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Functional and Engineering Methods of Upgrading the Quality of Induction Traction Electric Motors

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Abstract

The results of research on the creation of induction traction electric motors (EM), which made it possible to improve their technical, environmental and operational qualities, are presented. The main criterion for assessing the quality of EM were selected vibration levels. The main research methods were experimental and statistical methods. New technologies for assembling bearing units assemblies and EM as a whole were studied. New design and technological solutions have been developed. The dynamic characteristics of structural vibration damping have been improved by methods of rational distribution of masses, stiffness and distance between supports of bearings. The achieved reduction in vibration levels makes it possible to predict an increase in the resource developed by the EM by 2-3 times.

Keywords: Vibration, Industrial electrical equipment, Diagnostics, Vibrodiagnostic, Electric car, Electric motor, Induction traction motors, Bearing, Electric motor vibration class.

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1. Introduction

The catastrophic deterioration of the ecology and the decrease in natural resources have become the reason for the development of energy generating and energy saving technologies. In this regard, all over the world there is a refusal from vehicles with internal combustion engines in favor of cars on electric traction [1]. New alternative sources of electrical energy for transport and its infrastructure are being developed [2–4].

One of the main components of electric vehicles is their traction electric motor (EM). The most widespread are induction traction electric motors. They have a number of advantages, namely: simple design, handle ability, relatively low cost, etc. [5, 6]. But despite the widespread use of induction EM, they do not always meet the reliability requirements imposed on them [7]. This is because traction EM operate in a wide range of rotational speeds and variable load modes of frequent stops and starts, and often in difficult road operating conditions. All this leads to high vibration loads of traction EM and, accordingly, to a decrease in their reliability and service life [8, 9].

According to statistics, induction EMs need to be repaired annually about 20-25% of the total number of installed [10]. The reasons for this are the quality of design, manufacture and operation of EM.

Repair of EM is often carried out with disassembly and replacement of parts. This negatively affects the reliability of their work. This is especially true for bearing units (BU) [11, 12]. In addition, as shown by studies of induction EMs with a power from 5.5 kW to 225 kW, the authors carried out [13], after repair and rewinding of EMs, their efficiency drops by 0.5-0.7%.

On the other hand, a decrease in EM vibration levels allows increasing their reliability, durability, resource, reducing the harmful effect on humans and the environment, and reducing the cost of maintenance of electric vehicles in general [14].

Existing national and international standards for rotating electrical machines. But they only set requirements for their vibration levels. The issues of design and manufacturing methods for EM, search for rating constructive and technological methods of achieving specified vibration levels are often solved in practice [15].

Complex theoretical methods for calculating the permissible and specified vibration levels are known [16–18]. They have little integrity, since they cannot take into account many of the design, technological and dynamic factors of the technical condition that exist in real EM. The influence of these factors on the real quality of EM requires a large amount of experimental work. These works consist in further development the EM to the design technical level.

In addition, often these methods are aimed at monitoring a single unit or element, for example, a BU or the size of an air gap, or the state of insulation, etc. [19–22]. Therefore, in practice, the achievement of a given technical level of EM requires a large amount of further developments. The success of these further developments depends on the criteria and research methods applied. This is shown by the results of studies carried out by the authors of [23]. Experiments have confirmed the relationship between the reliable operation of the EM and the alignment accuracy of their rotors. The authors claim that up to 50% of rotary machine failures are due to improper alignment of their rotors. That is, the violation of the technological conditions for assembling EM is one of the reasons for the decrease in their resource.

Therefore, the purpose of this work is to develop constructive and technological methods for increasing the reliability and resource of traction asynchronous EM by experimental methods.

This work is a continuation of the research presented by the authors [7, 24]. These publications discuss the main EM defects. It was revealed that the main EM defects are caused by: BU; electromagnetic system; imbalance in rotating parts. Based on research, Figure 1:

- rational design parameters of BU and electromagnetic systems have been developed;
- it is determined that the vibration level should not exceed the values limited by the right line AB. Exceeding the vibration levels above the permissible values leads to additional vibration loads;
- a decrease in vibration levels below right line AB leads to an increase in the integrity and service life of the EM;
- a decrease in vibration levels by 8 dB below the AB line leads to a transition of the technical quality level to a higher level D, E or F.

