CHARACTERISTICS OF AN ELECTRORHEOLOGICAL DAMPER IN A VIBRATION INSULATOR

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A theoretical analysis and experimental study have been performed on the vibrations of a single-mass system having a viscous-friction damper involving an electrorheological suspension whose viscosity and elasticity are adjusted by means of an external electric field.

Viscous-friction dampers are effective in insulating precision apparatus from low-power low-frequency vibrations (vibrational background), such as electron and optical microscopes, interferometers, microanalytical balances, and weighing systems. The damping is widely adjustable [1], while a single device may damp vibrations in several directions simultaneously, and the insulation is of active type. Particular interest attaches to dampers whose characteristics can be remote-controlled over wide ranges such as by electric fields. The working medium is an electrorheological (ER) suspension [2, 3].

Here we consider a system based on an electrorheological damper (ERD). The object is a body of mass M coupled to the base via an elastic suspension having identical compression and shear moduli together with parallel dampers with identical elastic characteristics and damping parameters (Fig. 1). The base vibrates in accordance with $X_1 = X_0 \sin\omega t$, and the relative displacement of the object is defined by

$$M\ddot{x} + F(X, \dot{X}) + kX = Mg\sin\omega t, \quad g = \omega^2 X_0. \tag{1}$$

Here k is the spring rigidity, g the amplitude of the base acceleration, and F the damper resistance force.

Similarly, we have an equation for a damper as shown in Fig. 1b, where the right-hand side takes the form $kX_0 \sin\omega t$.

It has been shown [4] that the solid particles form bridges in an ERD at small vibration amplitudes, which join the oppositely charged electrodes. The adhesion forces and the bridge strength are dependent on the field strength. The shear deformation of the bridge framework linked to the electrodes produces an elastic component F_e in the resistance when the damper piston oscillates, this being coupled to the object. The pressure rise ahead of the piston produces a viscous component F_v of the resistance force. The elastic component is proportional to the number of bridges N per unit surface and the tension Φ in them, as well as the side surface area S_s of the piston and the shear X/h: $F_e = N\Phi S_s(X/h)$. The viscous component is proportional to the volume of liquid displaced by the piston and to the pressure gradient in the gap: $F_v = v(\partial P/\partial X)$.

The pressure gradient is determined by the mean flow speed V in the gap, which is related to the piston speed by the mass balance condition for the liquid: $V = (S/S_g)(\partial X/\partial t)$, where S and S_g are the area of the end surface of the piston and the cross section of the gap. A viscoelastic electrorheological suspension gives the resistance force

$$F = aX + b\frac{dX}{dt}. \tag{2}$$

The values of a and b are related in a complicated fashion to E and to the piston displacement and speed. As E increases, the number of particles in the bridges increases, as do the adhesion forces, and a increases. In strong fields, the viscous component is determined only

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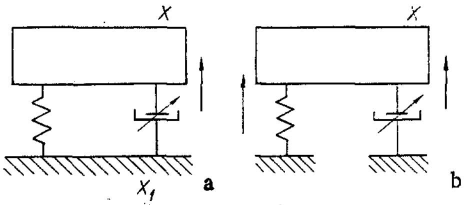

Fig. 1

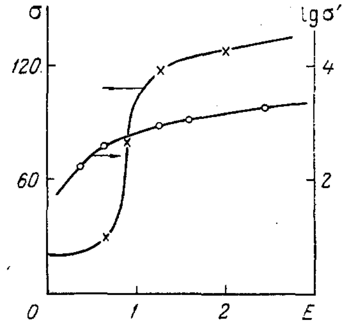

Fig. 2

Fig. 1. The apparatus: parallel connection of damper (a) and serial connection (b).

Fig. 2. Effects of electric field strength on the shear force σ and elastic modulus σ' for a 40% suspension of diatomite in transformer oil: $\dot{\gamma} = 10\ \mathrm{sec}^{-1}$, $\omega = 0.628\ \mathrm{sec}^{-1}$, E in kV/mm, and σ in Pa.

by the hydrodynamic resistance produced by the motion of the dispersion medium through the framework of solid particles. The quantity σ = NΦ corresponds to the detachment force in the electrorheological suspension. This has been measured for diatomite in transformer oil with a modified cell in a coaxial cylindrical Reotest-2 viscometer (Fig. 2).

