Adaptive Control of Magnetorheological Quarter-Car Suspension Model using Normalized LMS Algorithm

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Abstract

The paper presents modification of NLMS algorithm and its application in vehicle vibration control using magnetorheological (MR) dampers. Dynamics of a quarter-car model which exhibits 2 degrees of freedom (DOFs) was analyzed. MR damper was included in the suspension model as an actuator with generated force limited to its dissipative domain which was defined using nonlinear cyclometric atan function. A fictitious nonlinear damper was constituted and excluded from MR damper model. Such modification allowed for assuming the MR damper to be capable of adding energy to the suspension system which is compliant with the requirements of NLMS algorithm. A simulator which takes into account continuous and discrete-time domains was implemented using Matlab programming environment. Analysis performed in time and frequency domain validated the NLMS based vibration control algorithm and shown its advantage over passive soft and hard suspension systems.

Keywords: quarter-car model, magnetorheological damper, adaptive control, Normalized LMS algorithm

1. Introduction

Intelligent materials such as magnetorheological fluid (MR) [1] play an important role in industrial applications, e.g.: vehicles, machines and large structures such as cable bridges or buildings which are subjected to strong earthquakes [1], [2], [3]. Due to significant dimensions of semiactive elements, they are mostly used in control of low frequency vibrations. MR damper, which is filled with MR fluid, is meant to be a compromise between non-adaptive passive dampers and significantly energy-consuming active ones. Moreover, parameters of MR dampers can be changed in real time with short response time equal to dozens of milliseconds [1].

MR dampers which are installed in road vehicles [2], [3] require sophisticated semiactive control schemes including additional information about the experimental suspension, road-induced vehicle excitation and the MR damper dynamics in order to improve ride comfort and roadholding factors. While modelling vehicle suspension dynamics, most often a vehicle is divided into quarter parts and model of each part is analyzed separately. Quarter-car models [1], [5] mostly exhibit 2 degrees of freedom (DOFs); the first DOF corresponds to vertical dynamics of the unsprung mass (i.e. wheels including tyres); the second DOF corresponds to the sprung mass (i.e. vehicle body part). Some control applications need the quarter-car model to be extended to half-car model [1], [4], [6] (which most often exhibits 4 DOFs) or full-car model [2], [7] (3 DOFs or 7 DOFs).

Generally, in mechanical applications feedback and feedforward control schemes can be distinguished [1]; moreover, the latter require system excitation to be measured or estimated. Another classification of control algorithms is made with respect to their adaptability rate. Adaptive algorithms can react to the changes of suspension parameters. Another possibility is to make the algorithm adjust in real time to the varying features of road-induced excitation signal which depends on longitudinal vehicle velocity as well as on road profile itself.

Least mean square (LMS) algorithm is widely used in noise control applications [8]. LMS algorithm is a special case of gradient descent optimization method. Normalized LMS (NLMS) control scheme is one of the extensions of LMS in which the power of the reference signal is additionally normalized [8]. The finite response filter (FIR) adjusted by LMS algorithm processes the reference signal which results in a control signal of an actuator included in the system.

Adaptive algorithms are one of the main fields of interest in semiactive vehicle suspension control. Nonlinear feedback adaptive control was presented in [9]. The author included in the algorithm an identification block which is responsible for estimation of suspension parameters. Consequently, an LMS optimization formula is used to adjust the current supplying an MR damper. However, such control approach limits the control current to constant values in steady-state.

The article is organized as follows. In the second chapter the quarter-car and MR damper models are defined. The third chapter refers to a modified NLMS algorithm which is dedicated to semiactive elements. In the fourth chapter simulation results of the adaptive control are reported. The fifth chapter concludes the research presented in the paper.

2. Modelling of magnetorheological quarter-car suspension

Presented research is dedicated to semiactive vibration control system [3] which is installed in an experimental all-terrain vehicle. Original dampers of the vehicle suspension were replaced with Lord's MR dampers. Additionally, the vehicle is equipped with dedicated controllers which are used for acquisition of measurement data and generation of control current which supplies MR dampers. Strong nonlinearities in vehicle suspension dynamics, changes in ambient temperature and influence of the driver cause the parameters of the vehicle suspension to vary during each ride. The NLMS adaptive control algorithm is proposed to deal with the above-mentioned phenomena.

