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VIBRATION DIAGNOSTIC OF A FRICTION PROCESS IN SLIDE BEARINGS

DIAGNOSTYKA DRGANIOWA PROCESU TARCIA W ŁOŻYSKACH ŚLIZGOWYCH

Keywords: slide bearing, diagnostic, RMS (root mean square), friction, vibration.

Abstract An analysis of the condition of technical objects is carried out by diagnostic systems, the purpose of which is to detect irregularities in their operation and to prevent damages. In slide bearings, it applies to the diagnostic of friction and thermal phenomena of mating friction pairs. Among many methods of bearing diagnostics, special attention should be paid to vibration diagnostic methods based on measurements of relative vibration parameters or on absolute vibration (displacement, velocity, or acceleration of vibration). Methods of the vibration diagnostic of bearings rely on periodic or continuous measurements of relative vibration parameters of the bearing housing in relation to the rotor (in the case of slide bearings the measurements of the bearing sleeve in relation to the shaft neck) or absolute vibration parameters of the bearing housing (i.e. the sleeve in the case of slide bearing). The article presents a method of vibration diagnostics of friction phenomena that occur during the operation of slide bearings under various lubrication and load conditions. There are presented methods of analysis and the interpretation of measurement data obtained as a result of the conducted slide bearing tests on the laboratory stand. A method for assessing a technical condition of the slide bearing friction pairs is proposed.

Słowa kluczowe: łożysko ślizgowe, diagnostyka, RMS, tarcie, drgania.

Streszczenie Rozpoznanie stanu obiektów technicznych jest realizowane przez systemy diagnostyczne, których celem jest wykrywanie nieprawidłowości w ich działaniu i zapobieganiu uszkodzeniom. W łożyskach ślizgowych dotyczy to diagnostyki zjawisk tarciowych i cieplnych współpracujących węzłów tarcia. Wśród wielu metod diagnostyki łożysk na szczególną uwagę zasługują metody diagnostyki drganiowej, bazujące na pomiarach parametrów drgań względnych bądź drgań bezwzględnych (przemieszczenie, prędkość lub przyspieszenie drgań). Metody diagnostyki drganiowej stanu technicznego łożysk polegają na okresowych lub ciągłych pomiarach parametrów drgań względnych obudowy łożyska względem wirnika (w przypadku łożysk ślizgowych panewki łożyska względem czopa wału) bądź parametrów drgań bezwzględnych obudowy łożyska (czyli panewki w przypadku łożyska ślizgowego). W artykule przedstawiono metodę diagnostyki drganiowej zjawisk tarciowych, jakie zachodzą podczas pracy łożysk ślizgowych przy różnych warunkach smarowania i obciążenia. Przedstawiono metody analizy i interpretacji wyników pomiarów otrzymanych w wyniku przeprowadzonych badań łożyska ślizgowego na stanowisku laboratoryjnym. Zaproponowano metodę oceny stanu technicznego węzła tarcia łożyska ślizgowego.

INTRODUCTION

Using monitoring systems enables the continuous assessment of technical conditions of a machine. It also allows pre-planning the dates of maintenance and repair, in particular, of devices from which a high level of reliability is required [**L. 1, 2**]. The breakdown of one element of a machine results in numerous economic losses due to the unplanned downtime [**L. 3, 4**].

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Contemporary diagnostic systems of a technical condition are based mainly on the measurement of vibroacoustic signal basic parameters, such as an effective value of vibration (RMS) and their frequency spectrum. The change in the value of controlled parameters is, in most cases, treated as a change in technical condition of machines [**L. 5, 6**].

The use of advanced diagnostic systems is particularly important in the diagnosis of bearing pair elements of machines whose rotational motion is the source of vibration with the value of accelerations depending on the bearings wear condition [**L. 7**]. In the bearing friction process, complex phenomena occur, the analysis of which requires using advanced tribological and diagnostic tests. One of the methods to describe these phenomena is to apply vibroacoustic diagnostics, aiming to take into account all possible mutual interactions between elements of the machine in order to determine its technical condition. Measurements of the effective value of vibration acceleration, which are used in the vibroacoustic diagnostics, allow the precise location of malfunctioning bearing pairs of machines and devices and play an important role in controlling their correct operation [**L. 8**].

