Journal of Achievements in Materials and Manufacturing Engineering

VOLUME 21
ISSUE 1
March
2007

Prototype magnetorheological fluid damper for active vibration control system

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Received 07.11.2006; accepted in revised form 15.11.2006

Analysis and modelling

ABSTRACT

Purpose: The paper presents a concept of a system for isolation from external vibration sources with use of a magnetorheological (MR) dampers.
Design/methodology/approach: Results of experimental studies of a prototype magnetorheological damper at various magnitudes of control current and the manner of modelling electromagnetic phenomena occurring in the damper are presented in this paper. The effect of magnetic field on magnetorheological fluid is modelled by the finite element method. The mathematical model of the system as well as the damper model are outlined along with the relevant control facilities. Numerical simulations were carried out for an exemplary excitation.
Findings: The elaborated damper and applied control algorithms substantially influences the values for velocities and accelerations. Incorporation of a controllable damper into the stabilization system significantly decreases displacements of the mass to be stabilized being the results of shocks and bumps caused by excitations w(t) as compared to similar displacement of the same mass when only a passive damper was used.
Research limitations/implications: For the future research it is necessary to improve characteristics of elaborated damper in order to improve its efficiency.
Practical implications: Many mechanical systems should separate from sources of vibrations. The active or semiactive vibration control systems offer a number of advantages as compared with passive systems so that better efficiency of vibration damping is assured.
Originality/value: The paper presents new concept of vibration damper with magnetorheological fluids and way of its application in industrial practice.
Keywords: Applied mechanics; Active vibration control; Magnetorheological fluid; Fluid damper; Machine dynamics

1. Introduction

Magnetic fluids, discovered by J. Rabinow in the 1940's, are used in many fields of engineering. Since the 1990's, when the manufacture of magnetic fluids was mastered, many devices using these fluids have been developed. New devices and new suggestions of their structural design keep on emerging. The features of magnetic fluids enable these devices to change properties of mechanical systems (stiffness, damping force) by controlling electrical quantities (supply voltage, current intensity). Due to their unique properties, these fluids are classified among non-Newtonian fluids.

Many mechanical systems should separate from sources of vibrations. In parallel to commonly used systems with passive damping appliances more advanced solutions that contain active or semi-active components present increasing popularity [8]. They offer a number of advantages as compared with passive systems so that better efficiency of vibration damping is assured.

2. Concept of a prototype MR damper

Magnetorheological dampers have recently become an object of intensive studies, both due to their interesting physical features, as well as due to their potential applicability to control damping in mechanical systems.

The designed linear magnetorheological damper is an actuator that allows controlling performance characteristics. The resisting force depends on piston speed and on strength of magnetic field in the working gap.

The structure of the damper has been developed on the basis of the analysis of strength properties and of dynamic phenomena for the concept adopted [1]. Material properties have been defined for the various components of the prototype structure (Fig. 1).

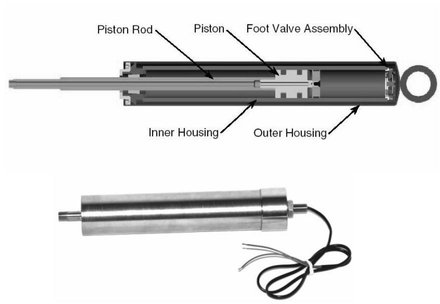


Fig. 1. Concept of a prototype twin tube magnetorheological damper

A number of assumptions were made at the design stage. The following criteria were taken into consideration:

- loads that must be carried by the system,
- geometric dimensions, restricted by the working environment,
- damping value,
- reliability of the system,
- range of temperatures at which proper functioning of the system may be initiated.

The structure of the damper is based on the double-barrelled model. Such design permits control of damping characteristics by means of magnetic field generated by an electromagnetic coil and by means of an equalizing suction valve.

2.1. Analysis of electromagnetic phenomena

Numerical calculations were done using Ansys computer software. The damper was modelled with quadrilateral 8-node finite elements with four degrees of freedom in the "Plane 53" type node applied for modelling magnetic field problems with axial symmetry (Fig. 2). The damper model was developed in a parametric form. The following simplifications were adopted in the modelling process:

1) magnetic flux caused by flow of current through the coil is so low that no saturation occurs in the damper circuit,
2) magnetic flux leakage is negligible.

The first assumption enabled calculations of a linear model (it was assumed that the relative magnetic permeability of the piston casing material is constant) and of a nonlinear model (material nonlinearity taken into account), whereas the second assumption enabled numerical analysis of the model.