2.1 Quarter-car suspension model (2 DOFs)

The experimental vehicle was conventionally divided into quarter parts and a selected part was modelled using a 2 DOFs mechanical system (Figure 1). The lower, unsprung mass of the model corresponds to one vehicle wheel and its assemblies. The upper, sprung mass maps a quarter of the vehicle body.

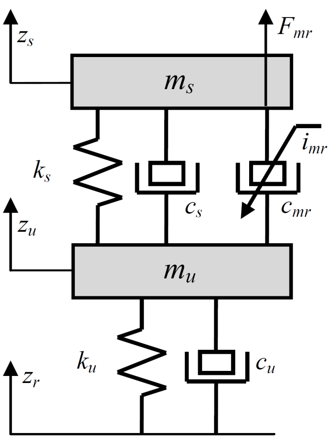

Fig.1. Quarter-car suspension model including MR damper.

The quarter-car model is defined using the following two differential equations which describe dynamics of the unsprung and sprung masses, respectively:

$$m_u \ddot{z}_u = -(k_u + k_s) z_u - (c_u + c_s) \dot{z}_u + k_u z_r + c_u \dot{z}_r + k_s z_s + c_s \dot{z}_s - F_{mr}, \quad (1)$$

$$m_s \ddot{z}_s = -k_s z_s - c_s \dot{z}_s + k_s z_u + c_s \dot{z}_u + F_{mr}, \quad (2)$$

where m_u and m_s denote unsprung and sprung masses, respectively. Symbols k_u, c_u and k_s, c_s denote stiffness and viscous damping of the suspension as well as of a vehicle wheel and its assemblies, respectively. Quarter-car model is subjected to the road-induced excitation signal denoted as z_r. Response of the sprung and unsprung mass to the excitation is described by vertical absolute displacement and denoted as z_u and z_s, respectively. Unsprung and sprung masses are additionally subjected to force generated by MR damper (marked in Figure 1 as c_{mr}) which is denoted by F_{mr} and depends on control current i_{mr}. Parameters of the quarter-car model were defined in [5] (presented in Table 1) and used in order to obtain simulation results presented in the current paper.

2.2 MR damper dedicated force constraints

Behavior of MR fluid subjected to stress excitation is known to be complex. Force-displacement and force-velocity characteristics obtained for MR damper reveal its significantly nonlinear dynamics. Inherent stability of semiactive systems is burdened with limitations of their functionality, i.e. dependence of generated force on instantaneous MR damper's relative piston motion. Force generated by MR damper is limited to energy dissipation which can be described using dissipative domain indicating instantaneous conditions of MR damper's activity. Dissipative regions are marked in a graphical illustration of dissipative domain (Figure 2). It can be stated that the semiactive damper is capable of generating a desired force if the sense of this force vector is opposite to the piston velocity vector.

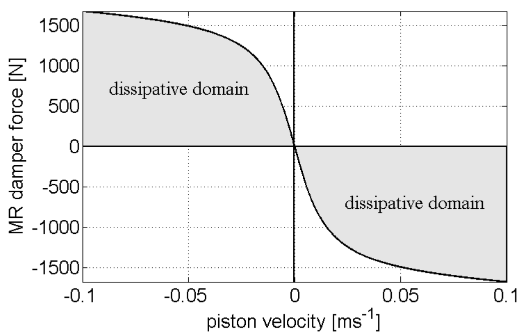

Fig.2. Graphical illustration of MR damper's dissipative domain.

