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Dynamic damping under introduction of additional couplings and external actions*

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Динамическое гашение колебаний при введении дополнительных связей и внешних воздействий***

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Introduction. The dynamic interaction features in mechanical oscillating systems, whose structure includes additional couplings, are considered. In practice, such cases occur when using various optional mechanisms and motion translation devices under the formation of technical objects. The study objective is to develop a method for constructing mathematical models in the problems of dynamics of the mechanical oscillating systems with optional devices and features in the system of external disturbing factors.
Materials and Methods. The techniques used to study properties of the systems and the dynamic effects are based on the ideas of structural mathematical modeling. It is believed that the mechanical oscillating system, considered as a design model of a technical object, can be compared to the dynamically equivalent automatic control system. The mathematical apparatus of the automatic control theory is used.
Research Results. A method for constructing mathematical models is developed. The essential analytical relations for plotting oscillating systems are obtained, which enable to form a methodological basis for the integral estimation and comparative analysis of the initial system properties in various dynamic states. Dynamic properties of the two-degree-of-freedom systems within the framework of the computer simulation are investigated. The implementability of dynamic oscillation damping mode simultaneously in two coordinates with the joint action of two in-phase kinematic perturbations in the mechanical oscillating systems is shown.
Discussion and Conclusions. The possibilities of new dynamic effects, which are associated with the change in the system

Введение. Рассматриваются особенности динамических взаимодействий в механических колебательных системах, в структуре которых имеются дополнительные связи. Практически такие ситуации возникают при использовании в формировании технических объектов различных дополнительных механизмов и устройств для преобразования движения. Цель исследования заключается в разработке метода построения математических моделей в задачах динамики механических колебательных систем с дополнительными устройствами и особенностями в системе внешних возмущающих факторов.
Методы, используемые для исследования свойств систем и изучения динамических эффектов, основаны на идеях структурного математического моделирования. Полагается, что механической колебательной системе, рассматриваемой в качестве расчетной схемы технического объекта, можно сопоставить эквивалентную в динамическом отношении систему автоматического управления. Используется математический аппарат теории автоматического управления.
Результаты исследования. Разработан метод построения математических моделей. Получены необходимые аналитические соотношения для построения частотных диаграмм колебательных систем, позволяющие сформировать методологическую основу для интегральной оценки и сравнительного анализа свойств исходных систем в различных динамических состояниях. Проведены исследования динамических свойств систем с двумя степенями свободы в рамках вычислительного моделирования. Доказаны возможности реализации в механических колебательных системах режимов динамического гашения колебаний одновременно по двум координатам при совместном действии двух синфазных кинематических возмущений.
Обсуждение и заключения. Отмечены возможности проявления новых динамических эффектов, которые связаны с изменением структуры системы при определенных формах динамических взаимодействий.

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structure under certain forms of dynamic interactions, are noted. The study is of interest to experts in machine dynamics, robotics, mechatronics, nano and mesomechanics.

Работа представляет интерес для специалистов в области динамики машин, робототехники, мехатроники, нано- и мезомеханики.

Keywords: structure diagrams, transfer functions,frequency plots, dynamic damping, additional couplings, joint interactions

Ключевые слова: структурные схемы,передаточные функции, частотные диаграммы, динамическое гашение колебаний, дополнительные связи, совместные взаимодействия.

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Introduction. Dynamic oscillation damping (DOD) is widely used in practice to change the state of technical objects and the local action on the forms of component interaction of mechanical oscillatory systems. In the papers [1–3], the features of the approaches to the implementation of the dynamic oscillation damping modes were considered, options of the design and engineering solutions and calculation technique of the system parameters were proposed.

A variety of dynamic tasks predetermines a wide variability of the proposed solutions, within the framework of which the dynamic features of the protected objects, the conditions of external disturbances of the original system and the design and engineering modes of the dynamic oscillation damping are considered [4–8].

