BG Pulse Journal USA
ISSN:1947-2536 http://www.bgpulseusa.com/index.php/bg
BG PULSE JOURNAL

STUDY OF FREE AND FORCED VIBRATIONS OF A MECHANICAL SYSTEM CONSISTING OF A VISCOELASTIC PLATE AND SUSPENDED MASSES

Jalolov Farrukh Bakhshullaevich
Asia International University, UZ
"General Technical Sciences" Department Teacher
jalolovfarruh2@mail.ru

Abstract. In forced steady or harmonic oscillations of the system, the dissipative properties of the system are manifested in resonance modes and lead to the adoption of limited values of resonance amplitudes. In non-stationary oscillations, the dissipative properties of the mechanical system also have an effective effect in determining the stress-strain state of the mechanical system. In vibrations in a forced harmonic state, the resonance amplitudes are considered a quantitative characteristic of the dissipative properties of the system, the intensity of which is manifested by a decrease in the resonance amplitude of the forced oscillations.

The main goal of this work is to study the dependence of the level of dissipative properties of a mechanical system on the geometric and physicomechanical parameters of the system. It was found that the global damping characteristics of a dissipative inhomogeneous mechanical system are characterized not only by the viscous properties of the system elements, but also by the interaction of free vibrations of various forms.

Keywords: wave diffraction, viscoelastic medium, deformable spherical body, free oscillations, frequency, viscosity.

1. Introduction

Let us assume that the integral in the hereditary relation describing the rheological properties of deformable elements has a small value. Since the integral has a small value, we have the opportunity to apply the freezing procedure. In this case, we obtain the following relations for deformable massless elements in a mechanical system:

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

For a body with deformable mass, the relationship between stress and deformation is as follows:

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

here

$$\bar{\lambda}_n = \lambda_n \left[1 - \Gamma_{n\lambda}^c(\omega_R) - i\,\Gamma_{n\mu}^s(\omega_R)\right] \qquad (3)$$

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

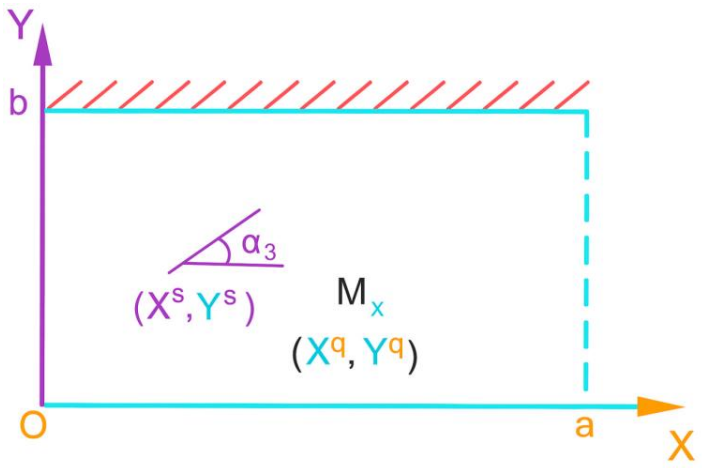
F_e – this i- point element stress, Δe - shift the given element; $\overline{C}_e, \sigma_{ij}, \varepsilon_{ij}$ - stiffness, stress, and strain of a deformable element with non-zero volume; $\Gamma_e^c, \Gamma_e^s, \Gamma_{n\lambda}^c, \Gamma_{n\lambda}^s, \Gamma_{n\mu}^c, \Gamma_{n\mu}^s$ – i- with a

mass of and n- Fourier kernel of the sine and cosine representation of the chi-dimensional parameter element; ω_R- the real part of the complex frequency.

In this case, it is difficult to find the frequency of free oscillations and the frequency of forced oscillations. In the first case, the abstract part of the complex free oscillation frequency gives the energy dissipation of the mechanical system corresponding to the frequency. For a viscoelastic system, the frequency of free oscillations is known to be a complex quantity. The abstract part of the complex frequency ω_I is determined by. The dimensionless damping coefficients are obtained as follows: $\delta_{\omega_R k} = \frac{\omega_{Ik}}{\omega_{Rk}} -$ relative frequency extinction coefficient; $\delta_{\pi k} = 2\pi\delta_{\omega Rk} -$ logarithmic decay decrement. Mathematical formulation of the problem and solution methodology. Let us assume that N rigidly assembled mass material points are mounted on a viscoelastic plate using K deformable viscoelastic elements. (1.12-rasm).

1.12- Fig. Calculation scheme for a plate with a fixed and suspended mass.