Most semiactive vibration control algorithms include additional inverse MR damper model which allows for linearizing nonlinearities indicated in force-velocity characteristics. However, performance of simulated and real algorithm is deteriorated by inaccuracy of the inverse model. Thus, application of inverse model is not considered in the current paper. However, semiactivity of MR damper as well as nonlinear shape of its dissipative domain is taken into account in the presented analysis. Reachable force which can be generated by MR damper is limited bilaterally by the velocity axis and nonlinear cyclometric function which is derived from the nonlinear MR damper [10], as follows:

$$F_{mr,max}(\dot{z}_{mr}) = F_{mr}(i_{mr}, \dot{z}_{mr})\big|_{i_{mr}=I_{max}} \\ = \alpha(I_{max}) \cdot \mathrm{atan}[\beta(I_{max}) \cdot \dot{z}_{mr}] + c(I_{max}) \cdot \dot{z}_{mr}, \quad (3)$$

where $\dot{z}_{mr} = \dot{z}_s - \dot{z}_u$ and denotes MR damper relative piston velocity. Limitation of MR damper force $F_{mr,max}$ depends on relative piston velocity and is obtained for boundary control current I_{max}. Parameters α, β and c depend on control current and were estimated based on MR damper dedicated experimental data [10].

3. Applications of NLMS in semiactive suspension control

Most adaptive algorithms, such as LMS or its extension - Normalized LMS, are dedicated to real systems which are equipped with active actuators [11]. In case of active suspension elements used in vehicle suspension control [4], the control force domain (equivalent to dissipative domain) covers more regions of the control space (compared to Figure 2) which indicates that the capability of control force generation is available regardless of the relative velocity of such actuators. Semiactivity of MR dampers makes application of LMS-like algorithms challenging and requires modification of the dissipative domain of MR damper from asymmetrical to symmetrical one.

3.1 NLMS adaptive algorithm

Normalized LMS (NLMS) algorithm is an optimization method used, among others, in adaptive feedforward control schemes in order to adjust parameters of a finite response filter which is intended to model a controlled plant. In the presented case a discrete-time control signal of the semiactive element denoted as $y_2(n)$ is obtained by filtering a reference signal $x(n)$. Response of the FIR filter and response of the plant to its excitation interfere with each other resulting in an error signal $e(n)$. The error signal is fed into the block of NLMS algorithm (Figure 4) which minimizes a squared error signal $e^2(n)$. The NLMS algorithm operates in discrete-time domain. Hence, time-continuous error and reference signals need to be sampled, as follows:

$$x(n) = \dot{z}_r(nT_{s,control}), \\ e(n) = -100 \cdot \dot{z}_s(nT_{s,control}). \quad (4)$$

Equation (4) shows that reference signal is assumed as the road-induced velocity excitation $\dot{z}_r$ and the error signal corresponds to the absolute vertical velocity of the sprung mass $\dot{z}_s$. The update period of the control algorithm denoted as $T_{s,control}$ is intentionally distinguished from the integration period $T_{s,model}$ of the simulated quarter-car model. Additionally, desired force generated by MR damper F_{alg} is assumed as a control signal $y_2(n)$ of the quarter-car system, as follows:

$$y_2(n) = F_{alg}(n) = \bar{h}(n) \cdot \bar{x}(n), \quad (5)$$

where $\bar{x}(n)$ = $[x(n), x(n\text{-}1), \ldots, x(n\text{-}M)]^T$ and parameters of the FIR filter denoted as $\bar{h}$ are adjusted according to the NLMS algorithm:

$$\bar{h}(n+1) = \bar{h}(n) + \frac{\mu \cdot e(n) \cdot \bar{x}(n)}{\bar{x}^T(n) \cdot \bar{x}(n)}. \quad (6)$$

Parameters vector $\bar{h}$ is adjusted in every control iteration which is denoted as n and the adaptation process can be influenced using an adaptation constant denoted as μ. Correction value is normalized using power estimation of the reference signal $x(n)$ located in the denominator which is shown in Equation (6).

3.2 NLMS based vibration control using MR damper

Classical NLMS algorithm assumes the reference and error signals as well as the control signal y_2 to be defined within a set of real numbers with mean value equal to zero. However, in case of semiactive elements control forces within the dissipative domain are either positive or negative valued depending on the sign of relative piston velocity. Such features require to include an additional control dedicated offset value which allows the NLMS to operate in range of real numbers. It is applied by fictitiously excluding the mean viscous damping of the MR damper from its model and including such nonlinear passive damper in the suspension model (Figure 3).

