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Dynamic dampers of vibrations of inherited-deformable systems with finite number of degrees of freedom

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Abstract. The problem of dynamic vibration dampers of inherited-deformable systems with finite number of degrees of freedom is considered. Rheological properties of spring (suspension) are taken into account using integral model with Koltunov-Rzhanitsin relaxation core. The behavior of the system with a damper is considered at free attenuation oscillations caused by the specified initial conditions, as well as at constant, pulse and periodic external impacts. The obtained results make it possible to conclude on the expediency of using dynamic dampers to reduce amplitude of oscillations, both in perfectly elastic and in inherited-deformable systems during transient processes. A computational algorithm based on quadrature formulas is used to solve the problem.

1. Introduction

The high level of vibration of machines often causes their fatigue damage and, in some cases, total destruction. In order to increase the reliability of machines, the vibration level must be reduced by setting off resonance zones or introducing various damping devices. In some cases, retraction from resonance zones may require changes in the stiffness and mass of machine design, which may be less advantageous than the use of dynamic vibration dampers of inherited-deformable systems.

Dynamic vibration dampers, like some additional devices introduced into the original design circuits of vibration protection systems, can be considered as one of the means of controlling the state of the protection object. Mathematical models of oscillatory systems in the form of structural diagrams of dynamically equivalent automatic control systems are shown to have certain advantages over conventional approaches based on the use of differential equations. Dynamic blanking in structural models is interpreted as introducing additional negative feedback circuits. Such circuits are formed on the basis of structural transformations of the initial model according to the rules of parallel and serial connection of spring [1-3].

It has been found that introduction of additional tuning masses significantly changes the type of amplitude-frequency characteristics, in particular, formation of such forms, which create the possibility of erosion of point frequencies of dynamic damping of oscillations until their representation in the form of zones with average-constant value of transmission coefficients of amplitude of vibrations from the source of disturbances to the object [4].

The urgency of the problem is constantly increasing due to the increase in the size of the structure, the increase in the speed of the machines, the tightening of sanitary and technological requirements to permissible vibration levels [5]. When investigating this problem, it is important to consider the non-elastic resistance of the damper and the protected system. In papers [6-9] non-elastic resistance is taken into account according to elementary theory of viscoelasticity and model "Complex stiffness" proposed by E.S. Sorokin [10]. These theories have serious disadvantages, they do not take into account the time factor associated with creep and stress relaxation. Hereditary viscoelastic theories are the most common when considering the material 's non-elastic resistances, as they simultaneously account for both internal friction and creep deformation and stress relaxation of the material. Therefore, the development of more efficient methods of calculating fluctuations taking into account their rational use in solving various tasks of inherited-deformable elements of machine structures is very relevant.

Problem definition. Let the protected structure with mass of m_1 be given, resting on nonlinear inherited-deformable spring (suspension) with instantaneous rigidity of c_1, which is under action of external load q(t) (Figure 1, System I). Consider the motion of system I, whose position is determined by the generalized coordinates $u_1(t)$. We attach to this system a special device with mass of m_2, which with the help of nonlinear inherited-deformable spring, with instantaneous rigidity of the c_2, is connected to the protected structure. Then we get system II with two degrees of freedom (Figure 1, System II), positions of which at any moment are determined by generalized coordinates $u_1(t)$ and $u_2(t)$.

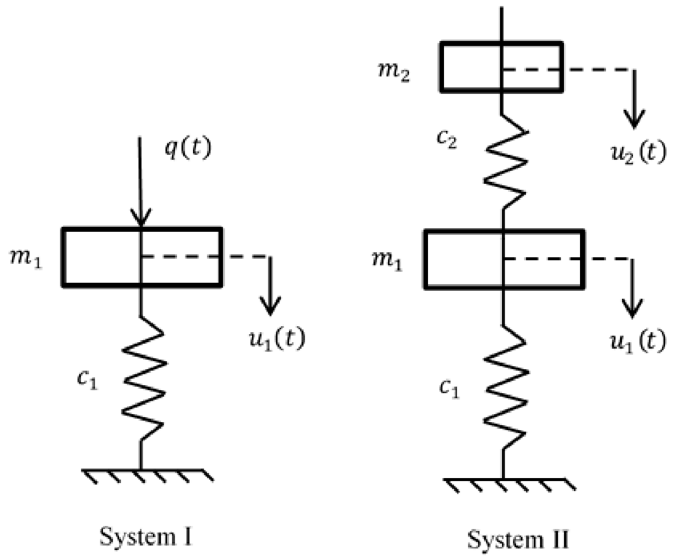

Figure 1. Investigated systems

Rheological properties of suspension are assumed to be different and subject to cubic nonlinear hereditary theory of viscoelasticity [11,12]. Then, according to the variation principle of inheritance theory of viscoelasticity [11], kinetic energy, potential energy and operation of external forces are calculated as follows:

$$T = \frac{1}{2}(m_1\dot{u}_1^2 + m_2\dot{u}_2^2), P = q(t)u_1$$

$$\Pi = \frac{c_1}{2}\left\{u_1\left[u_1 + \frac{\gamma_1}{2}u_1^3 - 2R_1^*(u_1 + \gamma_1 u_1^3)\right]\right\} +$$

$$+\frac{c_2}{2}\left\{(u_1 - u_2)\left[u_1 - u_2 + \frac{\gamma_2}{2}(u_1 - u_2)^3 - 2R_2^*[u_1 - u_2 + \gamma_2(u_1 - u_2)^3]\right]\right\}$$

here $\gamma_i, (i = 1,2) -$ is a physical nonlinearity coefficient that is less than zero $(\gamma_i < 0)$ for a soft material and greater than zero $(\gamma_i > 0)$ for a rigid material. $R_i^* -$ are Volterra integral operators for which

$$R_i^* f(t) = \int_0^t R_i(t-\tau)\, f(\tau) d\tau,$$

here $R_i(t) = \varepsilon_i e^{-\beta_i t} t^{\alpha_i - 1} -$ is a core of inheritance having weakly singular Abel-type features.