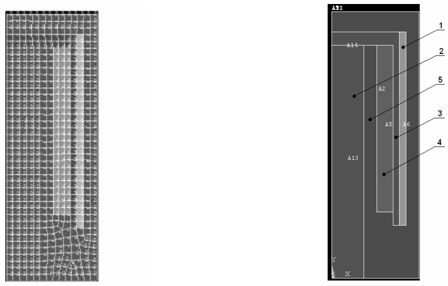
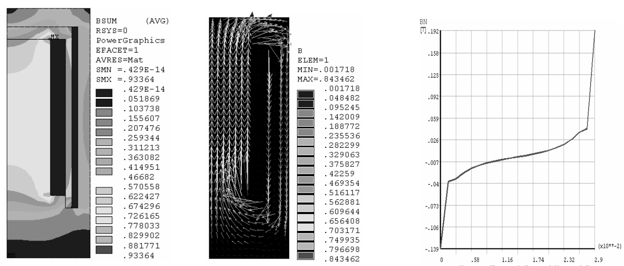


Fig. 2. Numerical model of MR damper

1 material representing gap filled with MR fluid,
2 material forming the core of electromagnetic coil,
3 material of coil screen,
4 material forming the winding of electromagnetic transducer,
5 material of the piston casing of the magnetorheological damper.

The presented examples of results (Fig. 3) have been obtained from numerical calculations made for a nonlinear model, which takes into account the real magnetisation characteristic of S295 grade steel. Distribution of magnetic field lines, values of magnetic flux density and corresponding magnetic field strengths have been obtained in the form of maps and in the form of a vector field, which enable determination of dynamic damping forces that occur in the model of magnetorheological fluid [6].



a) b) c)

Fig. 3. Examples of results of numerical calculation: a) magnetic field strengths form of maps, b) magnetic field strengths form of a vector field, c) values of magnetic flux vs. distance of gap filled

Magnetic flux density increases linearly from the start of gap length to the point where coil winding starts. It then decreases nonlinearly down to zero in the middle of the total gap length.

2.2. Experimental tests

Experimental studies included a number of tests made with an MTS dynamic testing apparatus (Fig. 4).

Tests of the prototype MR damper were carried out at various magnitudes of control current passing through the coil circuit: from 0A to 5.5A (Fig. 5-6). For each set control current, the input was in the form of a sine function, and the speed varied between 0.01 m/s to 0.2 m/s.

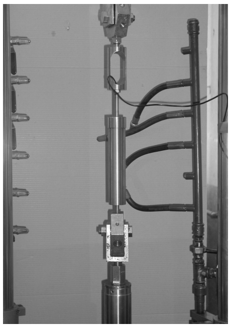


Fig. 4. Dynamic testing apparatus

Examples of characteristics for two values of control current are shown below:

a) current equal to 0 [A],

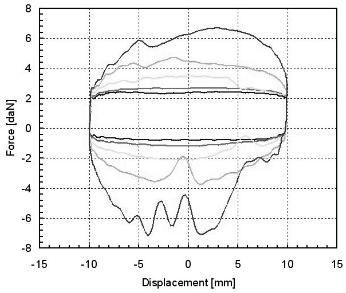


Fig. 5. Examples of results of tests, force vs. displacement

b) current equal to 5.5 [A].

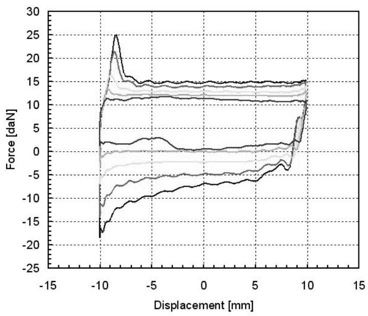


Fig. 6. Examples of results of tests, force vs. displacement

Changes seen in the extremely compressed position (-10 mm; 5.5 A) result from the interaction of magnetic field with the MR fluid.

3. Numerical simulations

The object represents a solid body that has one degree of freedom and its mass *m* incorporates weights of a real object. The adopted model of the vibrating system is a parallel combination of an elastic member with the spring rate *k* and a suppressing member which is a magnetorheological damper with a variable damping force. The bearing structure that generates excitations *w(t)* of kinematic character pursuing to enforce displacements (Fig. 7).

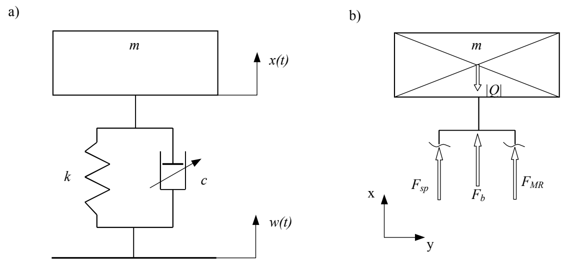


Fig. 7. Scheme of the model

To determine the force that is exerted by the damper the model of a magnetorheological damper was applied as proposed by Spencer for the first time [3, 7]. The reason for selecting that model was easy availability of papers that provide necessary values for essential parameters of the model. Thus behavioral simulation of a specific damper becomes feasible. The Spencer's model of a MR damper is presented in Fig. 8 [4, 5].

