

Optimization of Electromechanical Drive Systems of Machines by the Dynamic Factor

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Abstract: A new round of development of modern mechanical engineering is considered, characterized by a continuous increase in drive power, movement speed and withstanding alternating loads with reduced vibration flows. A study of the nature of vibrations was carried out when optimizing dynamic processes for newly designed machines, where the process of the influence of dynamic loads on them was examined with satisfaction and an assessment was made of changes in reliability and durability parameters. Complex methods have been developed and implemented to modernize the optimization of industrial processes in machines using methods of combined parameters with a dynamic coefficient for electromechanical drives, which are an urgent problem and meet modern requirements for the development of domestic mechanical engineering. An analysis of the applied new technique with reduced metal consumption and the lowest dynamic properties was carried out, due to which an indicator of increased productivity and durability of machines was obtained. The obtained values for optimizing dynamic machines using a reduced coefficient made it possible to obtain a new machine with combined parameters operating at optimal modes with given values of quality, vibration resistance and wear resistance. The authors propose to use a new method in the production of electromechanical and mechanical devices and equipment, which makes it possible to reduce unnecessary vibrations during the operation of electric drives and increase their stability and reliability.

Key words: vibration, dynamic coefficient, electromechanical transmission, load, differential equation

The complex operation of improved mechanisms and machines assembled with an electric drive, representing a complex multifaceted structure of a dynamic system, participating in mechanical and automated operations using electromagnetic drives of a modernized type, is considered.

The use of an improved technique for mechanical processes requires minor calculations that will increase the system's performance to specified parameters and reduce undesirable combinations of defect-forming parameters. The design of these machines takes into account some features of their design, due to which the

necessary values of the parameters of the dynamic model will be obtained taking into account the requirements of time.

In the crucial areas of modern technical progress, especially in the theory of automatic machines and automatic lines, optimization of machine dynamics and transmissions is of particular importance. The complex operation of improved mechanisms and machines equipped with electrical transmissions, participating in mechanical and automated operations with the introduction of modernized types of electromagnetic transmissions, forms a multifaceted structure of the dynamic system. The study of forced oscillations and dynamic load conditions of electromechanical transmission systems is one of the main steps in the optimal design of machines [Samidov and Aliyev, 2007; Samidov and Samidov, 2003].

The use of advanced technology in mechanical processes requires the application of calculation and optimization methods that increase the performance of the system to the specified parameters and reduce undesirable combinations of parameters that lead to defects. Optimization of dynamic system parameters of gear transmissions and transmissions of metal-cutting machines gives important results in terms of limiting dynamic loads on machines and increasing reliability [Samidov and Aghayev, 2007; Aliyev et al., 2025; Adamiya and Mikhailenko, 1989].

In the design of these machines, constructive features are taken into account, thanks to which the necessary parameters of the dynamic model are obtained, anticipating the time requirements. These approaches, which are based on the fundamentals of the theory of mechanisms and machines [Levitskaya and Levitsky, 2003] and integral methods in dynamics [Golubentsev, 1999], are supported by mathematical optimization theories. In particular, dynamic programming [Bellman, 2011], mathematical theory of optimal processes [Pontryagin et al., 1976], formulation of optimal design problems with the application of ECM [Sobol and Statnikov, 1977] and applied nonlinear programming methods [Ximmelblau, 2008] allow for the accurate selection of parameters of the dynamic model.

For this, it will be necessary to consider the main parameters of the newly prepared methodology responsible for the quality indicator. Whereas the design of new machines and devices will require changing imperfect cycles and supplement them with newer and more perfect ones in a certain range for a given process. Taking into account the change in the design forms (K_m) of the newly manufactured mechanism, it is necessary to take into account the values of the n -th component of the obtained parameters, which has a certain dynamic form and limited values (b_j). Due to the fact that the design of the newly designed device K_m and the values of b_j can be changed only in a certain spatial range, then such a restriction can be written as follows:

$$K_m = b_j \leq b_j < (j = 1, 2, \dots, n \rightarrow \infty) \quad (1)$$

The restrictions that the authors are guided by in (1) can be considered as an improved model of a closed loop for elastic oscillatory systems limited by dynamic loads [1,2].

The optimization system of a modern dynamic structure, a newly manufactured structure, must have characteristic values (1), differing from those available according to the principle: optimally-high-quality-reliably with minimal costs and high economic effect [4]. Here, an important factor is the reduction of the main defect-forming factors: vibration, oscillations, displacements.

Therefore, the studies conducted by the authors are aimed at designing and using improved machines and mechanisms with stable characteristics to various loads, vibrations and oscillations, increasing the working potential of the device and reducing the risk of breakdowns by 30-32%, which will solve almost one of the problems most often encountered in mechanical engineering, aviation and shipbuilding industries. This approach of the authors to solving pressing problems is relevant today.

