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# CALCULATION OF SHOCK VIBRATION SUPPRESSORS OF UNILATERAL ACTION

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**Abstract**

The method of calculation of shock spring vibration suppressors of unilateral action at harmonic disturbance is considered. The problems arising at mathematical model operation of such systems are noted. In the available techniques, usually assume that during change of external indignation there is one impact of an object and a suppressor. Cases in which it is impossible to use the specified assumption are given. The calculation algorithm considering a possibility of several impacts for one frequency period is offered. Laws of motion of an object and suppressor are defined by addition of the equations of the compelled and free fluctuations. The stereomechanical model of blow is applied to accounting of shock interaction of bodies. The technique allows calculating time between impacts, the period of change of coordinates of bodies and the number of impacts for frequency period. The example of a duty of a shock suppressor at which for frequency period there are several impacts of an object and suppressor is given

**Keywords**

*Shock vibration suppressor, free fluctuations, forced oscillations, coefficient of restitution, characteristic equation, partial frequency*

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**Introduction.** The creation of high-performance and high-speed machines leads to an increase in the vibrations of their components and parts. Increased vibroactivity violates the laws of the links motion, increases the dynamic loads on the parts and the engine, and has a harmful effect on the human body.

Vibration activity can be reduced by various methods:

– balancing the mechanism and balancing the rotor;

– changing in the frequency spectrum of the structure;

– connection an additional elastic system to the structure (dynamic damping);

– installation the elastic system between the protected object and the source of oscillations (vibration isolation).

This article discusses the reduction of vibrations using shock suppressors.

A characteristic feature of shock vibration suppressors is the presence of collisions between the suppressor and the protected system. By the principle of operation, they are divided into suppressors of the floating, spring and pendulum types; by the gap size, they are divided into the suppressors with positive and zero gaps and tension. The advantages of shock suppressors include simplicity of design and ease of operation.

Spring shock suppressors of unilateral action are usually used for damping oscillations with a frequency of 2 to 10 Hz and amplitude varying within wide limits. If the system is excited by harmonic force, then the suppressor parameters are selected from the condition of the highest efficiency at resonance [1]. Disturbing force may have an unstable frequency. Motors with variable speed cause such an effect on an object, for example. In the system, the initial gap $\Delta_0$ (between the suppressors and the object in the undeformed system) is usually taken to be close to zero. In this case, the effectiveness of damping and the impossibility of oscillations without collisions are provided. Negative initial clearance is used in systems in which it is necessary to turn on a suppressor at a given level of excitation [1].

**Task definition.** Shock suppressors relate to systems with one-way links [2]. The work of [3–11] is devoted to the calculation of such suppressors, in which the fitting method [4], the approximate harmonic linearization method [7], etc. were used for the solution. In a number of works [6, 8–11], a stereomechanical impact model is presented. Such a model can be used for suppressors, in which the time of impact of system elements is short compared with the period of oscillation of the external load. According to this model, instantaneous pulses replace the impact on the oscillatory system of body collisions. The magnitude of the pulses depends on the recovery rate of the speed, taking values from zero to one.

In [9–11], the law of motion of an object and a suppressor is determined by summing the equations of forced and free vibrations. Forced oscillations occur under the action of an external periodic force and free oscillations arise from the action of an instantaneous shock pulse. The magnitude of the pulse is determined by the theory of direct central impact [12]. The necessary equations are first written in a complex form, and then transformed into real form using algebraic transformations. The vibrations of the shock suppressors are considered for one period of external influence. It is assumed that during this period only one collision of bodies occurs. Arbitrary constants are determined from the condition of periodicity. By the time of calculation, the method considered in [11] is much more efficient than the method of fitting [4, 13].

Analysis of the shock suppressors with a negative initial gap $\Delta_0$ shows that for small amplitudes of the disturbing force the shock suppressor does not turn on (the object and the suppressor move together), and for large amplitudes it works with one blow during the period $T$ of the disturbing force. In the transition from small to large amplitudes, regimes can appear in which several body collisions occur over the period $T$, and the oscillation period of the system $T_c = kT$, where $k$ is integer. In this case, the technique proposed in [11] cannot be used. Since, firstly, when using it at a certain time interval, the calculated gap $\Delta(t)$ between the object and the suppressor may be less than zero, which does not correspond to the condition of their interaction $\Delta(t) \geq 0$. Secondly, equation expressing the condition of periodicity of oscillations may not have a solution.