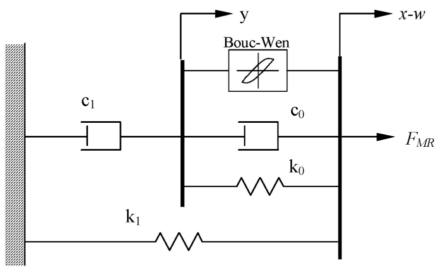


Fig. 8. The Spencer's model of a MR damper

The reaction force that is exerted by the damper can be calculated from the relation (1).

$$F_{MR} = c_1 \dot{y} + k_1 (x - w - x_0) \tag{1}$$

where respective

$$\dot{y} = \frac{1}{(c_0 + c_1)} \left[\alpha z + c_0 (\dot{x} - \dot{w}) + k_0 (x - w - y) \right] \tag{2}$$

$$\dot{z} = -\gamma |\dot{x} - \dot{w} - \dot{y}| z |z|^{n-1} - \beta (\dot{x} - \dot{w} - \dot{y}) |z|^n + A(\dot{x} - \dot{w} - \dot{y}) \tag{3}$$

Parameters α, c_1, and c_0 depend on voltage that is applied to the damper

$$\alpha = \alpha_a + \alpha_b u \, , \, c_1 = c_{1a} + c_{1b} u \, , \, c_0 = c_{0a} + c_{0b} u \tag{4}$$

The delay interval that expires until rheological fluid reaches the required viscosity due to voltage variation is defined by the equation:

$$\dot{u} = -\eta (u - v) \tag{5}$$

Parameter values for the adopted model were taken from experimental tests (Fig. 5-6).

3.1. Development of a dynamic model with control algorithm

Due to simplicity of implementation as well as presumption that stabilizing of the position is of primary importance, the developed system involves control algorithms that bases on Skyhook-type control, namely Continuous Skyhook [2]. The characteristic feature of the control algorithms consists in the rule that the damper supply voltage is proportional to the velocity of the stabilized mass $\dot{x}$. Mathematical notation of that algorithm is expressed by the equation below:

$$\dot{x}(\dot{x}-\dot{w})>0\,;u=a\dot{x}\,;\dot{x}(\dot{x}-\dot{w})<0\,;u=0 \qquad (6)$$

where: u - supply voltage as applied to the damper solenoids, a - adjustable gain factor.

The developed dynamic model along with its control is presented in the form of a flow chart (block diagram) in Fig. 9.

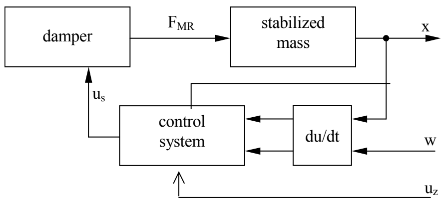


Fig. 9. Flow chart of the developed model. F_{MR} – force produced by the magnetorheological damper, u_z – supply voltage, u_s – control voltage

3.2. Numerical simulations

For numerical calculation the following parameters were adopted: k = 7500 N/m, m = 150 kg, a = 85.

Examples of characteristics displacement stabilized mass vs. time are shown in Fig. 10-11.

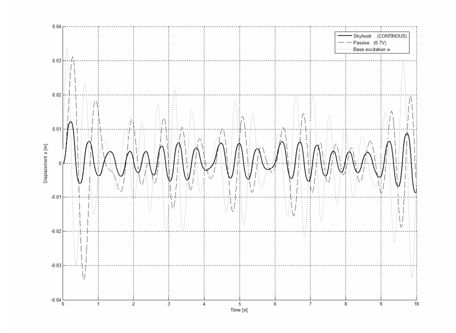


Fig. 10. Random Displacement Test

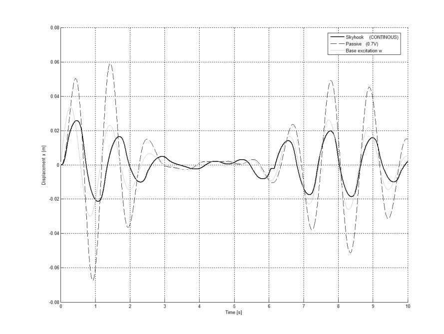


Fig. 11. Random Displacement Test

4. Conclusions

The elaborated damper and applied control algorithms substantially influences the values for velocities and accelerations as obtained from simulations. As timings presented in Fig. 10, 11 confirm, incorporation of a controllable damper into the stabilization system significantly decreases displacements of the mass to be stabilized being the results of shocks and bumps caused by excitations *w(t)* as compared to similar displacement of the same mass when only a passive damper was used. For the future research it is necessary to improve characteristics of elaborated damper in order to improve its efficiency.

Acknowledgements

The work was done with the supply of the research program 4 T07D 005 29.

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