

The object of research is the process of dynamic balancing of rotors on a balancing machine in the process of restoring electric machines.

The work is aimed at improving the quality of balancing rotors in the process of major repairs of traction electric motors for electric trains. The problem addressed was the quality of balancing the rotors of electric machines on stationary balancing machines. According to the conventional balancing technology, the rotor to be balanced is mounted on the supports of the balancing machine with support surfaces that usually have mechanical defects. These defects cannot be eliminated by machining due to the peculiarities of rotor repair technology. Theoretical and experimental studies of the effect of damage to the rotor support surfaces on the balancing parameters have been carried out. It has been proven that the properties of the bearing surfaces of the rotor during its balancing on a balancing machine significantly affect the results of determining the imbalance. At the same time, the difference in the mass values of balancing loads can reach 25 %. This is because damage to the bearing surfaces of the rotor generates false signals unrelated to the imbalance.

In order to increase the accuracy of determining the mass of balancing loads during rotor balancing, it is proposed to improve the technological process of balancing. The improvement involves the inclusion of a frequency filter in the signal conversion chain of acceleration sensors. The filter is designed to separate signals with a frequency greater than the rotational frequency of the rotor.

A condition for the practical application of the research results is the expediency of introducing a frequency filter of signals of acceleration sensors with a threshold frequency of filtering signals that exceeds the rotational frequency of the rotor into the schematic diagram of the balancing machine

Keywords: railroad transport, vibrations of electric machines, imbalance, mechanical balancing of rotors, balancing machine

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IMPROVING THE TECHNOLOGICAL PROCESS OF BALANCING ELECTRIC MACHINE ROTORS ON A BALANCING MACHINE

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1. Introduction

One of the most common defects of electric machines, which limits their service life, is the mechanical imbalance of the rotor. The presence of an imbalance of the rotor of the electric motor, as a rule, leads to vibrations, the level of which is one of the main factors affecting the reliability of traction electric motors. The imbalance of the armatures (rotors) of electric machines is the main source of vibrations that are transmitted to the body of railroad passenger cars and worsen their comfort indicators. For this reason, great attention is paid to issues of diagnosis and ways to eliminate the imbalance of rotors of electric motors during their production, operation, and repair. The causes of imbalance in the rotors of electric motors are the result of many features of the design and technical operation of electric motors. In general, the causes of imbalance can be divided into two groups: the first is an imbalance formed during the production or repair of electric motors; the second is an imbalance arising in operation. The cause of the imbalance of the first group can be defects in the manufacture and assembly of the rotor core, uneven laying of electrical windings, uneven distribution of insulating materials over the rotor volume, etc. The causes of the imbalance of

the second group are operational changes in the geometric and physical and technical parameters of the rotor. The main parameters are uneven wear and destruction of rotor elements; aging of electrical insulation; the appearance of residual deformations during uneven heating of individual elements of the rotor, etc.

Balancing the rotor is a mandatory final technological operation at the stage of engine acceptance tests in the process of its production or repair. Balancing is performed on balancing machines based on monitoring the level of vibrations during rotor rotation. The balancing process includes two stages:

- 1) determining the parameters of imbalance;
- 2) its elimination by installing balancing loads or removing a part of the rotor body.

The rotor is considered balanced if the imbalance does not exceed the maximum permissible value defined by DSTU ISO 21940-21:2017. Mechanical vibration. Rotor balancing (ISO 20816-1:2016. Mechanical vibration. Measurement and evaluation of machine vibration).

Scientific research aimed at increasing the accuracy of balancing is important for improving the reliability of electric machines, which in turn is of great importance in the practice of operating traction machines of electric transport.

2. Literature review and problem statement

Work [1] reports the results of a thorough analysis of the causes of electric motor malfunctions, in particular, the causes of rotor imbalance. The paper states that armature residual unbalance is the most common fault in rotating electrical machines. Among the causes of imbalance are the following: production errors; anisotropy of material properties; thermal deformation; wear during operation, etc. In [1], the issue of the influence of internal factors of the balancing process on its quality remained unresolved. The reason is that the balancing process of an exclusively new rotor with no bearing surface damage was considered.

The desire to reduce critical failures of electric motors has contributed to the development of a new trend in maintenance (TO) and repair of rotating electric machines. It consists in replacing the corrective approach to maintenance with a preventive one. The trend is based on operational monitoring of the electric motor system [2]. The prospect of this trend is also confirmed in work [3]. In particular, the relevance of the development of systems of built-in diagnostics of electric drives is proven, which consists in the ability to detect malfunctions at an early stage. This could reduce the magnitude of the harmful effects of electric motor failure. The conclusions drawn in works [2, 3] made it possible to systematize the causes of rotor imbalance and confirm the prospect of considering internal factors that affect balancing results. However, works [2, 3] do not analyze internal disturbances that may occur in the operation of the electro-mechanical drive and affect the results of rotor imbalance monitoring.

As evidenced by the data given in paper [4], various types of mechanical and electrical malfunctions cause noises and vibrations of different amplitudes and frequencies compared to the normal noise of equipment operation. Study [4] gives examples of some electrical and mechanical faults in induction motors and how these faults affect motor vibration. Each type of fault produces vibrations of a certain frequency. It is claimed that vibration spectrum analysis could identify these specific frequencies and predict certain faults through them. The level of danger of the development of individual malfunctions is assessed by the degree of approximation of the vibration amplitude to the set maximum permissible level. A possible model of two degrees of freedom for electric motors is presented. On the basis of mathematical modeling, vibration spectra for the main types of malfunctions were obtained. However, the work does not consider the influence of internal disturbances that occur during the balancing process on the balancing machine. In particular, cases where the bearing surfaces of the rotor are damaged have not been considered. Therefore, the vibrational frequencies associated with this defect have not been resolved.

The most likely cause of failure of electric motors was found in [5] based on a statistical analysis of the operation of asynchronous electric motors. Such a reason turned out to be rotor imbalance, that is, the presence of an imbalance acquired during operation. It was also theoretically justified and experimentally confirmed that imbalance could be the cause of significant vibrations of engine elements. The work claims that the vibration response, which contains information about the existing defects of electric machines, could be measured directly under the operating mode of the electric motor without the need to intervene in its design. The prom-

ising technology of diagnosing electric motor defects based on the analysis of vibration parameters has been proven. But the use of the proposed method is a means of built-in diagnostics and does not solve the issue of eliminating imbalance in the process of repairing electric motors. Thus, there is a fundamental limitation of the proposed method for correcting the operational unbalance of the rotor. The issue of balancing accuracy was not resolved due to the fact that the vibration response associated with internal disturbances of the technological process of balancing on the balancing machine was not considered.