The article presents the possibilities of vibroacoustic diagnostics of a slide bearing based on the measurement of the effective value of vibration accelerations. The study focuses on the development of a method for assessing the technical condition of a bearing pair using the frequency analysis method based on vibration acceleration signal, taking into account an influence of the rotational speed change as well as the system load. The method was developed in such a way that it could be implemented in continuous monitoring systems. Therefore, in order to obtain the analysis results in real time, the method should not be too computationally complex. For diagnosing processes and tribological phenomena that occur during friction of slide bearings, there are proposed decomposition procedures of vibration acceleration signals in the domain of frequency. Based on the methods of the decomposition of vibration signals, there were carried out tests of friction processes in a slide bearing operating under different loads and different friction conditions, and there were presented conclusions regarding the effectiveness of the proposed diagnostic method.

TESTING STAND

The laboratory stand was built as a part of the work to enable testing and measurements of vibration accelerations, torque, rotational speed, and temperature in a slide bearing. The laboratory stand was completed in the Department of Machinery Construction and Operation at the AGH University of Science and Technology in Kraków. The construction of the test stand and the principle of its operation are shown in the diagram on **Figures 1** and **2** [**L. 9, 10**].

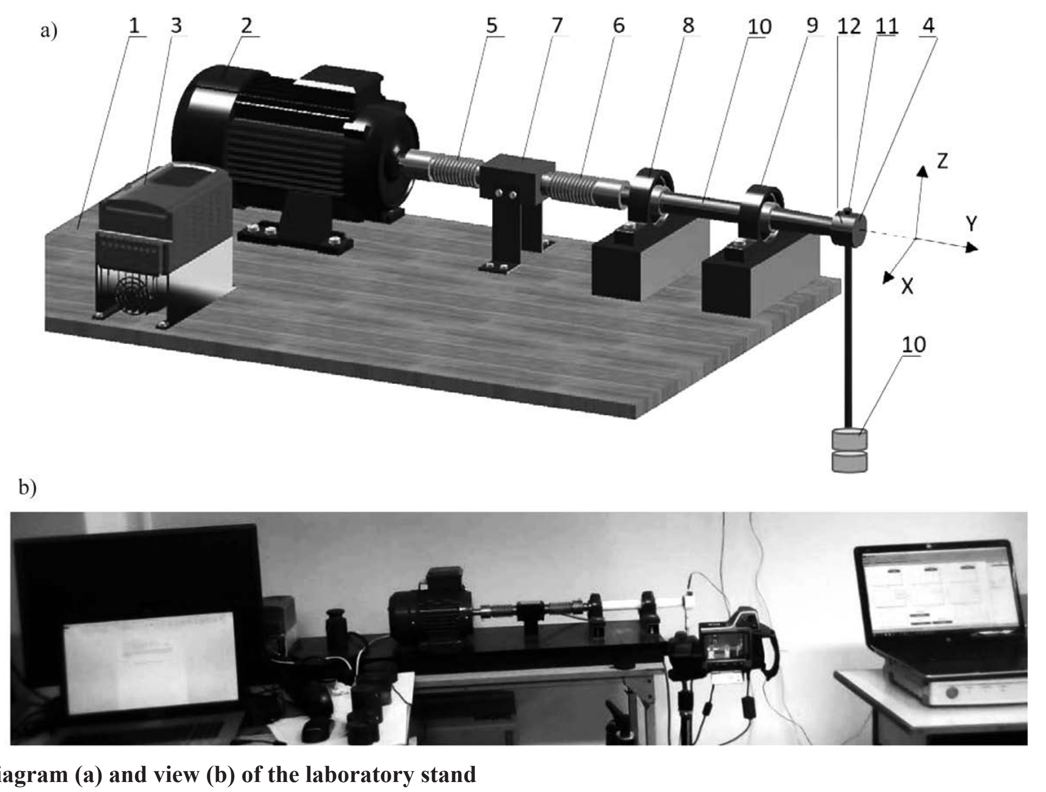

Fig. 1. Diagram (a) and view (b) of the laboratory stand
Rys. 1. Schemat (a) i widok (b) stanowiska laboratoryjnego

