Vehicle System Dynamics

ISSN: 0042-3114 (Print) 1744-5159 (Online) Journal homepage: http://www.tandfonline.com/loi/nvsd20

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To cite this article: Tomas Zubek , Milan Apetaur & Jan Skliba (2004) Transient Properties of a Magnetorheological Damper – Discussion to the Paper by Seung-Bok et al.: “H_∞ Control Performance of a Full-Vehicle Suspension Featuring Magnetorheological Dampers”, Vehicle System Dynamics, 41:4, 323-330, DOI: 10.1080/00423110412331315187

To link to this article: http://dx.doi.org/10.1080/00423110412331315187

Published online: 02 Feb 2010.

Vehicle System Dynamics
2004, Vol. 41, No. 4, pp. 323–330

Transient Properties of a Magnetorheological Damper – Discussion to the Paper by Seung-Bok et al.: ''H_∞ Control Performance of a Full-Vehicle Suspension Featuring Magnetorheological Dampers''

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SUMMARY

A discussion in terms of dynamic properties of a magnetorheological damper is lead to those in the denoted paper. Results of experimental evaluation of the properties of a commercially available magnetorheological damper are presented. A possibility of use of the damper in practical applications is also considered.

1. INTRODUCTION

Use of magnetorheological dampers (MRD) has recently attracted much attention in different applications. The reason of this popularity is a simple possibility of the damping force control by the input current regulation with relatively small power consumption combined with very small temperature sensitivity. The paper [1], to which this discussion is lead, deals with the use of MRDs for semi-active vehicle suspension.

All papers dealing with MRDs known to us describe the behavior of the damper by a simple model (see e.g., [1, 2]), where the damping force F (N) is dependent on the relative velocity v (m/s) of the two damper's ends and on the current I (A) applied to the coil generating magnetic field. It is expressed as

$$F = F_{visc} + F_0 \mathrm{sgn} v + F_d \mathrm{sgn} v \qquad (1)$$

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10.1080/00423110412331315187$22.00

where

- F_{visc} is the viscous component due to the fluid transfer inside the damper, usually described by $F_{visc} = bv$ with b (N s/m) being the viscous damping coefficient,
- F_0 is value of the passive "friction type" component when $I = 0$,
- F_d is value of the active "friction type" component added to F_0 when $I \neq 0$.

The active component is generally described as dependent on the input current only $F_d = F_d(I)$, that is, independent on time. It can be verified that Equation (1) represents very satisfactorily the MRD's properties for I = const or for relatively slow changes of the input current I. We denominate these control states as "quasistatic damper control states". However, if faster damping force changes are demanded, that is, if faster input changes occur, transient properties of the control chain and MRD must be taken into account. We denominate that as "dynamic damper control states". The value of the active component F_d must then be considered as time dependent $F_d(t)$.

This contribution shows briefly results of measurements of dynamic properties of a commercially available MRD intended for the use in suspended vehicle seats. Its maximum stroke is 58 mm, maximum input current is $I_{max} = 0.7$ A which excites maximum damping force $|F_{max}| \cong 2.1$ kN. Its general layout corresponds to the common praxis and its principle is similar to the MRD shown in [1] (Fig. 1). Our measurements were undertaken for the sake of general recognition of MRD's suitability for the use in damping control. No theoretical analysis of the MRD's dynamic properties was undertaken.

2. PROPERTIES OF THE MEASURED MRD IN QUASISTATIC CONTROL STATES

Quasistatic properties of the measured MRD were very close to those shown in [1] (Fig. 2), thus they will be discussed only very briefly.

Our experiments were done with harmonic stroking with frequencies up to 5 Hz and amplitudes up to 20 mm (maximum relative velocities of the MRD's ends up to 0.63 m/s).

It can be said in general, that

- nearly symmetrical MRD's behavior for $v > 0$ and $v < 0$ was observed,
- the influence of the MRD's ends relative velocity on the total damping force F is small, viscous damping coefficient in Equation (1) was $b \cong 0.44$ kN s/m,
- dynamic component $F_d(I, t)$ in Equation (1) can be approximated as $F_d(I) \cong$ const. I,
- magnitude of the passive component F_0 is relatively high with $F_0 \cong 0.12$ kN.