DOD is used in the tasks of protecting instrumentation systems and tooling [9, 10]. However, some aspects of evaluating the dynamic properties of vibration protection systems have not obtained a proper level and detail of representations in the formulation of research tasks and parameter analysis for dynamic absorbers. This may be due to the introduction and use of additional links, considering the characteristics of external influences, as well as the effect of the simultaneous joint action of several external disturbing factors.

In this paper, we develop a method of constructing mathematical models and the formation of dynamic oscillation damping effects in the chain mechanical oscillating two-degrees-of-freedom systems.

I. Background. The original system is shown in Fig. 1. It represents two inertia members (m_1 and m_2), which are interconnected by the elastic elements with rigidities (k_1, k_2, k_3) and additional links in the form of motion translation devices (MTD) with reduced masses (L_1, L_2, L_3). The system performs small oscillations under the action of external in-phase harmonic effects. Resistant forces are not considered.

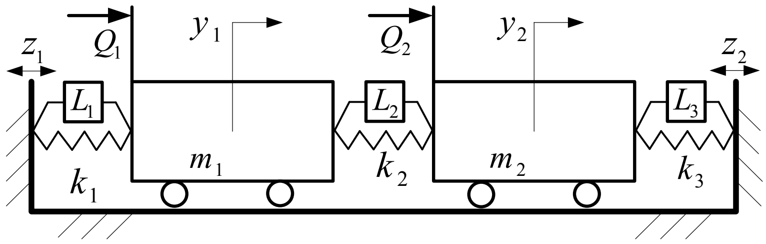

Fig. 1. Flowsheet of technical object in form of two-degrees-of-freedom lumped system

To describe the motion, the coordinate system (y_1, y_2) is used in a fixed basis. Suppose that the kinetic and potential energies of the system are determined by the following expressions:

$$\mathrm{T} = \frac{1}{2}m_1\dot{y}_1^2 + \frac{1}{2}m_2\dot{y}_2^2 + \frac{1}{2}L_1(\dot{y}_1 - \dot{z}_1)^2 + \frac{1}{2}L_2(\dot{y}_2 - \dot{y}_1)^2 + \frac{1}{2}L_3(\dot{y}_2 - \dot{z}_2)^2, \quad (1)$$

$$\Pi = \frac{1}{2}k_1(y_1 - z_1)^2 + \frac{1}{2}k_2(y_2 - y_1)^2 + \frac{1}{2}k_3(y_2 - z_2)^2. \quad (2)$$

The set of differential equations of the system motion in the time domain is obtained on the basis of the formalism of the Lagrange equations of second kind, and it has the form:

$$(m_1 + L_1 + L_2)\ddot{y}_1 + y_1(k_1 + k_2) - \ddot{y}_2 L_2 - k_2 y_2 = L_1\ddot{z}_1 + k_1 z_1 + Q_1, \quad (3)$$

$$(m_2 + L_2 + L_3)\ddot{y}_2 + y_2(k_2 + k_3) - \ddot{y}_1 L_2 - k_2 y_1 = L_3\ddot{z}_2 + k_3 z_2 + Q_2. \quad (4)$$

After the Laplace transformations under zero-initial conditions [11], the equation set (3)–(4) can be represented by a structural mathematical model in the form of a circuit that is dynamically equivalent to the automatic control system [12, 13] shown in Fig.2.

