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DYNAMIC BEHAVIOR OF THE STRESS - STRAIN STATE OF A DISSIPATIVE MECHANICAL SYSTEM

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Abstract. The paper considers two modes of operation of the system - natural and forced oscillations. Natural oscillations are understood as movements in which all points of the system oscillate with the same frequencies and damping indices (but with different complex amplitudes). It is assumed that there are no external influences during natural oscillations. Forced oscillations occur under stationary (periodic) and non-stationary external influences. The oscillation mode (steady or unsteady) depends on external influences.
Keywords: natural oscillations, frequency, steady state, strength, vibration, mechanical systems, object, damped oscillations, resonance

Аннотация. В статье рассматриваются два режима работы системы - собственные и вынужденные колебания. Под собственными колебаниями понимаются движения, при которых все точки системы колеблются с одинаковыми частотами и показателями затухания (но с разными комплексными амплитудами). Предполагается, что во время собственных колебаний внешних воздействий нет. Вынужденные колебания возникают при стационарных (периодических) и нестационарных внешних воздействиях. Режим колебаний (устойчивый или нестационарный) зависит от внешних воздействий.
Ключевые слова: собственные колебания, частота, установившийся режим, прочность, вибрация, механические системы, объект, затухающие колебания, резонанс.

Introduction

In many cases, solvying the problem of dynamic strength and stability is related to the study of the elements\` rheological properties as well as the interaction of different forms of vibration of the considered mechanical system. In the works [1,2,3,4], the problems of interaction of the elements of a mechanical system with each other and with the medium are considered. Such mechanical systems are often used to protect objects from vibrations and impacts. Research in the field of protection of various objects from vibrations and shocks was intensively developed in the early 70s of the last century, among which one could name the works [5,6]. Modern radio-mechanical complexes and navigation equipment placed on mobile objects (aircraft, ships, self-propelled vehicles, cars, etc.) are exposed to a complex complex of destabilizing factors.

These factors, in particular, include vibration and shock loads arising from changes in the speed of movement of moving objects with screw, turboprop and jet engines, acoustic influences, etc. Aerodynamic forces also cause an increase in vibration of components and parts of moving objects [7,8].

Studies of the strength and dynamic stability of dissipative (dissipatively homogeneous and inhomogeneous) mechanical systems are still far from exhaustive solutions. The control of resonant oscillations of dissipative mechanical systems using active methods has not yet been definitively solved.

Problem statement and medoc and solutions.

The dynamic behavior of the stress-deformable state of a dissipative mechanical system consisting of deformable and non-deformable bodies is considered. The relationship of stresses and deformations for the elements of a mechanical system satisfies the linear hereditary Boltzmann-Aviary relations, which we take as:

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$$s_{ij} = 2\sigma(e_{ij} - \int_{-\infty}^{t} R(t-\tau)e_{ij}(\tau)d\tau) \quad (1)$$

where, s_{ij}- is the stress deviator σ_{ij}, e_{ij}- is the strain deviator $\varepsilon_{ij}, \sigma = \sigma_{ij}/3$, R -is a weakly singular relaxation kernel, taken as

$$R(t) = Ae^{-\beta t} \cdot t^{\alpha-1};$$

here, *E*- is the instantaneous modulus of elasticity, A, α and, ai β- are dimensionless parameters.

The parameters of the relaxation core and the instantaneous modulus of elasticity are determined from quasi-static experiments by the methodology described in [9].

General variational formulation of the dynamics of dissipative mechanical systems.

Consider a mechanical system consisting of N rigid and K deformable elements connected to each other and to the base (or environment) by S viscoelastic elements:

$$S = S_1 + S_2.$$

Deformable elements of the system are made of viscoelastic or two-component bodies. The physical properties of viscoelastic materials are described by linear hereditary Boltzmann-Volterra relations with integral differences of inheritance nuclei [10, 11].

Some of the deformable elements may be elastic; in this case, the nuclei of heredity describing rheological properties elements are identically equal to zero. A system in which the rheological properties of deformable elements are identical (the nuclei of the heredity of the elements are equal to each other) will be called dissipatively homogeneous, and a system with different rheological characteristics will be called dissipatively heterogeneous.

In the special case, when there are no external influences, the system's own damped oscillations are considered, in the presence of external influences - forced. The main problem is the study of dissipative (damping) properties of the system as a whole, as well as the study of its stress-strain state. With free oscillations, dissipation is reduced to the attenuation of natural oscillations. The attenuation rate quantifies the dissipative properties of the system: the higher the attenuation rate, the higher the dissipation.

With forced steady-state oscillations of the system, its dissipative properties manifest themselves in resonant modes and lead to finite values of resonant amplitudes. In the case of non-stationary oscillations, dissipative properties are manifested when determining the stress-strain state of the system.

