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A STUDY OF A SEMIACTIVE CAR SUSPENSION ADAPTIVE CONTROL ALGORITHM EMPLOYING A NON-PARAMETRIC MR DAMPER MODEL

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ABSTRACT: This paper deals with an implementation of a non-parametric magneto-rheologic (MR) damper model to the adaptive control algorithm of the semi-active car suspension. AA convergency into the optimal control in real time allows implementation of constrains (like max. allowed travel, max. allowed tire force) into the system through penalization functions. Thereby, a constrained optimization is turned to an unconstrained optimization. This approach allows minimize compromises necessity resulting from obligations to ride comfort, suspension travel and tire holding.

KEYWORDS: adaptive control, car suspension, non-parametric MR damper model.

1. INTRODUCTION

In general, a car suspension is a non-linear dynamical system subjected to an unknown vibration source, which depends on a road nature and a vehicle speed. Regarding a semi-active car suspension dealing with a MR damper, proper choice of a control algorithm is a main factor to reach maximal suspension performance. The suspension performance is given by the optimization criteria [2, 5]. Such control algorithm has to include an accurate and a numerically effective mathematical model of a MR damper to achieve a necessary suspension performance. The possible kind of way is to bring sufficiently robust AA which should be able to perform sufficient control of the non-linear system subjected to a wide-spectrum of non-stationary random excitation sources and also should be able to adapt in terms of real time changes in a system through on-line identification. [3]

2. A NON-PARAMETRIC MR DAMPER MODEL

Song [4] has brought a non-parametric MR damper model through analytical functions in combination with a differential equation, which provides a hysteresis in a damper characteristic.

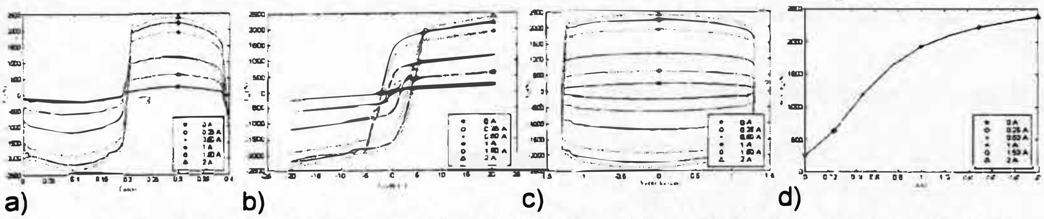


Fig. 1: Experimentally measured characteristics of the MR damper, a) force–time, b) force–velocity, c) force–displacement, d) maximum force–current.

Concerning a non-parametric MR damper model it is necessary to describe an each aspect of the damper behavior separately through a proper chosen mathematical functions. In order to describe the force-velocity characteristic Fig. 1(b), it is obvious that some hyperbolic function would describes real damper behavior. Such function could be in form of (1). The bending in the force-velocity characteristic near zero velocity is caused by a MR fluid passing through yield stress [5], whereas the earlier mentioned hyseresis is caused by a MR fluid dynamics raised by compressed gas in the system which basically behaves as an air spring.

$$S_b(v_p) = \frac{\left(b_0 + b_1\left|v_p - v_0\right|\right)^{b_2(v_p - v_0)} - \left(b_0 + b_1\left|v_p - v_0\right|\right)^{-b_2(v_p - v_0)}}{b_0^{\,b_2(v_p - v_0)} + b_0^{\,-b_2(v_p - v_0)}} \tag{1}$$

$b_0 > 1$, $b_1 > 0$, $b_2 > 0$ are parameters resulting from a damper identification, v_p is a damper's plunger velocity and v_0 is a constant. Other property of the MR damper, which is required to be described, is related to saturation of a magnetic field. Saturation of magnetic field causes non-linear dependency of maximum damping force related to the applied control current I Fig. 1(d). This non-linear dependency can by approximated using the polynomial function in the form

$$A_{mr}(I) = \sum_{i=0}^{n} a_i I^i , \tag{2}$$

where A_{mr} is the maximum damping force, a_i are coefficients of polynomial resulting from approximation, n is a degree of the polynomial and I is the applied control current. Considering the expressions (1), (2) the resulting damping force is