Paper [6] could be considered a development of vibration monitoring ideas for effective prediction of malfunctions in an electric motor. The principles of their identification proposed in the paper are based on the uniqueness of the vibration spectra of certain malfunctions. The amplitude and frequency of vibration change when the operating conditions of the machine change. The ideas presented in the work could be useful for improving the quality of balancing on balancing machines. However, the proposed principle of identification of mechanical faults does not consider the internal noise of the balancing machine during balancing.

Works [7, 8] are a continuation of the method of analysis of harmonic spectra of vibration to determine various malfunctions based on comparison with the spectra corresponding to the normal operation of the electric motor. Work [7] develops the principles of vibration measurement using strain gauges. In addition, the paper made an important conclusion about the influence of the characteristics of the engine mounting base on the amplitude of the frequency components of the spectrum of vibration harmonics. But, apart from indicating this influence, the work does not contain its quantitative characteristics. Paper [8] considers a similar method of detecting malfunctions in traction electric motors of urban electric transport by comparing their vibration spectra with reference spectra. The main defects, the appearance of which is accompanied by a high level of engine vibration, are identified – rotor imbalance and misalignment. They are related to the frequency of rotation of the rotor and its multiples, the so-called rotor harmonics. The experiments confirm that any defects that occur in electric machines and are determined by the general level of vibration could be decomposed into a spectrum that would consist of n components, starting with rotating frequencies and ending with high-order harmonics. As the harmonic component increases, the spectrum decays exponentially. The work claims that a diagnostic sign of rotor imbalance is a proportional increase in the amplitudes of rotor harmonics. At the same time, the exponential law of the distribution of the levels of rotor harmonics is preserved. Implementation of the ideas presented in works [6–8] requires high-precision and bulky measuring equipment: sensors, converters, recorders, and analyzers. This makes it impossible to implement them on serial electric drives of small and medium power. Thus, the described method could be used only for initial diagnosis with subsequent correction of imbalance on stationary balancing machines. When comparing rotor vibration spectra with reference ones, vibrations associated with internal disturbances of the technological process of balancing are not considered, which does not make it possible to solve the issue of the quality of balancing during rotor repair.

Structural diagrams of balancing machines for measuring the vibration of rotors and analysis of methods for diag-

nosing the technical condition of high-speed drums are given in paper [9]. It was concluded that the method of dynamic two-plane balancing is the most effective for high-speed machine drums. To substantiate the method, the criterion of clarity and simplicity of calculation operations and carrying out balancing with the least number of starts was adopted. However, with this approach, the accuracy of measurements is pushed to the background, which is not acceptable from the point of view of the quality of balancing.

Study [10] presents an overview of classical balancing methods used to eliminate rotor imbalance. The methods of balancing were analyzed: the method of influence coefficients; method of modal balancing. The classification of the latest balancing methods is also considered. The course of research, stages of work, advantages and disadvantages of these methods are described in detail. However, the question of the quality of balancing during stationary tests remained outside the analysis. Therefore, the influence of the state of the rotor base system on the supports on the quality of balancing was not considered.

Work [11] provides a detailed review of rotor balancing methods and devices for its implementation. In particular, the advantage of “floating” elastic roller supports of the balancing machine over rigid ones is proven. However, all conclusions are based on the assumption of an ideal state of the rotor bearing surface.

Paper [12] considers a balancing system that uses pendulum supports of a balancing rotor. This is claimed to significantly improve balancing accuracy compared to conventional rigid roller bearing systems. However, work [12] did not pay attention to an important issue – the appearance of two subsystems with different characteristics of their own oscillations. In particular, it is not taken into account that the support pendulums could oscillate with different amplitudes and with a phase shift, which makes it difficult to evaluate the balancing results.

Our review of the literature [1–12] reveals that the principle of resting the rotor on the supports of the balancing machine could affect the accuracy of balancing. In particular, the results of the unbalance determination are influenced by internal disturbances not related to the unbalance. One of these disturbances is the excitation that occurs in the contacts of the bearing surfaces of the rotor and the roller bearings. Therefore, the study of the influence of the quality of the rotor bearing surface on the roller bearings on the results of the balancing process is appropriate. Depending on the obtained results, technical solutions could be searched for to improve the technological process of balancing the rotors of electric machines on a balancing machine.

3. The aim and objectives of the study

The purpose of our study is to determine the influence of the disturbances associated with the properties of the rotor support surfaces during balancing on the results of determining the imbalance. This will make it possible to improve the quality of balancing due to more accurate determination of the masses of balancing loads.

To achieve the goal, the following tasks were set:

- to perform a theoretical comparison of the results of balancing the rotor with different quality of support surfaces;
- to test experimentally the effect of damage to the rotor support surfaces on balancing results.

4. The study materials and methods

The object of research is the process of dynamic balancing of rotors on a balancing machine in the process of restoring electric machines.

The main hypothesis of the study assumes that the system of resting the rotor on the supports of the balancing machine could introduce internal disturbances that are perceived by vibration sensors and negatively affect the accuracy of the measurement results.

In the process of analysis of the technology of balancing the armatures of electric machines, elements of the theory of unbalance of the rotating rotor were used. The mathematical model of the oscillations of the balancing rotor is built on the basis of the classical Lagrange differential equation of the II kind. To integrate the system of differential equations, the Given-Odesolve procedure from the Mathcad computing package (developer – Parametric Technology Corporation, USA) was used. Experimental elastic-dissipative characteristics of the floating supports of the balancing machine are included in the mathematical model. The damping coefficient of the supports for the mathematical model is determined by the method of the theory of free oscillations as follows:

- 1) the rotor is deflected in the horizontal direction on roller floating supports for a distance of 10–12 mm and released;
- 2) the rotor carries out free oscillations with damping until a complete stop;
- 3) oscillation parameters are recorded in the form of an oscillogram of oscillations.