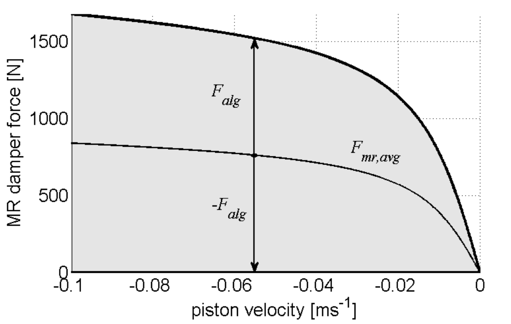

Fig.3. Modification of MR damper dissipative domain dedicated to NLMS algorithm.

The fictitious passive damper is described using MR damper model (3), as follows:

$$F_{mr,avg}(\dot{z}_{mr}) = \frac{1}{2} F_{mr}(i_{mr}, \dot{z}_{mr}) \Big|_{i_{mr}=I_{max}} . \quad (5)$$

The algorithm will generate real numbered control force which mitigates vibrations of the sprung mass (Figure 3) and it will indirectly make the FIR filter map both the quarter-car dynamics as well as fictitious passive nonlinear damper. Block diagram of the NLMS algorithm dedicated to magnetorheological suspension system is presented in Figure 4. The reference signal $x(n)$ is fed into the FIR filter denoted as $H(z^{-1})$ as well as it is assumed as an excitation signal of the quarter-car model. Parameters of filter H are adjusted according to the NLMS algorithm based on reference $x(n)$ and error $e(n)$ signals. The algorithm is modified by adding offset block denoted as $c_{mr,avg}$ which emulates behavior of fictitious nonlinear passive damper. Desired force generated by the fictitious damper directly depends on relative velocity of MR damper piston.

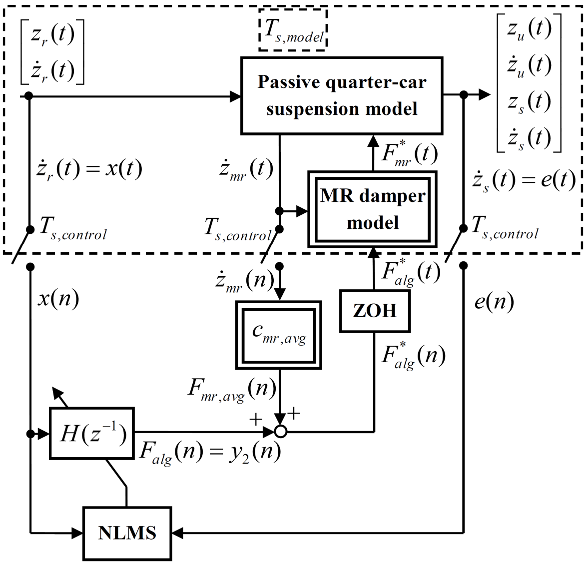

Fig.4. Block diagram of NLMS based control scheme dedicated to MR dampers.

4. Results

Simulation based research was done using Matlab programming environment. Quarter-car model and NLMS algorithm dedicated to MR dampers were implemented according to the block diagram (Figure 4). The control system is divided into continuous-time and discrete-time domains. Response of the quarter-car model was evaluated for sinusoidal road-induced excitation signal of frequency f_r which was defined for different experiments within frequency range from 1.0 to 15 Hz with resolution of 0.1 Hz. The continuous-time quarter-car model was simulated for integration period $T_{s,model}$ varying from 0.07 ms to 1 ms over different experiments which depends on the excitation frequency in order to retain constant number of simulated excitation sinusoidal cycles.

The implemented quarter-car model is described by differential Equations (1)-(2) and includes operational limitations of MR dampers described in Equation (3). Parameters of the quarter-car model are listed in Table 1. Modal analysis of the quarter-car model was performed; it indicates two damped resonant frequencies which correspond to vibrations of unsprung and sprung masses and are equal to 13.78 Hz and 1.98 Hz, respectively. Damping ratios of both quarter-car vibration modes were also estimated and are equal to 0.36 and 0.10, respectively.

Tab.1.

