EFFECT OF EXTERNAL ELECTRIC FIELD ON AMPLITUDE—FREQUENCY CHARACTERISTICS OF ELECTRORHEOLOGICAL DAMPER

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The amplitude—frequency characteristic of a viscous damper was experimentally studied and theoretically analyzed on the basis of a physical model of an elastic structural skeleton forming in an external electric field.

The operation of highly sensitive devices (electron microscopes, optical microscopes, interferometers, microanalytical balances, calibrating systems, etc.) becomes difficult when vibrations caused by random actions occur. This necessitates equipping such devices with intricate vibration protection systems.

The effectiveness of using these protective systems for shielding machines and mechanisms against the effects of random vibrations is determined by the transfer ratio β, namely ratio of the vibration amplitude of the isolated device to the vibration amplitude of the base

$$\beta = \frac{A_\mathrm{d}}{A_\mathrm{b}} = \sqrt{\frac{1+(\delta/\pi)^2\,(f/f_0)^2}{(1-f^2/f_0^2)^2+(\delta/\pi)^2\,(f/f_0)^2}}. \tag{1}$$

By varying the damping (logarithmic attenuation decrement δ, which characterizes the kinetic stability of the system) and the frequency of natural vibrations, one can attain the optimum transfer ratio in each individual case.

The greatest difficulties arise in the design of protection against weak low-frequency vibrations by providing the necessary strong attenuation. The possibility of passive vibration isolation is inadequate in this case. Active vibration protection, generally controlled through application of the feedback principle to nonlinear stiffness or damping in accordance with conventional methods of suppressing mechanical vibrations is, on the other hand, technically very difficult in this case, so that new methods of shock absorption and damping are eagerly sought. An effectiveness analysis of various dampers in the design of a vibration isolation system specifically for electron microscopes, a system which must ensure a small transfer ratio for weak low-frequency vibrations and a strong attenuation of natural vibrations, indicates promising results attainable with viscous dampers. The advantages of viscous hydraulic dampers [1] are the adjustability of the logarithmic decrement over a wide range by use of fillers with various viscosities or by use of two-component systems [2], the possibility of simultaneous damping of vibrations in various directions by the same device, the feasibility of active vibration isolation, and last but not least, the simplicity of construction. Particularly interesting are dampers whose characteristics can be regulated over wide ranges by remote control as, for example, by means of an external electric field. Dampers of this kind are based on the use of an electrorheological suspension as working medium [3, 4].

We will consider the simplest model of a damper. A piston of length L and cross-sectional area s moves at velocity v inside a cylinder filled with fluid, displacing a volume sv of the latter. This produces a pressure rise ahead of the piston. The clearance between piston and cylinder is usually small in comparison with the length and the radius: $h \ll L$, s/Π (Π denoting the perimeter of the piston cross section). For the purpose of analyzing the motion of the fluid in clearance, one can consider the flow with pressure gradient dp/dx during motion of the piston at velocity v and the flow rate of fluid per unit clearance area sv/

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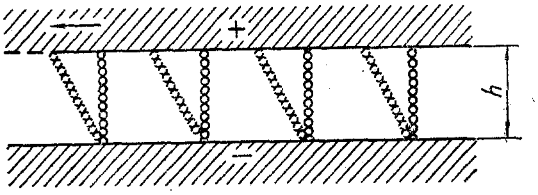

Fig. 1. Schematic diagram depicting elongation of bridges of solid phase in electrorheological suspension during motion of electrode.

Π. When $s/\Pi h \gg 1$, then the flow almost follows Poiseuille's law. Calculating the force of resistance to the piston motion requires determination of the shearing stress τ_w and the pressure rise $\Delta P = (\partial p/\partial x)L$ ahead of the piston. The equation of motion for a hydraulic damper is [5]

$$m\frac{d^2x}{dt^2} + qx + F_h = 0, \quad F_h = \tau_w \Pi L + \Delta ps.$$

The motion of a Newtonian fluid in the clearance is described by the equation

$$\rho\frac{\partial u}{\partial t} = \eta\frac{\partial^2 u}{\partial y^2} - \frac{\partial p}{\partial x}, \quad u|_{y=0} = 0, \quad u|_{y=h} = v. \tag{2}$$

In the case of random natural vibrations of the piston, which occur in practical damping by means of a viscous fluid, the Reynolds number is small and the inertia in Eq. (2) can be disregarded. Then,

$$F_h = r\,dx/dt, \text{where } r = \eta\frac{sL}{h^2}\left[12\left(\frac{s}{\Pi h} + 1\right) + 4\frac{\Pi h}{s}\right]. \tag{3}$$

When $s/\Pi h \gg 1$, then the shearing stress at the lateral surface of the cylinder is low and the pressure rise ahead of the piston is the main contributor to the resistance force.