The experiments were carried out on the EI2000 balancing machine manufactured by NORMA-UA LLC (Ukraine). The measuring and recording system of the machine is equipped with the following elements:

- acceleration sensors in elastic supports of the ADXL202 type (Analog Devices) with a measurement range of 0.1–500 m/s²; frequency range 0–100 Hz;
- analog-digital converter KADL-6;
- computing and recording equipment based on the Windows 8 OS with a test protocol output device.

The manufacturing plant has embedded in the EI-2000 balancing machine the possibility of performing balancing by two methods: the method of balancing with test loads and the method of constant influence coefficient.

The first method of balancing involves performing three trial runs in order to determine the dynamic coefficient of influence (DCI). In this method, the DCI is a variable value. In the second method, DCI has a constant value (according to factory settings, DCI=15). The indicated coefficient is related to the technical characteristics of the machine. In addition, the weight of the rotor and the distance between the supports of the machine are entered into the program. For the experiments, the first method was used as it has a greater accuracy in determining the weight of the balancing loads and the installation location.

The software of the balancing machine provides that the experimental session is conducted by continuously recording readings for five seconds. According to factory settings, the periodicity of taking readings is 50 thousand measurements per second. Based on the measurement results, an experiment protocol is formed according to the template embedded in the machine program. The reliability of the balancing results is guaranteed by systematic checks by DP “UKRMETRTESTSTANDART” (Ukraine).

5. Results of investigating the influence of properties of the rotor support surfaces on the results of determining the imbalance

5.1. Theoretical comparison of the results of rotor balancing with different quality of support surfaces

Imbalance of the rotor of the electric motor occurs when the main axis of inertia of the rotor does not coincide with the axis of its rotation. It is known that three types of imbalances are possible. Static imbalance occurs when the main axis of inertia and the axis of rotation are parallel and do not coincide. Torque imbalance is manifested when the main axis of inertia and the axis of rotation intersect at the center of mass of the rotor. Dynamic imbalance occurs when the main axis of inertia and the axis of rotation do not intersect at the center of mass, or do not intersect at all, that is, they cross [10]. The last type of imbalance is the most typical of the armatures (rotors) of traction motors.

The main vector of the centrifugal force of inertia of the rotor during its rotation around the axis of rotation is determined from the following formula [10]:

$$\vec{F} = m \cdot \omega^2 \cdot \vec{r}, \quad (1)$$

where m is the mass of the rotor; ω – angular speed of rotation of the rotor; r – eccentricity of the rotor – distance from the axis of rotation to the center of mass of the rotor (m); $r = \sqrt{x_c^2 + y_c^2}$; x_c, y_c are the coordinates of the center of mass of the rotor C in the plane coordinate system Oxy .

Components of the centrifugal force of inertia of the rotor along the axes Ox, Oy :

$$F_x = m \cdot \omega^2 \cdot x_c; \quad F_y = m \cdot \omega^2 \cdot y_c. \quad (2)$$

Fig. 1 shows the diagram of the forces acting on the rotor in the presence of the inertial centrifugal force of imbalance F .

The balancing method was defined quite a long time ago and is currently classic [10]. At the first stage of balancing, owing to the capabilities of the balancing machine, reactions in the resistances are determined. When rotating the rotor around the z axis at the first stage of the balancing operation before installing the balancing loads, the system of balance equations takes the following form:

$$\begin{cases} \sum X = -X_A - X_B + m \cdot \omega^2 \cdot x_c = 0; \\ \sum Y = -Y_A - Y_B + m \cdot \omega^2 \cdot y_c = 0; \\ \sum M_x = X_A \cdot a - X_B \cdot b - m \cdot \omega^2 \cdot x_c \cdot z_c = 0; \\ \sum M_y = Y_A \cdot a - Y_B \cdot b + m \cdot \omega^2 \cdot y_c \cdot z_c = 0. \end{cases} \quad (3)$$

Theoretically, the system of equations (3) has no solution due to the fact that the number of unknown system of equations ($X_A, X_B, Y_A, Y_B, x_c, y_c, z_c, a, b$) exceeds the number of equilibrium equations. Thus, the system of equations is only a mathematical notation of the experiment, as a result of which the unknown reactions of the supports X_A, X_B, Y_A, Y_B are determined.

These reactions are input parameters in the second stage of balancing for further determination of the masses of the balancing loads and their locations. Fig. 2 shows the diagram of forces acting on the rotor after installation of balancing loads. A condition for high-quality balancing is the absence of dynamic reactions on the rotor supports A and B after installation of the balancing loads.

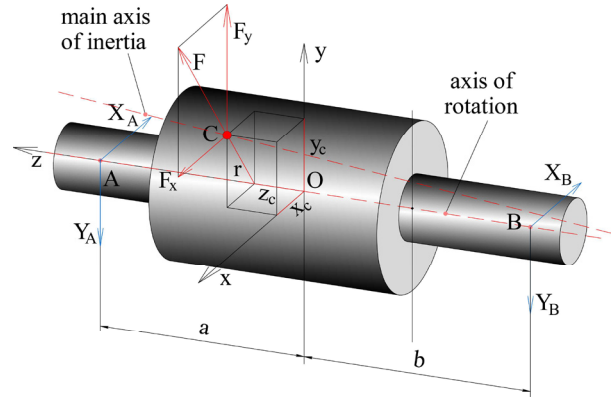


Fig. 1. Estimation scheme of the rotor in the presence of imbalance

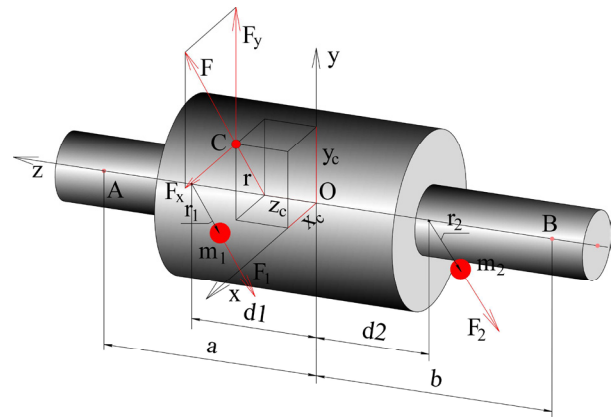


Fig. 2. Estimation scheme of the rotor after installation of balancing loads

In essence, the balancing loads m_1 and m_2 replace the reactions in the supports by their action, so one can build a simple system of equations – the algorithm for determining the loads m_1, m_2 :

$$\begin{cases} A' + B' - m_1 \cdot \omega^2 \cdot r_1 - m_2 \cdot \omega^2 \cdot r_2 = 0; \\ A' \cdot a - B' \cdot b - m_1 \cdot \omega^2 \cdot r_1 \cdot d_1 + m_2 \cdot \omega^2 \cdot r_2 \cdot d_2 = 0. \end{cases} \quad (4)$$

In equations (4):

$$A' = -\sqrt{X_A^2 + Y_A^2}; \quad B' = -\sqrt{X_B^2 + Y_B^2}, \quad (5)$$

where X_A, Y_A, X_B, Y_B are components of reactions in supports determined at the first stage of balancing.