A statistical analysis was carried out of ten bipolar induction EM of the ANU92-2 type with a power of 90 kW on rolling element bearings QE3 [7, 12]. He showed that with one production technology, large variations in vibration levels between minimum and maximum values were obtained. Greater instability (variation) of vibration levels is due to design and manufacturing defects.

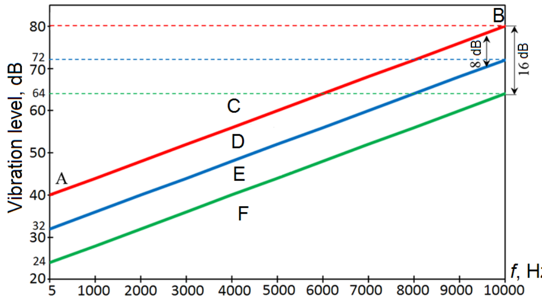

Figure 1. Ranges of electric motor vibration classes

Therefore, it is necessary to solve the following tasks to reduce the vibration of the designed EM:

- determine the optimal distance between supports of bearings;
- to implement the results of research on optimization of landings and clearances of parts of BU;
- to improve the accuracy of assembly of BU and the motor as a whole;
- to reduce vibrations of magnetic origin by choosing the shape of the rotor slot;
- improve the dynamic characteristics of EM.

2. Methods, tools and methodology of experimental research

The theoretical research was based on: vibration methods; analysis; comparison; statistical processing of vibration spectrums of analogs, developed models of EM structures; modeling; generalization.

Vibration methods for assessing the technical level of EM were chosen, since they are the most perfect and integrity [24, 25].

Empirical methods were: observation, natural experiment, measurement, comparison, generalization, processing of research results.

The sources of design defects and EM production technology were determined in one-third octave and narrow-band vibration spectrum with a bandwidth of 3-10% in the frequency range from 5 to 10 kHz in idle and under load modes.

For the initial positions of the evaluation of the EM vibration parameters, the well-known close to proportional dependence of the change in vibration levels (dB) on their technical state, load, rotation frequency, clearances, kinematic and geometric parameters of the EM parts was taken.

Subject of research: traction induction electric motors of the ANU92 type with a short-circuited rotor, with a power of 90 kW. Adjustment of the vibration characteristics of these EMs was carried out by introducing rational design and technological changes with subsequent full-scale vibrodiagnostic tests. The obtained vibration characteristics were compared with the reference vibration values and analogs.

As it is known, when shock excited respond undamped frequency of parts and components close to the excitation source. Therefore, the response was measured at the control points on the flap in the direction of application of the perturbing force.

The detection of the places of unsuccessful distribution of natural frequencies, bending vibrations, mobility in the mating details of the EM in statics was determined by the appearance of individual peaks of the amplitudes of resonance phenomena. The recognition of forced vibration from the resonant frequency in operation was determined by changing the rotational speed or abruptly stopping the rotation. If, with a slight change in the rotation speed, the vibration level of the source under study decreases, then the cause is resonance.

In more complex situations, it was established how the EM vibration changes when stopped. In the presence of resonance and an abrupt stop of the EM, the speed drops rapidly, and the decrease in vibration levels at the resonant frequency is delayed. If there is resonance particles, then the vibration of the source under study is reduced with a slight change in the rotation speed.

The tuning of natural frequencies and forced vibration frequencies was carried out by increasing rigidity and adding mass, increasing damping in order to reduce the response of the EM to the forced frequency. This was achieved, for example, by changing the number and location of the ribs, as well as their location relative to the propagation of vibration waves; a change in the number of rotor slots in order to redistribute the energy of forced forces in the region of high vibration frequencies, which have a lesser effect on bearing wear and on the EM resource as a whole.

The reduction of the amplitudes of the forced forces was carried out by detuning the natural frequencies of the parts from the frequency of the forced forces, by reducing bending vibrations by separating the vibration waves by installing ribs of various shapes and angles relative to the stator and rotor axis.

The number of body ribs determines not only the number of natural frequencies, but also their relative position in the vibration spectrum. In general, it can be seen that the more ribs, the greater the separation of vibration waves and natural frequencies of parts (body), the rigidity of the body and the shield. However, too much stiffness reduces the isoelasticity of the system.

