seat with the MR damper is shown in Fig. 21. Potential damping states for the RD-1005 damper in the controllable range of 0.0-0.20 A are shown in Fig. 22.

6.2. Environment for real-time controllers development

The integrated design and control environment makes use of the state-of the art tools for simulation and modelling. Real-time control for an MR damper employed in the seat was developed using the toolbox RTW (Real

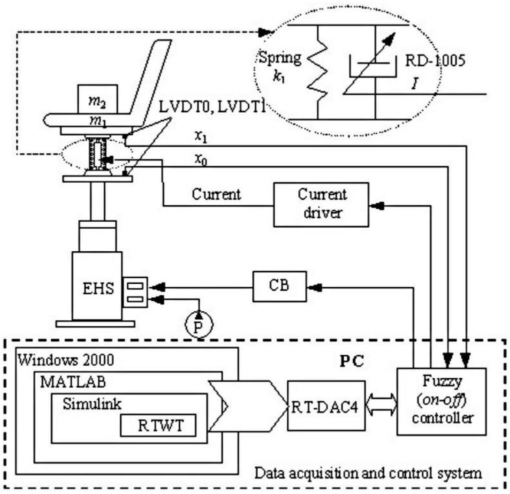

Fig. 20. Experimental setup for testing the driver's seat with the RD-1005 damper

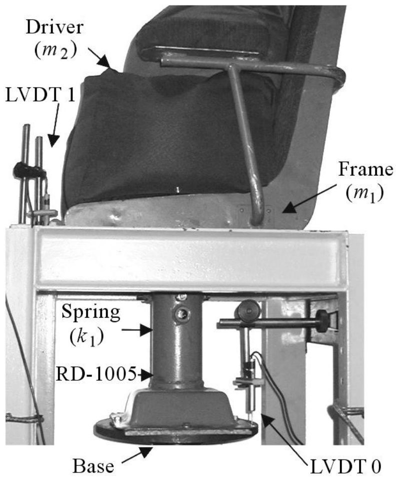

Fig. 21. General view of the driver's seat – ready for tests

Time Workshop) with the extension RTWT in the MATLAB/Simulink environment. A diagram of real-time task creation in this environment is shown in Fig. 23.

The device driver blocks close the feedback loop while moving from simulations to experiments. They include procedures in the language C. Real-time controllers were developed through compilation and linking stages, in a form of a dynamic link library (DLL), which was to be downloaded into memory and started-up in MS Windows 2000.

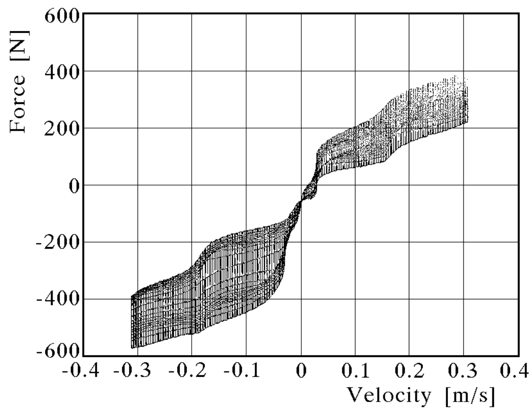

Fig. 22. Damping states for the RD-1005 damper

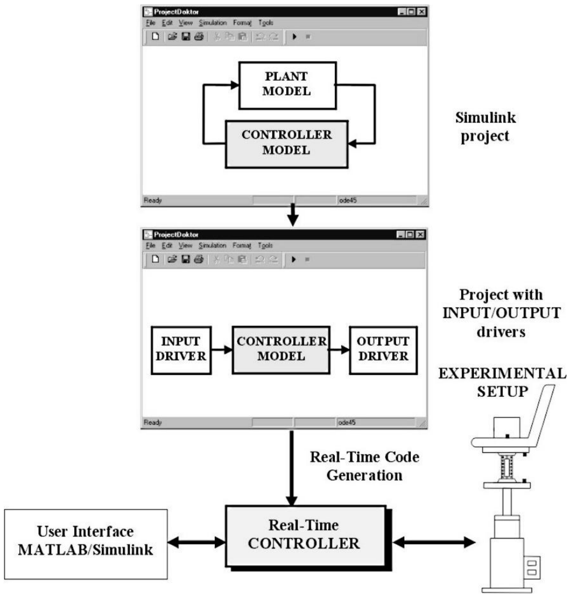

Fig. 23. Diagram of real-time task creation in the MATLAB/Simulink environment

6.3. Experiments in open loop and feedback systems

The first stage involved experiments in the open loop system configuration to determine the characteristic of acceleration transmissibility vs. frequency. Tests performed for base excitations with sine signals of the frequency 1-12 Hz and amplitude 1.5×10^{-3} m revealed that the resonance frequency of the seat is about 5 Hz (see Fig. 24).