Fig. 3 shows the estimation scheme for balancing the rotor on a machine with floating supports.

The mathematical model of the oscillating rotor balancing system is built on the basis of the classical Lagrange equation of the II kind:

$$\frac{d}{dt} \frac{\partial T}{\partial \dot{q}_i} + \frac{\partial Q}{\partial \dot{q}_i} + \frac{\partial P}{\partial q_i} = F_i, \quad (6)$$

where T, P are the kinetic and potential energies of the oscillating system, respectively;

Q is the dispersion function of vibration dampers;

F_i – generalized external power excitation;

q_i, \dot{q}_i – generalized coordinates and generalized velocities.

The following generalized coordinates are adopted for the estimation scheme:

- $q_1=x_c$ – linear movement of the center of mass of the rotor along the axis Ox ;
- $q_2=y_c$ – linear movement of the center of mass of the rotor along the Oy axis;
- $q_3=\lambda$ – angular displacement of the rotor in the Oxz plane around the Oy axis;
- $q_4=\xi$ – angular movement of the rotor in the Oyz plane around the Ox axis.

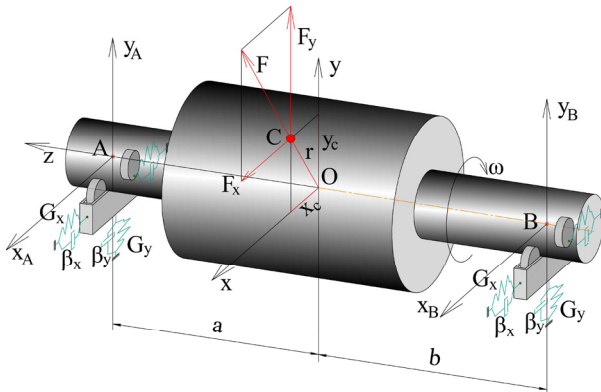


Fig. 3. Calculation diagram of balancing on a balancing machine with roller supports

Accordingly, the system of generalized velocities is accepted for displacement:

$$\dot{q}_1 = \dot{x}_c; \quad \dot{q}_2 = \dot{y}_c; \quad \dot{q}_3 = \dot{\lambda}; \quad \dot{q}_4 = \dot{\xi}.$$

The kinetic energy of the system associated with the movement of the rotor in the adopted coordinate system is determined by the following expression:

$$2T = m \cdot (\dot{x}_c^2 + \dot{y}_c^2) + J \cdot (\dot{\lambda}^2 + \dot{\xi}^2), \tag{7}$$

where m, J are, respectively, the mass of the rotor and the moment of inertia of the balancing rotor around the Oy axis.

The potential energy of the system is related to the presence of elastic elements G and depends on the movements of the rotor at points A and B :

$$2P = G_x [(x_c - a \cdot \lambda)^2 + (x_c + b \cdot \lambda)^2] + G_y [(y_c + a \cdot \xi)^2 + (y_c - b \cdot \xi)^2], \tag{8}$$

where G_x, G_y are the horizontal and vertical stiffness of the roller support.

Dissipation function of vibration dampers:

$$2Q = \beta_x [(\dot{x}_c - a \cdot \dot{\lambda})^2 + (\dot{x}_c + b \cdot \dot{\lambda})^2] + \beta_y [(\dot{y}_c + a \cdot \dot{\xi})^2 + (\dot{y}_c - b \cdot \dot{\xi})^2], \tag{9}$$

where β_x, β_y are the horizontal and vertical damping coefficients of the roller support.

$$\frac{\partial T}{\partial \dot{x}_c} = m \cdot \dot{x}_c; \quad \frac{\partial T}{\partial \dot{y}_c} = m \cdot \dot{y}_c; \quad \frac{\partial T}{\partial \dot{\lambda}} = J \cdot \dot{\lambda}; \quad \frac{\partial T}{\partial \dot{\xi}} = J \cdot \dot{\xi}; \tag{10}$$

$$\frac{\partial P}{\partial x_c} = G_x [2x_c - (a-b) \cdot \lambda];$$

$$\frac{\partial P}{\partial y_c} = G_y [2y_c + (a-b) \cdot \xi];$$

$$\frac{\partial P}{\partial \lambda} = G_x [(a^2 + b^2) \cdot \lambda - x_c \cdot (a-b)];$$

$$\frac{\partial P}{\partial \xi} = G_y [(a^2 + b^2) \cdot \xi + y_c \cdot (a-b)]. \tag{11}$$

$$\frac{\partial Q}{\partial \dot{x}_c} = \beta_x [2\dot{x}_c - (a-b) \cdot \dot{\lambda}];$$

$$\frac{\partial Q}{\partial \dot{y}_c} = \beta_y [2\dot{y}_c + (a-b) \cdot \dot{\xi}];$$

$$\frac{\partial Q}{\partial \dot{\lambda}} = \beta_x [(a^2 + b^2) \cdot \dot{\lambda} - \dot{x}_c \cdot (a-b)];$$

$$\frac{\partial Q}{\partial \dot{\xi}} = \beta_y [(a^2 + b^2) \cdot \dot{\xi} + \dot{y}_c \cdot (a-b)]. \tag{12}$$

Generalized external power excitations are components of the centrifugal force of inertia, which is generated by the imbalance of the rotor and are determined by the expressions:

$$F_x = F \cdot \cos(\omega t); \quad F_y = F \cdot \sin(\omega t), \tag{13}$$

where ω is the angular speed of rotation of the rotor during balancing.