The influence of the EM rigidity on the vibration characteristics was determined by changing the distance between bearings supports, the thickness, the number and location of the radial and annular ribs of the EM body and shield.

The accuracy of the manufacturing technology was increased by providing the capabilities of coaxial boring of the stator magnetic circuit and the bearing seats, their assemblies in the EM housing, by changing the assembly technology of the BU and EM as a whole.

The required location of the part along the length of the shaft, the rib on the machine bodies is determined experimentally by reducing the resonant vibration at a given frequency with the free movement of the rib model along the body or part along the shaft. To determine the undesirable redistribution of vibration, the found installation location on the shaft or the model of the rib on the body is refined by the change in the EM vibration spectrum over the entire spectrum: from 5 Hz to 10 kHz, since vibration redistribution to another undesirable region of the spectrum is possible.

3. Improvement of the dynamic properties of EM

The improvement of the dynamic properties of EM was studied in the following ways:

- adding or removing from the mechanical part of the system any combination of structural stiffeners;
- increase or decrease in mass;
- adding rigidity of the connection;
- adding a damping element.

The research was carried out experimentally on prototypes and new designs of EM by methods of a series of test shock excitations of units or a complete EM with a tool hammer 8202 and vibration analyzers 2033, 2034 of the firm "Brüel & Kjær".

A more detailed description of the experimental setup is presented in the authors' work [12].

Evaluation of the effectiveness of changes made to the design and manufacturing technology to reduce vibrations was adopted to reduce vibration levels below the permissible values of right line AB (Figure 1) to classes E and F.

The technical level of changes introduced in the technology of manufacturing and assembly of EMs was evaluated according to their vibration classes D, E, F and the accuracy of minimum and maximum values of vibration levels and standard deviations from their average values.

The vibration levels of the studied EM were determined in decibels by the maximum root-mean-square values of the vibrational acceleration on a stand with vibroacoustic decoupling, excluding the influence of noise at the points of measurement of engine vibration.

Acceleration $a=3\cdot10^4$ m/s^2 is taken as a conditional zero vibration level.

The engine was installed on vibration damper of the AKCC type. Vibrating contactor of type 4376 was installed on the head of the screw fixing the EM to the vibration dampers.

3.1. Reducing EM vibration levels due to the distribution of mass and stiffness

The main source of EM forced vibration is rotating parts and assemblies. Becours, it is advisable to reduce the ratio of the mass of the rotating parts to the mass of the electric motor, that is, to increase the mass of the engine body. However, an increase in mass without changing the design and technological parameters can lead to a decrease in vibration only in certain bands of the spectrum and, above all, in the low-frequency region. Therefore, it is rational to distribute mass and stiffness in the very source of vibration manifestation.

The result of experimental studies on the effect of changing the mass of bearing parts and the rigidity of assemblies are shown in Figure 2 and Figure 3.

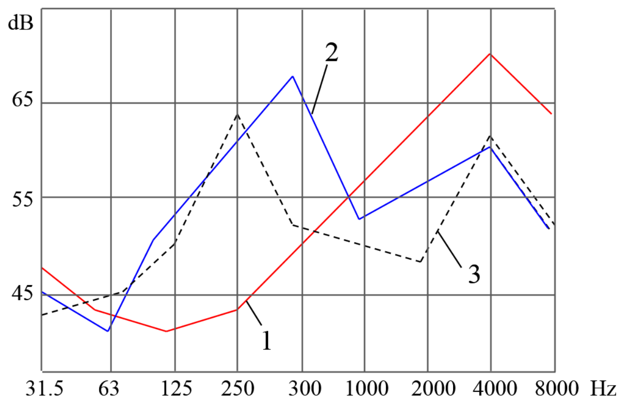

Figure 2. Spectrograms of vibration of rolling bearings type 74-310ЕУШ2: 1 – in the radial direction; 2 – in the axial direction; 3 – with an added weight of 0.5 kg

