

ПОВЫШЕНИЕ НАДЕЖНОСТИ И БЕЗОПАСНОСТИ ЭКСПЛУАТАЦИИ ТРАНСПОРТНЫХ СРЕДСТВ КАК ЗАДАЧА ОЦЕНКИ И ФОРМИРОВАНИЯ ДИНАМИЧЕСКИХ СОСТОЯНИЙ ТЕХНИЧЕСКИХ ОБЪЕКТОВ

Елисеев С.В., Кузнецов Н.К., Елисеев А.В., Лонцих П.А.

Ключевые слова: режимы динамического гашения колебаний, частота динамического гашения колебаний, связность колебаний, частотная диаграмма.

Аннотация. Рассматриваются задачи динамики транспортного объекта, ориентированные на повышение надежности и безопасности эксплуатации в условиях вибрационных внешних возмущений. Цель исследования заключается в разработке метода построения математических моделей для оценки динамических состояний и определения условий получения динамических эффектов «обнуления» движений по отдельным координатам. Используются методы структурного математического моделирования. Научная новизна работы заключается в возможности оценки формирования динамических состояний механических колебательных систем при введении дополнительных связей. Разработана технология перехода от передаточных функций систем к уравнениям состояний для оценки частот динамического гашения колебаний, что связано, в частности, с тем, что при внешнем кинематическом возмущении объекта одновременно возбуждаются обе парциальные системы.

IMPROVING THE RELIABILITY AND SAFETY OF VEHICLE OPERATION AS A TASK OF EVALUATING AND FORMING THE DYNAMIC STATES OF TECHNICAL OBJECTS

Eliseev S.V., Kuznetsov N.K., Eliseev A.V., Lontsikh P.A.

Keywords: dynamical damping of oscillations, the frequency of dynamic oscillation damping of the coherence of oscillations, frequency chart.

Abstract. The problems of transport object dynamics aimed at increasing the reliability and safety of operation in the conditions of external vibration disturbances are considered. The purpose of the research is to develop a method for constructing mathematical models for evaluating dynamic States and determining the conditions for obtaining dynamic effects of «zeroing» movements on individual coordinates. Methods of structural mathematical modeling are used. The scientific novelty of the work consists in the possibility of evaluating the formation of dynamic States of mechanical oscillatory systems with the introduction of additional connections. The technology of transition from transfer functions of systems to equations of States for estimating the frequency of dynamic damping of vibrations has been developed, which is due, in particular, to the fact that both partial systems are simultaneously excited when an external kinematic perturbation of an object occurs.

I. Introduction. Transport means play an important role in ensuring the normal functioning of various branches of production and economic activity [1,2]. In this regard, railway transport is no exception, which requires increasing attention to its problems, given the need to transport increased volumes of goods with increased loads on the running systems and the need to move at increased speeds. Many issues of developing ways and means to improve the safety of locomotives and their components and aggregates are considered in the works of domestic scientists [3,4]. Much attention is paid to transport dynamics, the interaction of rolling stock with the rail track, which forms an increased attention to ensuring the reliability of electric locomotives ' traction engines [5,6]. On the other hand, the

information and analytical base related to the detailed understanding of the features of dynamic interactions of traction engines that operate under complex dynamic loading conditions is also expanding [7,8].

The presented work reflects the results of theoretical developments related to the development of a number of new approaches and ideas about the possibilities of evaluating, monitoring and forming dynamic States of traction and axial-suspension engines. The authors develop original methods of structural mathematical modeling.

II. Some general terms. The axial-axle suspension traction motor functions in a fairly saturated dynamic environment which is formed in several directions. On the one hand, a wheelset interacts with a rail track, which can be considered as a source of the so-called kinematic disturbance $z_1(t)$. The basic design scheme of the traction engine is shown in Fig. 1 It shows the relative position of the acting disturbances and the principal arrangement of elastic elements with reduced stiffnesses k_1 and k_2 . It is supposed that the basic diagram reflects the traction motor symmetry properties, i.e. the design circuit is flat; equivalent transformations are possible between power factors [9-11].

III. Features of mathematical model construction. A schematic diagram of an axial-axle suspension traction motor is, in the general case, a mechanical oscillatory system, as shown in Fig. 1.