Parameters of the implemented quarter-car based simulation environment

Simulation environment		
$T_{s,model}$ = (0.07; 1.0) ms	$T_{s,control}$ = 10 ms	
Quarter-car model		
m_u = 12.5 kg	m_s = 87 kg	
k_u = 90000 Nm^{-1}	k_s = 16000 Nm^{-1}	
c_u = 470 Nsm^{-1}	c_s = 300 Nsm^{-1}	
Modal analysis of the quarter-car model		
Sprung mode	f_{d1} = 1.98 Hz	ξ_1 = 0.10
Unsprung mode	f_{d2} = 13.78 Hz	ξ_2 = 0.36
MR damper model		
α = 1056.9 N	β = 88.2 sm^{-1}	c = 1367.0 Nsm^{-1}
NLMS based vibration control algorithm		
μ = 0.7		

The NLMS algorithm makes use of the quarter-car model responses sampled with sample period $T_{s,control}$ which is constant over different experiments and is equal to 10 ms. Parameters of the control algorithm, mainly adaptation constant, are also listed in Table 1.

Preliminary experiments which indicate correctness of application of NLMS algorithm in semiactive suspension control were performed for 3 control stages activated consecutively in real time. The time domain analysis of sprung mass vibrations was performed for both resonant frequencies dedicated to unsprung and sprung masses (Figures 5 and 6). Initial 6 seconds of time diagrams which are related to both experiments show results for deactivated vibration control algorithm. Resonance behavior and high damping of the sprung mass for both sprung and unsprung dedicated resonant frequencies are in line with expectations. During the second stage of control, which is activated within time period from 6th to 12th second, the fictitious nonlinear damper block $c_{mr,avg}$ is activated. However, execution of the rest of the algorithm, strictly adaptation procedure, is still suspended. The third period lasts from 12th to 18th second and shows results of full execution of the algorithm. In case of the first resonant frequency, mitigation of sprung mass vibrations is slightly improved (Figure 5). In case of the second resonant frequency, damping factor of sprung mass vibrations is expected to reach the result obtained for the first control stage (Figure 6). However, results indicate that the parameters of the algorithm are not optimal and its performance needs to be improved.

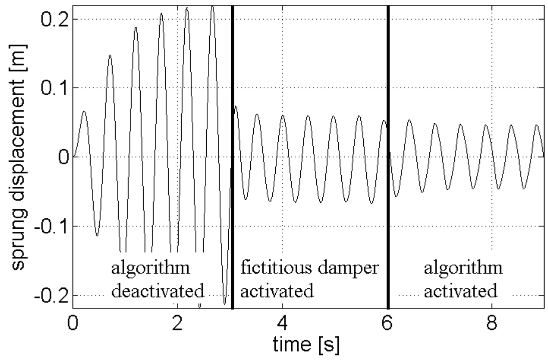

Fig.5. Different running modes of the NLMS based vibration control in time domain – sprung mass vertical displacement, 1.98 Hz excitation frequency.

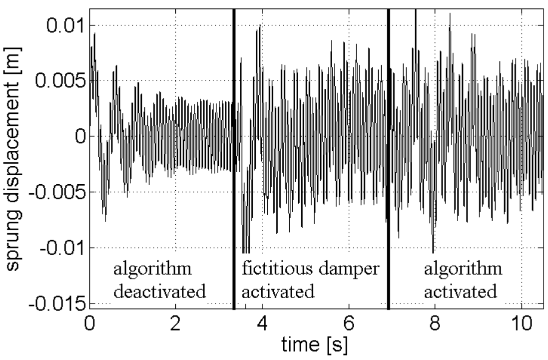

Fig.6. Different running modes of the NLMS based vibration control in time domain – sprung mass vertical displacement, 13.78 Hz excitation frequency.

Changes detected in vibrations level of unsprung mass are presented in Figure 7 for the experimental case corresponding to the first resonant frequency. Priority of the semiactive suspension control which is set on mitigation of sprung mass vibrations is reached at the cost of mitigation of unsprung mass vibration. For the last control period, when the third control stage is activated, significant deterioration in vibrations mitigation of unsprung mass can be noticed.