The Lagrange equation for the system in question is:

$$\frac{\partial L}{\partial u_i} - \frac{d}{dt}\frac{\partial L}{\partial \dot{u}_i} = 0,\ i = 1,2 \qquad (1)$$

here $L = \Pi - T - P$;

$$\frac{\partial L}{\partial u_1} = c_1(1 - R^*)(u_1 + \gamma_1 u_1^3) + c_2(1 - R^*)[u_1 - u_2 + \gamma_2(u_1 - u_2)^3] - q(t);$$
$$\frac{\partial L}{\partial \dot{u}_1} = -m_1 \dot{u}_1; \qquad \frac{\partial L}{\partial \dot{u}_2} = -m_2 \dot{u}_2; \qquad (2)$$
$$\frac{\partial L}{\partial u_2} = -c_2(1 - R^*)[u_1 - u_2 + \gamma_2(u_1 - u_2)^3].$$

By adjusting (2) to (1), we obtain:

$$\begin{cases} m_1 \ddot{u}_1 + c_1(1 - R_1^*)(u_1 + \gamma_1 u_1^3) + m_2 \ddot{u}_2 = q(t) \\ m_2 \ddot{u}_2 - c_2(1 - R_2^*)[u_1 - u_2 + \gamma_2(u_1 - u_2)^3] = 0. \end{cases} \qquad (3)$$

The system of nonlinear weakly singular integral-differential equations (IDE) (3) describes a mathematical model of the problem of dynamic vibration dampers of systems with nonlinear inherited-deformable suspensions.

In case of transient process, initial conditions shall be added to IDE system (3), i.e.:

$$u_i(0) = \alpha_{0i},\ \dot{u}_i(0) = \alpha_{1i}, \quad i = 1,2. \qquad (4)$$

The IDE system (3) is quite general: if the suspensions of the protected structure are perfectly elastic, then $R_1^* = 0$; if the problem is linear, then $\gamma_1 = \gamma_2 = 0$.

2. Methods

Assuming $\tau = \omega_0 t,\ \omega_0 = \sqrt{\frac{m_1}{c_1}}$, we write the equations of vibrations of the protected system and the damper in a dimensionless form:

$$\begin{cases} \ddot{u}_1 + (1 - R_1^*)(u_1 + \gamma_1 u_1^3) + \nu \ddot{u}_2 = q_0 \cdot q(t) \\ \ddot{u}_2 - \mu^2(1 - R_2^*)[u_1 - u_2 + \gamma_2(u_1 - u_2)^3] = 0. \end{cases} \qquad (5)$$

The parameters $= \frac{m_2}{m_1}$, $\mu^2 = \frac{c_2}{m_2}\omega^2$ play a major role in the theory of dynamic vibration dampers (DVD), and are hereinafter referred to as the relative mass and setting of the damper. DVD is a device in which an inertia force occurs that reduces the vibration level of the protected structure. The parameters of the damper should be selected so that they substantially reduce the amplitude or completely extinguish the forced oscillations determined by the first generalized coordinate in the main system I with one degree of freedom, in the case where, in the absence of the damper, a resonance phenomenon would occur at harmonic loads.

The main task of the transient process is that with the help of a computational experiment to find DVD parameters significantly increasing the rate of attenuation of the transient process. Installation of dynamic damper with inherited-deformable suspensions should significantly increase energy dissipation in the system and positively affect transient modes of forced oscillations of the protected structure taking into account and without taking into account inherited-deformable properties of the suspension. To trace this process, consider a numerical solution to the system of nonlinear weak-singular IDEs (5).

By solving the system (5) under initial conditions (4) by the method given in [13, 14], we have:

$$u_{1n} = \alpha_{01} + \alpha_{11} t_n + q_0 \sum_{i=0}^{n-1} A_i (t_n - t_i) q_i -$$

$$-\sum_{i=0}^{n-1} A_i (t_n - t_i) \{u_{1i} + \gamma_1 u_{1i}^3 + \nu f^2 [u_{1i} - u_{2i} + \gamma_2 (u_{1i} - u_{2i})^3]\} +$$

$$+\sum_{i=0}^{n-1} A_i \Gamma_1 (t_n - t_i)(u_{1i} + \gamma_1 u_{1i}^3) + \nu f^2 \sum_{i=0}^{n-1} A_i \Gamma_2 (t_n - t_i)[u_{1i} - u_{2i} + \gamma_2 (u_{1i} - u_{2i})^3]; \quad (6)$$

$$u_{2n} = \alpha_{02} + \alpha_{12} t_n - f^2 \sum_{i=0}^{n-1} A_i \Gamma_2 (t_n - t_i)[u_{1i} - u_{2i} + \gamma_2 (u_{1i} - u_{2i})^3];$$

Here $t_n = n \cdot \Delta t,\ (n = 0,1,2,\dots);\ \ u_{1i} = u_1(t_i);\ \ u_{2i} = u_2(t_i);\ \ q_i = q(t_i);\ A_0 = A_n = \frac{\Delta t}{2};\ A_j = \Delta t,\ \ j = \overline{1, n-1}.$

$$\Gamma_p (t_n - t_i) = \int_0^{t_n - t_i} (t_n - t_i - \tau) R_p(\tau) d\tau,\ \ (p = 1,2); \quad R_p(t) = \varepsilon_p e^{-\beta_p t} \cdot t^{\alpha_p - 1}.$$

3. Results and discussion

A computer program for numerical implementation of the developed calculation algorithm under arbitrary external influences has been compiled. The behavior of the system with a damper at free attenuation oscillations caused by the specified initial conditions is considered. Figures 2-5 show an influence of quencher on free fluctuations of a system under entry conditions of $u_i(0) = 0{,}4\ \ and\ \ \dot{u}_i(0) = 0,\ (i = 1,2)$.

Here and hereinafter, solid and dashed lines correspond to the solution of the problem without a damper ($\nu = 0$) and with a damper.