The system of differential equations describing the rotation of the rotor on roller supports, in Cauchy form, takes the following form:

$$\begin{cases} \ddot{x}_c = \frac{-G_x [2x_c - (a-b) \cdot \lambda] - \beta_x [2\dot{x}_c - (a-b) \cdot \dot{\lambda}] + F \cdot \cos(\omega t)}{m}; \\ \ddot{y}_c = \frac{-G_y [2y_c + (a-b) \cdot \xi] - \beta_y [2\dot{y}_c + (a-b) \cdot \dot{\xi}] + F \cdot \sin(\omega t)}{m}; \\ \ddot{\lambda} = \frac{-G_x [(a^2 + b^2) \cdot \lambda - x_c \cdot (a-b)] - \beta_x [(a^2 + b^2) \cdot \dot{\lambda} - \dot{x}_c \cdot (a-b)]}{J}; \\ \ddot{\xi} = \frac{-G_y [(a^2 + b^2) \cdot \xi + y_c \cdot (a-b)] - \beta_y [(a^2 + b^2) \cdot \dot{\xi} + \dot{y}_c \cdot (a-b)]}{J}. \end{cases} \tag{14}$$

In the system of support of the balancing rotor, there is a rolling contact on the support rollers, which, in the presence of defects of the contact surfaces, could give additional dynamic excitations. As for the support rollers, the quality of their rolling surfaces could be considered ideal. In the case when a new rotor is balanced with polished surfaces of the support necks, their quality is also not in doubt. But when we are dealing with balancing the rotor in the repair process, it should be taken into account that there are always slight damages on the surfaces of the rotor necks associated with the previous assembly and disassembly of rolling bearings. Eliminating these damages is impossible due to the danger of loosening the press connection of the bearings with the rotor shaft.

To determine the possible impact of damage to the surfaces of the rotor necks on the quality of its balancing, an

additional external kinematic excitation in the form of a groove on the rotor support surface was introduced into the mathematical model. The groove had the following parameters: width – Δ and depth – h . Considering that the groove contacts the support rollers twice per revolution of the rotor, the current value of the groove depth, as the amplitude of the kinematic excitation, is determined by the procedure:

$$hx(t) = \begin{cases} h & \text{if } \sin\psi < \sin(\omega t) < 0; \\ 0 & \text{otherwise.} \end{cases} \quad (15)$$

For the object of theoretical research, the parameters of the RT-51D traction electric motor armature and the experimental parameters of the elastic supports of the balancing machine were used as input parameters. The integration of the system of differential equations (14) was performed using the Given-Odesolve procedure from the computer mathematical package Mathcad. Zero values of independent variables (displacements and velocities) were taken as initial integration conditions. This is justified by the fact that the role of the main exciting factor – dynamic “shocks” from the groove on the rotor’s support surface – was investigated. To obtain sufficient accuracy of the results, the integration step was equal to 10^{-4} s. The integration time of two seconds was sufficient, given that the oscillations become stable after the first second.

Fig. 4 shows the results of solving the system of differential equations (14) in the form of time dependences of displacements $x(t)$ and $y(t)$ along the Ox and Oy axes, respectively. Two cases are presented: *a* – rotor bearing surface without damage – “smooth” rotor; *b* – the rotor has damage on the bearing surface in the form of a groove with a width of $\Delta=1.5$ mm and a depth of $h=0.2$ mm.

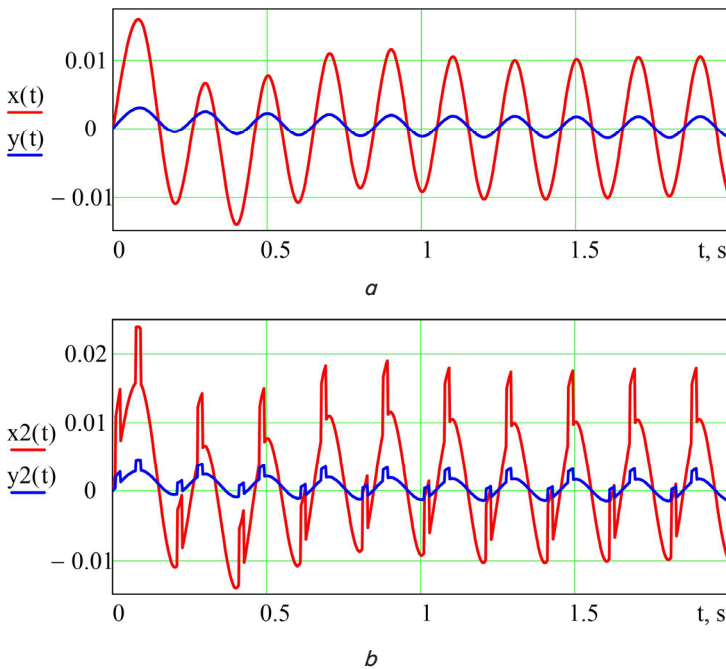


Fig. 4. Calculated dependences of horizontal (x) and vertical (y) movements of the rotor on time (t): *a* – for the case of a “smooth” rotor; *b* – for a rotor with damage on the rotor bearing surface

To determine the degree of influence of damage on the rotor bearing surface on the error of balancing results in

a wide range of unevenness parameters, calculations were performed for the following field of Δ and h values: $\Delta=1.0$; 2.0 mm; $h=0.1$; 0.25; 0.5; 0.75; 1.0 mm.

Fig. 5 shows a geometric interpretation of the influence of the parameters of damage to the rotor support surface (groove width Δ and depth h) on the error in determining the imbalance parameters in comparison with a “smooth” rotor.