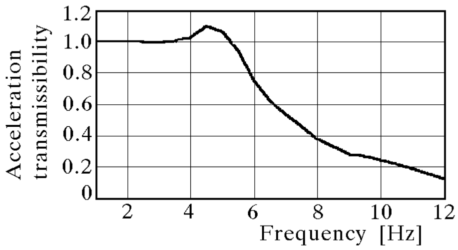

Fig. 24. Acceleration transmissibility in the open loop system

Similar experiments were conducted for feedback system configurations. These tests followed an experimental program to check the operation of real time *on-off* and fuzzy controllers. The seat frame velocity, and relative $\dot{x}_1$ velocity, $(\dot{x}_1 - \dot{x}_0)$ were the input signals which, like the acceleration signals, were reproduced by using derivative blocks of MATALB/Simulink. The sampling rate for real-time tasks was 1000 Hz. The constant c_1 for the *on-off* controller was taken as 0.10 and the base of rules for the fuzzy controller was assumed as that given in Table 1. Selected time patterns of control, velocity product and frame velocity for a sine displacement-input with the frequency 8 Hz, provided in Fig. 25 and Fig. 26, bespeak correct operation of both feedback system configurations.

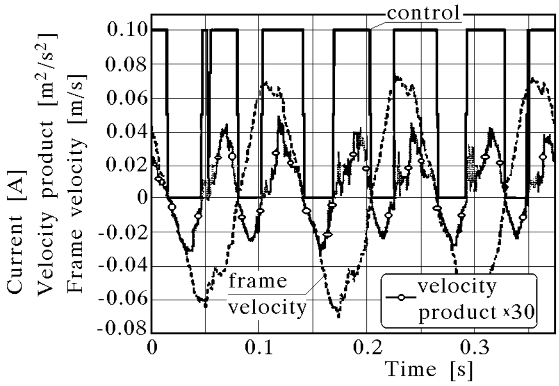

Fig. 25. Time patterns of control, velocity product, frame velocity in the feedback system with the *on-off* controller

Note that in the case of the *on-off* controller there are only two control values (i.e. levels of the applied current): 0.000 A and 0.100 A, while in the case of the fuzzy controller, the level of the applied current may assume any value from the range 0.000-0.060 A.

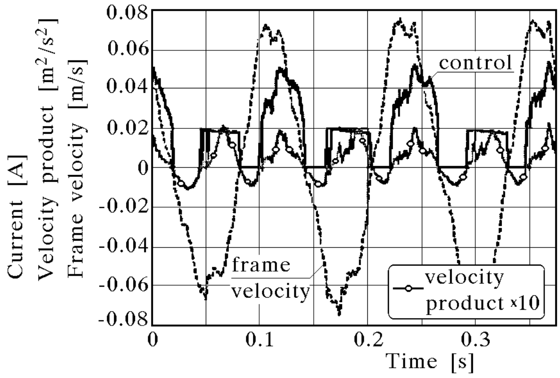

Fig. 26. Time patterns of control, velocity product, frame velocity in the feedback system with the fuzzy controller

In the second stage, frequency characteristics of acceleration transmissibility vs. frequency in feedback systems containing *on-off* and fuzzy controllers were determined (see Fig. 27).

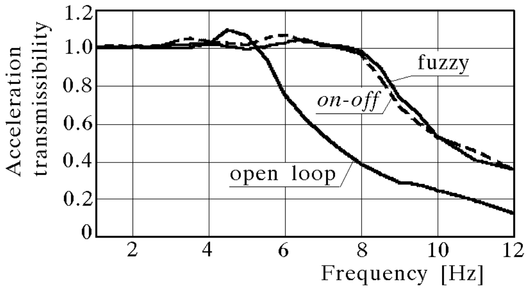

Fig. 27. Acceleration transmissibility in the open loop and feedback system with *on-off* and fuzzy controllers

When comparing these characteristics with that obtained in the open loop system, it is apparent that better performance is achieved in feedback system configurations. That was also proved in the time domain; see Fig. 28, where time patterns of acceleration for the seat frame at a near-resonance frequency (about 5 Hz) are provided.

Figures 27 and 28 do not adequately capture the better performance achieved thanks to application of the feedback system with the fuzzy controller. To demonstrate that, zoomed sections of time acceleration patterns for open loop and feedback systems with *on-off* and fuzzy controllers are compared in Fig. 29.