**The purpose of this work** is to develop a method for calculating shock suppressors using an stereomechanical model of impact and taking into account the possibility of several impacts in one oscillation period.

The considered technique is a development of the method used by A.V. Dukart [11]. Unlike works [5, 6, 11], calculations are performed in a complex form, which significantly reduces the amount of analytical transformations.

**Mathematical model.** Consider a vibration protection object with one degree of freedom; to reduce oscillations of an object; a shock spring suppressor of unilateral action is used. The design scheme is shown in Fig. 1. The following notations are entered in the diagram: *1* is the object to be protected; *2* is the vibration suppressor; $m_1$, $m_2$ are the masses of the object and suppressor; $c_1$, $c_2$ are stiffness coefficients; $k_1$, $k_2$ are the coefficients of viscous friction; $P(t)$ is the function of external influence; $x_1$, $x_2$ are the coordinates of the mass of the protected object and the mass of the oscillation suppressor in a fixed coordinate system; $\Delta$ is the gap between the suppressor and the object. The coordinates of the bodies are measured from the position in which the elastic elements are in the undeformed state.

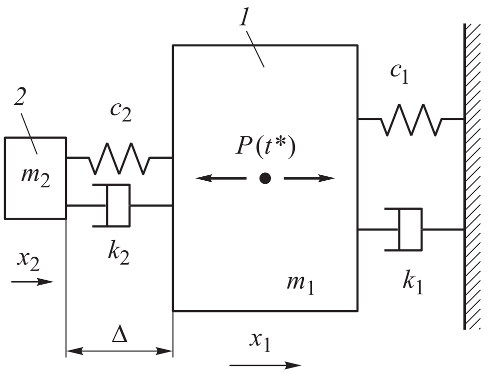


**Fig. 1.** The design scheme of shock spring suppressor of unilateral action

The law of motion of this system between body collisions is determined by the addition of the equations of forced and free vibrations:

$$x_1(t) = x_{1p}(t) + Sx_{1s}(t); \quad x_2(t) = x_{2p}(t) + Sx_{2s}(t), \tag{1}$$

where $x_{1p}(t)$, $x_{2p}(t)$ are the laws of forced oscillations of an object and suppressor, made under the action of force $P\,(t^*)$; $x_{1s}\,(t)$ and $x_{2s}\,(t)$ are the laws of free oscillations of the object and suppressor from the impact of a single shock pulse; $S$ is the impulse value at the moment of impact $t^* = t_y^*$; $t = t^* - t_y^*$ is the time counted from the moment of impact.

Let us determine the laws of motion of bodies from the action of two opposite single impulses $S = 1$. In the interval between collisions, the system performs free oscillations, the differential equations of which have the form

$$\begin{aligned} &m_1\ddot{x}_{1s} + k_1\dot{x}_{1s} + k_2(\dot{x}_{1s} - \dot{x}_{2s}) + c_1x_{1s} + c_2(x_{1s} - x_{2s}) = 0;\\ &m_2\ddot{x}_{2s} + k_2(\dot{x}_{2s} - \dot{x}_{1s}) + c_2(x_{2s} - x_{1s}) = 0. \end{aligned} \tag{2}$$

Equations (2) form a system of two linear homogeneous differential equations with constant coefficients. The solution to this system is as follows [14]:

$$x_{1s} = e^{\lambda t},\ x_{2s} = Ae^{\lambda t}, \tag{3}$$

where $\lambda$ is some unknown constant.