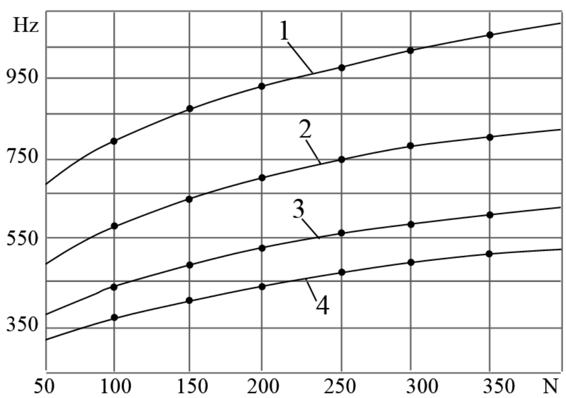

Figure 3. Dependences of the resonance frequency of the outer bearing race 74-310ЕУШ2 on the axial interference force: 1 – without added mass; 2 – with an added weight of 0.3 kg; 3 – 0.8 kg; 4 – 1.3 kg

You can redistribute mass by assigning a mate type to parts. An increase in landing interference leads to a decrease in the natural frequencies of the node.

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The stiffness of the BU can also be changed by adjusting the axial elastic interference of the bearings in the supports.

Bearing preload increases the rigidity of the BU and creates conditions for normal bearing operation. It is known that a correctly selected value of the preload force of the bearings reduces wear on rolling bearings and increases the reliability and durability of the bearings.

With the introduction of an axial tightness, the conditions for "sealing" the supports change. Depending on the value of the spring pressure, the static stiffness of the rotor changes and the natural frequency changes by 20-30%. This circumstance should be borne in mind when calculating the EM vibration activity. Overestimation of the preload force of the bearings when tuning out the natural frequencies of the shaft and rotor system should be avoided, since the interference force affects the durability of the bearing. Excessive interference is dangerous, as it causes pinching of the balls, overloading of the rolling element bearings and increased heat generation.

An increase in the rigidity of the unit causes a decrease in vibration and contributes to an increase in the natural frequencies of the EM rotor, which makes it possible to adjust the natural frequencies of the rotor and EM vibration.

The flexibility of the rotor BU can be reduced by increasing the stiffness of the shaft and the distance between the supports. However, it is necessary to maintain the optimal isoelasticity of the structure.

Studies of the distance between the supports of the BU were carried out on a model of an ANU electric motor with a capacity of 90 kW at idle due to a stepwise change in the length of the primer bottle-top seating surfaces. The diameter of the shaft journal for bearings is 65 mm, for the package of the magnetic circuit 80 mm, the rotational speed is 3000 rpm.

Increasing the rigidity of the EM rotor by reducing the distance between the supports of the BU reduces the levels of tooth vibration of the rotor and the housing (800-3150 Hz, Figure 4). But already with a distance between the supports of 450 mm, the isoelasticity of the BU decreases. Vibrations excited by rolling of the rolling elements increase (160-315 Hz, Figure 4). Insufficient, as well as "excessive" rigidity of the system reduces the reliability of the BU.