With the damper filled with an electrorheological suspension and placed under action of an electric field, the damper characteristics depend on the electric field intensity $E = U/h$ as well as on the ratio A/h.

An external electric field applied to the electrodes causes particles of the solid phase in suspension to agglomerate into bridges whose strength and bond to an electrode depend on the electric field intensity E. Under a hydraulic pressure head, or during motion of one electrode, there occurs elongation (Fig. 1) or displacement of these bridges and, as has been demonstrated [6, 7], their breakdown with subsequent restoration when the distance between particles exceeds the critical magnitude of ∼40 A. At high concentrations of suspended particles the bridges form a dense integral skeleton. The forces of cohesion between particles in the direction of the electric field are much stronger than those in the direction perpendicular to it. When the piston vibrates with small amplitude $A/h \ll 1$, then the distance between particles increases insignificantly and the bridges do not disintegrate. The skeleton is rigidly coupled to the electrodes and will, as the piston moves through distance x, depart from the direction of the electric field by an angle $\varphi = x/h$, which will give rise to an elastic restoring force $F = N\Phi\Pi Lx/h$. As the electric field intensity increases, the stress in a bridge increases proportionally to E^2 up to some limit [6, 7]. The dependences of stress σ on the electric field intensity and on the particle concentration are shown in another study [8]. The stress $\sigma = N\Phi$ corresponds to the breakaway force (proportional to τ_0).

The pressure rise ahead of the piston during motion of the latter in the cylinder can be estimated from the relation

$$\Delta p = \left(\frac{s}{\Pi h} + \frac{1}{2}\right)\frac{L}{k}\frac{dx}{dt}.$$

In the case of low rates in a strong electric field it is permissible to assume that k depends only on the composition of the suspension, the clearance width, and the electric field intensity E. For an electrorheological suspension we then have

$$F_h = ax + b\frac{dx}{dt},$$

where

$$a = N\Phi \frac{\Pi L}{h}; \quad b = \frac{sL}{k}\left(\frac{s}{\Pi h} + \frac{1}{2}\right). \tag{4}$$

The interparticle interaction force decreases upon elongation of the bridges [6, 7], which causes a to decrease as x increases.

Inserting expressions (3) and (4) into Eq. (2) with a = const yields the expressions for damped vibrations of the piston in an electrorheological suspension

$$x = A\exp(-\varepsilon_s t)\cos w_s t, \quad \varepsilon_s = b/2m, \quad w_s = \sqrt{\frac{a+q}{m} - \varepsilon_s^2}, \tag{5}$$

and in a viscous fluid

$$x = A\exp(-\varepsilon_f t)\cos w_f t, \quad \varepsilon_f = \frac{r}{2m}, \quad w_f = \sqrt{\frac{q}{m} - \varepsilon_f^2}. \tag{6}$$

Relations (5) and (6) reveal that natural vibrations of the piston are damped at higher frequency in an electrorheological suspension than in a viscous fluid. This difference is attributable to the elasticity of the skeleton formed by particles in suspension.

When damping in an electrorheological suspension is weak, $\varepsilon_s \ll \sqrt{(a + q)/m}$, then the solution to Eqs. (1) and (3) with $a = a_0 - a_1 x^2$ $(a_1 > 0)$, taking into account that the tension on bridges decreases during their motion, can be obtained by the Bogolyubov—Mitropol'skii method of slowly varying amplitudes [9]

$$x = A\exp(-\varepsilon_s t)\cos\left\{w_0 t - \frac{3a_1 A^2}{8bw_0}\,[1 - \exp(-2\varepsilon_s t)]\right\}, \quad w_0 = \sqrt{\frac{a_0 + q}{m}}.$$

According to this expression, the piston passes through the equilibrium position (x = 0) at the instant of time

$$t_k \approx \frac{\pi}{2w_0}(2k+1) + \frac{3a_1 A^2}{8w_0^2 b}\left\{1 - \exp\left[-\frac{\pi\varepsilon_s}{w_0}(2k+1)\right]\right\}, \quad k = 0, 1, 2, \ldots$$

In the case of large vibrations $A/h \gg 1$, even in a strong electric field, the lifetime of a bridge is much shorter than the vibration period. For describing the dependence of shearing stress on flow velocity one can use the results of measurements of steady Couette flow [6, 7] $\frac{\partial u}{\partial y} = \varphi(|\tau|)\,\mathrm{sign}(\tau)$, where the flow function $\varphi(|\tau|)$ depends on the electric field intensity. In the inertialess approximation the shearing stress varies linearly across the channel: $\tau = (\partial p/\partial x)(y - h/2)$. Disregarding any motion of the channel walls when $s/\Pi h \gg 1$, we obtain the equation

$$\frac{s}{\Pi}\frac{dx}{dt} = 2\left[\int_0^{h/2} y\varphi\left(\left|\frac{\partial p}{\partial x}\right| y\right) dy\right]\mathrm{sign}\left(\frac{\partial p}{\partial x}\right) \tag{7}$$

relating the pressure gradient to the piston velocity. According to Eq. (7), the pressure gradient depends on the piston velocity in the manner

$$\frac{\partial p}{\partial x} = F\left(\left|\frac{dx}{dt}\right|\right)\mathrm{sign}\left(\frac{dx}{dt}\right). \tag{8}$$

