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CONTROLLING THE VIBRATION OF A SEAT SUSPENSION SYSTEM WITH THE USE OF A MAGNETO-RHEOLOGICAL DAMPER

STEROWANIE DRGANIAMI UKŁADU ZAWIESZENIA SIEDZISKA Z WYKORZYSTANIEM TŁUMIKA MAGNETO-REOLOGICZNEGO

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Summary

A model and control strategy of a semi-active seat suspension system with a magneto-rheological (MR) damper, chiefly intended to protect construction machine operators from vibration acting in one of the horizontal directions, has been presented. The parameters of the model shown were determined experimentally as functions of the damper control current, for which a control algorithm imitating the operation of a sky-hook damper (a damper hanging from an inertial reference point) was

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developed. Afterwards, the model proposed was verified on a test stand with an electrohydraulic actuator and then the semi-active seat suspension system was experimentally tested with the participation of a human subject. Results of the work carried out have been presented in the form of power spectral densities of vibration accelerations and transfer functions of the suspension system. Based on a qualitative assessment of the test results obtained, the operation of the semi-active seat suspension system was found to be far more effective than that of a conventional system in the excitation frequency range under consideration. Noteworthy is the fact that the highest effectiveness of the system operation was achieved at the resonance frequency, corresponding to that of the passive system. This is confirmed by the tabulated values of vibration transfer coefficients and maximum relative displacements of the seat suspension system in the conventional passive version and in the semi-active version provided with an MR damper.

Keywords: vibration control, seat suspension system, magneto-rheological damper

Streszczenie

W pracy zaprezentowano model oraz strategię sterowania semi-aktywnym układem zawieszeniem siedziska z tłumikiem magneto-reologicznym (MR), którego podstawowym zadaniem jest zapewnienie ochrony operatorów maszyn roboczych przed drganiami w jednym z poziomych kierunków oddziaływania. Parametry przedstawionego modelu wyznaczono eksperymentalnie w funkcji natężenia prądu sterującego pracą tłumika, dla którego opracowano algorytm sterowania naśladujący działanie tłumika zawieszonego w inercjalnym punkcie odniesienia (ang. sky-hook damper). Kolejno dokonano weryfikacji zaproponowanego modelu na stanowisku badawczym ze wzbudnikiem elektro-hydraulicznym oraz przeprowadzono badania eksperymentalne semi-aktywnego układu zawieszenia siedziska z udziałem człowieka. Wyniki zrealizowanych prac zaprezentowano w postaci gęstości widmowych mocy przyspieszenia drgań oraz funkcji przenoszenia układu zawieszenia. Na podstawie oceny jakościowej otrzymanych wyników badań stwierdzono, że działanie semi-aktywnego zawieszenia siedziska jest znacznie skuteczniejsze od układu konwencjonalnego w rozpatrywanym zakresie częstotliwości wymuszenia. Jednak największą skuteczność działania osiąga się w przypadku częstotliwości rezonansowej, odpowiadającej układowi pasywnemu. Potwierdzają to zestawione tabelarycznie wartości współczynników przenoszenia drgań i maksymalnych przemieszczeń względnych zawieszenia w przypadku konwencjonalnego układu pasywnego oraz układu semi-aktywnego z tłumikiem MR.

Słowa kluczowe: sterowanie drganiami, zawieszenie siedziska, tłumik magneto-reologiczny

1. Introduction

The effectiveness of operation of vibration-reducing systems can be raised by using semi-active suspension systems with controllable spring or damping elements. Thanks to such systems, low-frequency vibration may be reduced, with good vibro-isolating properties being simultaneously maintained at higher excitation frequencies. In many cases, the spring characteristics of vibro-isolation systems are controlled by means of pneumatic systems [1, 2], where variable stiffness of the system is obtained by connecting an additional reservoir. On the other hand, electro-rheological (ER) [3, 4] or magneto-rheological (MR) [5] dampers are often used to control the damping characteristics of the system.

The objective of the work described herein was the construction of a model of an MR damper for the needs of controlling the horizontal vibration of a semi-active seat suspension system. This work was to result in determining the vibro-isolating properties of the system in order to improve the possibilities of protecting construction machine operators from the harmful impact of mechanical vibration.