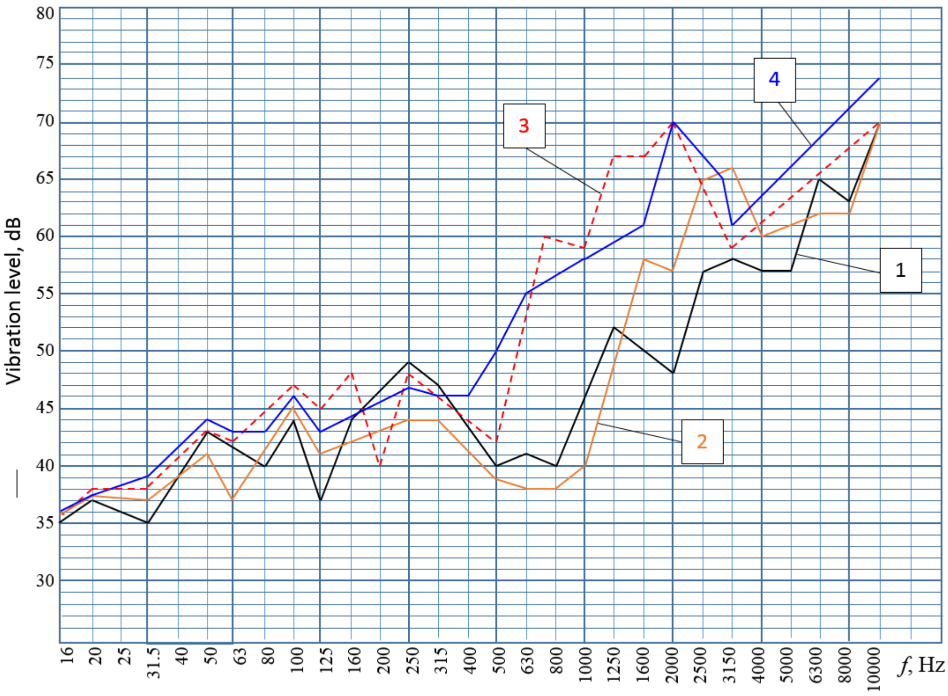

Figure 4. Vibration characteristics of the EM model at the distances between supports: 1 – 450 mm; 2 – 465 mm; 3 –560 mm; 4 – 725 mm

The issue of reducing the vibration generated by the imbalance of the rotating parts was solved by installing a balancing ring on both sides of the EM shaft (Figure 5). The rotor was rebalanced on the stand separately and as part of the EM. Weights in grooves 1, 2 move along the diameter in holes 3 radially (not shown).

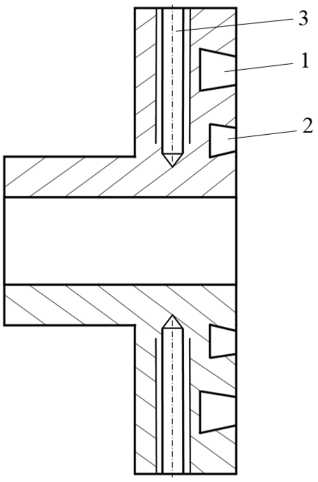

Figure 5. Balancing ring for ED rebalancing in assembly: 1, 2, 3 – places of installation of weights for "coarse", "medium" and "fine" rebalancing, respectively

3.2. Improvement of the technology of assembly of EM bearing units

3.2.1 Existing technology of detailed assembly of BU on the EM shaft

The assembly is carried out in the horizontal position of the rotor shaft 1 with sequential assembly of parts: a primer bottle-top with a bearing 4 and further parts 5-11, Figure 6. Before installation, the primer bottle-top with the bearing is heated in a thermostat to 90 ˚C. After that, the rotor with the BU is installed in the stator. Mounting the shields on both sides of the stator housing on the cap 4 of the BU and fixing the shields to the stator housing (not shown in Figure 6). The processing of the seating surfaces of these shields and the stator housing is carried out as on a separate part. This technology has significant disadvantages:

- does not provide a tight fit of the inner ring of the bearing to the shoulders 2 of the landing journal 3 of the shaft;
- in the process of attaching the cover with 6 screws to the primer bottle-top, the bearing can move;

the primer bottle-top cools down before it can be fixed with nut 11.

As a result, the rigidity of the BU decreases, the inner ring may be skewed on the shaft, and the linear dimensions of the bearing position on the shaft are violated.

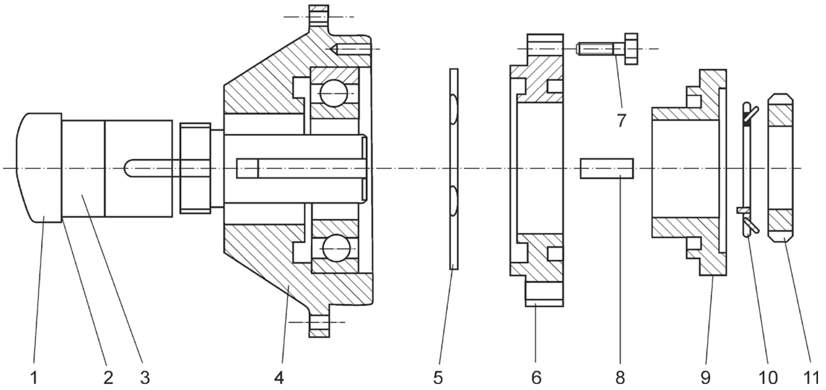

Figure 6. Detail assembly of the EM rotor BU: 1 – rotor shaft; 2 – shaft shoulders for fitting the bearing on the journal 3; 4 – primer bottle-top; 5 – wave spring; 6 – cover; 7 – screw; 8 – key bolt; 9 – balancing ring; 10 – lock washer; 11 – nut for fixing the position of parts on the shaft

This technology does not provide the required accuracy of the BU and the position of the rotor magnetic circuit in the stator. It also neutralizes the high quality of bearings QE6, QE8.