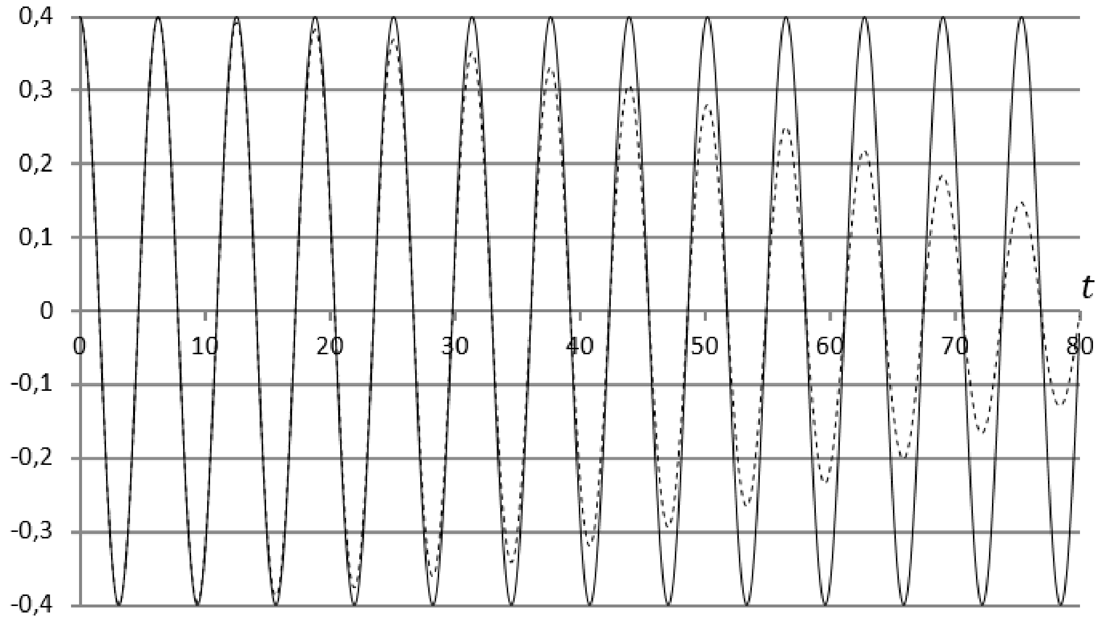

Figure 2. $\alpha = 0.8;\ \ \beta = 0.05;\ \ \varepsilon = 0;\ \ \gamma = 0;\ \ \nu = 0.001\ u_1(t)$

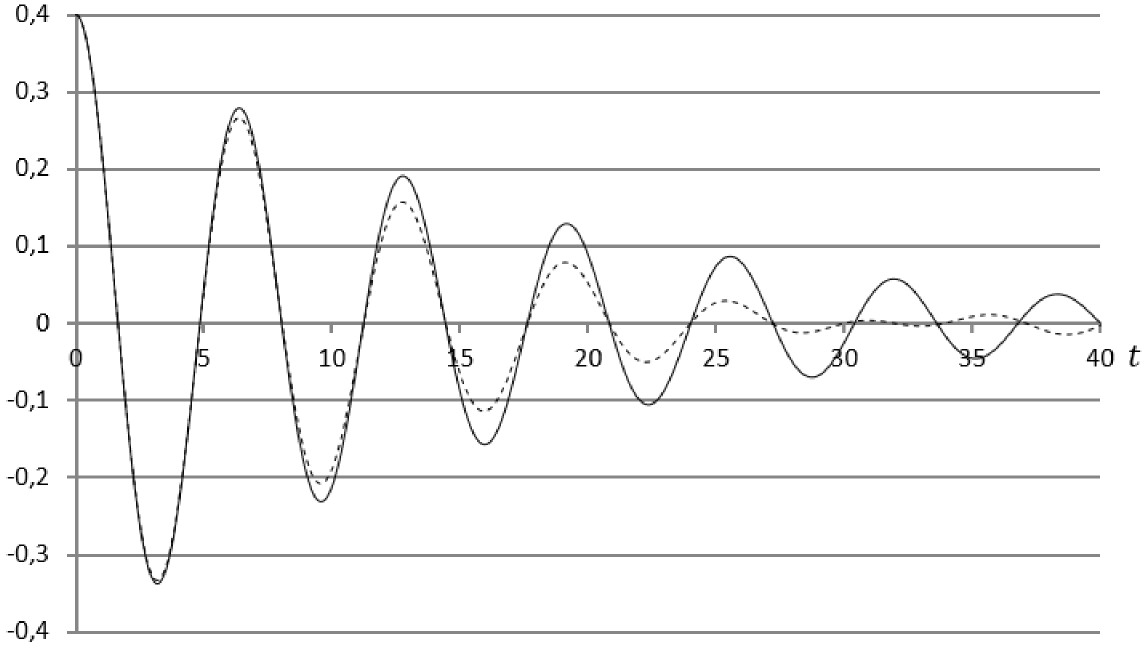

Figure 3. $\alpha = 0.8; \ \beta = 0.05; \ \varepsilon = 0.1; \ \gamma = 0; \ \nu = 0.01$

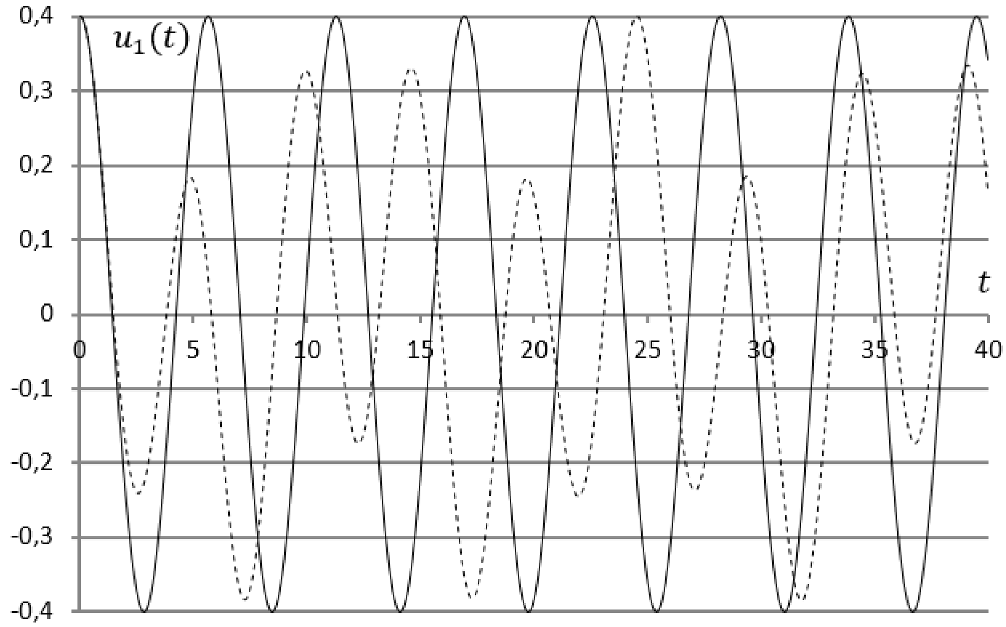

Figure 4. $\alpha = 0.8; \ \beta = 0.05; \ \varepsilon = 0; \ \gamma = 2; \ \nu = 0.001$