The stand was built on a specially prepared platform 1 allowing stable fixing of individual components to the floor. The laboratory stand for performing measurement tasks consisted of an electric motor 2 with the designation MS80 and parameters: rated power P = 1,1 kW, rated speed n_n = 1380 rev/min. The motor operation was controlled by a frequency converter 3 with the designation E1000. The drive from the motor was transferred to the sliding bearing pair 4 by bellow couplings 5 and 6 and a torque converter 7 with the designation T20WN/10. To measure the rotational speed of the electric motor 2, the torque converter had an additional built-in optical converter of rotational speed. The shaft was mounted on two roller bearings 8 and 9. The transverse load of the slide bearing pair was set by weights attached to the loading rod 10. In the tests, the slide bearing was loaded with a gradually increasing force from the initial value F_0 = 12.45 N, through forces F_1 = 22.26 N and F_2 = 32.07 N, to the force of value F_3 = 41.87 N. Loads of the slide bearing corresponded to the average pressures: p_0 = 0,034 MPa, p_1 = 0,062 MPa, p_2 = 0,089 MPa, and p_3 = 0,116 MPa. Vibration measurements of the slide bearing pair were carried out using the triaxial accelerometer 11. The accelerometer was mounted on the bearing sleeve by a screw joint. The measurement of working temperature of the slide bearing pair was made using a thermocouple 12 fixed in a specially drilled hole. The applied solution allowed for the measurement of the slide bearing working temperature near the contact area of the rotating shaft neck and the sleeve of the tested slide bearing. The detailed design of the slide bearing pair 4 (**Fig. 1**) is shown on **Figure 2**.

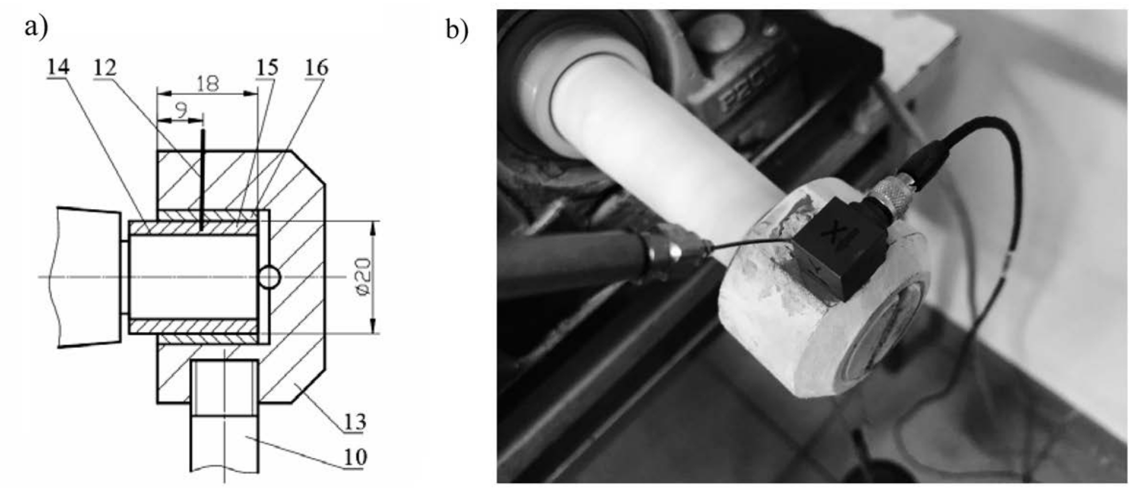

Fig. 2. Cross-section of slide bearing (a), and its view with measurement sensors (b)
Rys. 2. Przekrój łożyska ślizgowego (a), oraz jego widok z czujnikami pomiarowymi (b)

The design of the slide bearing pair 4 from **Figure 1** is shown in detail in **Figure 2** and consists of an outer housing 13, in which is mounted the tight fitted sleeve 16, made of bronze in granulation 30 µm with 10% addition of graphite. On the shaft neck 14 was pressed the bushing 15 made of steel grade C45 of hardness 220 HB and slide surface roughness R_a = 0.063 µm. The rotary movement occurs on the sliding surface between the sliding bushing 16 and the sleeve 15. The detailed description of the stand design can be found in the publication [**L. 11–13**].