Therefore, when developing a new E right line, changes were made, first of all, in the structural parameters of BU, according to the results of experimental studies with changes in the technology of assembly of BU.

3.2.2 The modified BU assembly technology

The modified technology for assembling the BU is shown in Figure 7. Assembly is performed in the vertical position of the rotor shaft. The bearing is pre-assembled into a one unit (Figure 7, a), heated in a thermostat to the normalized temperature, and then vertically installed on the shaft journal (Figure 7, b), followed by fixation (Figure 7, c).

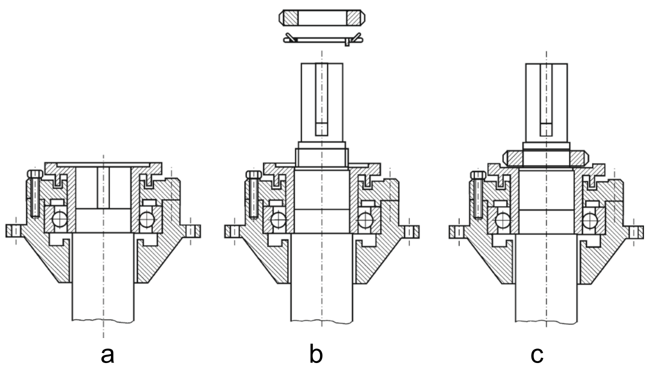

Figure 7. Assembling the BU: a – assembling the bearing in the primer bottle-top; b – installation of the BU with the primer bottle-top on the shaft journal; c – mounting on the shaft

The basis for making changes in the structural parameters of BU is based on the experimental studies of the authors of this work [12].

The described assembly technology ensures that the bearing adheres to the shoulders of the shaft 2 (Figure 7). With the vertical position of the shaft, the possibility of misalignment of the bearing rings under the influence of the weight of the BU is reduced, and the time for adjusting the fixation of the bearing unit with a nut increases (Figure 7, c). This made it possible to improve the quality of QE3 bearings in comparison with QE2 bearings. It was also possible to reduce the scatter of the levels of bearing vibrations in comparison with bearings QE2.

4. Results and discussion

4.1 Improved design and manufacturing technology of the stator housing and EM rotor

Improvement of the dynamic characteristics of the EM housing (Figure 8) was carried out by increasing the rigidity of the flange (1), the number and distribution of external (2) and internal ribs (3), radial ribs of the ventilation channel of the housing (4), shield (6) and annular ribs housing (5).

The number of ribs 2, 3 made it possible to dismember and reduce the amplitude of bending vibrations of the flange; of ribs 4, 7 of the ventilation duct and shield 6 – to dismember their bending vibrations. The best dismemberment of bending vibrations was achieved with five and seven ribs. This number of ribs made it possible to damp vibrations and, in general, determine the required rigidity of the body. The ribs 2 were distributed around the circumference of the flange, and their best location was the places between the ribs 3. The optimal location of the ribs was determined by the characteristics not only of damping of body vibrations, but also by the vibroacoustic qualities of the structure, such as vibration conductivity and sound insulating ability. With an increase in the ratio of the thickness of the flange 1 to the thickness of the body by 3-5 times, the vibration drop on it reaches up to 10 dB.