2. Model of a semi-active seat suspension system

Fig. 1 shows a model of a semi-active seat suspension system used to protect construction machine operators from vibration exposure in a horizontal direction.

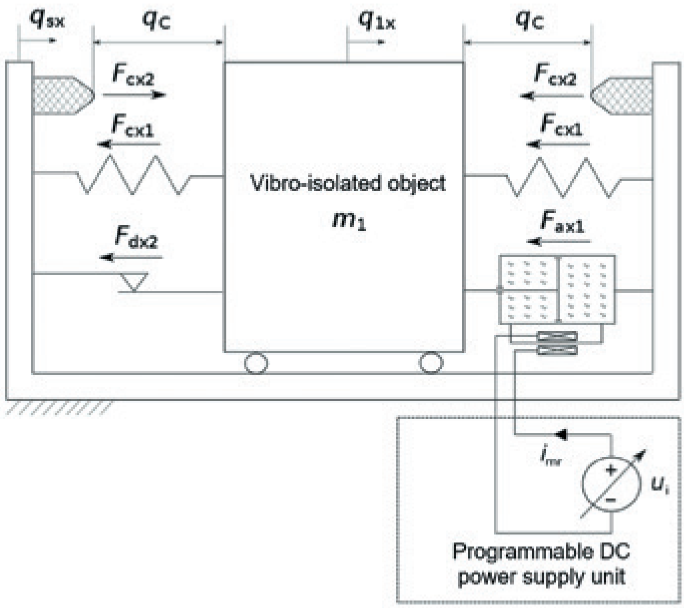

Fig. 1. Physical model of a semi-active seat suspension system used to protect construction machine operators from vibration exposure in a horizontal direction

To establish the position of equilibrium, two tension springs have been provided in the system, acting in opposite directions on the vibro-isolated object; the reaction forces exerted by the springs have been denoted as F_{cx1}. When the suspension system displacement exceeds the predefined limits, the system may come into undesirable contact with one of the limit stops, which will then exert a reaction force of F_{cx2}. The phenomena related to the friction present in the kinematic pairs generate a force of F_{dx2} during the movement of the vibro-isolated object relative to the machine being in motion. The damper reaction force F_{ax1} is a force that controls the operation of the seat suspension system and its characteristics vary depending on the electric current flowing through the solenoid. The phenomenon of changes in the viscosity of the magneto-rheological fluid in the working gap due to modifications to the intensity of electromagnetic field results in changes in the local resistance to fluid flow and, in consequence, makes it possible to control the damping force in the system.

The equation of motion of the semi-active seat suspension system may be written as follows:

$$m_1\ddot{q}_{1x} = -2F_{cx1} \pm F_{cx2} - F_{dx2} - F_{ax1} \tag{1}$$

where: m_1 is the mass of the vibro-isolated object and q_{1x} is the displacement of the said object in the longitudinal direction, i.e. in the direction of machine motion. The characteristics describing the main forces acting in the seat suspension system (F_{cx1}, F_{cx2}, F_{dx2}) were determined experimentally and their models have been presented in publication [6].

This work was dedicated to modelling the characteristics of the force produced by the magneto-rheological damper at various values of the current controlling the system operation (Fig. 2). The damper reaction force was described in accordance with the Bouc-Wen model of the structure [7], in which a damping element and spring element are included and a hysteresis is represented by the following equation:

$$F_{ax1} = d(\dot{q}_{1x} - \dot{q}_{sx}) + c(q_{1x} - q_{sx}) + \alpha z + f_0 \tag{2}$$

where: d and c are damping and stiffness coefficients, respectively, describing the viscoelastic properties of the system modelled, α is a quantity that characterizes the damper friction force, governing the hysteresis height, and f_0 is the force generated by the damper in the state of static equilibrium. Variable z is related to the formation of the hysteresis loop and is described by the following equation [7], according to the Bouc-Wen model:

$$\dot{z} = \delta(\dot{q}_{1x} - \dot{q}_{sx}) - \beta(\dot{q}_{1x} - \dot{q}_{sx})|z| - \gamma z|\dot{q}_{1x} - \dot{q}_{sx}| \tag{3}$$

where: δ, β, and γ are coefficients that define the shape of the hysteresis loop.