The measurement system was built on the basis of the measuring amplifier Spider8 made by the company Hottinger Baldwin Messtechnik. It has measurement channels that enable the integration of sensors with voltage, current, resistance, and bridge systems' outputs in full and half-bridge configurations. It also has functions that allow direct connection and temperature measurements using typical thermoelectric and thermo-resistance sensors. **Figure 3** shows a block diagram of the measurement system used in laboratory experiments.

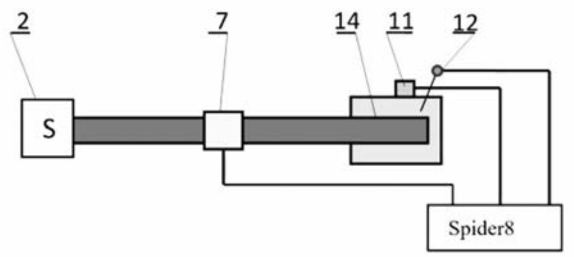

Fig. 3. Block diagram of the measurement system
Rys. 3. Schemat blokowy systemu pomiarowego

The measurement system includes a system for measuring torque and rotational speed using the T20WN/10 torque converter, made by the company

HBM, marked in **Figure 2** by the symbol 7. The torque is set by the motor 2. The torque gauge has integrated measurement systems: strain gauge for non-contact torque measurement and an optical system for rotational speed measurement. The measuring range of the torque gauge is 10 Nm and the uncertainty of measurement is 0.2% at a maximum rotational speed 10000 rev/min, while the measuring range of the revolution counter is 3000 rev/min with a measurement resolution 1/360 rev/min. To measure vibration accelerations, 3-axis accelerometer in CCLD technology of type 4528-B-001 made by the company Brüel&Kjaer 11 were used. The measuring range of the accelerometer is 70 m/s^2 in the frequency range from 0.3 Hz to 12800 Hz. The accelerometer works together with a dedicated amplifier CCLD type 1704-A-001 of the company B&K. The total measuring range of the acceleration measurement channel in each axis is set to 10 m/s^2. The temperature of the slide bearing sleeve, in the direct vicinity of the sliding surface (**Fig. 2**), was measured using needle thermoelectric thermometer 12 of type K, placed in a hole made in the sleeve.

The measurements were planned in such a way as to perform them in the frequency range correlated with the rotational speed. For the experimentally determined value of 1496 rev/min of the rotational speed, the frequencies are at 24.9 Hz and its higher harmonic values. Therefore, in order to limit the influence of disturbances and components of higher frequencies, the assumed sampling rate of the measurement system was 600 Hz with the antialiasing filter of the limit frequency of 100 Hz.

As a part of the experiments, the tested slide bearings were lubricated with lubricant SN 100, and non-lubricated bearings operating under dry friction conditions. An operation of the bearing in dry friction conditions is recognized as improper, leading to its damage. This slide bearing operating mode was applied to determine differences in the value of measured parameters related to the properly operating bearing under lubrication conditions. The bearing operation was tested in a steady state at the rotational speed n = 1496 rev/min. The experiments were carried out in such a way that, in the first stage, the system operated without the slide bearing in order to achieve the determined thermal condition of ball bearings. In the second stage, the bearing sleeve was put on the shaft neck and there was applied the initial load of F_0 = 12.45 N imposed in the middle of the sleeve housing, perpendicular to the bearing axis. In this way, the loaded bearing was operating until reaching the state in which the value of its temperature was stabilized. Then, in the first stage, the load of the slide bearing was increased to the value of force F_1 = 22.26 N and, as in the previous stage, the measurement was made to the state of the next stabilization of its temperature at the increased load. The operation was repeated by increasing the load in Stages 2 and 3 to the value of forces F_2 = 32.07 N and F_3 = 41.87 N.