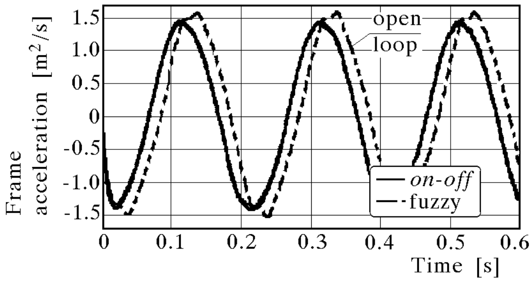

Fig. 28. Time patterns of frame acceleration in the open loop system and feedback systems with *on-off* and fuzzy controllers

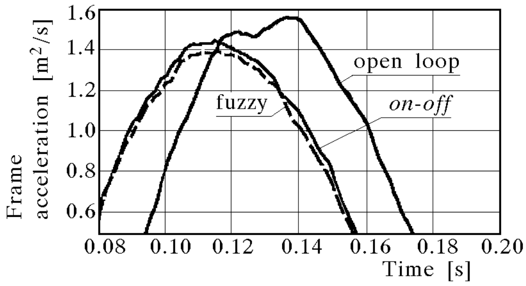

Fig. 29. Time patterns of frame acceleration – zoomed sections

Experiments performed in the feedback system with fuzzy controllers, when the number of rules went up from 5 to 9, revealed no significant improvement of system performance, which is in agreement with the statement given in Subsection 5.3.

To estimate the performance of feedback system configurations, the following factors were introduced: f_1, f_2, Δf, DP, S_1 and $S_1 + S_2$, (see Fig. 30). The computed values of performance factors are compared in Table 2.

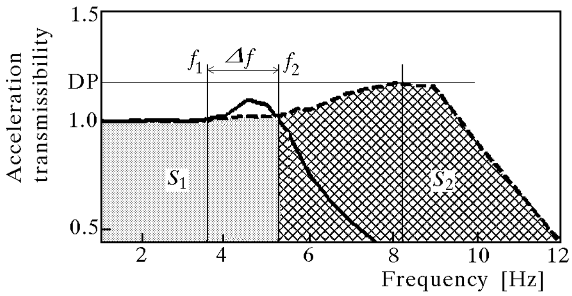

Fig. 30. Performance factors

Table 2. Performance factors of feedback systems with *on-off* and fuzzy controllers

Performance factor	*On-off* controller	Fuzzy controller
f_1 [Hz]	4.07	4.02
f_2 [Hz]	5.14	5.25
Δf [Hz]	1.07	1.23
DP	1.076	1.043
S_1	8.170	8.080
$S_1 + S_2$	11.086	11.627

Note that the fuzzy controller has better features (wider frequency control range and smaller values of DP, S_1) than the *on-off* controller.

7. Conclusions

The paper reports research on fuzzy control applied to an MR damper employed in a driver's seat support. It also indicates a potential application of fuzzy logic that can be used in semi-active dampers. The criterion for estimation of the system performance was based on the sprung mass acceleration.

The fuzzy controller was developed on the expert knowledge acquired in the course of experiments. It appears to be flexible as its parameters may easily be altered through changes of membership functions and rules. The number of 5 rules for the fuzzy controller was established as that giving satisfactory improvement of the feedback system performance in comparison with that containing an *on-off* controller.

The vibration control of the seat (with no cushion) for both feedback system configurations was effective throughout a frequency range of 3-5 Hz. Advantages of the fuzzy algorithm in comparison to the *on-off* algorithm are revealed basing on the performance factors provided in Table 2.

In the experiments on the feedback system with the *on-off* controller, the presence of the chattering effect was observed. As a result, the control signal could be produced in the states when velocity product oscillated round the zero value (the effect of non-zero off state damping). When assuming the dead zone, the chattering effect could be eliminated (Sapiński and Rosół, 2003).

The research work is now focused:

- on fuzzy control for an MR damper in the seat with a cushion (2 DOF system),
- introducing two switching criteria for the MR damper (one principle based on the sprung mass acceleration and the second on relative displacement of the seat frame and shaker base).

Acknowledgement

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Sterowanie rozmyte tłumikiem MR w zawieszeniu fotela kierowcy

Streszczenie

W artykule przedstawiono zastosowanie logiki rozmytej do sterowania tłumika magnetoreologicznego (tłumika MR) w zawieszeniu fotela kierowcy. Fotel zamodelowano jako układ o jednym stopniu swobody, do którego dołączono uogólniony model tłumika MR uwzględniający fluktuacje pola magnetycznego. W symulacjach komputerowych porównano działanie układu ze sprzężeniem zwrotnym z regulatorem rozmytym i regulatorem dwupołożeniowym. Wyniki symulacji zweryfikowano w badaniach laboratoryjnych. W tym celu zrealizowano regulatory czasu rzeczywistego (rozmyty i dwupołożeniowy) w środowisku projektowania i sterowania MATLAB/Simulink. Zalety regulatora sprawdzono eksperymentalnie.

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