Studying and analyzing literary sources and existing scientific developments devoted to optimizing the dynamics of machines and mechanisms [1-7, 9, 10], we can undoubtedly say that the problem remains and it must be studied, researched, analyzed and certain decisions must be made to resolve it. But considering and analyzing which processes most often arise during its analysis, the authors divided it into two main branches:

-1- the optimal method of programming dynamics [2].

-2- mathematical theory of process control [5].

Each of the considered methods and theories has a certain pattern

which are characterized by their direction for resolution and obtaining the necessary values of dynamic optimization. Such a method of analysis allows choosing a combined one from random, directed and combined directions, offering to choose several options and directions for programming and mathematical modeling of dynamic processes of specific machines and devices [5].

Considering and analyzing the multi-criteriality of dynamics and the optimality of the designed machines, it is possible to select such options from existing methods that are multifunctional and universal, and mathematically modulating them in an elastic system for specific machines and mechanisms under maximum loads, it is possible to obtain such quality criteria that were specified in the programming process [4, 9].

The purpose of the study is to develop and apply in practice a multi-purpose scientific methodology that has the best design achievements, taking into account the requirements of the industry, and also has reliability, durability and economic efficiency.

This approach will make it possible to concretize and evaluate the results of the scientific work of the authors, which is aimed at completely new developments in the field of optimal design of electromechanical drive systems of machines with a minimum dynamic coefficient. In addition, the authors set a goal to further improve the main technological and flow cycles, reduce the metal consumption of products and automate the process.

Considering and analyzing the results of the study on the processes occurring in the system of dynamic and static loads with an elastic system, a certain pattern of the transient process is revealed, where it is possible to calculate the largest dynamic coefficient. This circumstance is substantiated by the fact that under a static load on an elastic body, a load acts equivalent to the force of action during a transient dynamic process with an underestimated coefficient.

$$\lambda_{My} = \frac{StM_{n,j+1}}{Md_{n,j+1}x} T \quad (2)$$

Material and methods of the study. The issues of machine and mechanism loading during operation are considered, where certain types of vibrations occur in transient processes in the electric motor drive, since the electric motor operation process is multi-stage. If we consider the example of Fig. 1, then here the four-stage condensate pump of the KsV brand, having high performance indicators, is used in viscous and chemical environments, works from the drive mechanism of the electric motor (Fig. 1). This shows that in any mechanism there is a transition drive, which must ensure its normal functionality, i.e., when connecting two or more mechanisms:

-a) with an elastic (rigid) connection

- lateral displacement must not exceed 0.05 mm according to the clock indicator,

- the bend in the connection - top-bottom must not exceed 0.02-0.03 mm according to the clock indicator,

- b) with a flexible connection

- lateral displacement must not exceed 0.08 mm according to the clock indicator,

- the bend in the connection - top-bottom must not exceed 0.03 mm according to the clock indicator.

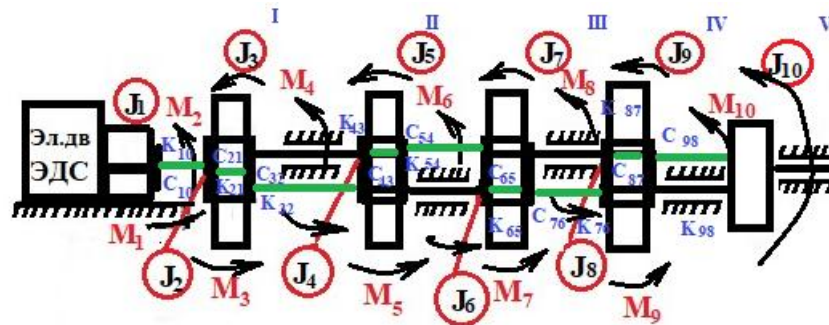


Fig. 1. Schematic diagram of the electromechanical operation of a four-stage pump drive

If the above values of the drive connection are not taken into account, it is impossible to reduce the vibration in the connections. As a result, the authors recommend adhering to these values in the future, as soon as we receive a new machine model corresponding to the calculated parameters, to improve it, that is, to integrate it to the model whose purpose was set in this article [2,3]. These recommendations can be useful in calculating two-, four- and five-stage machines and mechanisms whose torsional vibrations exceed the torque of the shafts (for example, in calculating and designing a marine internal combustion engine in conjunction with a gearbox).

It is necessary to take into account that when calculating dynamic models of some machines and mechanisms, these recommendations may not be suitable due to differences in the optimality of the design and its purpose [1, 6]. In this regard, by analyzing the results of theoretical research and the obtained practical values for mechanisms with a system of uniform and non-uniform torsional oscillations, the oscillatory circuit of which depends on the applied energy and the energies arising in the mechanism itself, it is possible to modulate, calculate and obtain a certain type of dynamic model diagram for a certain type of drive (Fig. 2).