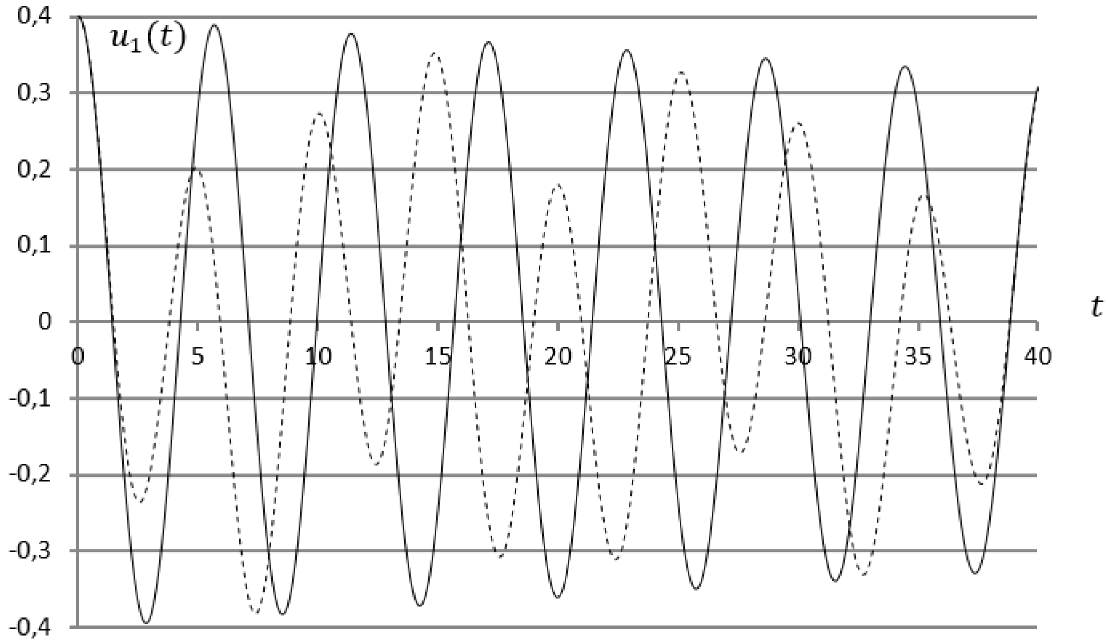

Figure 5. $\alpha = 0.8$; $\beta = 0.05$; $\varepsilon = 0.01$; $\gamma = 2$; $\nu = 1.2$

DVD can be used not only in the case of attenuation free oscillations caused by initial conditions, but also in the case of constant, pulse and periodic external effects. Here, the optimum parameters of the damper are suitably selected from the condition that during the final period of time the fluctuations of the main mass caused by said external effects must be reduced to a predetermined level. In such cases, the transition process accompanied by the beats may be more acceptable than uniformly fading over an infinite period of time.

Figures 6-9 show the effect of DVD on forced oscillations at: $q_0 = 1$; $q = 0.07$; $\nu = 0.001$; 0.05; 0.075; 1; $W_0 = 1$; $q_0 = 1$; $u_1(0) = 0.4$; $u_2(0) = 0.4$; $\dot{u}_1(0) = 0$; $\dot{u}_1(0) = 0$. Similar results are shown in Figures 10-15 at: $q_0 = 1$; $q(t) = sin\theta t$; $\theta = 0, \frac{\pi}{2}, 1$.

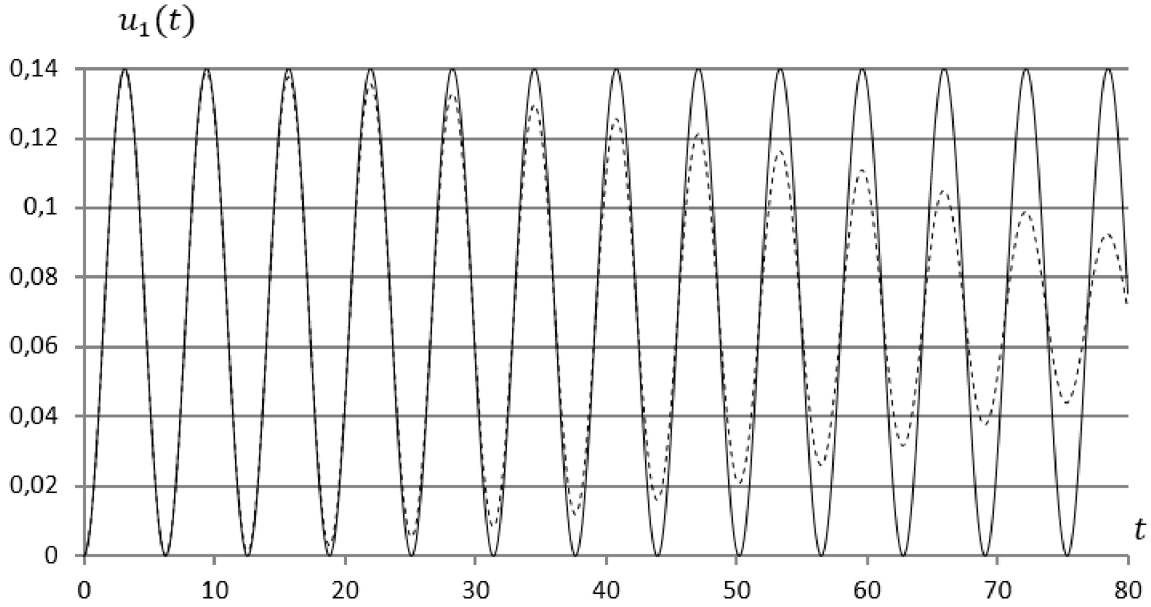

Figure 6. $\alpha = 0.8$; $\beta = 0.05$; $\varepsilon = 0$; $q = 0.07$; $\gamma = 0$; $\nu = 0.001$