TEST RESULTS

Test results of the moment of friction M_{f0} for the slide bearing and the total moment of friction Mf as a function of time are shown in **Figures 4** and **5** for a bearing lubricated with SN 100 lubricant and in **Figures 10** and **11** for a bearing operating under dry friction conditions. The construction of the stand in which the torque converter is placed between the motor and two rolling bearings and the tested slide bearing allows that the measured total friction moment M_f presents the total moment of friction of rolling bearings and the tested slide bearing. The moment of friction M_{f0} of the slide bearing was determined as the difference between the total moment of friction M_f and the moment of friction of roller bearings M_{fr}, measured after a disassembly of the slide bearing, according to the following formula:

$$M_{f0} = M_f - M_{fr} \tag{1}$$

where

M_{f0} – moment of friction of the slide bearing,

M_f – total moment of friction of the unit of roller bearings and the slide bearing,

M_{fr} – the moment of friction of roller bearings.

The acceptance of such a methodology to determine the moment of friction of the slide bearing was dictated by difficulties of measuring the moment of friction directly on a slide bearing. According to the authors, the applied methodology of measurement for the accepted measuring range and the high accuracy and resolution of the moment measurement is sufficient to describe the tested phenomena taking place in a slide bearing. In addition, during the experiments, the temperature of the roller bearings was also controlled, which remained stable for the applied load changes. This may indicate a low impact of the applied load changes on the value of the moment of friction in roller bearings.

The measuring points marked on all diagrams determine the time intervals of periodic measurements with a time interval 5 minutes for a bearing lubricated by SN100 lubricant and 2 minutes for a bearing operating under dry friction conditions. At any time of the measurement, 20-second signal sequences were recorded. In the case of measurements of the moment of friction and temperature, their average value was determined for each of the recorded 20-second sequences, and the obtained results are presented in **Figures 4–6** and **10–12**. In the case of measurements of the vibration acceleration, their effective values were determined for each of the recorded 20-second sequences, and the obtained results are presented in **Figures 7–9** and **13–15**.

Vertical lines marked with numbers 1, 2, and 3 determine the moments in which, in the successive stages of the experiment, when the load of the tested slide bearing was increased, corresponding to the values F_1 = 22.26 N, F_2 = 32.07 N, and F_3 = 41.87 N.

Figures 4–9 show the results of the experiment in operating conditions of a lubricated slide bearing. Measuring points marked on the diagrams determine measuring moments with a 5-minute interval.

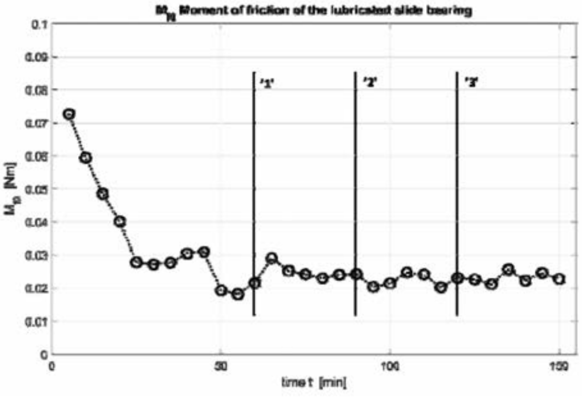

Fig. 4. Course of the moment of friction *Mf*$_0$ of the slide bearing with lubrication

Rys. 4. Przebieg momentu tarcia *Mf*$_0$ łożyska ślizgowego ze smarowaniem

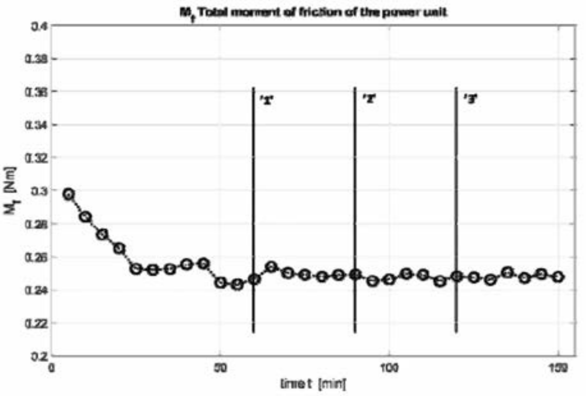

Fig. 5. Course of the total moment of friction *Mf* of driving unit – bearing with lubrication

Rys. 5. Przebieg całkowitego momentu tarcia *Mf* zespołu napędowego – łożysko ze smarowaniem