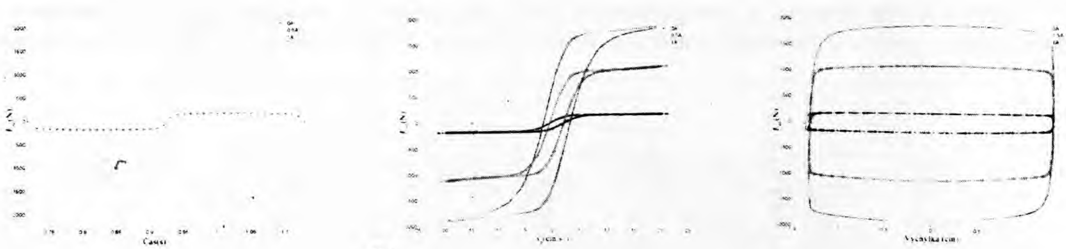
$$F_s(I, v_p) = A_{mr}(I) S_b(v_p) . \tag{3}$$

Hysteresis of the force-velocity characteristic should be considered as phase lag of damping force according to velocity. This phase lag can be achieved through a first order filter in the form

$$\dot{x} = -f(I)x + h_3 F_s , \tag{4}$$

where x is a state variable , h_3 is a constant and $f(I)$ is a function of current which establishes size of phase lag. After this consideration the final damping force is

$$F_{mr}(I, v_p) = f(I)x + h_4 F_s(I, v_p) . \tag{5}$$



b)

Fig. 2: Numerically evaluated characteristics of the MR damper model, a) force–time, b) force–velocity, c) force–displacement.

To ensure stability of the filter it is obvious that inequality $f(I)>0$ must be fulfilled. According to the real MR damper characteristic Fig.1, The results from the mentioned non-parametric model are entirely similar Fig.2. Since the state variable x is a continuous function, easy to differentiate, numerical efficiency is much better in compare with parametric models [5]. Since this model makes it possible to derive a gradient of the control quantity (current I) according to the controlled quantity, this model can be advantageously implemented into real time adaptive control algorithms.

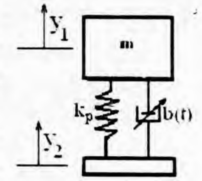


Fig. 3: 1DOF suspension model.

2. AN ADAPTIVE CONTROL ALGORITHM

In sense of simplicity, the simple one DOF model is considered Fig.3. The dynamics of this system can be described by the equation

$$m\,\ddot{y}_1 + F_{mr}(v_p, I) + k_p\, y_{12} = 0\,, \tag{5}$$

where $v_p = \dot{y}_1 - \dot{y}_2$ and $y_{12} = y_1 - y_2$. In this case the linear spring of the stiffness k_p and the mass of the weight m are constant. However, it is necessary to notice that the following AA isn't limited by those linear members with constant parameters [3]. Main aim of the AA is to provide suitable control of the system in such way that performance index (7), including the weighted single optimization criteria, is minimized.

$$J = p\,\ddot{y}_1^2(k) + q\,y_{12}^2(k)\,, \tag{7}$$

where p and q are weighting coefficients and k is a step. The control current I applied to the MR damper is adjusted in real time through the gradient search method by differentiating (7) with respect to the control current I

$$I(k+1) = I(k) + \mu\left[-\frac{\partial J}{\partial I(k)}\right], \tag{8}$$

$$\frac{\partial J}{\partial I(k)} = 2p\,\ddot{y}_1(k)\frac{\partial \ddot{y}_1(k)}{\partial I(k)} + 2q\,y_{12}(k)\frac{\partial y_{12}(k)}{\partial I(k)}\,, \tag{9}$$