The variations in damping coefficient d and friction force coefficient α as functions of control current i_{mr} are described by polynomial models in the form as follows [8]:

$$\begin{aligned} d &= a_1 i_{mr} + a_0 \\ \alpha &= b_2 i_{mr}^2 + b_1 i_{mr} + b_0 \end{aligned} \tag{4}$$

where: α_1, α_0, b_2, b_1, and b_0 are coefficients of the approximating polynomial functions.

The dynamic properties of the damper were represented in the model by a first-order inertial term [9] with a time constant t_{mr}, the value of which defines the rate of growth (or drop) in the force following a change in the value of the control signal, according to the equation:

$$t_{mr}\dot{i}_{mr} + i_{mr} = k_{mr}u_i \tag{5}$$

where: k_{mr} is the static gain of the magneto-rheological damper and ui is the damper force control signal.

The unknown model parameters were determined experimentally for the specific engineering design of the MR damper and their numerical values have been specified in Table 1.

Table 1. Numerical MR damper parameter values determined experimentally

Symbol	Value	Unit
c	1000	N/m
$f0$	0	N
δ	400	–
β	300	m^{-2}
γ	100	m^{-2}
a_1	40	Ns/(Am)
a_0	900	Ns/m
b_2	–1	$N/(A^2m)$
b_1	20	N/(Am)
b_0	60	N/m
t_{mr}	0.01	s
k_{mr}	0.42	A/V

3. System to control the semi-active seat suspension system

In the semi-active systems, the damping force is often modelled as being proportional to the absolute velocity of the vibro-isolated object [4]. Such a control method consists in simulating the force generated by a sky-hook damper (a vibration energy-dissipating system hanging at one of its ends from an immobile reference point). Unfortunately, such a damper fixing method is impracticable in the case of vehicle seat suspension systems because of the absence of a reference point that would be immobile in relation to the vehicle [10]. Therefore, a control algorithm was adopted in this work, which imitated the operation of a passive damper fixed to an inertial reference point.

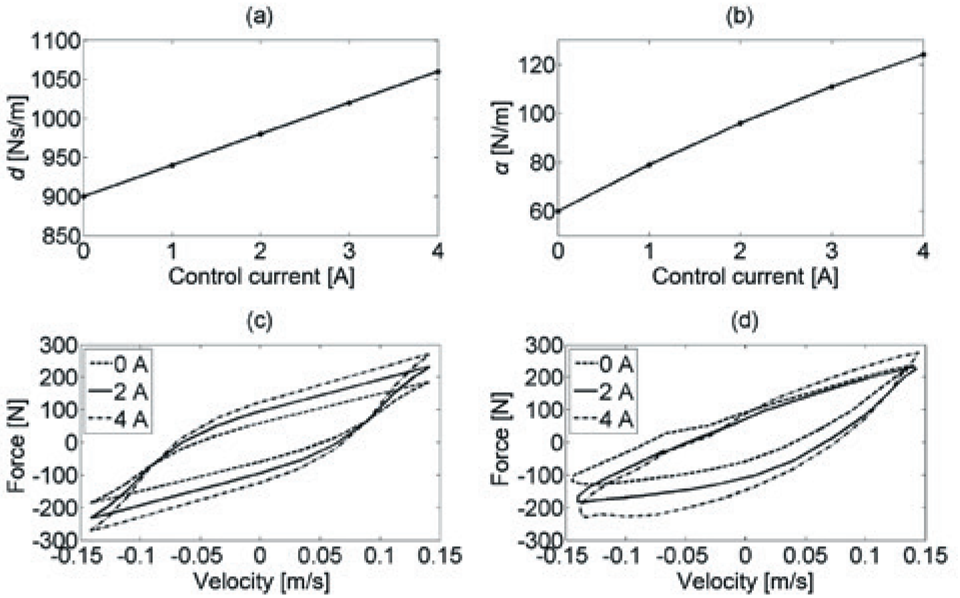

Fig. 2. Damping coefficient vs. control current (a), coefficient that governs the hysteresis height vs. control current (b), and characteristics of the magneto-rheological damper reaction forces at various values of the control current, obtained from computer simulations (c) and experimental measurements (d)