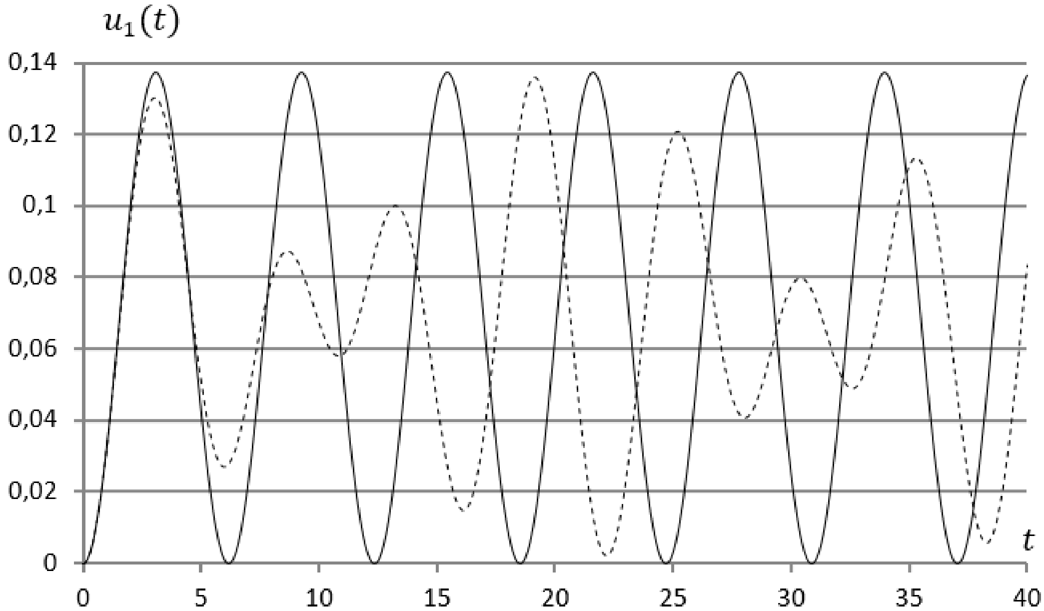

Figure 7. $\alpha = 0.8$; $\beta = 0.05$; $\varepsilon = 0$; $q = 0.07$; $\gamma = 2$; $\nu = 0.001$

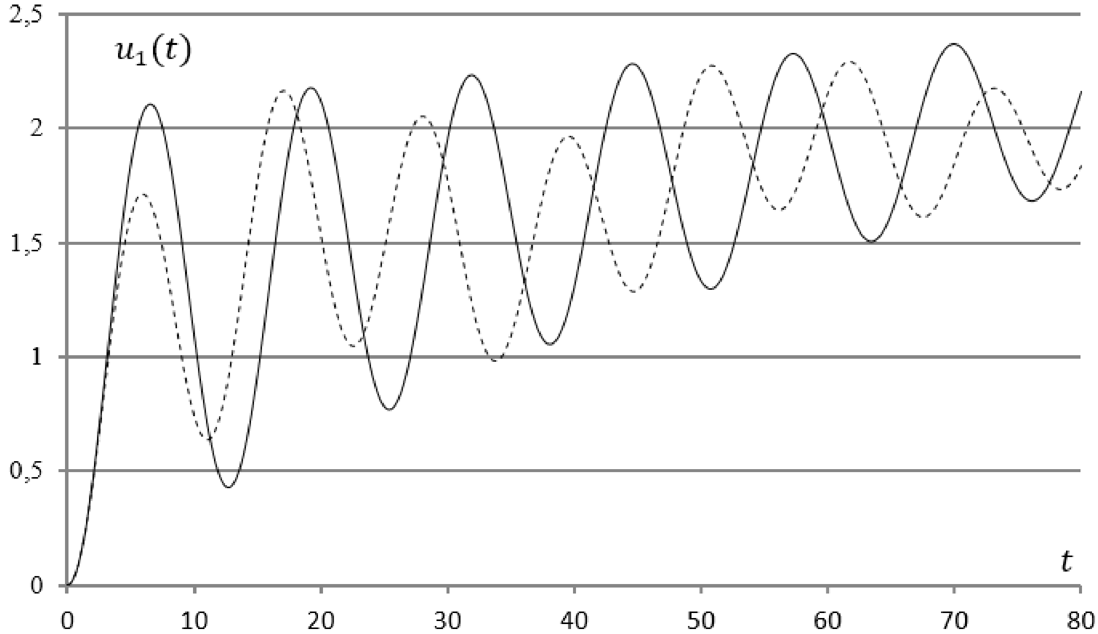

Figure 8. $\alpha = 0.8$; $\beta = 0.05$; $\varepsilon = 0.05$; $q = 1$; $\gamma = 0$; $\nu = 1.5$