Substituting the solution (3) into the system (2) and reducing the resulting expression by $e^{\lambda t}$, we obtain the characteristic equation and the formula for determining the coefficient $A$:

$$d_5\lambda^4 + d_4\lambda^3 + d_3\lambda^2 + d_2\lambda + d_1 = 0; \tag{4}$$

$$A = \frac{2h_2\lambda + \omega_2^2}{\lambda^2 + 2h_2\lambda + \omega_2^2}, \tag{5}$$

where $d_1 = m_1\omega_1^2\omega_2^2$; $d_2 = 2m_1\left(\omega_2^2h_1 + \omega_1^2h_2\right)$; $d_3 = m_1\left(\omega_1^2 + 4h_1h_2\right) + \omega_2^2(m_1 + m_2)$; $d_4 = 2\left[m_1h_1 + h_2(m_1 + m_2)\right]$; $d_5 = m_1$; $\omega_1 = \sqrt{c_1/m_1}$ is the natural frequency of oscillation of an object without a suppressor; $\omega_2 = \sqrt{c_2/m_2}$ is the partial damping oscillation frequency; $h_1 = k_1/(2m_1)$, $h_2 = k_2/(2m_2)$ are attenuation coefficients.

If the characteristic equation (4) has simple roots, the general solution of system (2) is a combination of linearly independent solutions of the fundamental system

$$\begin{aligned} x_{1s}(t) &= \sum_{j=1}^{4}\alpha_j e^{\lambda_j t};\\ x_{2s}(t) &= \sum_{j=1}^{4}\alpha_j e^{\lambda_j t}. \end{aligned} \tag{6}$$

Here $\alpha_j$ are arbitrary constants determined from the condition of periodicity of motion; $\lambda_j$ are roots of the characteristic equation (4); $A_j$ are coefficients determined by the formula (5).

We assume that the functions $x_{1s}(t)$ and $x_{2s}(t)$ change with a period equal to the period of change of the external force $T$. Having assumed $t = 0$ at the moment of body impact, we get the periodicity conditions:

$$\begin{aligned} &x_{1s}(T) = x_{1s}(0); \quad m_1\dot{x}_{1s}(T) = m_1\dot{x}_{1s}(0) - 1; \\ &x_{2s}(T) = x_{2s}(0); \quad m_2\dot{x}_{2s}(T) = m_2\dot{x}_{2s}(0) + 1. \end{aligned} \tag{7}$$

Substituting (6) into (7), we obtain a linear system of algebraic equations with complex coefficients, from which we define $\sigma_j$:

$$\begin{aligned} &\sum_{j=1}^{4} \alpha_j (e^{\lambda_j T} - 1) = 0; \\ &m_1 \sum_{j=1}^{4} \alpha_j \lambda_j (e^{\lambda_j T} - 1) = -1; \\ &\sum_{j=1}^{4} \alpha_j A_j (e^{\lambda_j T} - 1) = 0; \\ &m_2 \sum_{j=1}^{4} \alpha_j \lambda_j A_j (e^{\lambda_j T} - 1) = 1. \end{aligned} \tag{8}$$

It should be noted that for the considered elastic systems, the roots $\lambda_j$ ($j = 1, \ldots, 4$) of the characteristic equation (4) and the corresponding values of $A_j$ and $\alpha_j$ are complex conjugate. Therefore, functions (6) expressing the response of the system to the action of two opposite single impulses are real.

Now let us consider the forced oscillations of the system, performed under the action of an external periodic force $P(t^*) = P_0 \sin \omega t^*$, where $\omega = 2\pi / T$. In this case, the external force $P(t^*)$ is added to the system of differential equations of oscillations of the object and suppressor:

$$\begin{aligned} &m_1\ddot{x}_{1p} + k_1\dot{x}_{1p} + k_2(\dot{x}_{1p} - \dot{x}_{2p}) + c_1 x_{1p} + c_2 (x_{1p} - x_{2p}) = P_0 \sin(\omega t^*); \\ &m_2\ddot{x}_{2p} + k_2(\dot{x}_{2p} - \dot{x}_{1p}) + c_2(x_{2p} - x_{1p}) = 0. \end{aligned} \tag{9}$$

To solve system (9), we apply the complex amplitude method [1]

$$\begin{pmatrix} -m_1\omega^2 + i\omega(k_1 + k_2) + (c_1 + c_2) & -i\omega k_2 - c_2 \\ -k_2 i\omega - c_2 & -m_2\omega^2 + i\omega k_2 + c_2 \end{pmatrix} \begin{pmatrix} \hat{x}_{1p} \\ \hat{x}_{2p} \end{pmatrix} = \begin{pmatrix} P_0 \\ 0 \end{pmatrix}. \tag{10}$$