The control system has two feedback loops: from the absolute velocity $\dot{q}_{1x}$ of the vibro-isolated object and from the relative velocity $\dot{q}_{1x} - \dot{q}_{sx}$ of the seat suspension system. The algorithm of controlling the magneto-rheological damper reaction force is represented by the following relation:

$$u_{mr} = \begin{cases} g_{sky}\left|\dot{q}_{1x}\right| & if \quad \dot{q}_{1x}\left(\dot{q}_{1x} - \dot{q}_{sx}\right) \geq 0 \\ u_{min} & if \quad \dot{q}_{1x}\left(\dot{q}_{1x} - \dot{q}_{sx}\right) < 0 \end{cases} \tag{6}$$

where: g_{sky} is the controller setting that governs the static gain of the feedback loop from the absolute velocity of the vibro-isolated object.

Signal u_i, which controls the damper force, is limited to the control voltage ranges adopted in the system design; this is expressed as follows:

$$u_i = \begin{cases} u_{min} & for \quad u_{mr} < u_{min} \\ u_{mr} & for \quad u_{min} \leq u_{mr} < u_{max} \\ u_{max} & for \quad u_{mr} \geq u_{max} \end{cases} \tag{7}$$

where: u_{min} is the minimum MR damper control voltage at which the lowest damper force is generated and u_{max} is the maximum MR damper control voltage at which the damper force reaches its highest value.

4. Verification of the model of a semi-active seat suspension system with an MR damper

To verify the model having been developed to represent a semi-active seat suspension system, experimental tests were carried out, where an electrohydraulic actuator was used to generate mechanical vibration. The actuator was to force movement of the seat suspension system under test by means of a stochastic input signal, whose spectral properties were close to those of white noise. In the work described herein, the input vibration was within a frequency range of 0.5-10 Hz and it was applied with various intensities, i.e. the root mean square (RMS) values of the system acceleration were 1.02 m/s^2, 1.36 m/s^2, and 1.91 m/s^2.

The test results, presented in the form of power spectral densities (PSD) of vibration accelerations and transfer functions of the suspension system (Fig. 3), were obtained for the system under test being loaded with 55 kg weights, which were fixed to the upper part of the seat suspension system.

Apart from qualitative assessment of the power spectral densities of vibration accelerations and transfer functions of the suspension system, the values of vibration transfer coefficients TFE_x and maximum relative displacements s_{tx} of the seat suspension system were brought together [11]. The relative simulation errors (relative differences between results of computer simulation and real measurements) were determined from the formulas:

$$\delta_{TFE_x} = \frac{\left|(TFE_x)_s - (TFE_x)_m\right|}{(TFE_x)_m} \tag{8}$$

$$\delta_{s_{tx}} = \frac{\left|(s_{tx})_s - (s_{tx})_m\right|}{(s_{tx})_m} \tag{9}$$

where: $(TFE_x)_s$ and $(s_{tx})_s$ are values of the criteria adopted, determined from the results of a computer simulation, and $(TFE_x)_m$ and $(s_{tx})_m$ are values of the same criteria, but determined from measurement results. The numerical values of these parameters have been specified in Table 2.

Table 2. Numerical values of the vibration transfer coefficient (TFE_x) and maximum relative displacement (s_{tx}) for the semi-active seat suspension system with an MR damper

Excitation intensity (RMS acceleration)	Simulation		Measurement		Relative error (absolute value)	
	TFE_x	s_{tx}	TFE_x	s_{tx}	TFE_x	s_{tx}
1.02 m/s^2	1.050	14.7 mm	1.018	14.8 mm	4.9 %	0.7 %
1.36 m/s^2	1.023	22.1 mm	1.016	21.5 mm	0.6 %	2.1 %
1.91 m/s^2	0.997	28.5 mm	1.012	30.6 mm	2.5 %	6.9 %