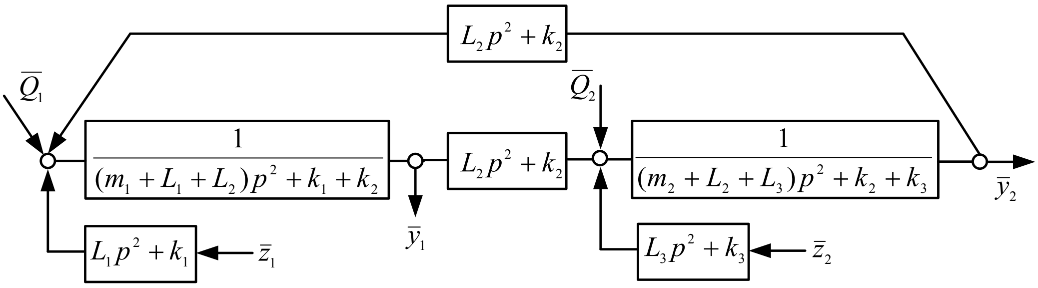

Fig. 2. Structural mathematical model of mechanical system shown in Fig. 1

II. Mathematical Models Development. The type of transfer functions depends on the nature of external disturbances, that is, on whether the disturbances are force ($\bar{Q}_1$ and $\bar{Q}_2$) or kinematic ($\bar{z}_1$ and $\bar{z}_2$). Further on, suppose that the force external actions ($\bar{Q}_1$ and $\bar{Q}_2$) have connectivity determined by the ratio

$$\bar{Q}_2 = \alpha \cdot \bar{Q}_1, \tag{5}$$

where α is coefficient of connectivity of external in-phase harmonic actions.

For kinematic effects ($\bar{z}_1$ and $\bar{z}_2$), it is assumed that

$$\bar{z}_2 = \beta \cdot \bar{z}_1, \tag{6}$$

where β is coefficient of connectivity of kinematic disturbances.

The connectivity coefficients (α and β) can have positive, negative, and zero values. Special cases of the effect of external disturbances can be considered through zeroing α and β properly.

1. The case of joint force perturbation at β ≠ 0 ($\bar{Q}_1 = 0$ and $\bar{Q}_2 = 0$) is considered. The transfer functions of the system in this case take the form:

$$\underset{\bar{Q}_1=\bar{Q}_2=0}{W_1(p)} = \frac{\bar{y}_1}{\bar{z}_1} = \frac{(L_1p^2+k_1)[(m_2+L_2+L_3)p^2+k_2+k_3]+\beta(L_3p^2+k_3)(L_2p^2+k_2)}{A(p)}, \tag{7}$$

$$\underset{\bar{Q}_1=\bar{Q}_2=0}{W_2(p)} = \frac{\bar{y}_2}{\bar{z}_1} = \frac{\beta(L_3p^2+k_3)[(m_1+L_1+L_2)p^2+k_1+k_2]+(L_1p^2+k_1)(L_2p^2+k_2)}{A(p)}, \tag{8}$$

where

$$A(p) = [(m_1+L_1+L_2)p^2+k_1+k_2]\cdot[(m_2+L_2+L_3)p^2+k_2+k_3]-(L_2p^2+k_2)^2 \tag{9}$$

is a system frequency standard equation.

When considering the transfer functions (7), (8), it is assumed that the dynamic mode of oscillation damping is determined by the conditions for zeroing numerators (7), (8). Coordinate may cause two frequencies of dynamic oscillation damping. Along the coordinate ($\bar{y}_1$), the occurrence of two DOD frequencies is possible. Along the coordinate ($\bar{y}_2$), it is also possible to implement two DOD modes due to the virtual existence of roots of the biquadratic frequency equation.

Assuming that the variable factor is β (coefficient of connectivity), a frequency diagram can be developed considering the following frequencies:

1. partial frequencies:

$$n_1^2 = \frac{k_1+k_2}{m_1+L_1+L_2}, \tag{10}$$

$$n_2^2 = \frac{k_2+k_3}{m_2+L_2+L_3}; \tag{11}$$

2. critical frequency of interpartial communication:

$$n_{\text{пар}}^2 = \frac{k_2}{L_2}; \tag{12}$$

3. DOD frequencies determined from the solution of equations in $\bar{y}_1$ coordinate:

$$p^4[L_1(m_2+L_2+L_3)+\beta L_2L_3]+p^2[L_1(k_2+k_3)+k_1(m_2+L_2+L_3)+\beta(k_3L_2+k_2L_3)]+ \\ +k_1(k_2+k_3)+\beta k_2k_3=0; \quad (13)$$

and in $\bar{y}_2$ coordinate:

$$p^4[\beta L_3(m_1+L_1+L_2)+L_1L_2]+p^2[\beta L_3(k_1+k_2)+\beta k_3(m_1+L_1+L_2)+k_1L_2+k_2L_1]+ \\ +\beta k_3(k_1+k_2)+k_1k_2=0. \quad (14)$$