Let be mentioned as an example that the total damping force at maximum relative velocity $v = 0.63$ m/s and maximum input current $I = 0.7$ A is $F = 2.1$ kN. Out of that

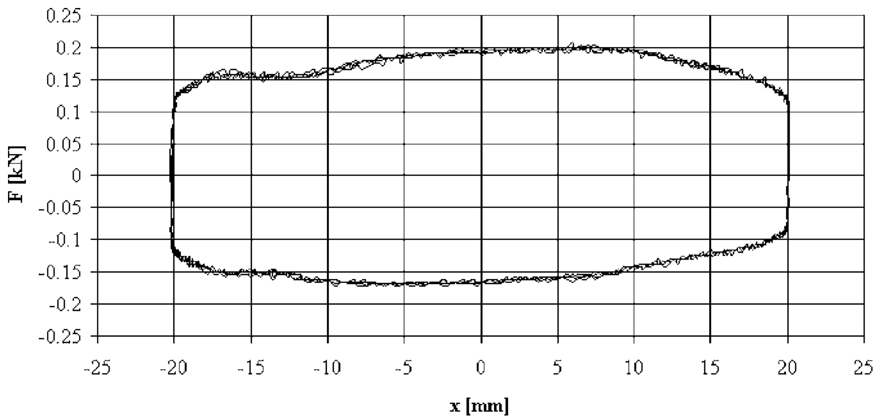


Fig. 1. Damping force-stroke diagram of the measured damper at harmonic stroking with $f = 1.25$ Hz with $I = 0$.

the portion of the viscous component F_{visc} amounts to approx. 14%, the portion of the passive "friction" F_0 to approximately 6% and the rest approximately 80% is due to the active F_d component governed by current I.

A typical force-stroke diagram for harmonic stroking with $f = 1.25$ Hz and $I = 0$ is shown on Figure 1. Diagrams with similar nearly rectangular force-stroke dependence, indicating prevailing "friction type" damping force development, are obtained for other measurement conditions as well.

3. PROPERTIES OF THE MEASURED MRD IN DYNAMIC CONTROL STATES

The damping force of a MRD must be controlled during every work cycle by the input current I changes as the viscous component F_{visc} is relatively unimportant and cannot be influenced by classical means known from hydraulic dampers. That means, for example, that if certain $F(x, v)$ damper characteristic is demanded, the input current must be controlled at every time instant. Because of that, the dynamic properties of this control and the time delays in the control chain are extremely important.

The general MRD's control chain consists of two basic sections.

The control input processes (e.g., acceleration processes in the suspension) are measured and evaluated by an ECU in the first section. The ECU's output is formed by control voltage $U(t)$ for the MRD control. This section of the actual control chain is not discussed here. It must be however emphasized that the time-delay between the actual measurement time of the input processes and the control voltage U reaction lies in the region of one to several milliseconds. It depends on the complexity of the used control algorithm and on the sophistication of the ECU's hardware and cannot be neglected in the control of dynamic systems where processes with higher frequencies are occurring.

The actual MRD control section is formed by voltage-current transducer which produces current driving the coil and thus the magnetic field applied to the MR fluid. The current applied is based on a value of input U(t). Each of these components has its own dynamic delay concerning its reaction to corresponding input.

Our measurements were directed towards the target of getting a general idea about the dynamic behavior of the particular MRD. They were carried out with triangular MRD's stroking. The stroking velocity during the motion interval between both inflexion points was nearly constant. Abrupt, nearly stepwise, changes of the control voltage U(t) between two preset values were executed in that interval. The damping force components F_{visc} and F_0 [Equation (1)] were therefore constant during the whole procedure and only the active component F_d was changing between two steady state values. This transient process was recorded.