In the case of forced steady-state oscillations, resonant amplitudes are a quantitative characteristic of the dissipative properties of the system, the intensity of which becomes higher by lowering the resonance of the amplitude of forced oscillations.

Two values are proposed for a quantitative sketch of the dissipative properties of the system as a whole: the minimum attenuation rate of natural oscillations and the maximum resonant amplitude. We introduce the concepts of global damping coefficient and global resonant amplitude. The dissipative properties of the system are determined primarily by the damping characteristics of the systems, which is completely inapplicable to dissipatively inhomogeneous systems.The study of the dependence of the level of dissipative properties of the system on its parameters is the main content of this work. It is established that the global damping characteristics of a dissipatively inhomogeneous system as a whole are determined not only (and not so much) by the viscoelastic properties of the system elements, but by the interaction of vibrations of various eigenforms, which are significantly determined by the structure, design, geometry, dimensions, the presence of elastic bonds, the mutual arrangement of the system elements as a whole.

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According to the work dynamics of dissipative mechanical systems, they are more realistically described by a generalized viscous model (1):

$$\tau\phi(t) = c_m\left[\phi(t) - \int_a^t R_c(t-\tau)\phi(\tau)d(\tau)\right] \qquad (2)$$

where *τ*- is the viscoelastic operator; c_m - is the instantaneous coefficient of the viscoelastic operator; R_c - is the relaxation kernel; *ϕ(t)* - are arbitrary time functions.

Mathematical formulation of the problem

Suppose that the integral terms in the hereditary relation (2) describing the rheological properties of deformable elements are small compared to the instantaneously elastic terms. Together with the assumption of the oscillatory nature of the motion, this will allow the freezing procedure to be applied [12,13,14], which leads to the following complex physical relations for deformable elements of zero volume:

$$F_e = -c_e\Delta e = -c_e[1 - \Gamma_e^c(\omega_R) - i\Gamma_e^s(\omega_R)]\Delta e \qquad (3)$$

For elements with distributed stiffness , this ratio will have the form:

$$\sigma_{ij} = \lambda_n\varepsilon_{ni}\delta_{ji} + 2\bar{\mu}_n\varepsilon_{ni}, \quad S = S_1 + S_2, \quad n = 1,2,3,\ldots,S\ , \qquad (4)$$

Here $\bar{\lambda}_n = \lambda_n\left[1 - \Gamma_{n\lambda}^c(\omega_R) - i\ \Gamma_{n\mu}^s(\omega_R)\right] \qquad (5)$

$$\Gamma_{\lambda,m}^c(\omega) = \int_0^\infty R_{\lambda,m}(\tau)\cdot cos\,\omega\,\tau d\tau \quad ; \quad \Gamma_{\lambda,\mu}^s\ (\omega) == \int_0^\infty R_{\lambda,\mu}\,(\tau)\,sin\,\omega\,\tau d\tau$$

F_e - is the force in the *i*-th concentrated element, *Δe* is the elongation of this element; $\bar{C}_e, \sigma_{ij}, \varepsilon_{ij}$ is its complex stiffness, stress and deformation in an element of non-zero volume; $\Gamma_e^c, \Gamma_e^s, \Gamma_{n\lambda}^c, \Gamma_{n\lambda}^s, \Gamma_{n\mu}^c, \Gamma_{n\mu}^s$ is the sine and the cosine of the Fourier image of the relaxation nuclei of the *i*-th concentrated element and the *n*-th distributed element; ω_R - is the real part of the complex oscillation frequency of the system.

In this case, the frequency will be complex with its own, real, as well as forced fluctuations. In the first case, the complex natural frequency is the frequency of damped oscillations; the imaginary part is the damping coefficient of the natural oscillations of the system. In the second case, ω_R coincides with the frequency of forced oscillations. For natural oscillations, the ratios (3), (4) are approximate, for forced ones - exact.

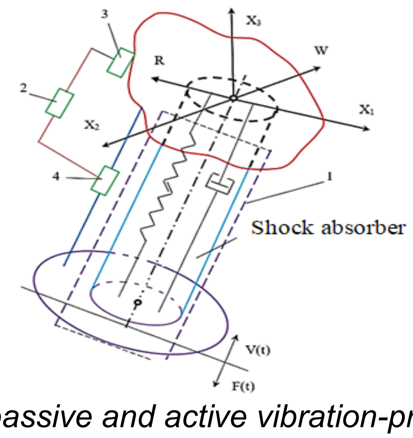

Fig.1. *Functional diagram of passive and active vibration-proof mechanical system.1-the passive part of the VPS (vibration protection system), 2-devices for signal conversion, 3-sensors, 4- the active part.*

When setting the problem of natural and forced oscillations of the system, the principle of possible displacements is used, according to which the sum of all active forces acting on the system, including inertia forces, is zero:

$$\delta A = \delta A_\sigma + \delta A_u + \delta A_F = 0 \qquad (6)$$

here

$$\delta A_F = -\sum_{n=1}^{S_2} \int_{v_n} \sigma_{ij}\, \delta\varepsilon_{ij}\, dV - \sum_{e=1}^{S_1} \Gamma_e\, \delta\Delta e$$

as an example, consider a body mounted on deformable supports (Fig. 1) or a dissipative mechanical system is considered in the place of the object (Fig.2).