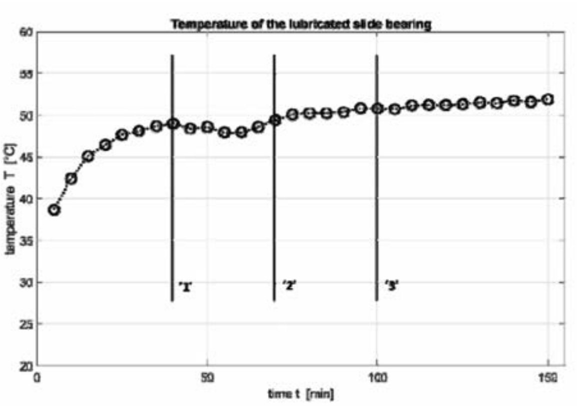

Fig. 6. Course of temperature *T* of slide bearing sleeve vibration – bearing with lubrication

Rys. 6. Przebieg temperatury *T* panwi łożyska ślizgowego – łożysko ze smarowaniem

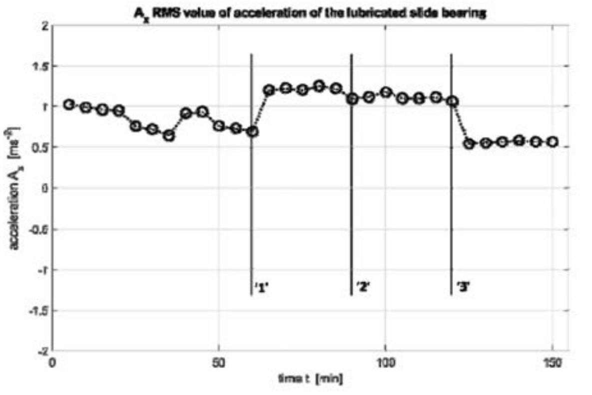

Fig. 7. Course of the effective value RMS of the acceleration A_x of slide bearing sleeve with lubrication

Rys. 7. Przebieg wartości skutecznej RMS przyspieszenia drgań A_x panwi łożyska ślizgowego ze smarowaniem

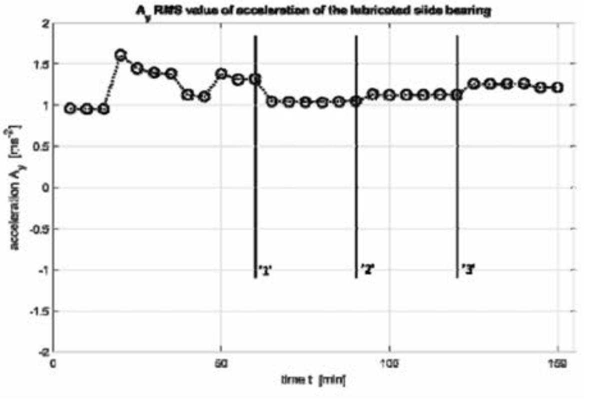

Fig. 8. Course of the effective value RMS of the vibration acceleration A_y of slide bearing sleeve with lubrication

Rys. 8. Przebieg wartości skutecznej RMS przyspieszenia drgań A_y panwi łożyska ślizgowego ze smarowaniem

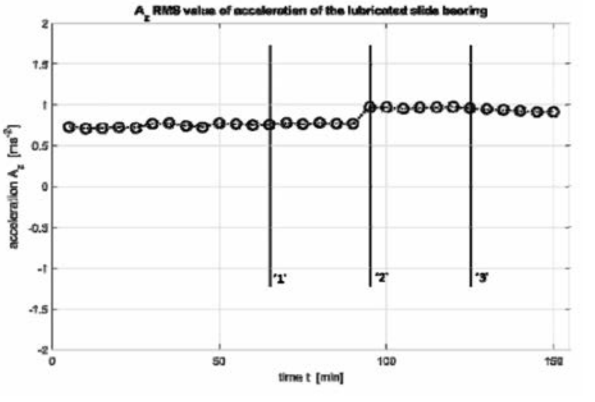

Fig. 9. Course of the effective value RMS of the vibration acceleration A_z of slide bearing sleeve with lubrication

Rys. 9. Przebieg wartości skutecznej RMS przyspieszenia drgań A_z panwi łożyska ślizgowego ze smarowaniem

Figures 10–15 show the results of the experiment for the slide bearing operating under dry friction conditions. Measuring points marked on the diagrams determine measuring moments with a 2-minute interval.