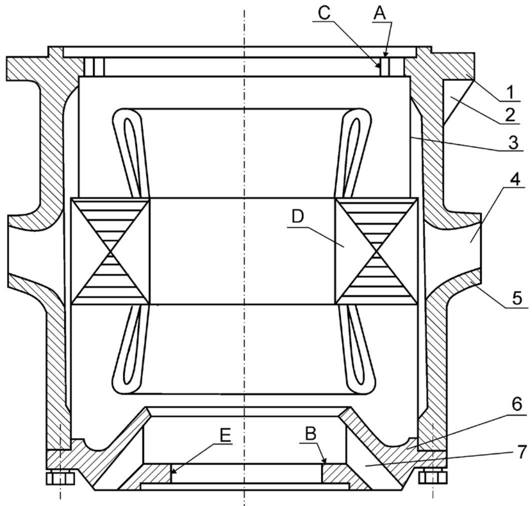

Figure 8. Modified stator housing design: 1 – flange; 2, 3 – ribs; 4 – radial ribs of the ventilation duct; 5 – annular ribs of the ventilation duct; 6 – shield; 7 – ribs of the shield ventilation ducts; A, B, C, D – supporting connecting surfaces of the stator housing; D – stator magnetic circuit package

The task of increasing the accuracy of EM manufacturing was solved by boring from one installation of the magnetic circuit D and places A, C, E and B for the installation of BU (Figure 8). The bore of the hole F of a separate shield for the bearing diameter and surface A/ was carried out from one setup (Figure 9). The increase in the alignment of the places of installation of the BU and the gap between the magnetic circuits of the rotor and stator was carried out by boring from one installation of seats for the installation of BU A, C, E, B and the stator magnetic circuit D. stator allowed to use bearings of noise class QE6, QE8.

In the general case, a decrease in magnetic vibrations is achieved by a well-chosen ratio of the number of stator and rotor slots, a correctly selected size of the air gap between the stator and rotor magnetic circuits, the bevel of the rotor slots, a decrease in the eccentricity of the air gap, and other design and technological solutions.

To reduce the vibrations excited by the rotor, the shape of the groove of the magnetic circuit is arc-shaped. The slope of the arc changed (when collecting sheets) from right to left along the length of the groove every 20 mm.

To exclude the coincidence of the natural frequencies of the bearings, the housing and the forced forces excited by the rotor, the number of rotor slots is increased to 90. High-frequency vibrations of the EM have less impact on the resource and they can be damped structurally better.

Separately assembled and controlled rotor (Figure 9) made it possible to improve the system of balancing the rotor in bearings separately on the machine and on the stand with rebalancing in the EM assembly.

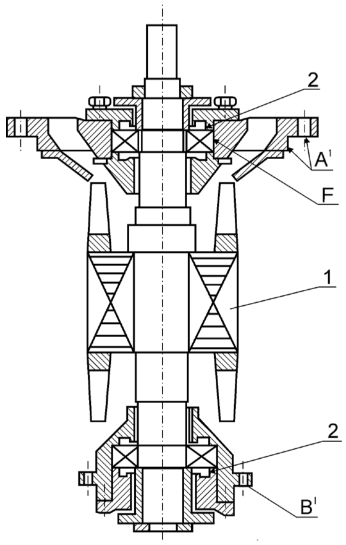

Figure 9. The rotor of the modified design: 1 – magnetic circuit; 2 – wave spring; A/, B/ – rotor connecting surfaces

The one-sided arrangement of the base surfaces A, C and B, E (Figure 8) in the stator housing and fastening of the BU A/, B/ of the rotor (Figure 9) made it possible to vertically assemble the EM (Figure 10) from two independently controlled units: housings with shield and rotor with BU.

The final operations of the EM assembly were fastenings 1, 2 – holes for fastening the rotor with screws in the stator housing (Figure 10).

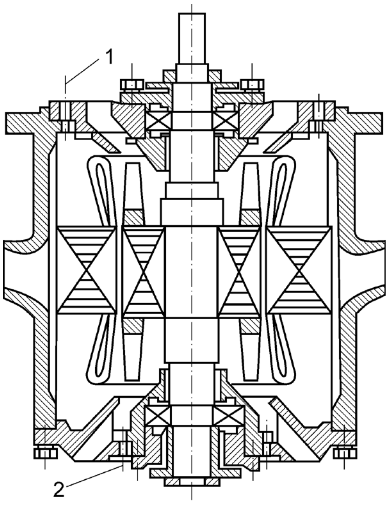

Figure 10. Sketch of the design of the developed EM: 1, 2 – holes for fastening the rotor with screws in the stator housing

4.2 Achieved vibration levels of the improved EM

Vibration characteristics of the new design and production technology on rolling bearings QE8 and with a distance between supports of 450 mm when powered from a mains frequency of 50 Hz are shown in Figure 11.