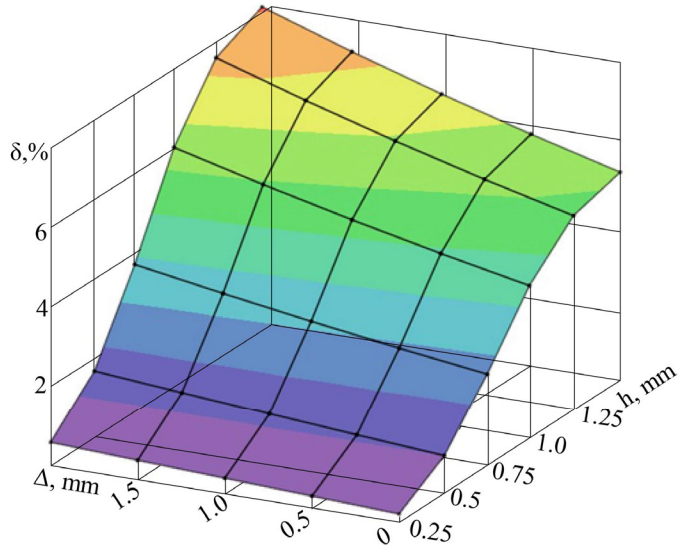


Fig. 5. Theoretical dependence of the error of determination of rotor imbalance parameters on the characteristics of damage to its support surface: δ – relative error of determination of imbalance parameters, %; Δ – width of the groove of damage to the supporting surface of the rotor, mm; h – groove depth, mm

The error in determining the imbalance parameters was estimated as the relative deviation of the constant values of the average amplitudes of the forced oscillations of the damaged rotor along the Ox axis in comparison with the average amplitudes of oscillations for the “smooth” rotor as a base variant.

5. 2. Experimental verification of the influence of properties of the rotor support surfaces on balancing results

5. 2. 1. Determining the elasticity and damping characteristics of roller supports

The rotor support system when balancing on two pairs of roller supports is the most widespread [13]. Roller supports – “floating” with the possibility of movement in the horizontal-transverse direction – axis Ox . In the same direction, accelerometers are installed to control accelerations. The software allows integration of acceleration data into velocity and displacement. The program also calculates the masses of the balancing loads and the phase angles corresponding to the maximum amplitudes of oscillations to determine the position of the load installation [11, 14].

Fig. 6 shows a photograph and diagram of a typical roller support of a pendulum “floating” type balancing machine. Transverse and vertical movement of supports is limited by stiffeners, and oscillations are damped by dampers. The transverse stiffness of supports G_x depends on the vertical load of the support by the weight of the rotor. The stiffnesses G_x and G_y were de-

terminated using a dynamometer and an indicator measuring head and were: $G_x=3.227$ kN/m; $G_y=5900$ kN/m.

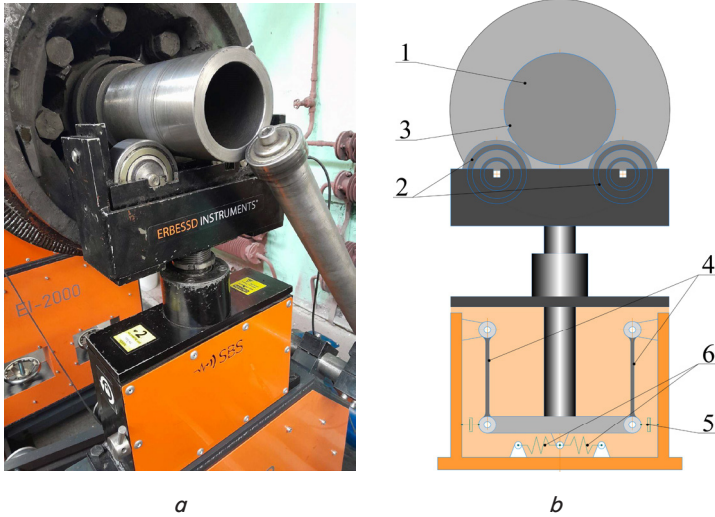


Fig. 6. Floating supports of the balancing machine: *a* – photograph of the floating support; *b* – rotor installation scheme on supports: 1 – balanced rotor; 2 – support rollers; 3 – groove-damage on the rotor support surface; 4 – pendulum support; 5 – dampers; 6 – springs

The damping coefficient of the supports in the transverse direction β_x was determined by the method of free oscillations. Fig. 7 shows a fragment of the protocol for recording free oscillations of the rotor on the support.

The logarithmic decrement of the damping of oscillations of elastic supports in accordance with DSTU 2473-94 “Mechanical oscillations. Terms and definitions” (ISO/TR 19201:2013(en). Mechanical vibration. Methodology for selecting appropriate machinery vibration standards):

$$\alpha = \ln \frac{A_n}{A_{n+1}}, \tag{16}$$

where A_n, A_{n+1} are adjacent amplitudes from the oscillogram recording of oscillations (Fig. 7).

According to the theory of classical oscillations, the damping coefficient could be found using the formula:

$$\beta_x = \frac{2m}{T} \ln \frac{A_n}{A_{n+1}}, \tag{17}$$

where T is the oscillation period, s; m is the mass of the rotor, kg.

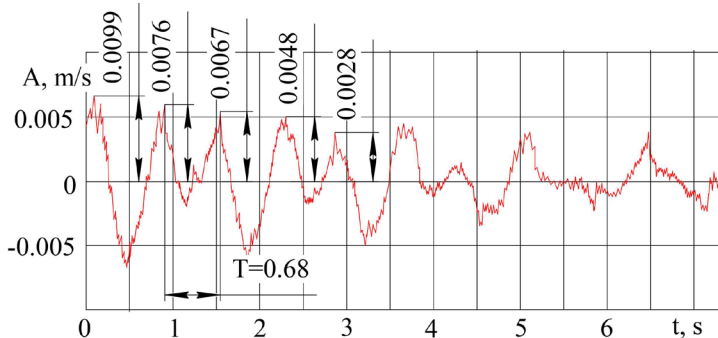


Fig. 7. Experimental oscillogram of free oscillations of the rotor along the axis Ox , obtained to determine the damping coefficient of the supports

The results of the calculations are summarized in Table 1.

Table 1

Results of calculating the damping coefficient of the supports in the balancing machine

Amplitudes of oscillation velocities A , 10^{-3} m/s	A_0	A_1	A_2	A_3	A_4	Mean values
Ratio x_n/x_{n+1}	1.39	1.30	1.13	1.40	1.71	1.39
Logarithmic decrement of oscillation attenuation α	0.33	0.26	0.13	0.33	0.54	0.33
Oscillation period T , s	0.68					
Rotor weight m , kg	783.5					
Damping coefficient β , N·s/m	760.5					

The resulting values G_x, G_y, β_x were used for calculations when integrating the system of differential equations (15).