Due to the risk of seizing, the bearing operating under dry friction conditions was additionally loaded only twice (lines 1 and 2).

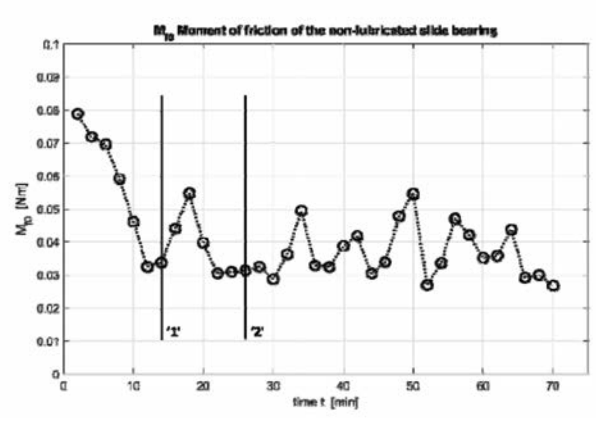

Fig. 10. Course of the moment of friction M_{f0} of the slide bearing without lubrication

Rys. 10. Przebieg momentu tarcia M_{f0} łożyska ślizgowego bez smarowania

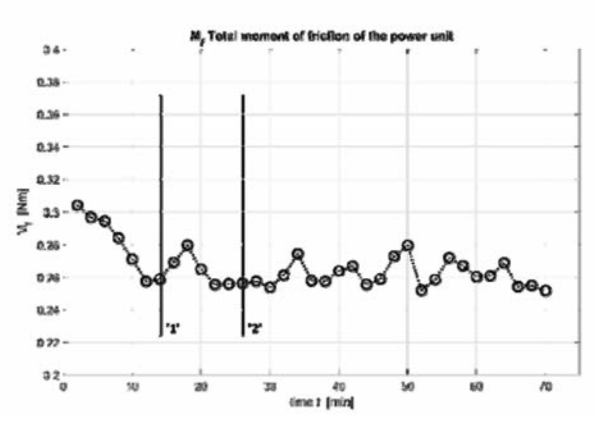

Fig. 11. Course of the total moment of friction M_f of driving unit – bearing with lubrication

Rys. 11. Przebieg całkowitego momentu tarcia M_f zespołu napędowego – łożysko bez smarowania

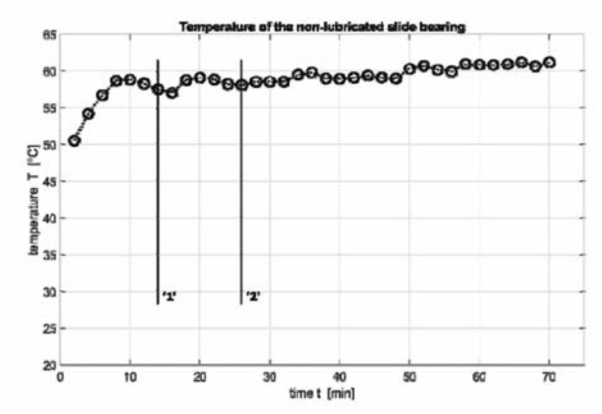

Fig. 12. Course of temperature T of slide bearing sleeve without lubrication

Rys. 12. Przebieg temperatury T panwi łożyska ślizgowego bez smarowania

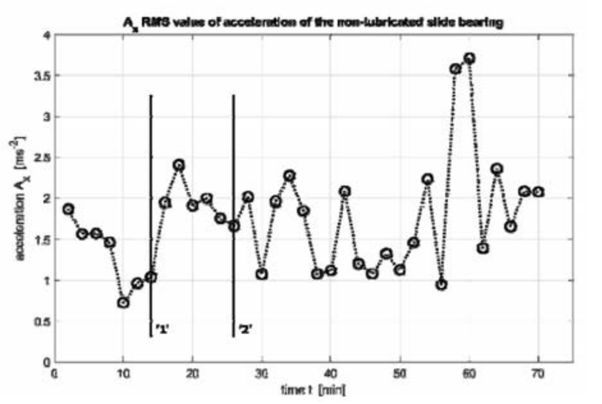

Fig. 13. Course of the effective value RMS of the vibration acceleration A_x of slide bearing sleeve without lubrication