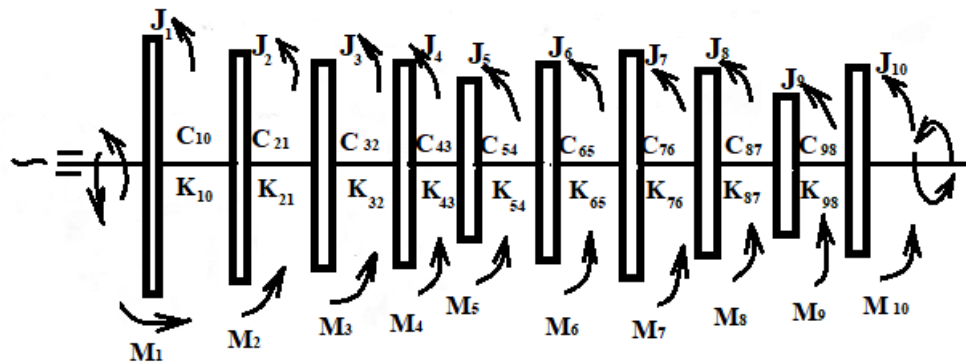


Fig. 2. Dynamic diagram of the electric drive model with a rigid connection

Here we consider the inertial moment of the rotating mechanism, with an elastic-rigid connection and the forces that were applied both from the source itself and from the mechanism itself, where we take the "Law of conservation and transformation of energy ..." as a basis. Then the kinetic energy (T) of the rotating body is converted into potential energy (P) and the resulting dissipative function (F), that is, the entropy in the system increases. Therefore, the oscillatory system in the elastic system has small damped oscillations and in the resulting medium, taking into account the influence of resistances, remain unchanged, as well as in work with the attenuation of the electric current of a closed circuit.

Taking into account the above and taking the "Law of conservation and transformation of energy ..." (LCTE) as a basis, where there are no maximum friction-destruction forces and no elastic deformation in the machine circuit, then the values for T , P and F will have the following form:

$$T = \frac{1}{2} \sum_{n=1}^k \sum_{j=1}^K a_{nj} \varphi_n \varphi_j;$$

$$P = \frac{1}{2} \sum_{n=1}^k \sum_{j=1}^K c_{nj} \varphi_n \varphi_j;$$

$$F = \frac{1}{2} \sum_{n=1}^k \sum_{j=1}^K k_{nj} \varphi_n \varphi_j; \quad (3)$$

where - a_{nj} c_{nj} k_{nj} composed of inertial, quasi-elastic and dissipative coefficients.

As the analysis of available literary sources has shown, a number of works are devoted to the general issue of constructing a dynamic model of machines. But the recommendations of these developments cannot always be directly applied to the calculations of dynamic models of specific machines, since the designs of machines differ greatly from each other, therefore, the approach and the objectives of the study are different. For example, when calculating torsional vibrations in transmissions of machines with an electric drive with a flexible connection, the torque of large-diameter and short-length shafts is significantly exceeded, whereas in diesel engines with an elastic clutch, these values can be neglected, due to the fact that it is small compared to the torque of the entire crankshaft

It follows that in an electromechanical circuit, the mass and moment of inertia of the main link and its rigidity, preventing its deformation due to vibration, can be determined by the formulas:

$$m_{n1} = \sum_{n=1}^{k=0.5} \left[m_n + 1 \left(\frac{V_{n-1}}{V_{n+1}} \right)^2 + J_n m_n \left(\frac{\omega_{n-0.5}}{V_{n+0.5}} \right)^2 \right] \quad (4)$$

$$J_{n1} = \sum_{n=1}^{k=0.5} \left[m_n + J_n \left(\frac{V_n}{\omega_n} \right)^2 + J_i \left(\frac{\omega_n}{\omega_n} \right)^2 \right] \quad (5)$$

$$P_n = \sum_{n=1}^{k=0.5} \left(P_n + m_n \frac{V_n \cos a_n}{V_n} + M_n \frac{\omega_n}{V_n} \right) \quad (6)$$

$$M_n = \sum_{n=1}^k \left(P_n \frac{V_n \cos a_n}{\omega_n} + M_n \frac{\omega_n}{\omega_n} \right) \quad (7)$$

$$c_{nt} = \left(\frac{V_n - V_n}{V_n} \right)^2 \quad (8)$$

$$M_1 \frac{1}{\omega_e M^k} + \frac{S_{k-M_1-1}}{M_2} = S \quad (9)$$

$$s_{nt} = s_{n\varphi} \omega_0 \left(1 - \frac{\omega_n}{\omega_n} \right)^2 \quad (10)$$

$$s_{n\varphi} = s_{n\varphi \omega_0} \left(\frac{1-n}{1 \pm \omega_n} \right)^2 \quad (11)$$

where m_n – is the mass in the electromechanical circuit;