From the written system of algebraic equations (10) we determine the complex amplitudes of the oscillations $\hat{x}_{1p}$ and $\hat{x}_{2p}$. The law of forced oscillations flowing in the system under the action of an external periodic force $P(t)$ is expressed in terms of the complex amplitudes obtained. Turning to the countdown of time from the moment of the collision of an object and a suppressor, we obtain:

$$\begin{aligned} x_{1p}(t) &= \left|\hat{x}_{1p}\right| \sin(\omega t + \varepsilon_1 + \varepsilon); \\ x_{2p}(t) &= \left|\hat{x}_{2p}\right| \sin(\omega t + \varepsilon_2 + \varepsilon), \end{aligned} \tag{11}$$

where $\varepsilon_1 = \arg(\hat{x}_{1p})$, $\varepsilon_2 = \arg(\hat{x}_{2p})$ are the initial phases of oscillation of the object and suppressor; $\varepsilon = \omega t_y^*$ is unknown phase corresponding to the moment of impact of the object and suppressor.

The position of the suppressor relative to the object is determined by the coordinate

$$x_{21}(t) = x_{21p}(t) + S x_{21s}(t), \tag{12}$$

where

$$x_{21p}(t) = x_{2p}(t) - x_{1p}(t) = \left|\hat{x}_{2p} - \hat{x}_{1p}\right| \sin(\omega t + \varepsilon_{21} + \varepsilon),$$

$$\varepsilon_{21} = \arg(\hat{x}_{2p} - \hat{x}_{1p}); \;\; x_{21s}(t) = x_{2s}(t) - x_{1s}(t).$$

Differentiating (12) and (6) by time, we obtain the formula for calculating the speed of the suppressor *2* relative to object *1* (Fig. 1):

$$\begin{aligned} &\dot{x}_{21}(t) = \dot{x}_{21p}(t) + S\,\dot{x}_{21s}(t); \\ &\dot{x}_{21s}(t) = \sum_{j=1}^{4} \alpha_j \lambda_j e^{\lambda_j t} - \sum_{j=1}^{4} \alpha_j A_j \lambda_j e^{\lambda_j t}; \\ &\dot{x}_{21p}(t) = \left|\hat{x}_{2p} - \hat{x}_{1p}\right| \omega \cos(\omega t + \varepsilon_{21} + \varepsilon). \end{aligned} \tag{13}$$

With a direct central strike [12] the momentum

$$S = D\dot{x}_{21}(T), \tag{14}$$

where $\dot{x}_{21}(T)$ is the relative velocity at the moment of impact; $D = \dfrac{m_1 m_2}{m_1 + m_2}(R+1)$ ($R$ is the impact recovery rate).

Substituting (13) into (14), we have

$$S = \frac{D\dot{x}_{21p}(T)}{1 - D\dot{x}_{21s}(T)}. \tag{15}$$

At the moment of the beginning of the strike ($t = T$), the gap between the object and the suppressor is zero $\Delta = \Delta_0 - x_{21}(T) = 0$. Taking into account (12) and (15), we obtain the formula for determining the unknown phase ε:

$$U_1 \sin(\varepsilon + \varepsilon_{21}) + U_2 \cos(\varepsilon + \varepsilon_{21}) = \Delta_0;$$
$$U_1 = \left|\hat{x}_{2p} - \hat{x}_{1p}\right|, \quad U_2 = \frac{D\, x_{21s}(T)}{1 - D\, \dot{x}_{21s}(T)} \left|\hat{x}_{2p} - \hat{x}_{1p}\right| \omega. \tag{16}$$

The resulting equation on the interval $t \in (0, T)$ has two solutions

$$\varepsilon = \begin{cases} \arcsin\left(\dfrac{\Delta_0}{\sqrt{U_1^2 + U_2^2}}\right) - \zeta - \varepsilon_{21}; \\ \pi - \arcsin\left(\dfrac{\Delta_0}{\sqrt{U_1^2 + U_2^2}}\right) - \zeta - \varepsilon_{21}, \end{cases} \tag{17}$$

where $\cos(\zeta) = \dfrac{U_1}{\sqrt{U_1^2 + U_2^2}}$; $\sin(\zeta) = \dfrac{U_2}{\sqrt{U_1^2 + U_2^2}}$.