It is expected that the movement of the object is described in a two-coordinate system (y_1, y_2, y_0 and φ) associated with a fixed basis. The system has a center of mass (p.O in Fig. 1); the object has a mass M and moments of inertia J relative to the center of mass; in turn, the center of mass is located at distances l_1, l_2 from the reference points at which the elastic elements k_1, k_2 are fixed. The synchronizing action of two perturbing factors has the arrangement of conditions ability for intense vibrations of elements, the occurrence of increased loads and intense changes in the shapes of distribution structure of the oscillations amplitudes of the object points of the [9-11].

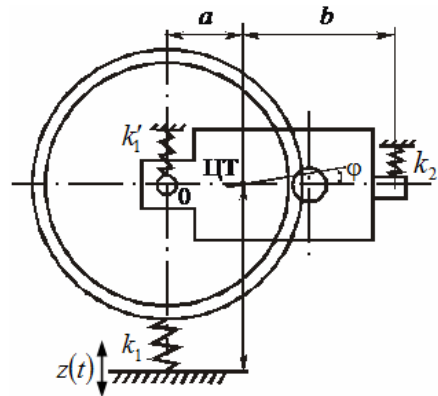


Fig. 1. Schematic diagram of an axial support traction motor

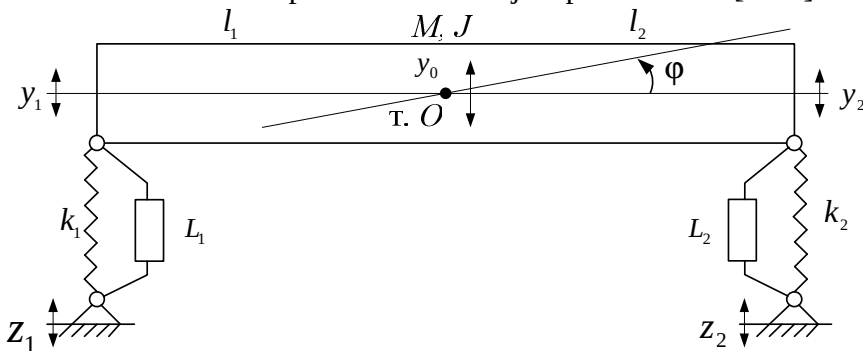


Fig. 2. The design diagram of the original object in Fig. 1 is considered as a mechanical oscillatory system with two degrees of freedom

Figure 2 shows the design circuit of the object considered as a mechanical oscillatory system with two degrees of freedom [9-11]. Fig. 2 gives the example of the motion translation device (UPD) in the form of a non-self-locking screw mechanism; equivalent moment of inertia UPD is defined by L .

The objective of the study is to develop a method for constructing mathematical models that provides the ability to evaluate and form dynamic states of an object.

IV. Mathematical models construction, features of subsystem interaction.

Note the structural mathematical models of a transport system that satisfies safety requirements can be interpreted in the formal symbolism of the electrical circuits theory. Dynamic constraint reactions can serve as state parameters of a mechanical oscillatory system, as well as well-known forms of transport safety assessment based on the kinematic parameters use in the problems of the technological transport systems dynamics. The constraints dynamic reaction at a selected point in the system is defined in the form of an operator as the displacement product by a dynamic stiffness unit and carries information about the features of resonant modes and dynamic damping of vibrations which may be applicable in the task of ensuring information security. To construct a mathematical model using the calculation scheme in Fig. 2, the approaches described in [9-11] are used. A structural mathematical model in the structural diagram form of a dynamically equivalent automatic control system is shown in Fig. 3.

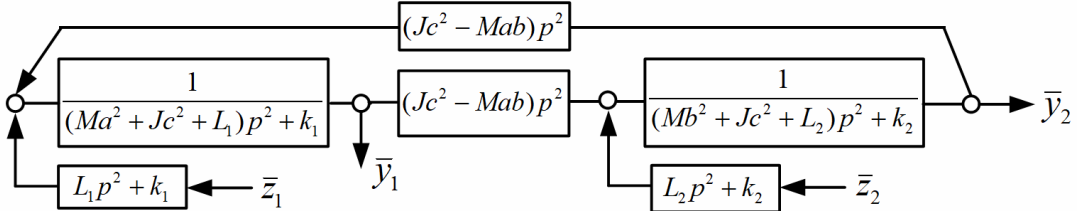


Fig. 3. The structural mathematical model of the original system according to Fig. 2 with kinematic disturbance

Fig. 3 shows a series of notations: $p = j\omega$ - complex variable ($j = \sqrt{-1}$), the $\leftrightarrow$ icon above the variable means its Laplace image; to derive the initial equations in operator form, the Laplace transforms are used for zero initial conditions.