J_H – is the moment of inertia of the main link of the electric circuit;

$m_n J_n$ – are the values of the masses and the moment of inertia of the n-th link during processing;

V_n, ω_n – is the uniform motion during rotation and the axial velocity vector of the n-th link;

a_n – is the angle formed by the driving force P_n and the vector velocity V_n ;

P_n, M_n – is the driving force of the mechanism during rotation and the moment of inertia acting on the n-th links during processing;

c_{nt} – is the uniformly directed force and axial rigidity of the n-th nature acting on the elastic link;

$snt, sn\varphi$ - are the coefficients of the force factor of linear and angular deformation.

Based on the above, we obtain general-order equations for torsional-oscillatory systems in the following form:

$$\frac{dn}{dt} = \frac{\partial T+1}{\partial \varphi_n} + \frac{\partial n+1}{\partial \varphi_n} + \frac{\partial F+1}{\partial \varphi_n} n = M_n(t) \quad (n = 1, 2, 3 \dots, k) \quad (12)$$

where $M_n(t)$ - is the value of the variation coefficient for spatial values. The dimension of this coefficient consists of the dimension of the main virtual work done divided by the total dimension of the coordinate system and related by Newton's law.

Taking into account the above and guided by the ZSPE, taking expressions (3) as the main one and substituting it into equation (12), we obtain universal values of the equation of motion of the torsional-oscillatory circuit, for the newly constructed model of the mechanism in the equilibrium state: $a_{11}\varphi_1 + \dots + a_{1k}\varphi_k + c_{11}\varphi_1 + \dots + c_{1k}\varphi_k + \dots + k_{11}\varphi_1 + k_{1k}\varphi_k = M_1(t)$. From here we calculate all the values of a , c and k and, based on the obtained values, we write the equation in the form of a general differential equation, where the values of M_k depend

on the excitation moment and are equal to:

$$M_k(t) = a_{kk}\varphi_k + c_{kk}\varphi_k + k_{kk}\varphi_k \quad (13)$$

The equation we obtained applies to a multi-connected dynamic system and if we transform it, we obtain an equation of the following form:

$$M_k(t) = \overline{M}(t) = a\varphi_k + k\varphi_k + c\varphi_k \quad (14)$$

where, a - is the coefficient of inertia in a free state;

k - is the dissipation coefficient of the natural oscillatory circuit;

c - are the dynamic values of the stiffness coefficient;

$M(t)$ - are the values of the matrix at the moment of excitation;

φ - are the obtained values of the matrix coefficients by coordinates.

Taking into account the obtained expressions (4-11), where a linear electromechanical system is considered, a branched network having a certain device with a mass whose values depend on the functionality of the dynamic system. The device shown in Fig. 1 is an electromechanical drive system that sets an inertialess shaft in rectilinear motion (Fig. 2). Analyzing and accepting the above, it can be substantiated that the differential equations of motion of the newly developed dynamic model, the diagram of which is shown in Fig. 2, will have the following form [9,10]:

$$\begin{aligned} \frac{1}{2\omega_e M_k} M_1 v_1 + \frac{S_k M_1}{2M_k} &= S v_0 \\ J_1 \varphi_1 + c_{10}(\varphi_1 - \varphi_2) + k_{10}(\varphi_1 - \varphi_2) &= M_1 \\ J_2 \varphi_2 + c_{10}(\varphi_1 - \varphi_2) + c_{21}(\varphi_2 - \varphi_3) &= k_{10}(\varphi_1 - \varphi_2) k_{21}(\varphi_2 - \varphi_3) = M_2; \\ J_{10} \varphi_{10} + c_{98}(\varphi_9 - \varphi_{10}) + k_{98}(\varphi_9 - \varphi_{10}) &= M_n; \end{aligned} \quad (15)$$

The obtained equations of the electromechanical system (15) have their own formulation and are classified by levels:

- the first equation of the system (15) is the equation of the dynamic characteristics of the engine
- the second and third equations of the system (15) are the equations of the rotor of the electric motor.