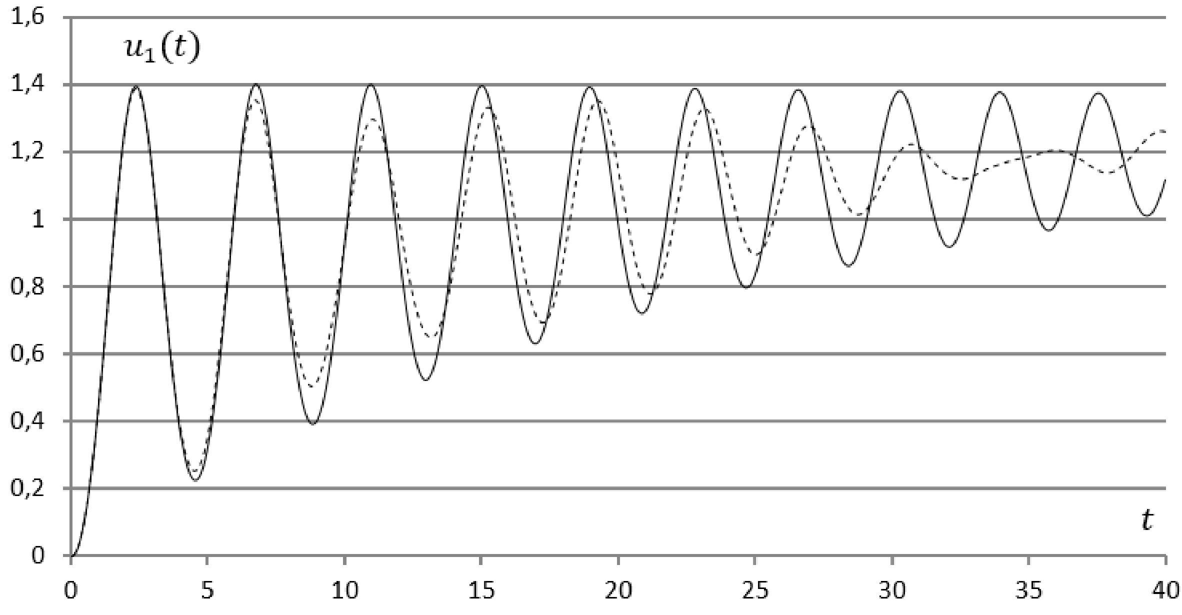

Figure 9. $\alpha = 0.8$; $\beta = 0.05$; $\varepsilon = 0.05$; $q = 1$; $\gamma = 0.5$; $\nu = 0.01$

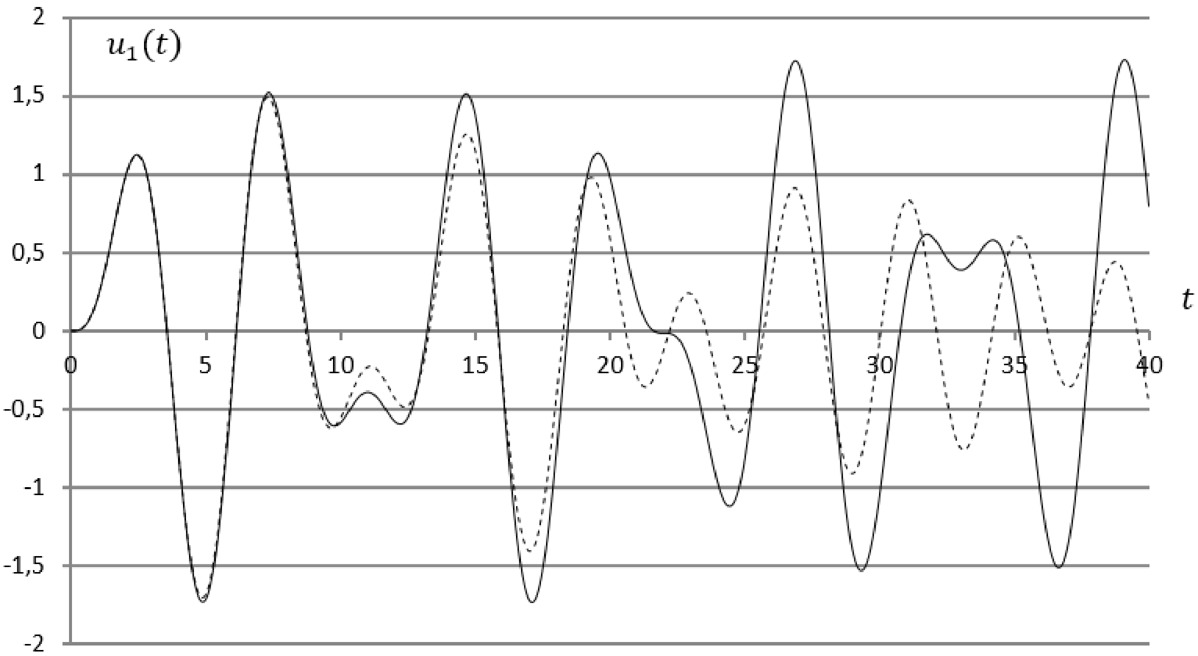

Figure 10. $\alpha = 0.8$; $\beta = 0.05$; $\varepsilon = 0$; $q = \sin\frac{\pi t}{2}$; $\gamma = 0$; $\nu = 0.01$

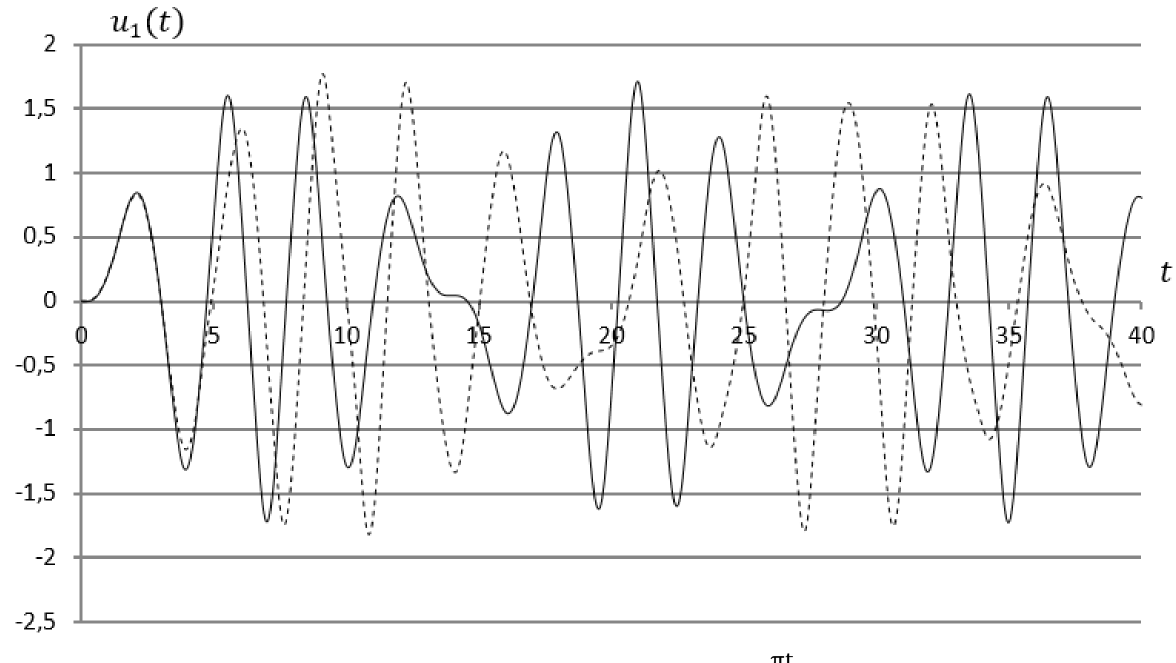

Figure 11. $\alpha = 0.8$; $\beta = 0.05$; $\varepsilon = 0$; $q = \sin\frac{\pi t}{2}$; $\gamma = 2$; $\nu = 0.1$