5. 2. 2. Comparison of experimental results of unbalance determination for rotors with different properties of contact surfaces

To experimentally verify the results of mathematical modeling, a series of experiments was conducted with the rotor of the RT-51D electric motor, which underwent major repairs with the replacement of electrical windings, impregnation and drying, and was prepared for balancing.

The general view of the balancing machine with the measuring and recording complex is shown in Fig. 8.

The experiment was conducted on a rotor “rejected” in the process of overhaul, for which it was possible to grind the bearing surfaces. The bearing surfaces of the rotor before grinding had a number of visually visible damage to the surfaces for mounting the rolling bearings (Fig. 9).

The experimental results were generated by the computer-recording system in the form of oscillograms of vibration-speeds of horizontal vibrations of the rotor at the points of support on elastic supports.

Fig. 10 shows examples of experimental oscillograms of rotor deflection speeds along the Ox axis for two cases:

- a) for a “smooth” rotor after grinding the support surfaces;
- b) for a rotor with damage on the rotor support surfaces on the supports of the balancing machine (before grinding the support surfaces).

During the tests, the following balancing parameters were determined: masses of balancing loads m_1, m_2 (kg) and phase angles for determining the position of balancing loads – α_1, α_2 (degrees). The balancing parameters were calculated by the built-in algorithm of the balancing machine software, which is the intellectual property of the manufacturer, so it is not given here.

Table 2 gives the results of determining the balancing parameters for two cases: a “smooth” rotor and a rotor with defects on the support surfaces

As could be seen from Fig. 10 and Table 2, there is a significant difference in the values of the obtained balancing parameters for rotors with different properties of the support surfaces.

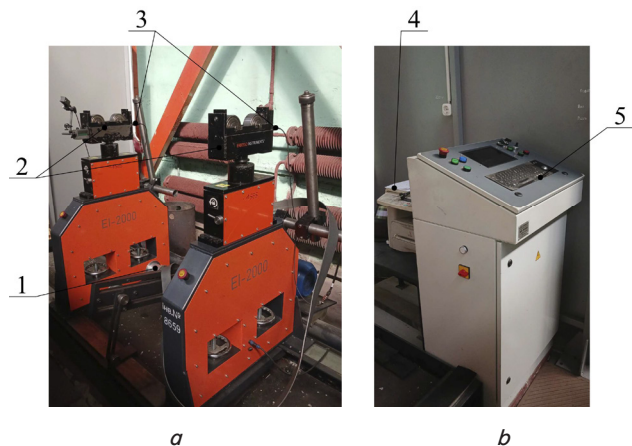


Fig. 8. General view of the balancing machine: *a* – modules of floating supports; *b* – measuring complex and registration devices: 1 – rotor rotation drive, which is balanced; 2 – elastic supports; 3 – acceleration sensors; 4 – a device for outputting test results; 5 – machine control panel

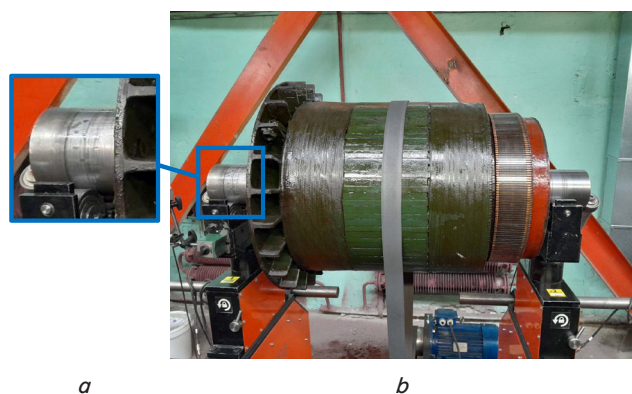


Fig. 9. The RT-51D electric motor rotor installed on the EI-2000 balancing machine: *a* – general view of the rotor support surface with visible damage to the rolling support surfaces; *b* – installation of the rotor on supports

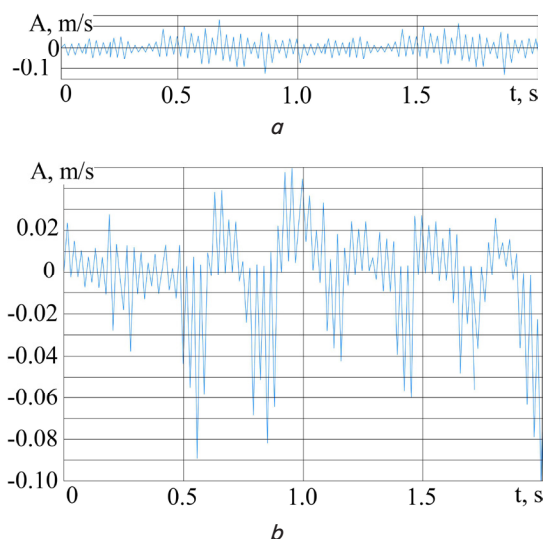


Fig. 10. Experimental dependences of the speed of rotor deflection along the Ox axis (A) in time (t): *a* – for a “smooth” rotor; *b* – for a rotor with defects on the support surfaces

Table 2

Results of determining the balancing parameters for the cases of a “smooth” rotor and a rotor with defects on the support surfaces

Parameters of the rotor support surface	Balancing weights, kg		Phase angles of balancing weights, degrees	
	m_1	m_2	α_{a1}	α_{a2}
“Smooth” rotor	0.147	0.102	31	328
Rotor with defects on bearing surfaces	0.202	0.097	241	216
The absolute difference of the defined balancing parameters	0.055	0.005	21	112
The relative difference of the defined balancing parameters	27 %	5 %	87 %	52 %

6. Discussion of results related to the influence of properties of the bearing surfaces of the rotor on the results of determining the imbalance