The first equation having the values M_k , C_k , v_1 , v_0 – are characterized by mechanical movement and are designated: M_k is the critical moment of the engine movement, S_k is its static slip $v_1 v_0 = v_n$ is the nodal speed of rotation of the engine. Then:

$$M_k = \lambda_m M_H v_n;$$

$$S_k = S_H \left\{ [\lambda_m + 5(\lambda_m - 1)S_H] \sqrt{[\lambda_m + 5(\lambda_m - 1)S]^2 - 1} - 1 \right\}$$

Here $-S_H$ – is the motor slip at the critical torque MN;

$-\lambda_m$ – are the values obtained at the critical torque;

$-Sv_0 = (\omega_0 - \omega_1)/\omega_0$; $\omega = \omega_0/p$ is the calculated rotor speed at idle;

$-v_0$ – is the nodal speed;

$-p$ – is the number of poles on the rotor;

$-\omega_0 l s = 2\pi f_0 l s$,

where, $-f_0$ – is the frequency of the power grid; l is the wavelength; s is the cross-section of the winding;

$-J_1$ – is the moment of inertial motion;

$-\varphi_1$ – is the angular displacement of the rotor at the maximum torque;

$-C_{10}$ – structural rigidity in an electromechanical system

$-k_{10}$ – damping coefficient for a rigid matrix system.

The resulting equation $Sv_0 = (\omega_0 - \omega_1)/\omega_0$ Sv , we differentiate with respect to time, taking into account the nominal values that must be obtained after the transformation and substitute it into the obtained second equation (15) $(\varphi_1 - \varphi_2) + k_{10}(\varphi_1 - \varphi_2) = M_1$, where we obtain an equation of the following type:

$$\frac{1}{J_1 \omega_0} [c_{10}(\varphi_1 - \varphi_2) + k_{10}(\varphi_1 - \varphi_2) - M_1 M_n] = S \quad (16)$$

Next, we exclude the values of S from the dynamic characteristic equation of the engine, substituting the values of equation (16) into equation (15) and obtain a mixed equation of the following type:

$$BTN\dot{M}_1 + BN\dot{M}_1 - M_{10} - \frac{k_{10}M_{10}}{c_{10}} = M_{n(t)} \quad (17)$$

where $M_{10} = c_{10}(\varphi_1 - \varphi_2)$ – elastic forces arising in the process of mechanical work;

$T = 1/\omega_0 Sk = 1/Skp\omega_0$ – value characterizing the magnetic constant over time, depending on the current strength in the circuit;

$B = TN/k_0$ – constant of the electromechanical system of the oscillatory circuit.

Studying the obtained research results and comparing them with the existing ones, the corresponding theoretical assumptions were made that in the process of work each newly developed system has its own design model, where in the design process, the main indicators of reliability and quality are calculated, and the specified parameters are laid down. In our case, the resulting dynamic model, made for an electromechanical system, has its own features and priority capabilities compared to the existing types.

The dynamic model developed and applied by the authors for an electromechanical system has n -th degrees of freedom, where each value is characterized by its components and depends on the connection of the engine, as well as on the static torsion coefficient.

Research results and their discussions. Based on the research results, an analysis was carried out that reveals the purpose of the work and shows the entire structure of the design obtained during the entire design and calculation process.

The process of studying the methodology of dynamic processes of machines and their optimization with the transition to modernization and creation of new modern systems from a multi-mass model to a three-mass model, possibly by the method of transformation, where in equation (15) transforming the differential equation of motion in coordinates and taking into account the moments of elastic forces acting on the electric motor, we mix them with three-mass elastic forces obtained in the coordinate system, we obtain a three-model elastic system of a dynamic model in a free state (Fig. 3).

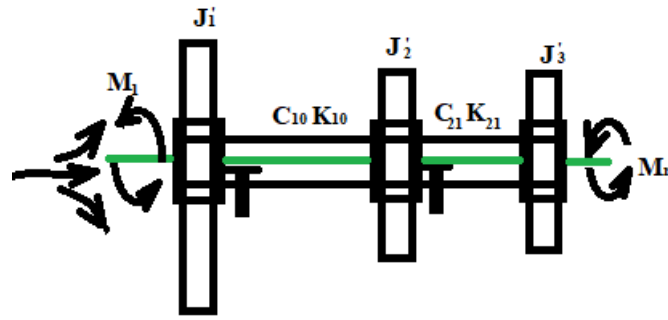


Fig. 3. Scheme of a newly designed three-mass elastic dynamic system in a free state

The resulting scheme of a three-mass elastic system shows the cyclicity of the frequency of free oscillations with a rigid connection and a closed loop. This allows you to control the periodicity of the dynamic system, taking into account the obtained values. From here, the resulting equation (17) must be differentiated and obtain a high-order equation with the variables M_1, M_{10}, M_{21}

$$\begin{aligned} \dot{M}_{10} + b_{10}^2 M_{10} - \frac{c_{10}}{J_1} M_1 - \frac{c_{10}}{J_2} M_{21} + \frac{k_{10}(J_1 + J_2)}{J_1 J_2} M_{10} - \frac{c_{10} k_{21}}{c_{21} J_2} M_{21} &= \frac{c_{10}}{J_2} M_2(t) \\ M_{21} + b_{21}^2 M_{21} - \frac{c_{21}}{J_2} M_{10} + \frac{k_{20}(J_2 + J_3)}{J_2 J_3} M_{21} - \frac{c_{21} k_{10}}{c_{10} J_2} M_{10} &= \frac{c_{21}}{J_3} M_3(t) - \frac{c_{21}}{J_2} M_2(t) \end{aligned} \quad (18)$$

Here the values of b_{n+1}^2 correspond to the calculated parameters of the current strength and depend on the cyclic frequency and have the form of an equation when the system is in a free state $\frac{c_{n+1}(J_1 + J_{n+1})}{J_{n+1}}$ with variables $n=1, 2, 3, \dots, k$.