The parameter ε, that have the positive momentum $s$ (15), is chosen.

Number 1 denotes the calculation algorithm considered earlier. If, as a result of the calculation, we obtain that $\Delta_0 \big/ \sqrt{U_1^2 + U_2^2} > 1$, then the oscillation periodicity equation (16) has no solution and, therefore, algorithm No. 1 cannot be used.

After calculations, it is necessary to check the gap values $\Delta(t)$ on the period of external influence $t \in (0, T)$. If at any time interval it turns out that the value $\Delta(t) < 0$, then the resulting solution incorrectly reflects the interaction of the object and the suppressor. In this case, the calculation algorithm based on the fit method [13] and the stereomechanical impact model [10, 11] can be applied. In the following, we denote this algorithm by No. 2.

The calculating algorithm No. 2.

1. $k = 1$. Initial conditions are set: the values of the coordinates $x_{1b}$, $x_{2b}$ and $\dot{x}_{1b}$, $\dot{x}_{2b}$ velocities of the object and suppressor. The coordinates must be set so that the gap $\Delta = 0$, and the speeds so that the object and suppressor pulses are equal, oppositely directed and lead to an increase in the gap. For example, $x_{1b} = 0$, $x_{2b} = \Delta_0$, $\dot{x}_{1b} = 1/m_1$, $\dot{x}_{2b} = 1/m_2$.

2. Determination of the law of motion of this system after the $k$-th collision $t > t_{ck}$:

$$x_{1k}(t) = x_{1p}(t) + \sum_{j=1}^{4} \alpha_j \exp\left[\lambda_j (t - t_{ck})\right];$$
$$x_{2k}(t) = x_{2p}(t) + \sum_{j=1}^{4} \alpha_j A_j \exp\left[\lambda_j (t - t_{ck})\right], \quad (18)$$

where $t_{ck}$ is the time value at the moment of the $k$-th collision $(t_{c1} = 0)$.

The coefficients $\alpha_j$ are found from the initial conditions that express the values of the coordinates and velocities of the bodies at the end of the $k$-th collision $t = t_{ck}$:

$$x_{1p}(t_{ck}) + \sum_{j=1}^{4} \alpha_j = x_{1b};$$
$$\dot{x}_{1p}(t_{ck}) + \sum_{j=1}^{4} \alpha_j \lambda_j = \dot{x}_{1b};$$
$$x_{2p}(t_{ck}) + \sum_{j=1}^{4} \alpha_j A_j = x_{2b};$$
$$\dot{x}_{2p}(t_{ck}) + \sum_{j=1}^{4} \alpha_j A_j \lambda_j = \dot{x}_{2b}. \quad (19)$$

3. The time value $t_{ck+1}$ at the time of the next collision is found from the condition that the gap between the bodies of the system is zero:

$$\Delta(t_{ck+1}) = \Delta_0 - \left[x_{2k}(t_{ck+1}) - x_{1k}(t_{ck+1})\right] = 0. \quad (20)$$

When solving equation (20), the root $(t_{ck+1} > t_{ck})$ is selected which is closest to the time value of the previous collision $t_{ck}$.

4. Determination of the shock pulse by the equation

$$S = \frac{m_1 m_2}{m_1 + m_2}(R+1)\left[\dot{x}_{2k}(t_{ck+1}) - \dot{x}_{1k}(t_{ck+1})\right]. \quad (21)$$

5. The calculation of the coordinates and speeds at the beginning of the motion after the ($k$+1)-th collision with $t = t_{ck+1}$:

$$x_{1b} = x_{1k}(t_{ck+1}); \quad x_{2b} = x_{2k}(t_{ck+1});$$
$$\dot{x}_{1b} = \dot{x}_{1k}(t_{ck+1}) + \frac{S}{m_1}; \quad \dot{x}_{2b} = \dot{x}_{2k}(t_{ck+1}) - \frac{S}{m_2}. \quad (22)$$

6. $k = k + 1$. The calculation is repeated from point 2.

Calculations continue until established oscillations occur.