To construct a frequency diagram, the following parameters of the model problem are accepted: m_1 = 10 kg; m_2 = 10 kg; k_1 = 5000 N/m; k_2 = 10000 N/m; k_3 = 15000 N/m; L_1 = 5 kg; L_2 = 10 kg; L_3 = 10 kg.

The modes of dynamic oscillation damping in $\bar{y}_1$ and $\bar{y}_2$ coordinates are determined not only by the parameters of elastic inertia members, but also by the specificities of the external actions formation, in particular, by the connectivity value.

From the equations (13), (14), corresponding DOD frequencies can be found showing that two dynamic oscillation damping frequencies can be found in each of $\bar{y}_1$ and $\bar{y}_2$ coordinates. The frequencies values, as it follows from (13) - (14), depend on the connectivity coefficient of kinematic disturbances (β). Fig. 3 shows the system frequency diagram.

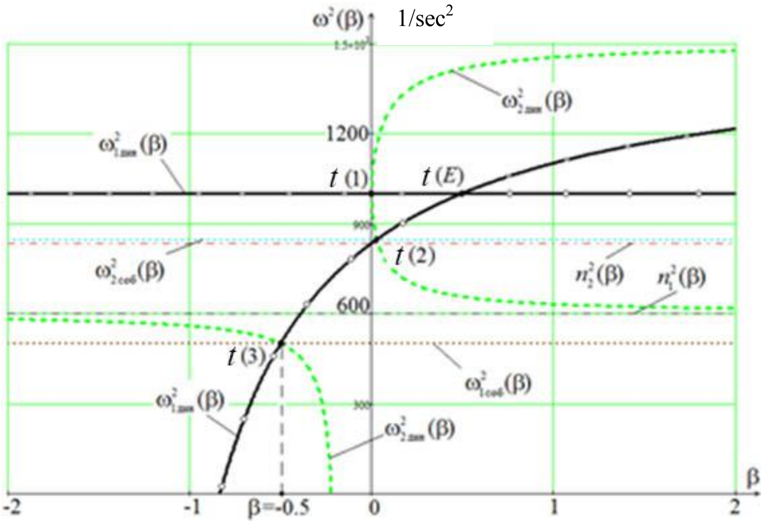

Fig.3. System frequency diagram shown in Fig. 1

In the diagram, the solid line (——) corresponds to $\omega^2_{1\,дин}(\beta)$ dependency graph. Since the frequency equations (13) - (14) are biquadratic, each of the equations has two roots. This is displayed in two graphs. For $\omega^2_{1\,дин}(\beta)$ graph of dependences, the solid line is marked with special symbols (×××), and for the second root, respectively (◦◦◦◦); $\omega^2_{1\,дин}(\beta)$ dependency graphs are touching in t. (*E*). Again, $\omega^2_{2\,дин}(\beta)$ dependency graphs are represented by dashed lines (— — —). $\omega^2_{2\,дин}(\beta)$ dependency graph consists of two non-contiguous blocks. Mutual intersections of $\omega^2_{1\,дин}(\beta)$ and $\omega^2_{2\,дин}(\beta)$ dependency diagrams occur in tt. (1), (2), (3). Each of the considered points determines the amplitude-frequency characteristics associated with the DOD modes features.