Next, we introduce the values of $d/dt p$ and the determinant $\Delta(p)$ into the high-order equation. We replace the obtained values with derivatives n and variables M_1, M_{10} and M_{21} , then we get:

$$\Delta(p) = nM^{n0} \quad (19)$$

where the coefficient $n = 0, 1, 2, 3, 4, 5$ and is expressed through the parameters of the system under consideration in the 0 - th, 1 - st, -2 - nd, 3 - rd, 4 - th and 5 - th degrees:

$$a_0 = \frac{(J_2 + J_3)k_{21}}{J_2 J_3} + n \frac{(J_1 + J_2)k_{10}}{J_1 J_2} = 1Tn \quad (20)$$

$$a_1 = b_{10}^2 + b_3^2 = \frac{(J_1 + J_2)(J_2 + J_3)k_{10}k_{21}}{J_1 J_2^2 J_3} - \frac{k_{10}k_{21}}{J_2^2} + \frac{(J_2 + J_3)k_{21}}{Tn \pm 1 J_2 J_3} + \frac{(J_1 + J_2)k_{10}}{Tn \pm 1 J_1 J_2} + \frac{1}{BTN} Mn \quad (21)$$

$$\begin{aligned} a_2 = & \frac{(J_1 + J_2)}{J_1 J_2^2} k_{10} b_{21}^2 n + \frac{(J_2 + J_3)}{J_2 J_3} k_{21} b_{10}^2 n + \frac{c_{10} k_{21}}{J_2^2} - \frac{c_{21} b_{10}}{J_2^2} + \\ & + \frac{b_{10}^2}{Tn} + \frac{b_{21}^2}{Tn} + \frac{(J_1 + J_2)(J_2 + J_3)k_{10}k_{21}}{T J_1 J_2^2 J_3} - \frac{k_{10} j k_{21}}{T n J_2^2} + \frac{(J_2 + J_3)k_{21}}{BTN J_2 J_3} + \frac{(J_1 + J_2)k_{10}}{BTN J_1 J_2} - \frac{k_{10}}{BTN J_1} \end{aligned} \quad (22)$$

$$a_3 = b_{10}^2 b_{21}^2 - \frac{b_{10}^2 c_{20}^2}{J_2^2} + \frac{(J_1 + J_2)}{TJ_1 J_2} k_{10} b_{21}^2 + \frac{(J_2 + J_3)}{TJ_2 J_3} k_{21} b_{10}^2 - \frac{c_{10} k_{21} c_{21} k_{10}}{TJ_2^2} +$$

$$+ \frac{b_{10}^2 c_{21}^2}{BTT} + n \pm 1.2 \frac{(J_1 + J_2)(J_2 + J_3) k_{10} k_{21}}{BTNJ_1 J_2^2 J_3} - \frac{k_{10} k_{21}}{BTNJ_2^2} n \pm 1.2 - \frac{c_{10} k_{10} k_{21} (J_3)}{BTNJ_1} \quad (23)$$

$$a_4 = \frac{b_{10}^2 b_{21}^2}{Tn} - n \frac{c_{10} c_{21}}{TJ_2^2} + \frac{(J_1 + J_2)}{BTNJ_1 J_2} k_{10} b_{21}^2 + \frac{(J_2 + J_3)}{BTNJ_2 J_3} k_{21} b_{10}^2 - \frac{c_{10} k_{21} c_{21} k_{10}}{TJ_2^2} +$$

$$+ \frac{c_{21} k_{10}}{BTNJ_2^2} n + \frac{k_{10}^2 b_{21}^2}{BTTN_1} - \frac{(J_2 + J_3)}{BTNJ_1 J_2 J_3} c_{10} k_{21} \quad (24)$$

$$a_5 = \frac{b_{10}^2 b_{21}^2}{BTN} - \frac{c_{10} c_{21}}{BTNJ_2^2} - \frac{c_{10} b_{21}^2}{BTNJ_1} \quad (25)$$

