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The research and accounting of aging processes of bearing unites

Abstract. *The basic aspects of aging of bearings as a part of bearing units of electric machines at their current operation are considered. The results of the accelerated experimental tests of bearing assemblies for reliability made it possible to research the influence of the main factors on their state and to obtain models that describe the change in their parameters during degradation. The obtained results allow in the future the assessment of the current state of bearing units based on the results of current monitoring of vibration and thermal parameters and carrying out maintenance and repair of rotating electric machines depending on their actual condition.*

Streszczenie. *Rozważono podstawowe aspekty starzenia się łożysk wchodzących w skład zespołów łożyskowych maszyn elektrycznych przy ich bieżącej eksploatacji. Wyniki eksperymentów przyspieszonych zespołów łożyskowych pod kątem niezawodności umożliwiło zbadanie wpływu głównych czynników na ich stan oraz uzyskanie modeli opisujących zmianę ich parametrów podczas degradacji. Uzyskane wyniki pozwalają w przyszłości na ocenę aktualny stan zespołów łożyskowych na podstawie wyników bieżącego monitoringu parametrów drganiowo-termicznych oraz przeprowadzania przeglądów i napraw wirujących maszyn elektrycznych w zależności od ich stanu faktycznego. (**Badanie i analiza procesów starzenia zespołów łożyskowych**)*

Key words: electric machine, bearing unit, reliability indicators, vibration, temperature.
Słowa kluczowe: maszyna elektryczna, zespół łożyskowy, wskaźniki niezawodności, drgania, temberatura

Introduction

A general feature of the aging of rotating electric machines (EM) during operation consists in the manifestation and development of their constructive and parametric asymmetry, both electrical and magnetic [1–3]. These changes must be taken into account in the implementation of various electrical technologies, especially those requiring high accuracy of parameter regulation [4, 5]. The main reasons for this include the spatial redistribution of the magnetic and electrical properties of the laminated cores and the presence of local damage to the windings in the form of turn-to-turn short circuits [6–8]. Additional changes are introduced by the presence of static or dynamic eccentricity of the rotor, which is mainly expressed via the value and features of its deflection. At the same time, changes in the properties of bearing unites (BU) are often not taken into account.

The main reason for this simplification consists in the fact that their basic elements – bearings are to be replaced during EM maintenance and overhauls, and the current deterioration of their properties, manifested in an increase in heating and vibration is believed to have practically no effect on the parameters and state of EM on the whole [9, 10]. In fact, these phenomena cause significant changes in the EM magnetic field due to the formation of an uneven dynamically changing air gap. They also affect the reliability of the windings due to the formation of an uneven distribution of the vector of the magnetic force radial component. That is, the state of the bearings directly determines the development of various types of asymmetry in EM and should be taken into account both in field calculation models and in calculating their reliability.

Accordingly, the purpose of the paper is to form a generalized approach to taking into account the determination of the state of bearings and the degree of its influence when the EM mean time between failures changes.

Theoretical provisions

The special features of the installation of single-row radial ball bearings used in the EM of the main dimensions of industrial series with a height of the axis of rotation up to 160 mm were assessed in accordance with the posed task [11]. It was suggested that the reliability of the BU to some extent depends on the correct choice of the fit. Thus, the inner rings of the EM bearings of the researched power range are fixed on the shafts, using the shaft flanges and tension fit (Fig. 1, a). This type of bearing mounting is used for shafts that rotate relatively to a stationary casing.

Since the EM provides for axial movement of the shaft support when the agreed position of the stator and rotor is acquired, the outer ring of the bearing is fixed in the axial direction only in the casing by one-sided fixing of the bearing with its cover (Fig. 1, b) [12].

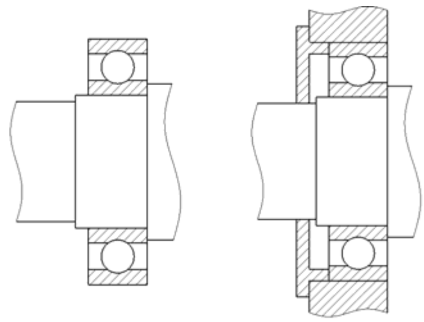

Fig. 1. Bearing mounting methods:
a) – on the shaft; b) – in a stationary casing

This is implemented by mounting the bearings on the EM shaft with tension, when their outer rings have a looser fit in the socket of the shield and are characterized by the possibility of axial movement of one of the bearings in the range of 0.5-1.5 mm to prevent jamming. Also at such installation the internal ring of the bearing is stretched after fitting on a shaft. Its deformation is directly proportional to the amount of tension and results in a decrease in the radial gap of the bearing by the value of $\delta 1 = 0.6\Delta$, μm. Excessive tension can cause an unacceptable reduction in the initial bearing gap. The absence of a gap leads to the clamping of the rolling elements between the rings and a tighter rotation, which sometimes jams the bearing. The situation is aggravated by the installation of bearings of a higher accuracy class than specified in the technical conditions in accordance with IM power, modes and conditions of their operation, which causes premature failure of the BU [13].