The vibration levels of the created EM meet the requirements for vibration of class E. The maximum vibrations affecting the EM resource are in the frequency range 50-315 Hz. The reasons for increased vibration in this frequency range are: imbalance of rotating parts (50 Hz); quality and frequency of power supply (100 Hz) from the mains and rotor misalignment in the stator housing. In the frequency range 160-315 Hz, the complex of reasons is as follows: the frequency of rolling of the rolling elements of the bearings; the value of the preload of the bearings in the supports; interference fit and clearances of bearing landings [12]. A significant reduction in EM vibrations in the frequency range 500-2500 Hz was achieved by improving the dynamic properties of the stator housing and the electromagnetic properties of the rotor. The spread of the EM vibration levels during repeated reassembly and measurements over the entire frequency spectrum (except for the 160-315 Hz region) was up to 2 dB. In the frequency range 160-315 Hz, it remains up to 3 dB. The number of measurements was repeated 4-5 times.

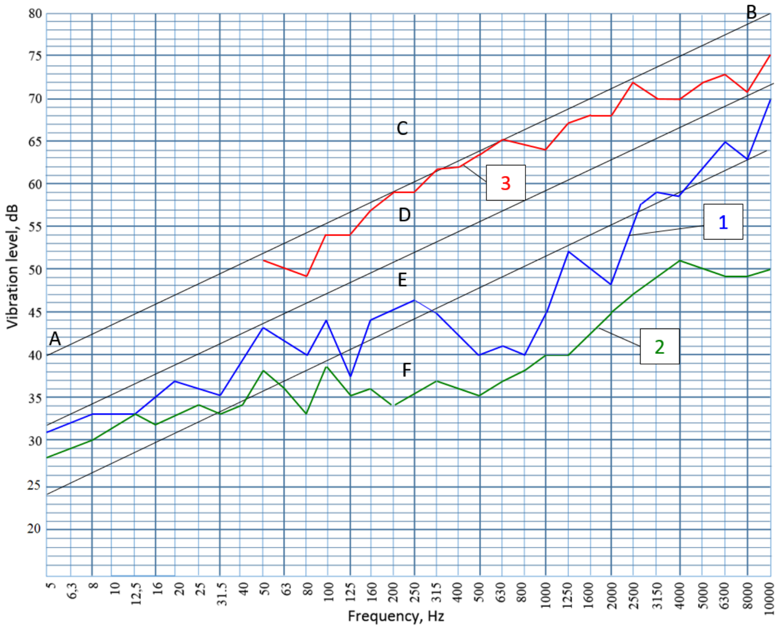

Figure 11. Vibration characteristics of EM: AB – levels of permissible vibrations; D, E, F – zones of vibration classes; C – zone of unacceptable vibration levels; 1, 2 – maximum vibration levels at a speed of 3000 rpm and 1500 rpm, respectively; 3 – vibration levels of the analogue on bearings QE3 at a speed of 3000 rpm

Vibration levels of class E do not affect human health, do not require special vibration isolation in the vehicle. The achieved reduction in vibration levels makes it possible to predict an increase in service life by 2–3 times in comparison with a class D analog (3, Figure 11).

Conclusions

The design of an induction electric motor and its manufacturing technology are proposed, which allow: to increase the accuracy of manufacturing and assembly of an electric motor from two independently assembled and controlled units; to reduce the labor intensity of assembly and to reduce the number of parts to be sorted out during inspection and replacement of bearings during repair.

Constructive and technological solutions have been developed that made it possible to create an induction traction EM with a power of 90 kW, with a rotational speed of 3000 rpm, which meets the requirements of class E with levels of permissible vibration below the straight line connecting the points (32 dB; 5 Hz) and (72 dB; 10 kHz). The achieved reduction in vibration levels by 8-16 dB below the permissible values allows predicting an increase in the resource developed by the EM, depending on the operating modes, by 2-3 times compared to class D. The created EM in terms of vibration level meets all the requirements of environmental standards.

The research results are intended for use in the production of induction traction motors.

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Conflict of interests.

The authors declare that there is no conflict of interests regarding the publication of this paper.

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