Function $F(|dx/dt|)$ can be evaluated either from Eq. (7) with data on Couette flow or directly from the measured dependence of flow rate on pressure head in the clearance at various levels of electric field intensity. The main feature of large vibrations is that, according to expression (8), the hydraulic drag

$$F_h = LsF\left(\left|\frac{dx}{dt}\right|\right)\mathrm{sign}\left(\frac{dx}{dt}\right) \tag{9}$$

depends only on the piston velocity. In the case of weak damping, as when massive bodies vibrate, the problem of motion can be solved by the method of slowly varying amplitudes. It has already been demonstrated [9] that with the drag force varying according to relation (9),

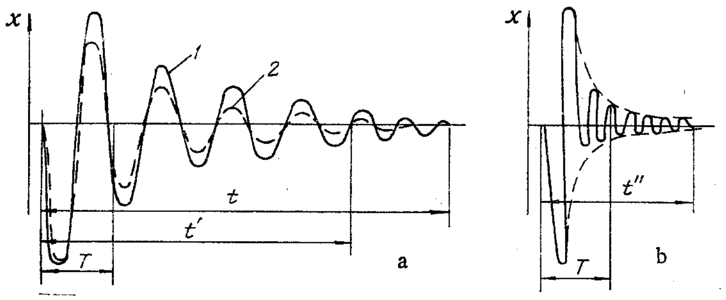

Fig. 2. Oscillograms of vibrations of electrorheological damper at U = 0 (a) and U = 6 kV (b): 1) PMS-1000 fluid; 2) electrorheological suspension.

a body performs damped vibrations $x = A(t) \cos \omega_f t$ at the frequency $\omega_f = \sqrt{q/m}$ with the vibration amplitude as function of time determined by the equation

$$\frac{dA}{dt} = \frac{-2}{m\omega_f} \int_0^{\frac{\pi}{2}} F_h (w_f A \sin z) \sin z dz. \tag{10}$$

Unlike during small vibrations (5), therefore, during large vibrations the frequency remains ω_f and does not depend on the electric field intensity. Only the amplitude attenuation (10) depends on the electric field intensity.

Accordingly, the theoretical analysis of the behavior of an electrorheological suspension in a damper in an external electric field reveals that the damper becomes stiffer, i.e., the frequency of natural vibrations increases with an electrorheological suspension instead of a viscous fluid.

A special experimental test stand has been built for verifying the proposed physical model and the results of theoretical analysis, on the basis of this model, of vibrations of an electrorheological damper. It includes a frame and loads with a total mass of 250 kg, the frame mounted horizontally on four vibration isolators in the form of compression springs with stiffness $q = 2.3 \cdot 10^4$ N/m. To this frame are rigidly coupled the piston-vibrators of hydraulic dampers.

The pistons were placed inside cylindrical cups on a separate base, each of them filled with an electrorheological suspension. The clearance between cylinder-stator and vibrator was set at 5 mm width, with a voltage from a dc source applied acroos it. The distance from the piston face to the cylinder bottom was 10 mm. Each cylinder had been furnished with a compensating cavity in the upper part, to prevent spillage of excess suspension during vibrations of the piston. Natural vibrations of the system were induced by an impact (approximately 10^4 N) in the direction perpendicular to the plane of the frame. Changes in vibrations of the system caused by the external force were sensed by a model SM-3 low-frequency pickup, which had been developed at the Special Design Office at the Institute of Physics of the Earth, Academy of Sciences of the USSR, and then through model TB-IV-SB galvanometers recorded on photographic tape of a model K115 oscillograph. The error of measurement of vibration parameters did not exceed 10%. The experiments were performed with pure transformer oil, polymethyl siloxan fluid, and 60% suspension of diatomite in transformer oil under normal conditions or in an electric field of $U \leq 12$ kV. The temperature was in all tests maintained within 20 ± 2°C.

A voltage was applied to the cylinders, and for 5 sec (the time necessary for complete formation of a structural skeleton, as determined experimentally under given conditions) a pulse force F was acting on the frame. The tests were performed with varying the magnitude of the initial perturbation, the composition of the fluid, and the intensity of the electric field.