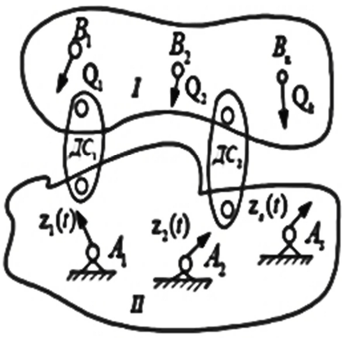

***Fig.2.** Schematic diagram*

Schematic diagram of the implementation of the vibration protection method by introducing additional links: a) the conditional division of the system into blocks I and II; b) the introduction of additional connections between blocks I and II (additional connections are designated DC_1, DC_2, etc.)

The projections of the displacement vector W $^{\rightarrow}$ on the coordinate axes are denoted by u,v, and w, and the projections of the force vector are respectively U,V, W.

Functional dependencies

$$U = U(u, \dot{u}), \quad V = V(v, \dot{v}), \quad W = W(w, \dot{w},$$

$$(\dot{u} = \frac{\partial u}{\partial t}, \quad \dot{v} = \frac{\partial v}{\partial t}, \quad \dot{w} = \frac{\partial w}{\partial t})$$

they are called the dynamic characteristics of the shock absorber. When analyzing the dynamics of vibration protection systems in a static equilibrium position, the following are taken:

$$U = V = W = 0, \quad \dot{U} = \dot{V} = \dot{W} = 0$$

Let a part of the surface of the body perform translational motion according to a given harmonic law, and on a part of the surface are given loads that vary in time according to a given harmonic law.

The tasks of vibration-proof objects with machinery in the time domain, in essence, are a dynamic analysis of the response of a mechanical system to the action of shock or vibration loads. Naturally, before conducting such an analysis, it is necessary to determine the input loads or displacements, geometric and physic-mechanical properties of the equipment, boundary conditions. Having such data, it is necessary to develop a dynamic (mathematical) model that will approximately represent real products and loads. As a rule, a dynamic model will only approximate the real object, because its exact representation will entail insurmountable mathematical difficulties in solving. The shape of the pulses and the mathematical expression are given below (Fig.2 and Fig.3).

A mathematical model of a semi-sinusoidal pulse

$F(t) = A\, sin\, \omega\, t$ at $0 \le t \langle \tau$, $F(t) = 0$ at $t \ge \tau$;

Rectangular pulse $F(t) = A$ at $0 \le t\langle \tau$, $F(t) = 0$ at $t \ge \tau$;

Triangular pulse $(t) = A$ at $0 \le t\langle \tau$, $F(t) = 0$ at $t \ge \tau$;

The affixing of shock pulses in the form of the simplest forms is not always justified. The representation of the shock process in the form of the frequency spectrum *F(t)* obtained by the Fourier transform leads to more accurate results.

$$F(\omega) = \int_{-\infty}^{\infty} F(t)\, e^{i\omega t} dt.$$

After an impact, a VAT occurs in the mechanical system, which can lead to permissible values of stresses and deformations. And when exposed to harmonic or vibrational influences, resonance occurs in the mechanical system

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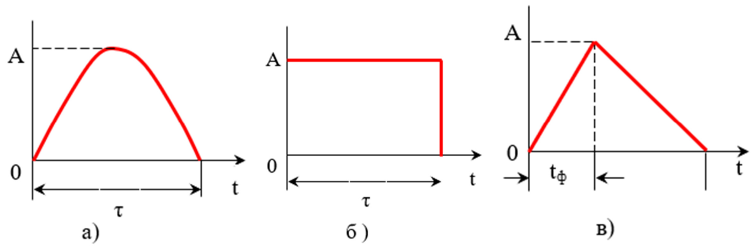

Fig.3. *The shape of shock pulses: a- semi-sinusoidal; b- rectangular; c- triangular.*

You can specify three types of dynamic models suitable for solving vibration protection problems in the time domain; models with distributed parameters; models with concentrated parameters; offset models [15,16].

Conclusion

A general variational mathematical formulation of the problem of dynamics of dissipatively homogeneous and inhomogeneous mechanical systems consisting of solid and deformable bodies is proposed. The relationship between stresses and deformations is taken into account using the Boltzmann-Volterra integral. Based on the method of Muller, Gauss, integral Laplace transform, a method for solving and an algorithm for the problem of natural and forced oscillations of dissipative mechanical systems consisting of solid and deformable bodies is proposed.

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