We obtained the values of the obtained polynomial in equation (19) by replacing the variables by the elastic moments M1, M10 and M21 found in equations (18), we expand the determinant p and replacing the polynomials with coefficients of the first, second and third order we obtain the equations of the system in the following form:

$$Q_n M_n - \left(\frac{k_{10} p}{c_{10} - n} + 1 \right) n \quad (26)$$

$$Q_1(M_n)_{(n=1,2,3)} = \left| \begin{array}{c} \frac{M_2(t) c_{10}}{J_2} \left[p^2 + n \frac{(J_1 + J_2) k_{10} p}{J_1 J_2} b_{10}^2 \right] - \left(\frac{c_{10} k_{21} p}{c_{21} J_2} + \frac{c_{10}}{J_2} \right) \\ \left| \left(\frac{M_3(t) c_{21}}{J_3} - \frac{M_2(t) c_{21}}{J_2} \right) - \left(\frac{c_{21} k_{10} p}{c_{10} J_2} + \frac{c_{21}}{J_2} \right) \right| \left[p^2 \frac{(J_2 + J_3) k_{21} p}{J_2 J_3} b_{21}^2 \right] \\ (BNT p^2 + BNp + n) \frac{M_2(t) c_{10}}{J_2} - \left(\frac{c_{10} k_{21} p}{c_{21} J_2} + \frac{c_{10}}{J_2} \right) \end{array} \right| \quad (27)$$

$$Q_2(M_n)_{(n=1,2,3)} = -\frac{c_{10}}{J_1} \left| \left| \frac{M_3(t) c_{21}}{J_3} - \frac{M_2(t) c_{21}}{J_2} \right| \left[p^2 \frac{(J_2 + J_3) k_{21} p}{J_2 J_3} b_{21}^2 \right] \left| \frac{M_2(t) c_{10}}{J_2} - \left(\frac{c_{10} k_{21} p}{c_{21} J_2} + \frac{c_{10}}{J_2} \right) \right| \right| \quad (28)$$

$$Q_3(M_n)_{(n=1,2,3)} = -\frac{c_{10}}{J_1} \left| \begin{array}{c} (BNT p^2 + BNp + 1)n - 1 \left(\frac{k_{10} p}{c_{10}} + 1 \right) \\ \left[p^2 + \frac{(J_1 + J_2) k_{10} p}{J_2 J_3} b_{10}^2 \right] \frac{M_2(t) c_{10}}{J_2} - \\ \left(\frac{c_{21} k_{10} p}{c_{10} J_2} + \frac{c_{21}}{J_2} \right) - \left(\frac{M_3(t) c_{21}}{J_3} - \frac{M_2(t) c_{21}}{J_2} \right) \end{array} \right| \quad (29)$$

We divide the calculated values for the first, second and third levels by the coefficients of the determinant of the constant electromechanical system $BTN = TJ_1/k_{\alpha}$ and expand them relative to the variables M1, M10, M21, taking into account the vibration from the oscillatory circuit depending on time and torque, we obtain an equation of the following type:

$$M_1^n + a_0 M_{31}^V + a_1 M_1^{IV} + a_2 M_1^{III} + a_3 M_1^{II} + a_4 M_1^I + a_5 M_{1...} = M_1(t)$$

$$M_{10}^n + a_0 M_{10}^V + a_1 M_{10}^{IV} + a_2 M_{10}^{III} + a_3 M_{10}^{II} + a_4 M_{10}^I + a_5 M_{10...} = M_{10}(t)$$

$$M_{21}^n + a_0 M_{21}^V + a_1 M_{21}^{IV} + a_2 M_{21}^{III} + a_3 M_{21}^{II} + a_4 M_{21}^I + a_5 M_{21...} = M_{21}(t) \quad (30)$$

From the values obtained above, it is clear that during operation, the electric motor is subject to certain types of oscillations. Having insignificant indicators of oscillatory vibrations that can be created by the impact of external forces in a three-mass elastic system, they have an insignificant effect on the operation of the electric motor. Such values can be neglected. But if the oscillations are stronger than the elastic forces of the machine, then it is necessary to consider their force of action and negative consequences. In our case, we neglect

insignificant values of oscillations and consider the mathematical calculation of the technological load that acts on the electric motor at $Mn = \text{const}$, as a result of which attenuation occurs in the mechanical part of the system at the initial point $a = 0$ and at the final point $z = n$. Since the system is always subject to these loads and has permissible values of variables (n), we can neglect them as a result of which we obtain the values of the equation for the oscillatory circuit with specified minimum vibration parameters in two axial directions:

$$M_{a,z,n+1}^V + a_1 M_{a,z,n+1}^{VI} + a_3 M_{a,z,n+1}^{II} + a_5 M_{a,z,n+1} = \frac{c_{10} c_{21} k_e}{T J_1 J_2 J_3} (M_2 + M_3) \quad (31)$$

Analyzing the results of the study, a discussion was held on the obtained values, where the main criteria for optimizing electric transmissions for dynamic components were considered and the main points that improve the reliability and practicality of the newly manufactured design were determined. Individual aspects of changing variable moments of static forces arising under the influence of external factors for oscillatory circuits for components M_1 , M_2 and M_3 were also considered, which depend on the mass J_1 , J_2 and J_3 and the action time (t). Here, in the process of discussion, the main values of the moments acting from the external circuit on the dynamic system were revealed, where the operation of the machine drive is taken into account, with its rigid and elastic connection, its efficiency in a free state.