Oscillograms depicting the process of vibrations of the frame with dampers filled with electrorheological suspension are shown in Figs. 2-3, along with their decoding, which reveal that damping in plain viscous fluid and in electrorheological suspension follows the same

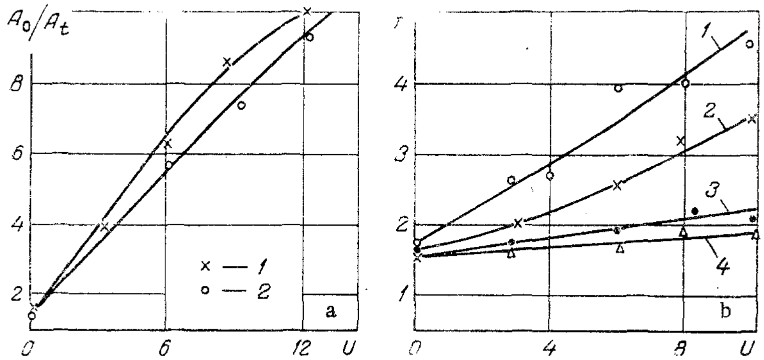

Fig. 3. Dependence of relative change in vibration amplitude (a) and frequency (b) of electrorheological damper on voltage, at various amplitudes of initial perturbation: a: 1) A_0 = 6.6 μm, 2) A_0 = 312 μm; b: 1) A_0 = 6.6 μm, 2) A_0 = 150 nm, 3) 1.75 nm, 4) 7.5 nm. U, kV; f, Hz.

qualitative trend when the initial displacement is the same. The pattern changes substantially upon application of an electric field (Fig. 2).

It is well known that the viscosity of an electrorheological fluid increases with increasing electric field intensity [10, 11]. Consequently, the logarithmic decrement $\delta^i = \ln [A_i/(A_i + 1)]$, where i is the number of oscillation cycles of the system, must increase (Fig. 3). In electrorheological suspensions, however, also the vibration frequency changes. In order to qualitatively estimate this trend, it is necessary to compare not the indicators corresponding to the same number of cycles, but those corresponding to the same instant of time.

The graph in Fig. 3a depicts the dependence of the ratio of vibration amplitudes at time t = 0.8 sec at various intensities of the applied electric field and for the same initial vibration amplitude A_0 = 6.6 μm or A_0 = 312 μm.

The almost linear dependence of ratio $A_0/A_t(U)$ on the voltage indicates that electrorheological suspensions have better damping characteristics and suppress natural vibrations completely much faster (5-6 times faster) as the voltage is raised.

Unlike with pure viscous and viscoplastic fluids, here the frequency of natural vibrations of the system can be regulated by means of an electric field. The rise of this frequency with rising voltage (Fig. 3b) agrees with the results of the preceding analysis based on the physical model of an elastic skeleton. This demonstrates a manifestation and amplification of elastic properties of electrorheological fluids in electric fields.

The change in vibration frequency with increasing initial vibration amplitude depends on the electric field intensity to a lesser degree. For A_0 = 6.6 μm the frequency at U = 10 kV has increased 300%, for instance, while for A_0 = 7.5 nm it has increased only 10%. These data confirm the conclusion that the period of first vibrations becomes longer as the initial displacement increases. They also agree with observations, made during study of flow processes in condenser-channels, that the electrorheological effect associated with the presence of an increasing transverse force becomes weaker as the piston velocity increases and the formation of bridges in the clearance subsides.

These trends are attributable to structural changes in electrorheological suspensions. In a damper with PMS-1000 fluid and a single clearance (simplest model) neither the logarithmic decrement δ nor the frequency f has been found to depend on the applied voltage over the A_0 = 50-850 μm range, which indicates that the magnitude of the restoring force associated with changes during vibrations of the system container is irrelevant here.

We thus conclude that using an electrorheological damper makes it possible to attain the main goal of vibration protection, namely, to more effectively than by conventional means reduce the amplitudes of vibrations and to appreciably shorten the damping time. Regulation by external means, with an electric field, makes it furthermore possible to suppress vibrations with various initial amplitudes to a given desired level. Use of electrorheological suspensions in viscous dampers extends the applicability of the latter, inasmuch as it becomes possible to introduce into the system an additional elastic element which is controllable. The combination of these advantages is particularly valuable in vibration isolation of electron-optical devices.

NOTATION

A_d, vibration amplitude of the device; A_b, vibration amplitude of the base; f, frequency of random vibrations; f_o, frequency of natural vibrations of the device; δ, logarithmic attenuation decrement; m, mass of the body; x, displacement; q, stiffness of the damper; t, time; F_h, hydraulic drag force; ρ, density of the fluid; η, dynamic viscosity of the fluid; y, transverse coordinate in the clearance; U, potential difference between cylinder body and piston; E, electric field intensity; h, clearance width; Φ, tension on a bridge; N, number of bridges; τ, shearing stress; P, pressure.

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