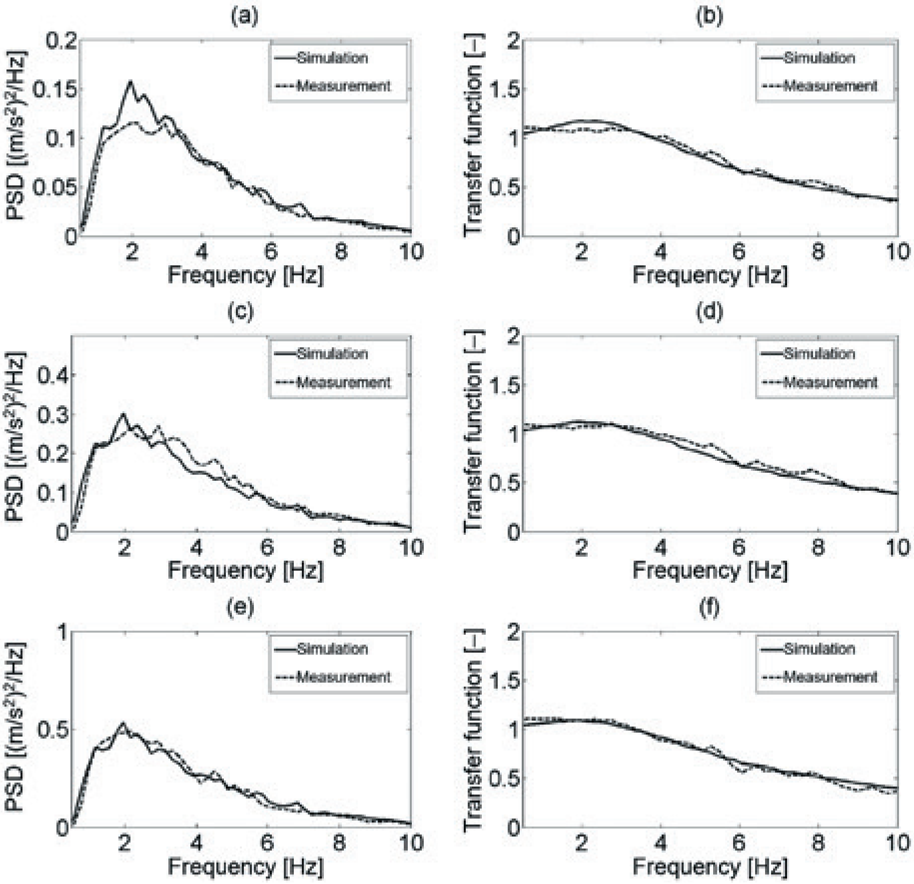

Fig. 3. Power spectral densities (PSD) of vibration accelerations and transfer functions determined for the semi-active seat suspension system with an MR damper from a computer simulation and real measurements at various excitation intensities, i.e. at RMS acceleration values of 1.02 m/s^2 (a-b), 1.35 m/s^2 (c-d), and 1.91 m/s^2 (e-f)

5. Experimental testing of the semi-active seat suspension system with the participation of a human subject

To verify the effectiveness of the seat suspension system design proposed, the system was subjected to experimental tests carried out with the participation of a human subject. The seat suspension system was loaded with a sitting human subject, whose body mass was 90 kg. The tests were carried out on a conventional passive system and on the semi-active seat suspension system with an MR damper, developed within this work. A comparison of the power spectral densities of vibration accelerations and transfer functions of the suspension system, measured at various excitation intensities, has been presented in Fig. 4.