Typical courses of the measured variables during such procedure are shown on Figure 2. The control voltage U(t) change instant is indicated by the vertical line. The

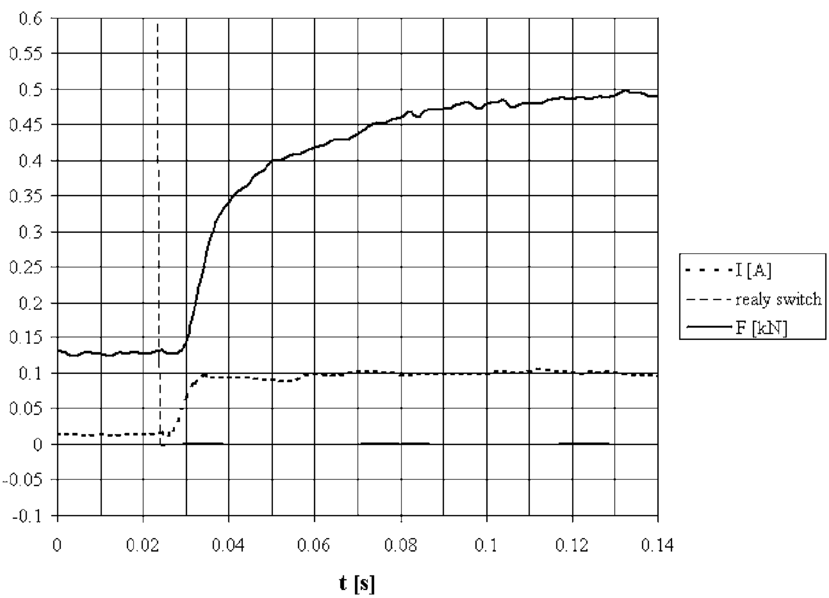


Fig. 2. Typical time dependence of input current I and damping force F for abrupt change of the control voltage U at steady stroking velocity (v = 33 mm/s).

current change begins with a delay of approximately 1 to 2 ms and reaches its new steady state value at approximately 8 to 12 ms. Its general course can be roughly approximated by a first order relation. The damping force growth begins with approximately 1 to 2 ms delay after the beginning of the current growth and has again a first order character.

Sophisticated mathematical description of this process would be evidently possible. We consider however that a simpler description of the active force $F_d(t)$ development is sufficient in most practical control problems. We are taking the control voltage U(t) from the ECU as input and we model the F_d transient properties by a first order system

$$\frac{d(F_d(t))}{dt} + \frac{1}{T} F_d(t) = \frac{1}{T} g(U(t)) \tag{2}$$

where T (s) is the time constant of the system and $F_{d,s} = g(U)$ is the steady state response.

The time constant T of the measured damper was evaluated for different measurements considering both relative velocity senses, increase or decrease of the damping force and different relative velocities of the MRD's ends. The time constant was in all cases determined from the time necessary to reach 90% of the steady state F_d change.

Time constant T is dependent on the relative velocity v of both MRD's ends as shown on Figure 3. Scatter of the results, especially at low relative velocities, is large. Reasons for that are unknown to us and may be caused by relative simplicity of the

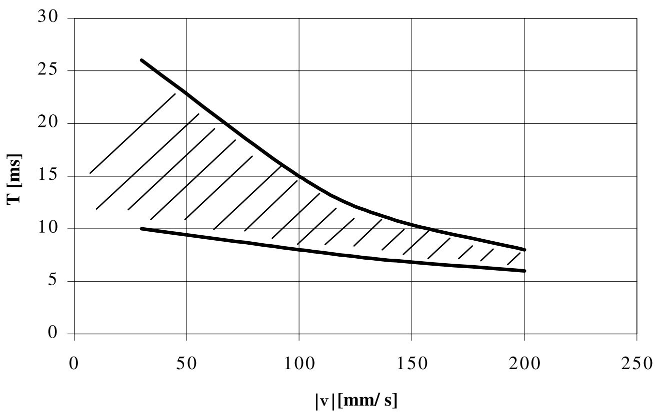


Fig. 3. Measured upper and lower limits of the time constant T in dependence on the MRD's ends relative velocity v.

used hardware for obtaining minimum cost of the product. Furthermore, the results for different measurement conditions were often not fully consistent. Maximum and minimum time constants observed are therefore indicated only. Diminishment of the damping force, that is, demagnetization of the fluid, however, seems to be faster than its rise especially at low relative velocities.

It is somewhat surprising, that time constant T drops with increasing relative velocity v to reach minimum value between 6 to 10 ms for $|v| > .15$ m/s.

As already mentioned, the measured damper is used in suspended driver seats in buses and trucks. It is working with average relative speeds v lower than 100 mm/s.