For the variables and coefficients considered above, calculated values were obtained based on a system of differential equations for a given circuit. The obtained calculated values of the coefficients, having their own values, also have estimated values. So for the dynamic coefficient λ_n ($n=1,2,3$) they are equal to:

$$\lambda_1 = \frac{M_{1max}}{M_{1CT}} = \frac{J_2^2 J_3 (b_{10}^2 b_{21}^2 J_1 J_2^2 - b_{21}^2 c_{10} J_2^2 - c_{10} c_{21} J_1)}{(c_{10} J_1 J_2^2 (b_{21}^2 J_2 M_2 + c_{21} J_2 M_2))} \dots = [B(\tau)] \quad (32)$$

$$\lambda_{10} = \frac{M_{10max}}{M_{10CT}} = \frac{J_2^2 J_3 (b_{10}^2 b_{21}^2 J_1 J_2^2 - b_{21}^2 c_{10} J_2^2 - c_{10} c_{21} J_1) M_H}{J_1 J_2^2 (b_{21}^2 J_2 J_3 M_2 + c_{21} J_2 M_2 - c_{21} J_2 M_2)} \dots = [B(\tau)] \quad (33)$$

$$\lambda_{21} = \frac{M_{21max}}{M_{21CT}} = \frac{c_{21} J_3 (b_{10}^2 b_{21}^2 J_1 J_2^2 - b_{21}^2 c_{10} c_{21} J_1)}{J_1 J_2^2 b_{10}^2 - c_{10} J_2} \dots = [B^{(1)}(\tau)] \quad (34)$$

Static coefficients arising in the electric drive system from external influences can be calculated by formula (12). Where all components of the values of n are considered and shown and, if necessary, it is possible to calculate and obtain parameters with the corresponding mechanical properties. Studying the obtained data, the values of which have the main goal and task, for a newly manufactured design, it can be said that, unlike static coefficients, which are also important and should be considered so that their values are minimal and to obtain the specified values of dynamic coefficients, characterized by λ_1 , λ_{10} , λ_{21} , a certain dependence of $B^{(\pm j)}(\tau)$ ($j=1,2,3$).

the transient process function is revealed, is associated with the expression

Analysis of research results. The research conducted by the authors is based on theoretical calculations and practical applications in practice. According to the calculated values, they proposed a design of a new dynamic model of an electromechanical drive with specified characteristics of reliability, durability with minimal weight qualities, but with a high efficiency. In addition, the authors considered the components of damped oscillations arising in the rigid circuit of the electromechanical drive and proposed an option to reduce them due to technological standards and requirements of the new method of connecting the drive to the mechanism taking into account the offset and fracture. Based on the results of the analysis, the authors obtained a model of a three-mass structure, the design of which has optimal parameters that meet the requirements of the time.

Conclusions.

The proposed multi-mass dynamic model and optimization methodology for electromechanical drive systems demonstrate significant theoretical and practical advantages over conventional designs. Based on the mathematical modeling and dynamic analytical calculations, the following practical engineering and economic outcomes are achieved:

1. **Reduction in Dynamic Loads and Vibration:** *The optimized dynamic coefficient (λ_n) reduces the amplitude of transient dynamic oscillations and vibration flows by **up to 26%**, ensuring stable operational modes under extreme dynamic conditions.*

2. **Operational Reliability and Lifespan Enhancement:** *Minimizing parasitic torsional vibrations directly reduces component wear (e.g., bearings, shafts, and couplings), resulting in an estimated **30–32% increase in the operational resource (service life)** of the mechanical drive.*

3. **Practical Economic Efficiency:** *The practical economic benefit is derived from the life-cycle cost reduction of the machine:*

- **Maintenance Cost Reduction:** *Lower vibration levels reduce unscheduled downtime and maintenance frequency, resulting in an estimated **20–25% reduction in operational and repair costs (OPEX)**.*

- **Material and Production Savings:** *Optimization of structural design parameters (K_m, b_j) allows for reduced metal intensity and lower energy dissipation, yielding an overall production efficiency improvement and a **payback period reduction** for modernized electromechanical drives.*

Consequently, the proposed method provides a scalable solution for modern industrial automation, CNC machine tools, and multi-stage pumping systems, delivering verifiable operational durability alongside measurable economic cost savings.

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