**The results of the calculations.** Calculations of dynamic suppressors are carried out in relative values [11]:

$$\mu = \frac{m_2}{m_1}; \quad \theta = \frac{\omega}{\omega_1}; \quad N = \frac{\omega_2}{\omega_1}; \quad \delta_1 = \frac{k_1}{2m_1\omega_1}; \quad \delta_2 = \frac{k_2}{2m_2\omega_1}; \quad x_{st} = P_0/c_1;$$

$$\overline{x}_1 = x_1/x_{st}; \quad \overline{x}_2 = (x_2 - \Delta_0)/x_{st}.$$

We consider the calculation results for the case when the condition of the effective suppressor setting is satisfied

$$\left(2\sqrt{c_2/m_2} = \sqrt{c_1/m_1} = \omega\right) : \theta = 1; \; \mu = 0,08; \; N = 0,5; \; \delta_1 = \delta_2 = 0,05;$$

$$x_{st} = 1; \; R = 0,7; \; \Delta_0/x_{st} = -5,8; \; \omega = 10 \text{ rad/s}.$$

Fig. 2, *a* shows graphs of changes in the relative coordinates of the object and the suppressor, obtained using algorithm No. 1.

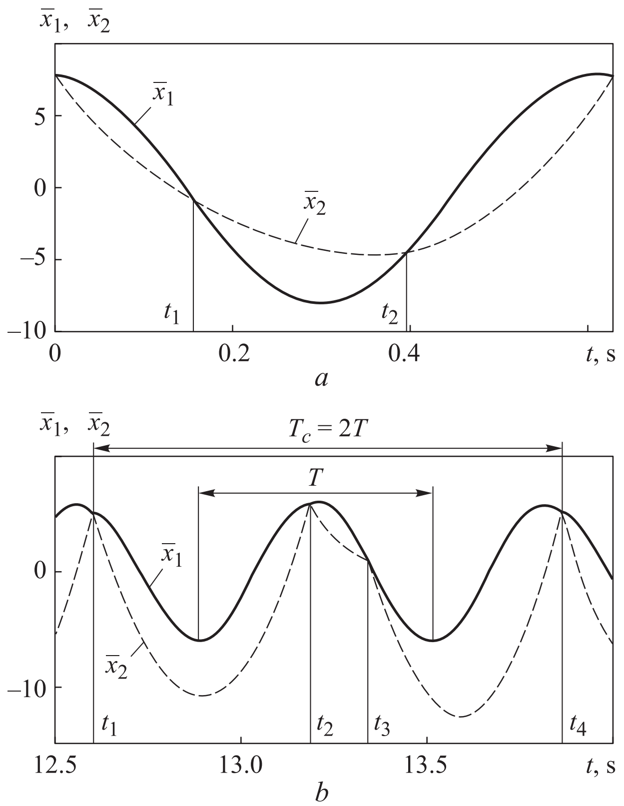


**Fig. 2.** The laws of motion of an object $\overline{x}_1(t)$ and a suppressor $\overline{x}_2(t)$, obtained using algorithm No. 1 (*a*) and No. 2 (*b*)

The gap between the object and the suppressor can be expressed in terms of relative coordinates $\Delta = (\overline{x}_1 - \overline{x}_2)\, x_{st}$. From the dependences presented it follows that the gap $\Delta$ takes on negative values when $t \in (t_1, t_2)$. Consequently, the use of algorithm No. 1 for calculating the shock suppressor leads in this case to obtaining the distorted laws of the motion of bodies.

From the dependences obtained using algorithm No. 2 (Fig. 2, *b*), it follows that the oscillation period of the system is equal to two periods of external influence $T_c = 2T$ and during this period $T_c$ there are three collisions of the object and suppressor at times $t_1$, $t_2$, $t_3$, woth $t = t_4$ the next cycle of interaction of the indicated bodies begins.

**Conclusion.** In a shock spring suppressor of unilateral action, a mode of operation may appear in which several body collisions occur over the period $T_c$ of the change in the coordinates of the system bodies. In this case, the period $T_c$ may not be equal to the period $T$ of the change in the external periodic force.

The specified mode of operation may occur at negative values of the initial gap between the object and the suppressor.

The proposed method of calculating shock a suppressors of oscillations takes into account the possibility of several body collisions during the oscillation period, allows identifying and, if possible, preventing the work of shock suppressors in the specified mode of operation.

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