Our results confirm the proposed hypothesis about the significant influence of the properties of the bearing surfaces of the rotors on the accuracy of determining their imbalance during balancing on balancing machines. This conclusion could be explained by the detected high level of internal mechanical interferences associated with irregularities on the bearing surfaces of the rotor. Internal disturbances are perceived by the sensors of the balancing machine and are a source of false signals and errors in determining the masses of the balancing loads and their location on the rotor. From Fig. 4, it could be seen that even a slight unevenness of the supporting surface of the rotor, for example, a groove with a depth of 0.1 mm, qualitatively affects the shape of the rotor oscillations in comparison with a “smooth” rotor. The presence on the oscillogram of Fig. 4 two “knocks” per rotation of the rotor is explained by the two-roller scheme of resting the rotor on supports. The theoretical dependences shown in Fig. 5 indicate that the error in determining the imbalance, associated with the presence of irregularities on the surface of the rotor, could reach 8 %. Given that in practice the irregularities on the bearing surfaces of the rotor after dismantling the rolling bearings are much deeper, the real amplitudes of false sensor signals could be much larger. This is confirmed by the experimental results shown in Fig. 10 and in Table 2 for two cases – for a “smooth” rotor and a rotor with irregularities. The experiment showed that the error in determining the balancing parameters due to rotor irregularities could reach 27 % – for the mass of the balancing loads m_1, m_2 and 87 % – for the phase angles for determining the position of the balancing loads α_{a1}, α_{a2} . Thus, it has been proven that the quality of the bearing surfaces of the rotor significantly affects the accuracy of unbalance determination on balancing machines.

The metrological cause of the imbalance has been established, which is related to the technology of determining the balancing parameters on the balancing machine. It could be considered a development of the classification of the causes of rotor imbalance given in [1]. Despite the fact that it is not a direct cause of imbalance, it affects the accuracy of its determination. Taking into account the error associated with internal disturbances of the technological process of balancing makes it possible to significantly improve the quality of rotor imbalance monitoring, which is not taken into account in [1].

The proposed improvement of the technological process of rotor balancing increases its quality. This is achieved by eliminating the assumption about the “ideality” of the rotor bearing surfaces on the roller supports, which are based on the technologies given in [2–7]. In contrast to [2], in which the vibrations of a partially damaged rotor were analyzed, it is proposed to take additional consideration of the disturbances associated with the process of measuring the balancing parameters. At first glance, it may seem obvious that there is a simple technological solution to this problem, namely, the use of high-quality rotor support surfaces for balancing, as suggested in [3, 4]. This solution really works in the case of a new rotor when it is manufactured. But when balancing the rotor during the overhaul of the electric motor, the solutions proposed in [3, 4] are less accurate. The reason for this is that the surfaces of the rotor necks, as a rule, have damage associated with the previous disassembly of the rolling bearings. It is not possible to eliminate irregularities on the rotor necks due to the danger of loosening the press connection of the bearings with the rotor shaft. A variant of expanding the capabilities of the methods outlined in [5] could be their use in combination with a stationary balancing machine. In this option, it is possible to take into account internal obstacles of the technological process of monitoring the imbalance. In contrast to [6], it is proposed to additionally take into account internal disturbances, the source of which could be the noise of the rotor suspension system on the balancing machine. Filtration of false signals from acceleration sensors in the process of rotor imbalance control [7] will make it possible to increase the reliability of the measurement results during their overhaul of the rotors.

Despite the widespread practice of restoring rotors, available results [8–12] do not take into account the technological features of overhaul of electric machines. In contrast to [8, 9], in which internal disturbances are not considered as false signs of imbalance, their consideration makes it possible to reduce the level of errors in diagnosing the rotors of electric machines. In contrast to [10, 11], in which the analysis of diagnostic signs of rotor imbalance excludes only operational factors, it is proposed to take into account the internal factors of the technological process of balancing. In particular, such factors could be excitation from the rotor support system on the supports of the balancing machine.

According to statistics, up to 80 % of electric cars on Ukrainian railroads are overhauled without mechanical processing of the rotor bearing surfaces. This is considered more cost-effective compared to the production of new electric machine rotors. The field of application of the proposed solutions is the technological process of balancing rotors in the process of overhauling electric machines.

The significant effect of internal interferences associated with the unevenness of the rotor support surfaces on the quality of their balancing on the balancing machine has been proved. It is proposed to supplement the classification of rotor imbalance factors with a new component – a metrological factor related to the technology of determining balancing parameters on a balancing machine. This factor is not a direct cause of imbalance but affects the accuracy of its control. It is proposed to improve the technological process of balancing. The essence of the improvement is the inclusion of the frequency filter balancing machine acceleration sensors in the signal conversion chain. The filter makes it possible to separate false signals associated with interference of any origin, except for imbalance.

The use of our results in practice is limited to the balancing of rotors on balancing machines during the overhaul of electrical machines.

Effective application of the results and proposed solutions became possible owing to the fulfillment of the following conditions. The first condition is sufficient accuracy of the input parameters of the balancing rotor and the balancing machine. The second is the relevance of the data regarding the determination of balancing parameters. The third is validation and reproducibility of balancing results through additional test trials.

A disadvantage of the research is that at this stage we are not ready to provide specific characteristics of the frequency filter to improve the balancing technology. The development of this research could be a more detailed analysis of a wide range of internal technological obstacles that affect the quality indicators of balancing. On this path, there are possible technical difficulties in the development of selective frequency filters for the separation of false signals of the balancing process on the balancing machine.

7. Conclusions

1. The properties of the bearing surfaces of the rotor when it is balanced on a balancing machine with a conventional system of support on roller supports significantly affects the results of determining the imbalance. At the same time, the deviation of the values of the determined masses of balancing loads, depending on the level of damage to the supporting surfaces of the rotor, could reach 8 %.

2. Experimental verification of the influence of rotor bearing surface damage on balancing results qualitatively confirmed the results of theoretical studies. Significant discrepancies in determining the balancing parameters of the rotor with damage to the bearing surfaces compared to the “smooth” rotor were revealed. They reach 27 % – for balancing loads and 87 % – for phase angles that determine their position. An increase in the quality of balancing on a balancing machine could be achieved by including in the signal conversion chain of acceleration sensors a frequency filter that filters out signals with a frequency greater than the rotational frequency of the rotor.

Conflicts of interest

The authors declare that they have no conflicts of interest in relation to the current study, including financial, personal, authorship, or any other, that could affect the study, as well as the results reported in this paper.

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Data availability

All data are available, either in numerical or graphical form, in the main text of the manuscript.

Use of artificial intelligence

The authors confirm that they did not use artificial intelligence technologies when creating the current work.

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