The type of test specimens was substantiated taking into account the conditions of bearing fitting in accordance with the geometric dimensions of the rotor, seat and calculations of axial and radial loads acting on the bearing assemblies from drive A and fan B, as shown in Fig. 2 [14]. Here R_A and R_B – respectively radial load from sides A and B; G_2' – the gravitational force of the rotor core with the winding; T_0 – the force of one-sided magnetic attraction when the rotor core is displaced; F_n – the transverse force caused by the transmission through the rigid coupling applied to the end of the shaft; a,b,l,c – linear dimensions of the rotor; Q – dynamic load; k_δ – the coefficient taking into account the nature of EM load; A – axial load; e – eccentricity; Y – the coefficient of reduction of the axial load to the radial one.

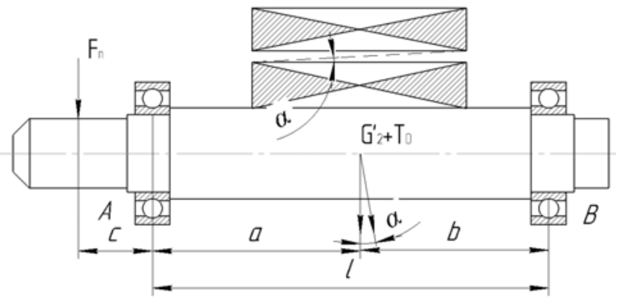

Fig. 2. A sketch of EM rotor shaft

According to Fig. 2, the EM is dominated by dynamic loads from the output end of the rotor shaft. They result from the different types of eccentricity possible in their operation. Thus, due to the subsidence of both bearing assemblies at the same time, static imbalance is possible, and unilateral subsidence of the EM or misalignment of the EM shafts and the actuator causes a dynamic imbalance. If their reasons consist in the loss of rigidity of the unit structure, primarily due to the damage to the bearings, conditions are created for further breaking of their seats. In any case, according to the system of forces shown in Fig. 2, the force of one-sided magnetic attraction increases and it can vary tenfold. Depending on the position of the rotor relative to the stator and the state of the BU, axial loads on the bearing can now increase with the radial ones.

The BU load is usually manifested in changes in the value of the transverse force and the force of one-sided magnetic attraction, and the nature of their impact on the supports is determined by the angle of its application. The maximum allowable value of this angle is determined by the size of the air gap, and the limit on it – by the limit angle of shear of the bearings outer and inner rings, which, in accordance with [15], is up to 4° for single-row self-aligning bearings, and from 1° to 8° for the non-self-aligning ones. At the same time, the change of this angle by 3° as a result of subsidence of one bearing assembly through non-center fastening of couplings causes additional increase in dynamic loading by 12 %.

Experimental research

During the research it was proposed to conditionally divide all the considered defects of BU into two groups. The first group included operational defects that cause the displacement of the center of the axis of the rotor rotation relative to the center of the axis of the bearing rotation (imbalance, shaft sagging, displacement of the couplings). The defects of the second, mounting group are caused by the skew of the bearing, the effect of which consists in the shift of the outer ring [16, 17].

The research specimen was represented by a physical model of the shaft from the side of the connection to the EM drive (Fig. 3). This approach made it possible to further separate the influence of the casing cover on the change of informative parameters [18].

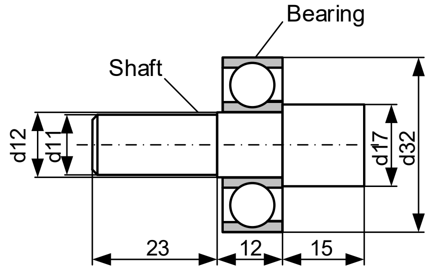

Fig. 3. The general view of the researched specimen of the BU

Such reasons for bearing failure as insufficient lubrication, improper installation, and the effect of bearing design were experimentally researched to solve the posed problems. 10 research specimens were worked out for the research. They were based on type 201 bearings, which were forcibly worn under certain conditions:

– the natural wear of a lubricated bearing, characterized by the presence of high-quality lubrication, no skew when installed, lower rotation frequency, minimum vibration and average temperature;

– the natural wear of a bearing without lubrication, characterized by no lubrication and skew when installed, lower rotation frequency, minimum vibration and average temperature;

– the natural wear of a bearing at high rotation frequency, characterized by the presence of high-quality lubrication, no skew when installed, double rotation frequency, minimum vibration and average temperature;

– the imitation of a defect of poor-quality installation of the bearing, characterized by the presence of high-quality lubrication, a skew when installed, lower rotation frequency, higher vibration value and its periodicity, average temperature;

– the imitation of a design defect, characterized by the presence of high-quality lubrication, no skew when installed, lower rotation frequency, higher vibration value and its periodicity, increased temperature;

– the imitation of the defect of the electromagnetic component, characterized by the presence of high-quality lubrication, lack of skew when installed, lower rotation frequency, higher value of unilateral periodic vibration in the direction of gravity, increased temperature;

– combined influence of defects.