The elastic parameters of the ER suspension have been measured as functions of field with a mechanical spectrometer whose working unit consisted of coaxial cylinders moving with forced sinusoidal oscillations (Fig. 2). As the amplitude increases, the bridge destruction time gradually shortens, even in a strong field, and it ultimately becomes much less than the vibration period. Then the contribution from the elastic component is unimportant, while the viscous component is determined by the pressure-flow characteristic for the stationary flow of the ER suspension in the cylindrical channel [5]. The flow-pressure relationship is nonlinear, being governed by the relationship between the field and the pressure. As E increases, the pressure corresponding to transition to the linear region increases.

This behavior enables us to analyze the frequency response qualitatively with allowance for the insulation provided by the damper. We derive the solution to (1) by the harmonic-balance method in the form $X = A\sin(\omega t + \phi)$. The amplitude and phase of the vibrations are determined from the condition for the orthogonality of the discrepancy between the functions $\sin(\omega t + \phi)$, $\cos(\omega t + \phi)$ [6]. We get

$$(\omega_0^2 - \omega^2) A + \frac{1}{M} f_1(A,\ \omega) = g\cos\varphi,\quad \frac{1}{M} f_2(A,\ \omega) = -g\sin\varphi. \tag{3}$$

Here $\omega_0^2 = k/M$ is the frequency of the natural oscillations,

$$f_1 = \frac{1}{\pi}\int_0^{2\pi} F(A\sin Z,\ \omega A\cos Z)\sin Z dZ = \frac{A}{\pi}\int_0^{2\pi} a\sin^2 Z dZ + \frac{\omega A}{\pi}\int_0^{2\pi} b\sin Z\cos Z dZ,$$

$$f_2 = \frac{1}{\pi}\int_0^{2\pi} F(A\sin Z,\ \omega A\cos Z)\cos Z dZ = \frac{A}{\pi}\int_0^{2\pi} a\sin Z\cos Z dZ + \frac{\omega A}{\pi}\int_0^{2\pi} b\cos^2 Z dZ.$$

Instead of g, one should take $\omega_0^2 X_0$ for the type of damper shown in Fig. 1b.

We now consider the mode of vibration in various states. In relatively strong fields, one can neglect the dependence of a and b on the piston displacement and speed. Then $f_1 = aA$ and $f_2 = Ab$. The theory of linear oscillations gives the frequency response of the damper, the resonant frequency, and the amplitude:

$$A = g \Big/ \left[\left(\omega_0^2 + \frac{a}{M} - \omega^2\right)^2 + \frac{\omega^2 b^2}{M}\right]^{1/2},$$

$$\omega_r = \left(\omega_0^2 + \frac{a}{M} - \frac{b^2}{M^2}\right)^{1/2}, \quad A_r = g \Big/ \frac{b}{M}\left(\omega_0^2 + \frac{a}{M}\right)^{1/2}. \tag{4}$$

In relatively weak fields, the contribution from the elastic component is small ($a \approx 0$), and the resistance coefficient is dependent only on the velocity: $b = b(|\omega A \cos Z|)$. The frequency response is then described by

$$(\omega_0^2 - \omega^2)^2 A^2 + \frac{1}{M^2} f_2^2(\omega A) = g^2, \; f_2 \simeq \frac{\omega A}{\pi} \int_0^{2\pi} b \cos^2 Z dZ.$$

If the damping is weak, the resistance force is much less than the inertial one, $f_2 \ll M\omega_0^2$, and the resonant frequency is unaltered, while the resonant amplitude is given by

$$f_2(\omega_0 A_r) \approx Mg, \quad \omega_r = \omega_0. \tag{5}$$

With moderate E, the bridge existence times are comparable with the oscillation period, and one cannot neglect the elastic component. One can assume as an approximation that the effective elasticity is dependent only on the displacement $\underline{a} \cong \underline{a}(|A \sin Z|)$, while the effective viscosity is dependent on the velocity $b = b(|\omega A \cos Z|)$. Then

$$f_1(A) \approx \frac{A}{\pi} \int_0^{2\pi} a \sin^2 Z dZ, \quad f_2(\omega A) = \frac{\omega A}{\pi} \int_0^{2\pi} b \cos^2 Z dZ.$$