Assuming for simplicity that $z_1 = z_2 = z$ from the block diagram in Fig. 3, you can get the transfer functions:

$$W_1(p) = \frac{\bar{y}_1}{\bar{z}} = \frac{(L_1 p^2 + k_1)[(Mb^2 + Jc^2 + L_2)p^2 + k_2] + (L_2 p^2 + k_2)(Jc^2 - Mab)p^2}{A(p)}, \quad (1)$$

$\bar{z}_1 = \bar{z}_2 = \bar{z}$

$$W_2(p) = \frac{\bar{y}_2}{\bar{z}} = \frac{(L_2 p^2 + k_2)[(Ma^2 + Jc^2 + L_1)p^2 + k_1] + (L_1 p^2 + k_1)(Jc^2 - Mab)p^2}{A(p)}, \quad (2)$$

$\bar{z}_1 = \bar{z}_2 = \bar{z}$

где $a = \frac{l_2}{l_1 + l_2}$, $b = \frac{l_1}{l_1 + l_2}$, $c = \frac{1}{l_1 + l_2}$,

$$A(p) = [(Ma^2 + Jc^2 + L_1)p^2 + k_1] \cdot [(Mb^2 + Jc^2 + L_2)p^2 + k_2] - [(Jc^2 - Mab)p^2]^2 \quad (3)$$

– is a characteristic frequency equation.

The system (Fig. 2) consists of two partial blocks having inertial connections between each other which brings a certain specificity to the system [9-11].

On the hypothesis that $\bar{z}_1 = \bar{z}_2 = \bar{z}$ we have the possibility of a functional relationship between the values of $L1, L2$ in the form

$$L_2 = \alpha \cdot L_1. \quad (4)$$

This kind of practical relationship is realized in the UPD on the basis of a non-self-locking screw mechanism where the flywheel nut can be braked by the controlled pressing of the brakeblock.

In this case, the expression for the transfer functions takes the form:

$$W'_{1(p)} = \frac{\bar{y}_1}{\bar{z}} = \frac{(L_1 p^2 + k_1)[(Mb^2 + Jc^2 + \alpha L_1)p^2 + k_2] + (\alpha L_1 p^2 + k_2)(Jc^2 - Mab)p^2}{A_1(p)}, \quad (5)$$

$$W'_{2(p)} = \frac{\bar{y}_2}{\bar{z}} = \frac{(\alpha L_1 p^2 + k_2)[(Ma^2 + Jc^2 + L_1)p^2 + k_1] + (L_1 p^2 + k_1)(Jc^2 - Mab)p^2}{A_1(p)}, \quad (6)$$

where

$$A_1(p) = [(Ma^2 + Jc^2 + L_1)p^2 + k_1] \cdot [(Mb^2 + Jc^2 + \alpha L_1)p^2 + k_2] - [(Jc^2 - Mab)p^2]^2. \quad (7)$$

For a general assessment of the capabilities of the system, a frequency diagram can be constructed.

V. A frequency system diagram building. To build the frequency diagram for the model task of the transport system while ensuring operational reliability and safety, the following parameters are selected $M = 7000$ kg; $J = 2000$ kg.m²; $a = 0.57$; $b = 0.43$; $c = 0.71$; $k_1 = 1000$ kN / m; $k_2 = 2000$ kN / m; $l_1 = 0.6$ m; $l_2 = 0.8$ m; $L_1 = 100$ kg.

The determination of the dynamic oscillation suppression frequency, at which the simultaneous dynamic oscillation suppression will occur, is identified by the curve crossing in the frequency diagram (Fig. 4). Note that the partial and eigen frequencies also depend on α .

1. Fig. 4, point (1) determines the frequency of dynamic oscillation suppression simultaneously along two coordinates $\bar{y}_1$ and $\bar{y}_2$. Such a point also determines the value of the coefficient of connectivity α .