The output end of the shaft was mounted in the chuck of the lathe in experimental research, as shown in Fig. 4.

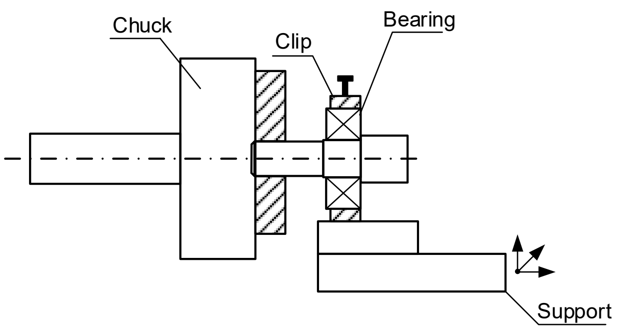

Fig. 4. Explanation of the special features of the specimen tests Clamping chuck Clamp Bearing Support

Different modes of operation were set by spatial movement of the support, according to the program of accelerated tests at variable radial load F_m, and the measurements of informative parameters were performed.

It was determined that the final experimental research of BA should include the following modes:

– two modes of friction (dry friction and friction with lubrication);

– two heating modes (natural and forced, performed by means of the adjustable industrial fan);

– two modes of rotation frequency (1500 rev/min and 750 rev/min).

– two modes of test time determined by the acceleration factor $K_{acc} = 50$, possible in this case, calculated on the basis of the experimental planning method (240 hours for complete tests to the final degradation mode and 80 hours for the estimated trend of changes in the determinants).

To provide the compliance with the modes of operation of the BU in the EM, meeting the conditions of proportionality of changes in temperature and load while providing the proper lubrication (if any in a particular experiment) was monitored.

The assessed indicators included the predicted mean-time-between-failures T and maximum displacement $\Delta\delta$ in the radial direction, which characterizes the value of the eccentricity and is a consequence of the development of beating. The test process was performed cyclically at intervals of 8 hours, at the end of each of which the vibration parameters and the value of the radial displacement were measured.

Upon completion of the research, the specimens were disassembled to identify the signs of emerging defects caused by operation and compare them with the characteristic manifestations of typical bearing damage.

This made it possible to fully research the change in the state of the tested parts of the BU in different modes of operation, and the assessment of the state of the bearings allowed linking the main informative parameters with the degree of development of the researched defects.

The generalization of the research results is illustrated in Fig. 5–6. According to the above, it was possible to clarify the relative change of vibration velocity ν^* (dependence 2) and upper level (maximum peaks) of shock pulses ν^*_{max} (dependence 1) at each stage of the development of BU defects. The upper limit of the allowable vibration velocity for the EM dimension was taken as the basic level, which corresponds to the tested specimens.

At the same time, a faster increase in value ν^*_{max} was confirmed in comparison with ν^* at stage 0–1, which additionally grows with increasing temperature due to more friction in the absence of lubrication, with a slower change ν^* compared with ν_{max} in the degradation zone 4–5.

It should be mentioned that value ν^*_{max} in this case exceeds the upper limit of the allowable vibration speed at stage 1–2, and this fact is stated earlier in the absence of lubrication, i.e. the link with bearing assembly temperature θ_b is evident, and ν^* reaches this value at stage 3–4 under the same condition.

Regarding temperature dependences, according to Fig. 6, it should be noted that its connection with the development of defects is not so clear. However, in any case, it is a determining factor in the wear of bearings in the operating time corresponding to stage 0–1 in Fig. 5. The lack of lubrication significantly affects this process, accelerating it by about 1.1…2.3 times (curves 1–2 in Fig. 6).