In this figure, we see the free and forced oscillations of the viscoelastic mechanical system. We use the principle of possible displacement when placing masses. For this, the sum of the variations of the work done by the active and passive forces acting on the mechanical system must be equal to zero.



$$\delta A = \delta A_\sigma + \delta A_I + \delta A_F = 0, \qquad (4)$$

here

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

$$\delta A_I = -\sum_{n=1}^{S_a} \int_{V_n} \rho_u \frac{\partial^2 \bar{u}_n}{\partial t^2}\delta\bar{u}_n dV - \sum_{k=1}^{N} m_k \frac{d^2\bar{u}_n}{dt^2}\delta\bar{u}_k - \sum_{k=1}^{N} I_k \frac{d\bar{\omega}_k}{dt^2}\delta\varphi_k \bar{l}, \bar{\omega}_k = \frac{d\varphi_k}{dt}\bar{l}$$

$$\delta A_F = -\sum_{n=1}^{S_2} \int_{v_n} \rho_n \bar{f}\delta\bar{u}_n dV + \sum_{n=1}^{s_1} \int_{v_n} \bar{f}_n \delta\bar{u}_n dV + \sum_{n=1}^{N} \bar{F}_n \delta\bar{u}_n + \sum_{k=1}^{N} \bar{m}_k \delta\bar{\varphi}_k$$

$\delta\varepsilon_{ij}$, $\delta\Delta e$ - Deformation of a dispersed body and changes in the displacement of mass elements attached to it; ρ_n- Collected at the point n- The density of a material or element; m_k- к- is the mass of a solid; $u, u_k, \delta u_1, \delta u_k$ - points of deformable material with distributed parameters and the displacement vector of elements collected at the point; f, ρ - the density of surface forces and mass forces acting on a deformable material with distributed parameters; V_n, E_n - distributed parametric deformable n- the volume of the material and its surface area; I_n - n- the central moment of inertia of a rigid body; F_m, M_k - k- The principal vector and principal moment of a rigid body. If the mass deformable element in the mechanical system described above consists of thin plates, then the work done in addition to equation (4) will have the following form:

$$\delta A_\sigma = -\int_V \sigma_{ij}\delta\varepsilon_{ij}dV,$$

$$\delta A_a = -\sum_{l=1}^{L} \sigma_l \ \delta\varepsilon_l ,$$

$$\delta A_m = -\rho h \int_\Omega \bar{U}(X_1, X_2, t)\ \delta\bar{U}d\Omega - \sum_{q=1}^{Q} M_q \bar{U}(X_1^q, X_2^q, t)\delta\bar{U}.$$

here ρ, h – The density of the plate material and its thickness, M_q -q- attached mass, $L^/$ The number of deformable supports, Q- number of suspended masses, V,Ω – Plate size and surface area accordingly, $\sigma_{ij}, \varepsilon_{ij}$ - stress and strain components respectively, $\sigma_{l^/}, \varepsilon_{l^/}$ - in line l - stresses and deformations in a deformable support, δ- generalized displacement variation. The second way to obtain a system of equations is to use the differential approach. The differential approach can be used mainly for systems with mass, such as spatial rods, plates. In this case, the differential equation of the mechanical system takes the following form :

$$\sum_{k=1}^{S_2} L_{jk}W_k - \sum_{k=1}^{S_2}\int_{-}^{t} R_{jk}(t-\tau)L_{jk}W(\tau)d\tau \pm \rho_j \frac{\partial^2 W_j}{\partial t^2} = \pm\rho f_j e^{i\lambda t}, \qquad (5)$$

here W_j - j- mass element displacement vector components; ρ_j - j- The density of elements with mass chi depends on the coordinate; f_j - mass force amplitude; $R_{jk}(t-\tau)$ - relaxation core, L_{jk} - differential operator in the spatial coordinate system. In the work, the above- mentioned free and forced oscillations of a mechanical system are studied. When studying free oscillations, we do not take into account external forces. Boundary conditions are not taken into account, which are automatically fulfilled in the variational formulation of problems. Forced oscillations occur under the influence of harmonic or static forces.

$$F(t) = F_0\ e^{-i\lambda t}, \qquad (6)$$

here F_0 - external load amplitude; λ – external load frequency. Obtained above (4) we are looking for a solution to the system of equations in the following form:

$$\bar{u}_k(\vec{x},t) = \vec{V}_k(x)\, e^{-i\omega t}, \qquad (7)$$

here ω - The complex frequency of free vibrations that we are looking for is found in the process of solving the problem. In forced vibrations, the frequency is a definite real quantity and is given by ($\omega = \lambda$); $\vec{V}(\vec{x})$ - is a complex quantity that gives the form of oscillations. If the mechanical system consists of a finite degree of freedom, then using Lagrange's differential equation of the second kind, we obtain the integro-differential equation of oscillations

$$\sum_{k=1}^{6N} \left(a_{jk}\ddot{q}_k(t) + C_{jk}\ q_k(t) - \int_{-}^{t} R_{ck}(t-\tau)q_k(\tau)d\tau\right) = f_j e^{-i\lambda t},\ j = 1,2,...,6N \qquad (8)$$

here a_{ij} - gives a positive definite quadratic form

$$\frac{1}{2}\sum_{i,j=1}^{n} a_{ij}q_i q_j .$$

2. Methods

2.1. Problem Statement and Methods of Solution

The problem is to find a periodic solution to the system of equations (8). If we represent the system of differential equations (8) in terms of normal coordinates, $R_{ij} = 0$ can be generalized for a viscoelastic mechanical system. If we introduce the generalized coordinate system as follows