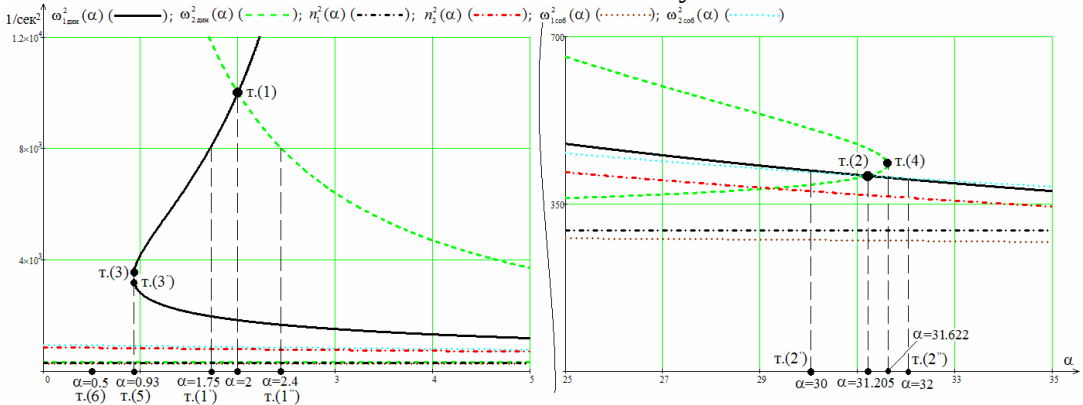


Fig. 4. The frequency diagram for determining the regimes of dynamic oscillation suppression under the condition $L1 \neq 0, L2 \neq 0$ and different values of the coupling coefficient α

2. For p (1), which corresponds to $\alpha = 2$ (on the horizontal axis), frequency response reflects the specific dynamic properties, namely that the frequency of external influences both coordinates $\bar{y}_1$, $\bar{y}_2$ simultaneously become equal to zero. Such a dynamic mode can be called "simultaneous mode dynamic damping of vibrations in two coordinates $\bar{y}_1$ and $\bar{y}_2$ ". The identification of such modes can be attributed to dynamic effects, which have scientific novelty that will eventually require some explanation because the present study is built on certain assumptions and conditions.

Conclusion. A method for constructing frequency diagrams of mechanical oscillatory systems has been developed. On their basis, studies of possible dynamic properties of systems under two perturbations in the modes of dynamic vibration damping have been carried out.

With two simultaneously acting perturbations, it shows the possibility of changing the amplitude-frequency characteristic of the system within a wide range. In particular, it is possible to create modes of dynamic damping of vibrations simultaneously on two coordinates, as well as forms of changing frequency characteristics when varying the connectivity coefficient. It is shown that changes in the dynamic properties of the system can be obtained by introducing additional links or devices for converting motion. Structurally, such links can be implemented on the basis of non-self-braking screw, gear or hinge mechanisms. The connectivity coefficient, in the physical sense, determines the ratio between the reduced masses of devices for converting motion.

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Елисеев Сергей Викторович – доктор технических наук, профессор, советник при ректорате по научной работе, Иркутский государственный университет путей сообщения, г. Иркутск	Eliseev Sergey Viktorovich – doctor of technical sciences, professor, advisor to the rectorate for scientific work, Irkutsk State Transport University, Irkutsk
Кузнецов Николай Константинович – доктор технических наук, профессор кафедры конструирования и стандартизации в машиностроении, Иркутский национальный исследовательский технический университет, г. Иркутск	Kuznetsov Nikolai Konstantinovich – doctor of technical sciences, professor of Department of design and standardization in mechanical engineering, Irkutsk National Research Technical University, Irkutsk
Елисеев Андрей Владимирович – кандидат технических наук, доцент кафедры математики, Иркутский государственный университет путей сообщения, г. Иркутск	Eliseev Andrey Vladimirovich – candidate of technical sciences, associate professor of Department of mathematics, Irkutsk State Transport University, Irkutsk
Лонцikh Павел Абрамович – доктор технических наук, профессор кафедры автоматизации и управления, Иркутский национальный исследовательский технический университет, г. Иркутск	Lontsikh Pavel Abramovich – doctor of technical sciences, professor of Department of automation and control, Irkutsk National Research Technical University, Irkutsk

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