In the usual formulation of studying the dynamic oscillation damping, that is, under the action of a single perturbing factor correlated with a certain coordinate, one DOD frequency is determined in the two-degrees-of-freedom system. Such a frequency is determined by the partial frequency values of that system block, the movement of which demonstrates the dynamic oscillation damping (that is, "zeroing" the value of the corresponding coordinate).

Under the action of several simultaneous disturbances, it becomes possible to implement two DOD modes in each of the coordinates for the system as a whole. When additional links are introduced into the system, in particular, on the basis of the MTD, specific properties occur when the dynamic oscillation damping becomes possible simultaneously in two coordinates.

III. Comparative analysis of dynamic properties of the systems in the DOD modes.

1. Fig. 4 shows frequency-response characteristics (FRC) of the system, which manifest themselves under the conditions corresponding to the intersection of $\omega^2_{1\,дин}(\beta)$ and $\omega^2_{2\,дин}(\beta)$ graphs in t. (1) in Fig. 3. The intersection

corresponds to the case when β = 0. The solid line (——) in Fig. 4 corresponds to $\frac{\bar{y}_1}{\bar{z}_1}(\omega)$ dependency graph; the dotted line (········) corresponds to $\frac{\bar{y}_2}{\bar{z}_1}(\omega)$ graph. The parameters of the system as a whole are also shown in Fig. 4.

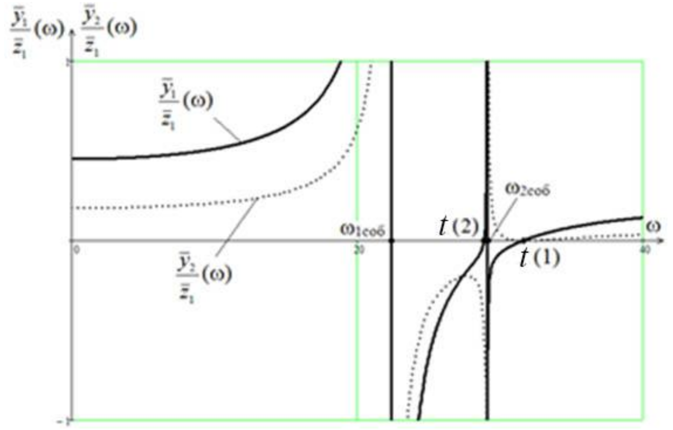

Fig. 4. Frequency-response characteristics of the system for parameters determined by t. in Fig.3

In t. (1) in Fig. 4, the frequency of the DOD mode is determined when $\bar{y}_1$ and $\bar{y}_2$ coordinate values are simultaneously "zeroed out".

It is possible to implement another DOD mode in $\bar{y}_1$ coordinate corresponding to $\frac{\bar{y}_1}{\bar{z}_1}(\omega)$ graph in t. (2) (Fig. 4). At this, the FRC reflect the properties of two-degrees-of-freedom systems. As follows from the FRC, it is possible to implement two DOD modes in tt. (1) and (2) in $\bar{y}_1$ coordinate. For the FRC corresponding to $\bar{y}_2$ coordinate, it is also possible to create two DOD modes at the double intersection of $\frac{\bar{y}_2}{\bar{z}_1}(\omega)$ graph by the abscissa line after $\omega^2_{2\,соб}$ eigenfrequency. Thus, with non-degenerate FRC for each of $\bar{y}_1$ and $\bar{y}_2$ coordinates, two DOD modes can be implemented; while at one of the frequencies, there is simultaneous DOD in two coordinates.

2. Fig. 5 shows the system FRC at $\beta = -\frac{1}{2}$, from which it follows that it becomes possible to restructure the system when one degree of freedom degrades at certain parameter ratios. In this case, in each of $\bar{y}_1$ and $\bar{y}_2$ coordinates (points (1) and (2) in Fig. 5), it is possible to implement the DOD modes, but this occurs when the system "degrades"; under the conditions when FRC ω → ∞ acquire limiting properties.