A weakly damped system has the resonant amplitude and frequency defined by

$$f_2(\omega_0 A_r) \approx Mg, \; \omega_r^2 \approx \omega_0^2 + \frac{f_1(A_r)}{MA_r}. \tag{6}$$

Then (4)-(6) enable one to examine the effects of the field on the frequency and amplitude. As E increases, the resonant amplitude at first decreases, while the resonant frequency is unaltered, because the effective viscosity is increased. The bridge lifetime is short and elastic framework deformation does not make itself felt. Then (5) is applicable. At moderate fields, the elasticity begins to have an effect. The effective viscosity gradually falls, since some of the bridges do not break during the oscillations. Here (6) applies. In strong fields, the viscous losses amount to the hydrodynamic resistance, i.e., cease to be dependent on the field. The elastic component alters because of the slow increase in the adhesion forces. The dependence of the adhesion force on the field has been examined in more detail in [7, 8]. Formula (4) applies for strong fields, where the resonant frequency increases slowly but the resonant amplitude gradually decreases. The field dependence of these is very weak. As the driving-force amplitude increases, one finds an increase in the E at which there is a shift between the above states, since the framework distortion increases with the piston vibration amplitude.

This model has been tested with a special system, which included an object to be isolated in the form of a rectangular plate of mass 3 kg on four passive vibrational isolators, namely springs of rigidity 3.5 kgf/cm. These were mounted on a base with harmonic vibration provided by the control unit in a VÉDS-10A electrodynamic tester giving an acceleration recorded by an IS-318 sensor. The damper body was mounted on the base and consisted of two coaxial cylinders with outside diameters of 55 and 30 mm filled with 40% ER suspension of diatomite in transformer oil. The piston consisted also of two cylinders of diameter 45 and 15 mm and of length 70 mm mounted on a rod rigidly coupled to the object. The gap between the cylinders for the body and piston was 5 mm, where the minimum distance from the lower edge of the piston to the bottom of the body was twice this. The damper body was fitted in the upper part with a compensating cavity to prevent excess suspension displaced by the piston from spilling. The cylinders (body and piston electrodes) were connected to a high-voltage source via an M 1109 microammeter to check for breakdown and determine the electrical losses in the ERD. The motion of the object in positioning mode can be controlled with a standard high-voltage stabilizer type VS-23, which produces a discrete signal in the range 0.5-10 kV. Tracking isolation mode can be based on the signal from a DS-13 sensor which records the object vibration amplitude, which operated with an automatically controlled voltage source producing a signal in accordance with the amplification law chosen: linear, logarithmic, or semilogarithmic. The amplitude of the input mechanical signal was recorded by a DS-13 sensor attached to the base. The vibration

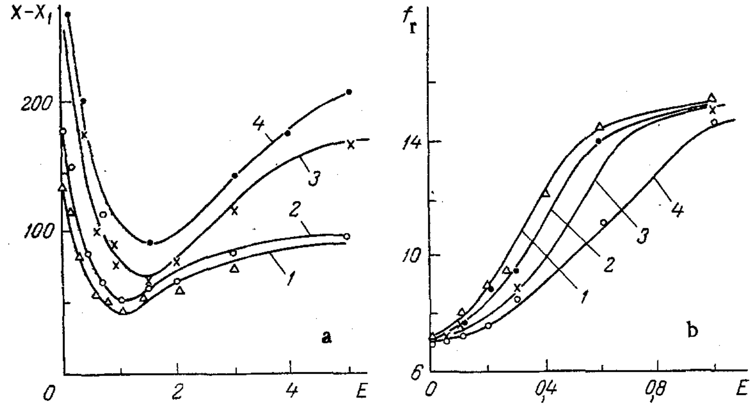

Fig. 3. Effects of electric field on relative amplitude (a) and frequency (b) of the system containing the ER damper at resonance for various vibrational accelerations g in m/sec^2: 1) 0.3; 2) 1; 3) 2; 4) 4; $X - X_1$ in μm, f_r in sec^{-1} (a: 0.2 E, kV/mm).