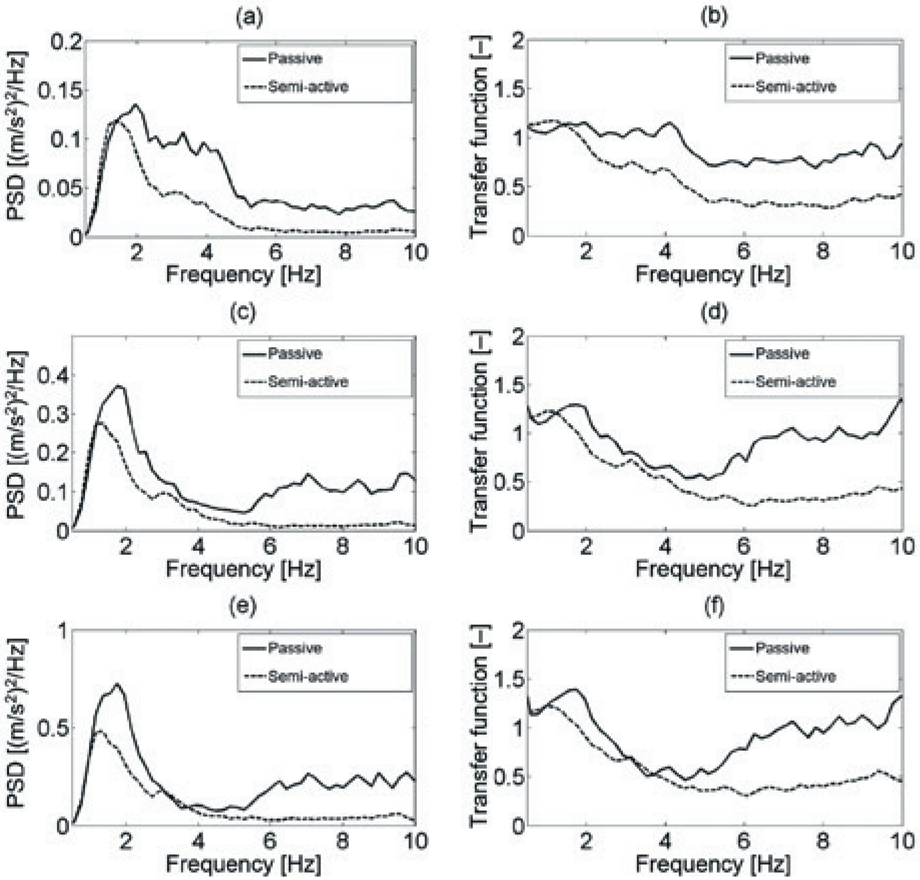

Fig. 4. Power spectral densities (PSD) of vibration accelerations and transfer functions measured for a passive seat suspension system and a semi-active seat suspension system with an MR damper at various excitation intensities, i.e. at RMS acceleration values of 1.02 m/s^2 (a-b), 1.35 m/s^2 (c-d), and 1.91 m/s^2 (e-f)

Based on a qualitative assessment of the test results obtained, the operation of the semi-active seat suspension system with an MR damper was found to be far more effective than that of a conventional system (Fig. 4). The conventional passive system increases the vibration amplitudes in the frequency range close to the natural frequency of the system, with the system effectiveness being quite low for the super-resonant frequencies. Conversely, the semi-active system having been developed effectively lowers the vibration amplitudes over the whole range of the excitation frequencies under consideration (0.5-10 Hz); thus, the protection of human beings from the harmful impact of vibration in the working environment becomes far more efficient. The measured values of the vibro-isolation criteria, i.e. vibration transfer coefficient TFE_x and maximum relative displacement s_{tx} of the seat suspension system, have been brought together in Table 3.

Table 3. Measured values of the vibration transfer coefficient (TFE_x) and maximum relative displacement (s_{tx}) for the seat suspension system

Excitation intensity (RMS acceleration)	Passive system		Semi-active system	
	TFE_x	s_{tx}	TFE_x	s_{tx}
1.02 m/s^2	1.050	18.2 mm	0.881	21.1 mm
1.36 m/s^2	1.046	25.1 mm	0.857	34.2 mm
1.91 m/s^2	1.071	28.2 mm	0.862	39.4 mm

6. Conclusions

The implementation of the strategy of controlling the vibration of a seat suspension system as proposed in this work will help to reduce the harmful impact of vibration on construction machine operators in a wide range of excitation frequencies. The measured values of vibro-isolation criteria clearly indicate that the effectiveness of operation of a semi-active seat suspension system is much better than that of the system of conventional design. This is proved by a reduction in the vibration transfer coefficient TFE_x by about 20 % in comparison with that of the passive system; however, the maximum relative displacement values grow then by even up to 30 %. This is because the semi-active vibration-reducing system requires that a wider range of relative displacements must be allowed in consideration of limited possibilities of controlling the magneto-rheological damper.

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