Summarizing analysis of the adaptive vibration control performance is illustrated using displacement transmissibility characteristics. Unsprung and sprung displacement transmissibility characteristics are evaluated as follows:

$$T_{ur/sr}(f_r) = \left.\frac{RMS(z_{u/s})}{RMS(z_r)}\right|_{f_r}, \quad (6)$$

where RMS stands for root mean squared value and each point of the characteristics is evaluated within 10 cycles of the sinusoidal suspension excitation z_r which is described as follows:

$$z_r(t) = 0.05 \cdot \sin(2\pi f_r t). \quad (7)$$

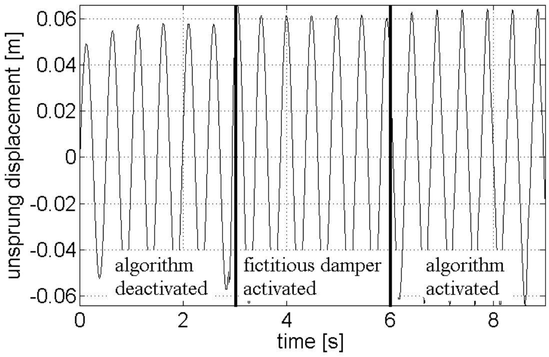

Fig.7. Different running modes of the NLMS based vibration control in time domain – unsprung mass vertical displacement, 13.78 Hz excitation frequency.

Results obtained for passive soft and hard as well as for the NLMS semiactive vibration control algorithms are presented in both transmissibility characteristics (Figures 8 and 9).

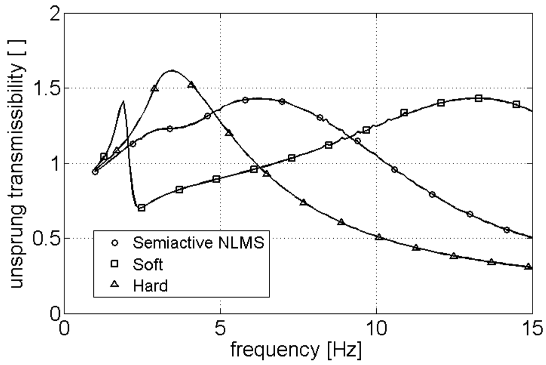

Fig.8. Unsprung displacement transmissibility characteristics for passive and semiactive suspension control.

Significant improvement of vibrations mitigation corresponding to sprung mass can be noticed in Figure 8 for both resonant peaks.

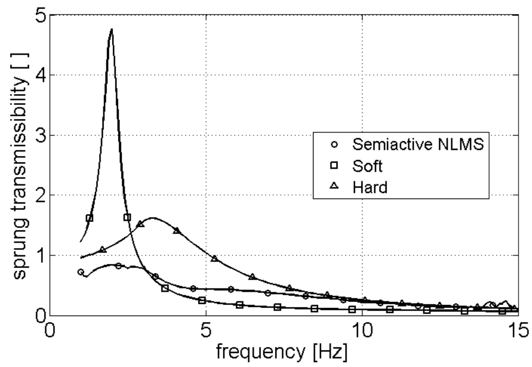

Fig.9. Sprung displacement transmissibility characteristics for passive and semiactive suspension control.

Mitigation of sprung mass vibrations is the main priority of the algorithm. Thus, mitigation of unsprung mass vibrations is deteriorated in comparison to hard suspension especially for the second resonant frequency.

5. Conclusions

Adaptive control algorithms are desired in real-time applications when experimental conditions as well as parameters of a plant are varying in time or cannot be precisely described. In such cases the adaptive control scheme can significantly improve performance of vibration control.

The article deals with application of NLMS algorithm in the control of vehicle suspension system modelled by 2 DOFs quarter-car and equipped with MR damper. Due to the inherent limitations of semiactive elements, classical NLMS algorithm was modified by adding fictitious nonlinear viscous damper which corresponds to the averaged damping of the MR damper. In such a case the MR damper can be assumed to be capable of adding energy to the system.

Time domain validation of the algorithm was performed for both vibration modes by analyzing vibrations level of unsprung and sprung masses. It was stated that adaptive algorithm efficiently mitigates vibrations of sprung mass in steady-state. The analysis performed in frequency domain shows that NLMS algorithm exhibits better sprung vibration mitigation than passive soft and hard suspensions.

Future analysis will be focused on improving performance of the algorithm in case of higher frequencies of the vehicle excitation. Presented approach dedicated to semiactive suspension systems will be extended to half-car and full-car models.

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