Paper [1] indicates on Figure 3 the "dynamic response characteristics" of the MRD used in the discussed control problem. Though it is not explained in the paper how this characteristics was obtained, a first order behavior with corner frequency about 10 Hz is evident. This corner frequency corresponds to time constant $T = 15.9$ ms which is in accordance with our measurements at lower relative velocities.

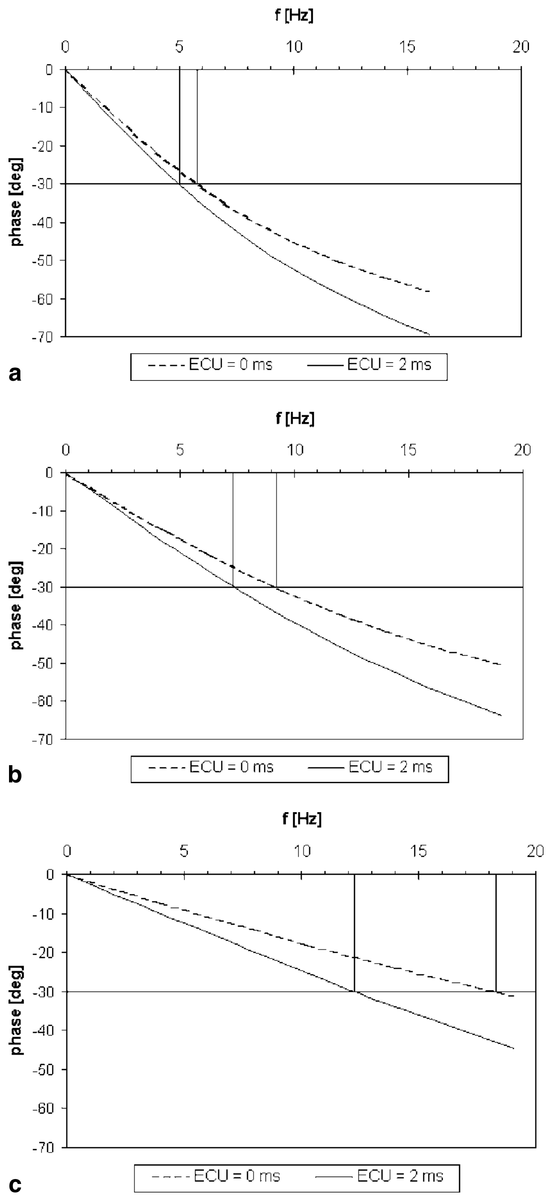
4. CONCLUSIONS

The value of the damping force of a MRD depends primarily on the control voltage applied. The frequency range of a MRD application in vibration problems is therefore limited by its speed of reaction to this control input, which can be expressed by its time constant T.

If harmonic stroking of the damper is assumed and therefore damping force change with the same basic period is demanded, the most important parameter indicating the suitability of the damper for its use at prescribed frequency is the phase shift between input voltage and damping force. If we further reduce the problem to harmonic voltage input $U(t) = U_0 \sin 2\pi ft$ and consider linear behavior of the damper ($T = \text{const}$, $g(U) = \text{const. } U$), this phase shift Φ (deg) can be easily determined using Equation (2).

Computed phase shifts at different input control frequencies for time constants $T = 15.9$ ms, $T = 10$ ms and $T = 5$ ms are shown on Figure 4a, b, c in the cases that no time-delay exists in the ECU and with time delay 2 ms in the ECU.

Let us assume that maximum phase delay for achieving an effective control must be for the desired frequency range in its absolute value smaller than 30°. The frequency ranges fulfilling these conditions for the time constants considered are shown. Let us further assume that we must take into consideration the 2 ms ECU delay. It can be deduced that for the seat damper control, where control frequencies lie below 5 Hz, time constant of the MRD smaller than approximately 16 ms is necessary. Full control of MRD in the frequency range 12 to 15 Hz, that is, in the frequency range



a

b

c

Fig. 4. Theoretical phase shifts between harmonic change of control voltage U and harmonic change of the value F_d of active “friction” component without (ECU 0 ms) and with (ECU 2 ms) consideration of the time delay in the ECU: (a) T = 15.9 ms, (b) T = 10 ms, (c) T = 5 ms.

of wheel vibrations in passenger cars, requires MRD's time constant smaller than 4 ms. This is not achievable with the MRD tested by us and it seems to be far below the possibilities of the MRD considered in [1].

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