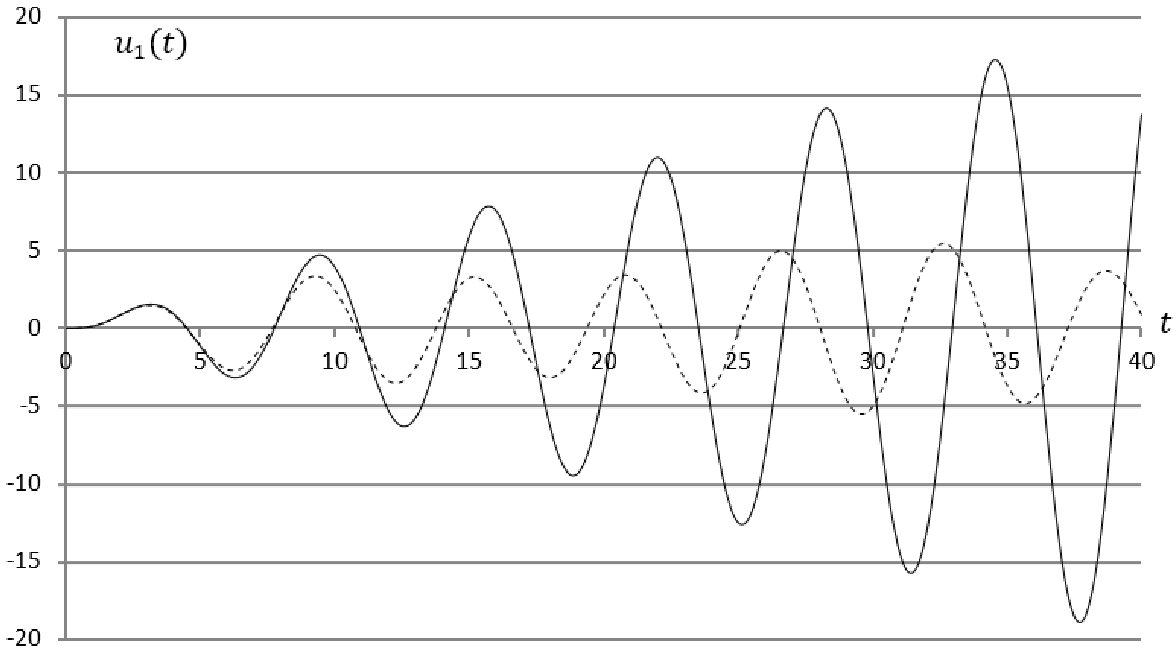

Figure 12. $\alpha = 0.8$; $\beta = 0.05$; $\varepsilon = 0$; $q = \sin\frac{\pi t}{2}$; $\gamma = 2$; $\nu = 0.1$

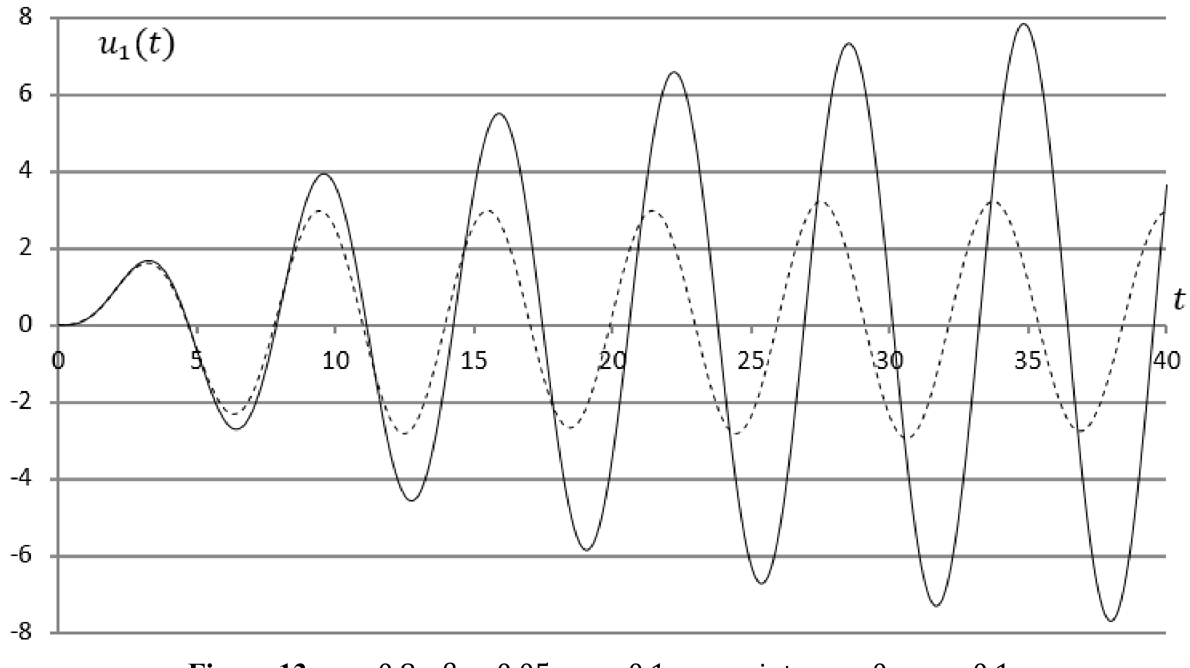

Figure 13. $\alpha = 0.8$; $\beta = 0.05$; $\varepsilon = 0.1$; $q = \sin t$; $\gamma = 0$; $\nu = 0.1$