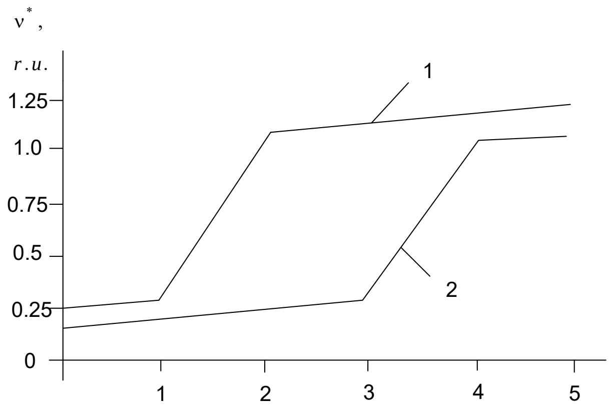

Fig. 5. Change of ν^* and ν^*_{max} during the development of bearing defects

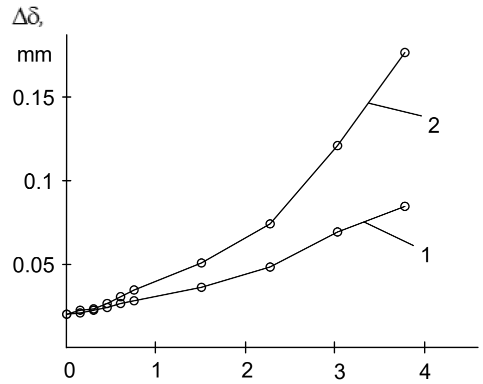

Fig. 6. Relationship between bearing temperature and change $\Delta\delta$

This effect increases up to 2.1…2.2 times with a simultaneous increase in vibration at the times corresponding to the areas after point 1.

A two-factor model was obtained as a result. It describes change $\Delta\delta$, μm depending on temperature θ_b and vibration ν of the form

$$\Delta\delta = b_0 + b_1 \frac{\theta_b}{\theta_{b\,max}} + b_2 \frac{\nu}{\nu_n} + b_3 \frac{\theta_b}{\theta_{b\,max}} \frac{\nu}{\nu_n} + b_4 \left(\frac{\nu}{\nu_n} \right)^2,$$

where coefficient b_0, b_1, b_2, b_3, b_4 for the researched BA are 0.0106; 0.01123; 0.001572; 0.001017; 0.0002947. Relevant generalizations were made regarding the use of the obtained results in the construction of BA reliability models within light series bearings.

In the course of the research, it was finally determined that the most informative parameters in BA analysis include the average value of the radial ν_r component of the vibration speed, temperature θ_b of the bearing assembly and some components of the direct spectrum of the vibration speed $\nu_i, ..., \nu_j$ in the radial and axial directions. In this case the most complete information on their condition is provided by the analysis of spectral characteristics.

The state of BU, according to [19, 20], can be expressed by the aging function of bearings of a certain series used in the EM of the corresponding power range. The results obtained for bearings of any size from the selected series can be transferred to bearings of other sizes, taking into account their proven similarity within the series. An appropriate correction was introduced for this purpose for the light series bearings researched in the paper, provided that the ratio $R_a/2Br$ of the reference load R_a to the cross-sectional area of the outer ring $2Br$ is maintained, where *B* – the width of the bearing, $2r$ – the height of the outer ring. The limit state of the bearing can be assessed by value $\nu c/(k2Br)$, where c – the number of damaged balls, k – the total number of balls in the bearing, correlated with the temperature θ_b of the bearing.

The research results proved the functional sufficiency of the two-factor reliability model of the form

$$T_b = b_0 + b_1 \frac{\theta_b}{\theta_{b\,max}} + b_2 \frac{\nu}{\nu_n} \frac{c}{k2Br} + b_3 \frac{\theta_b}{\theta_{b\,max}} \frac{\nu}{\nu_n} \frac{c}{k2Br},$$

where $\theta_{b\,max}$ – limit allowable bearing heating temperature; ν_n – permissible level of vibration speed in BU normal state. This model takes into account the change in the final service life at different states of the bearing assemblies, according to changes in temperature θ_b and vibration RMS ν_n.

The values of coefficients b_0, b_1, b_2, b_3, obtained in the course of accelerated tests, amounted to 14156, –1409, –26003, –3701.

In the case of bearings of other series, the corresponding geometrical parameters should be used, which directly take into account the overall strength of the structure.

Conclusions

1. The efficiency of determining the state of bearing assemblies due to the relative indicators of vibration components and overheating temperature has been proved, which makes it possible to clearly characterize the stages of their degradation and to create reliable models for determining the predicted mean time between failures and maximum radial displacement.
2. The primary model of reliability of the bearing assembly on the basis of the radial single-row bearings used in induction motors of small and average power is developed.
3. The possibility and principles of generalization of the results obtained for bearings of arbitrary size of the researched series with bearings of other sizes have been determined, taking into account the peculiarities of their similarity within the series due to geometric parameters that directly take into account the overall strength of the structure.
4. The development of this area enables a more accurate predicting the reliability and passing to the maintenance and repair of electric machines in the actual state.

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