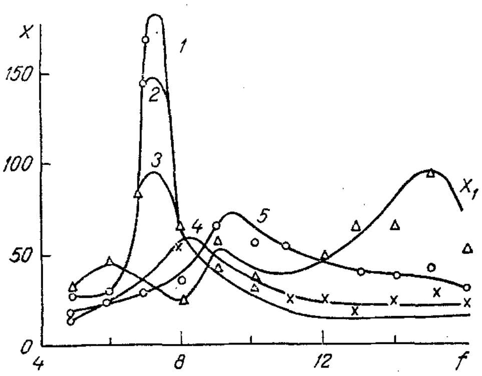

Fig. 4. Frequency-response curves for the system containing an ER damper with various control inputs E in kV/mm: 1) 0; 2) 0.04; 3) 0.08; 4) 0.12; 5) 0.3; X in μm.

frequency was recorded with a 43-35 frequency meter, while the waveform was recorded by a low-frequency sensor of seismic type, K001, working with an N-0044.1 light-beam oscillograph. The experiments were performed over the range 4-40 Hz, which covered the natural frequency range of the object, with accelerations of 0.3-4 g and with variation in the static parameters (object mass and spring rigidity) and with variable field strengths, i.e., with variable working parameters for the viscosity and elastic modulus.

Two methods were used to determine the frequency response: pointwise, i.e., at fixed frequencies, and with continuous variation. In the first case, the intervals were so small that no details in the characteristics were overlooked, so the curves reflected the resonant peaks, intervals not more than 0.5 Hz. When the frequency response is determined at low frequencies with continuous sweep, the period of the variation should not be less than 150 sec [9] in order to ensure steady-state vibrations.

Figure 3a shows the effects of the field on the amplitude of the relative vibrations in Fig. 1a; even in weak fields, where the viscous component is large, there is a marked reduction in the amplitude as the field increases down to a level comparable with the input base amplitude. As E increases further, the resonant amplitude increases, but in strong fields ($E > 1$ kV/mm) the initial values of A_r ($E = 0$) are not obtained, since the viscosity of the ER suspension continues to alter in spite of the increase in the elastic characteristics, and the static damping behavior persists. As the loading intensity g increases, the form of $A_r(E)$ does not alter qualitatively, but the points of inflection shift to higher fields, i.e., the electroelasticity effect is less pronounced. At high amplitudes ($X \gg h$), the hydraulic resistance is dependent only on the piston speed, being independent of E. The resonant frequency increases with E (Fig. 3b). The frequency response (Fig. 4) shifts to the right as far as a limit corresponding to the state of the working medium in which all the particles are rigidly linked in the framework and the attractive forces between adjacent particles are maximal. The change in elastic parameters in the field is clearly illustrated by the waveforms for the relative oscillations of object and base, where one can see a phase shift between the input and output signals, which increases with the field strength.

When the piston displacement is small, the bridges do not detach from the walls of the piston and cylinder, and all these regularities persist with certain reductions. For example, Fig. 3 shows that the transfer coefficient $\beta = |X/X_0|$ increases with g.

These experiments confirm the theoretical predictions on ER viscous-friction systems; one can use the known viscoelastic ER parameters as functions of field and vibration intensity to estimate the parameters of the principal isolating unit (the damper) for given transfer coefficients and damping decrements for forced and random vibrations. For example, one can estimate the working area of the damper from the condition for equality of the driving force and the hydrodynamic resistance. For Fig. 1a we have

$$\tau_0 S = M\omega^2 X_0,$$

and for Fig. 1b

$$\tau_0 S = kX_0.$$

The prototype used in the experiments employed $X \leq h$, and we determined τ_0, the shear stress at which the bridges detached from the walls, from rheological experiments (Fig. 2), which gave $S \approx 30.0$ cm². The piston may be solid or consist of a set of cylindrical or plate electrodes.

The results show that this ER system provides effective damping of harmful vibrations in conjunction with remote control on a given law (tracking or positioning). The stability is improved under any conditions, since the ER suspension enables one to set specified parameters if the static characteristics alter [10] while reducing the electrical energy consumption by comparison with existing pneumohydraulic systems, since the leakage current between the electrodes is extremely small ($I < 10^{-5}$ A).

NOTATION

M, mass of object, kg; h, gap between damper electrodes, m; ω_0, natural frequency, sec^{-1}; ϕ, phase shift between input and output displacements; E, electric field, kV/mm; A, amplitude of resonant oscillations; X, relative oscillation amplitude; X_1, amplitude of base oscillations; $\dot{\gamma}$, velocity, sec^{-1}.

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