$\dfrac{\partial J}{\partial I(k)}$ is the sensitivity of the performance index with respect to the control current and $\dfrac{\partial \ddot{y}_1(k)}{\partial I(k)}$, $\dfrac{\partial y_{12}(k)}{\partial I(k)}$ is the sensitivity of the controlled variables with respect to the control current I. If the mass weight and the spring stiffness are known in an each step k, than by differentiating (6) with respect to I

$$m\frac{\partial \ddot{y}_1}{\partial I} + \frac{\partial F_{mr}(v_p, I)}{\partial I} + k_p\frac{\partial y_{12}}{\partial I} = 0\,. \tag{10}$$

Employing the non-parametric model of the MR damper described by (3), (4), (5), after differentiating those equation with respect to the current I, the sensitivity of the damping force with respect to the current I is obtained

$$\frac{\partial F_s}{\partial I} = A_{mr}(I)\frac{\partial S_b(v_p)}{\partial I} + S_b(v_p)\frac{\partial A_{mr}(I)}{\partial I}, \quad \begin{cases} \dfrac{\partial \dot{x}}{\partial I} = -\dfrac{\partial f(I)}{\partial I}x - f(I)\dfrac{\partial x}{\partial I} + \dfrac{\partial (h_3 F_s)}{\partial I} \\ \dfrac{\partial F_{mr}}{\partial I} = \dfrac{\partial f(I)}{\partial I}x + f(I)\dfrac{\partial x}{\partial I} + \dfrac{\partial (h_4 F_s)}{\partial I} \end{cases} \tag{11-12}$$

After substituting $\dfrac{\partial F_{mr}(v_p, I)}{\partial I}$ from (12) in the equation (10), the sensitivity $\dfrac{\partial \ddot{y}_1(k)}{\partial I(k)}$, $\dfrac{\partial y_{12}(k)}{\partial I(k)}$ is obtained. After substituting the evaluated sensitivity into the equation (9) the performance index sensitivity $\dfrac{\partial J}{\partial I(k)}$ with respect to the control current is obtained. Finally, substituting the performance index sensitivity in the equation (8) leads to the needed control current setting in a step $k+1$.

2.1 System stability

A controlled suspension, which is using a MR damper as an actuator, is able to dissipate energy, but not to generate any. In this sense the mentioned system can't by unstable.

Concerning AA stability, for an every relevant type of excitation exist just the one global minima of the performance index whereupon only the one optimal control exists. A proof of this proposition is referred to [3].

2.2 An extended formulation of the performance index

As mentioned earlier, the AA converges into the optimal control in real time Fig.4. This property allows to defines the performance index in following way

$$J = p\,\ddot{y}_1^2(k) + q(y_{12}^2)\,y_{12}^2(k), \tag{13}$$

a)

b)

Fig. 4: The 1DOF model response to the sweepsine, a) acceleration, b) control current setting using AA.

where $q(y_{12}^2)$ can be considered to be the penalization function. It appears from this that single parts of the performance index can be weighted instantaneously in real time accordingly to actual driving conditions in contrast to estimated approach based on root mean square (RMS) values as the skyhook and the optimal control deal with. Also the mass acceleration $\ddot{y}_1$ can be weighted with respect to excitation frequency using a frequency filter in order to make provision of human body sensitivity to individual frequencies.

4. CONCLUSION

Since AA converges to the optimal control independently from excitation type and car speed, is able to deal with non-linear system, is able to adapt to changes of system parameters in cooperation with an on-line identification and also is able to change weighting of the individual parts of the performance index, that this strategy appears to be quite robust beside commonly used strategies. In order to design an open loop control system it might be advantageous to make so called polynomial model of the MR damper [1]. In this model whole behavior of the MR damper has been described by two polynomials. Since these two polynomials describe the mentioned hysteresis in the characteristic, differential equation isn't employed. Consequently the expression $I = I(F_{mr}, v_p)$ can be explicitly derived.

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