$$q_j = \sum_{k=1}^{n} b_{ik}\theta_k ,$$

the system takes the following form

$$\ddot{\theta} + \omega_i^2\theta_i - \int_{-}^{t} \bar{R}_{ik}(t-s)\,\theta_k(s)\,ds = Q_{i0}\sin pt \ , (i=1,...,n)$$

here θ_k - normal coordinates, b_{jk} - are coefficients or matrix elements that satisfy the following condition $\det[b_{jk}] \neq 0$, ω_i- frequency of free vibrations of an elastic system, Θ_{0i}- The generalized force amplitude corresponding to the normal coordinate takes the following form

$$\theta_{Oi} = \sum_{k=1}^{n} b_{ki}\theta_{ok}$$

$\bar{R}_{ik}$– relaxation core (affecting function):

$$\bar{R}_{ik} = \sum_{j=1}^{n} R_{ij}\theta_{jk} .$$

here a_{jk} - is symmetric real-valued and represents generalized mass; C_{jk} - a symmetric matrix representing the generalized instantaneous elastic modulus; q_k - complex generalized coordinates; f_i - complex generalized force amplitudes; λ - is the frequency of the external force, and the actual magnitude.

When the frequency of free vibrations is studied, the right-hand side of equation (8) is equal to zero, then the solution to the equation is as follows:

$$q_j = A_j e^{-i\omega t}, \quad j = 1,...,6 \qquad (9)$$

here $\omega = \omega_R + i\omega_I$ - The frequency of free vibrations is a complex quantity.

If we substitute the solution (9) into equation (8), we obtain the following system of homogeneous algebraic equations with complex coefficients:

$$\sum_{k=1}^{6N} (\bar{C}_{jk}(\omega_R) - \omega^2 c_{jk}) A_k = 0, \quad j = 1,2,...,6N \ . \qquad (10)$$

For a system of homogeneous algebraic equations to have a nonzero solution, its principal determinant must be zero. As a result, the real part of the complex frequency (11) ω_R we obtain a non-linear, non-integral transcendental equation

$$\left|\bar{C}_{jk}(\omega_R) - \omega^2 c_{jk}\right| = 0 \ . \qquad (11)$$

3. Numerical results and their analysis

The above transcendental equation (11) is solved using the numerical Muller method. The conservative solution of the problem is taken as the initial value in the first approximation. This transcendental equation (11) is solved using the Gauss method at each iteration when solving the Muller method. Therefore, it is not necessary to expand the determinant of (11). The roots of this equation are as follows

$\omega_k = \omega_{Rk} - i\omega_{Ik}$,

$\omega_{n-k} = \omega_{Rk} + i\omega_{Ik}$, $(k=1,...,n)$ (12)

here $\omega_{Ik}>0$ and $\omega_{Rk}>0$– are real numbers, $\omega_{Ik}>0$ represents the extinction coefficient.

If (8) the solution of the system of differential equations representing forced oscillations is sought as follows

$$q_j = A_j e^{-i\lambda t} \qquad j = 1,...,6N \qquad (13)$$

here A_j - is the complex amplitude to be found. If we substitute the solution (13) into the equation, we obtain the following system of algebraic equations with non-homogeneous complex coefficients:

$$\sum_{k=1}^{6N} (C_{jk}(\lambda) - \lambda^2 a_{jk}) A_k = f_j \ . \qquad (1.21)$$

The last system of equations is solved by the Gauss method.

If the mechanical system under study is subjected to an unsteady load, then the differential equation is solved by Laplace substitution

Conclusions

1. The study of resonance phenomena in dissipative homogeneous and inhomogeneous mechanical systems consisting of material points and plates under the influence of harmonic and non-stationary loads, depending on the energy intensity, remains as relevant as ever;
2. It has been found that the study of the vibrational stability of structurally (dissipatively uniform or non-uniform) inhomogeneous mechanical systems has not yet been fully resolved;
3. The issue of controlling resonant vibrations of dissipative mechanical systems using active methods is also a pressing problem.

Literature

1. Gorshkov A.G. Non-stationary interaction of plates and shells with solid media// Ed. RAN. Preschool Education. - 1981. - No 4. - Pp. 177-189.

2. Avliyakulov N.N., Safarov I.I. Modern problems of statics and dynamics of underground pipelines.-Tashkent, 2007.-306 p.

3. Bozorov M.B., Safarov I.I., Shokin Yu.I. Numerical modeling of oscillations of dissipatively homogeneous and non-homogeneous mechanical systems. SO RAN, Novosibirsk, 1966.- 188p.

4. Safarov I.I. Oscillations and waves in dissipative-inhomogeneous media and structures. Tashkent, Fan, 1992.- 250p.

5. Safarov I.I., Kayumov S.S. Distribution and diffraction of waves in dissipatively inhomogeneous cylindrical deformable mechanical systems. Tashkent: FAN, 2002.