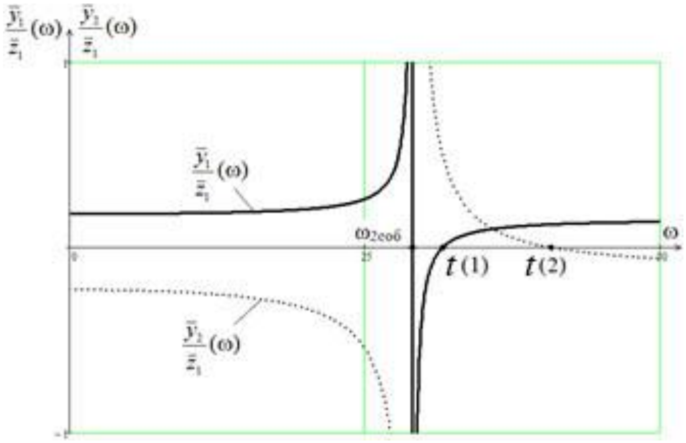

Fig. 5. Frequency-response characteristics of the system at $\beta = -\frac{1}{2}$ corresponding to t. (3) in Fig.3

3. Fig. 6 shows the system FRC at $\beta = \frac{1}{34}$ which corresponds to t. (2) in Fig. 3. For the given state of the system determined by the parameter values, a structural transformation of the system is also characteristic; the system

"degrades" to the status of a one-degree-of-freedom system. For the system on a whole, in each of the coordinates, the DOD implementation is possible, which corresponds to t. (1) and t. (2) in the graphs shown in Fig. 6.

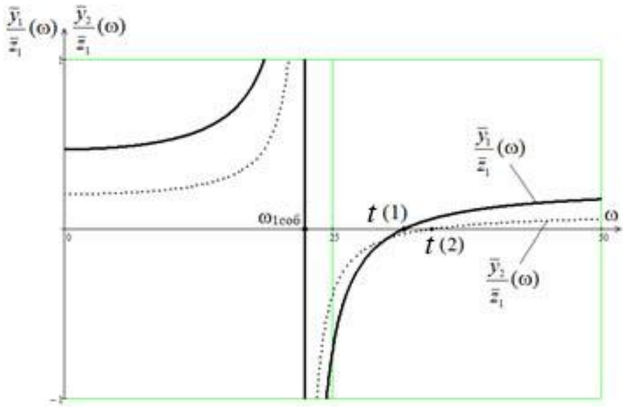

Fig. 6. Frequency-response characteristics of the system at $\beta = \frac{1}{34}$ corresponding to t. (2) in Fig. 3

In the high frequency area ($\omega \to \infty$), the system acquires limiting properties; at this, in comparison with the previous example, when $\beta = -\frac{1}{2}$, the oscillation amplitudes ratio will have a different sign at $\beta = \frac{1}{34}$. Consequently, changes in system parameters under the simultaneous action of two forces can change drastically the dynamic properties of the mechanical oscillatory systems.

Conclusion. The simultaneous action of external disturbances in the presence of additional links in the system, implemented by the MTD under the kinematic disturbance, can have a great impact on the change in the dynamic properties of the mechanical oscillatory systems with several degrees of freedom. So, the authors in this paper have obtained the following research results:

1. A technique of constructing mathematical models based on the use of methods of structural mathematical simulation, in which the mechanical oscillatory system is compared to the dynamically equivalent automatic control system, is proposed;
2. A technique for constructing frequency diagrams that allows for the integral estimation of the interdependence of frequency characteristics is proposed for the case when the system parameters and their perturbation conditions for various power factors change;
3. The analytical conditions for the implementation of the DOD modes simultaneously in two coordinates under the action of two interlinked disturbing factors are obtained;
4. The possibilities to control the structural states when the original mechanical oscillatory system can change the number of degrees of freedom and the system of its dynamic properties are proposed.

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