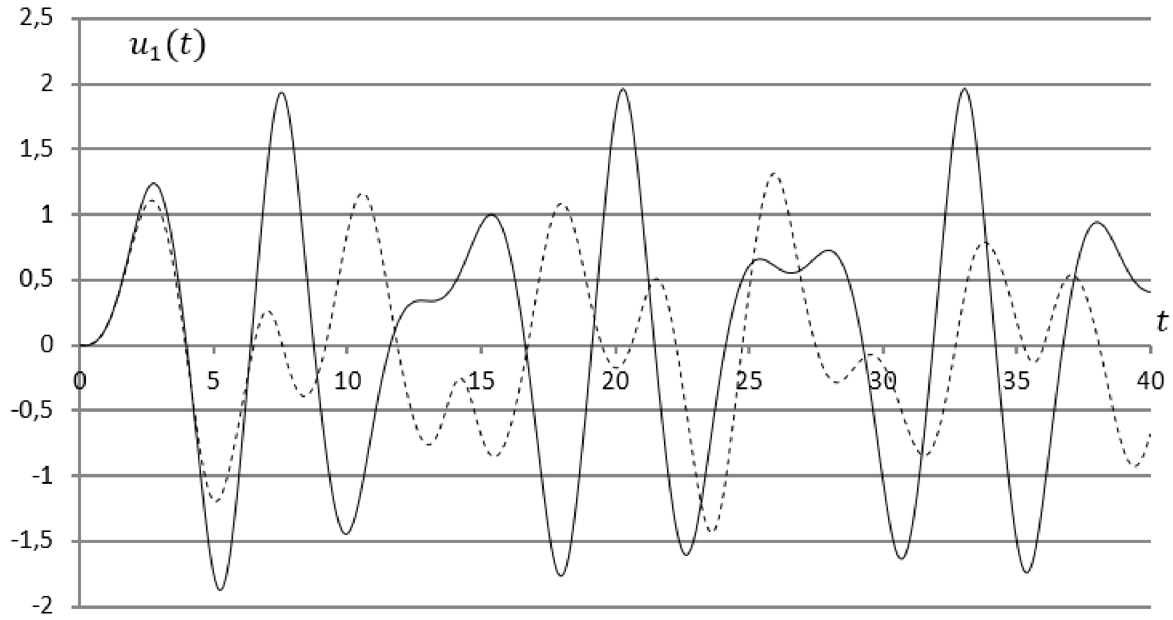

Figure 14. $\alpha = 0.8$; $\beta = 0.05$; $\varepsilon = 0$; $q = \sin t$; $\gamma = 0.5$; $\nu = 0.7$

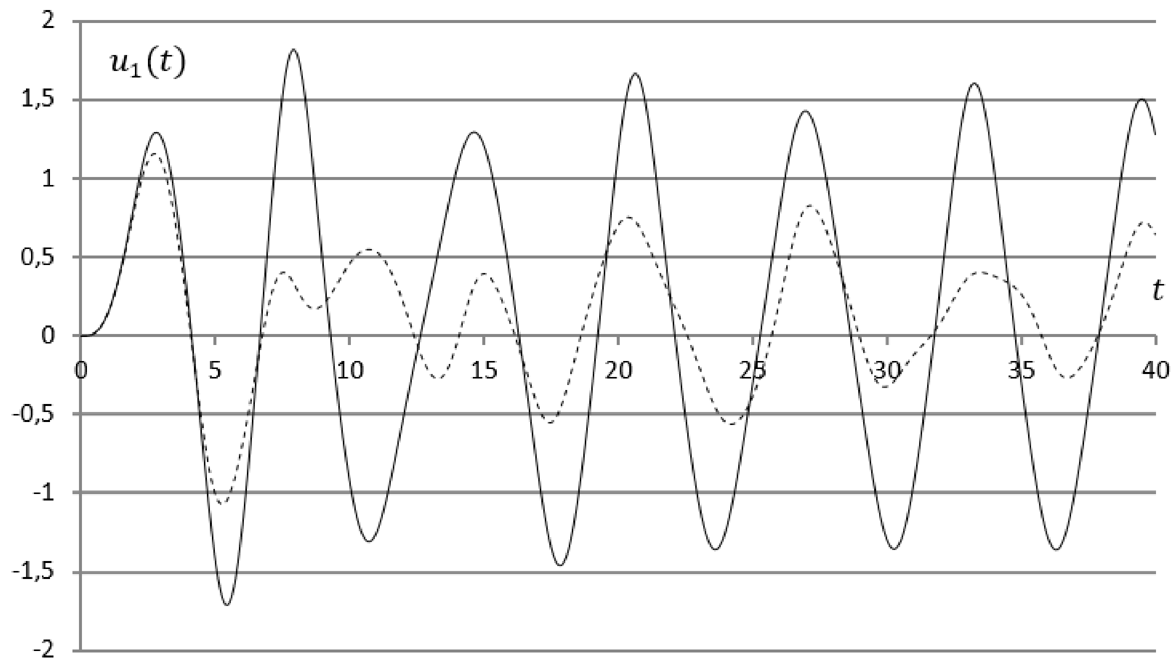

Figure 15. $\alpha = 0.8$; $\beta = 0.05$; $\varepsilon = 0.1$; $q = \sin t$; $\gamma = 0.5$; $\nu = 0.7$

4. Conclusions

Thus, the solution of vibration damping problems is connected with the need to carry out repeated calculations in the process of optimization of damper parameters. Therefore, in some cases it is advisable to carry out preliminary calculations according to simplified calculation schemes to determine the estimated efficiency and parameters of the vibration protection system. The use of circuits capable of obtaining a solution in closed form or using algorithms of type (6) is of great interest. It is these possibilities that provide a significant part of this work, not to mention, of course, when the design is directly reflected by the inherited-deformable models discussed herein. The

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obtained results make it possible to conclude on the expediency of using dynamic dampers to reduce amplitude of oscillations, both in perfectly elastic and in inherited-deformable systems during transient processes.

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