Rys.13. Przebieg wartości skutecznej RMS przyspieszenia drgań A_x panwi łożyska ślizgowego bez smarowania

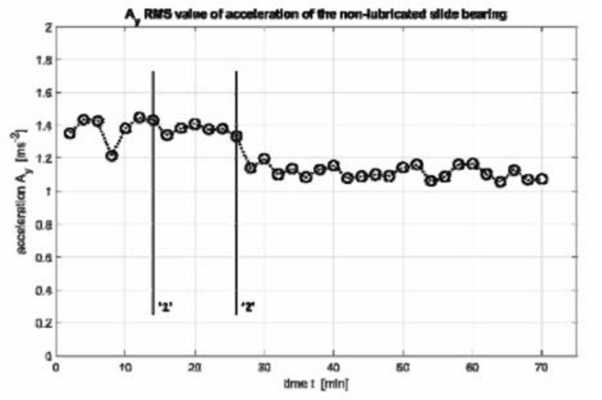

Fig. 14. Course of the effective value RMS of the vibration acceleration A_y of slide bearing sleeve without lubrication

Rys. 14. Przebieg wartości skutecznej RMS przyspieszenia drgań A_y panwi łożyska ślizgowego bez smarowania

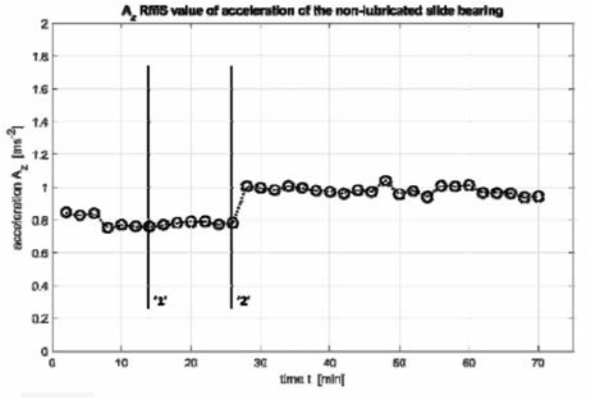

Fig. 15. Course of the effective value RMS of the vibration acceleration A_z of slide bearing sleeve without lubrication

Rys. 15. Przebieg wartości skutecznej RMS przyspieszenia drgań A_z panwi łożyska ślizgowego bez smarowania

Figure 16 is a summary of measurement results of the effective values RMS for 24.9 Hz, 49.8 Hz, 747 Hz, and 99.6 Hz frequency components of the vibration acceleration A_{xc} for the lubricated slide bearing driving unit, and on **Figure 17** for the slide bearing operating under dry friction conditions.

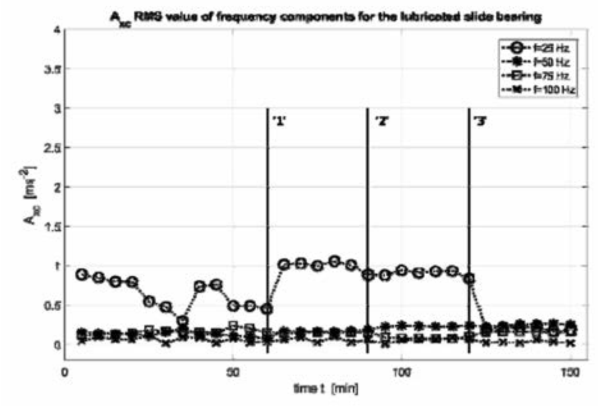

Fig. 16. Values RMS for 24.9 Hz, 49.8 Hz, 74.7 Hz and 99.6 Hz frequency components of the acceleration A_{xc} for the lubricated slide bearing

Rys. 16. Wartości RMS składowych częstotliwościowych 24,9 Hz, 49,8 Hz, 74,7 Hz i 99,6 Hz przyspieszenia A_{xc} dla łożyska ślizgowego ze smarowaniem

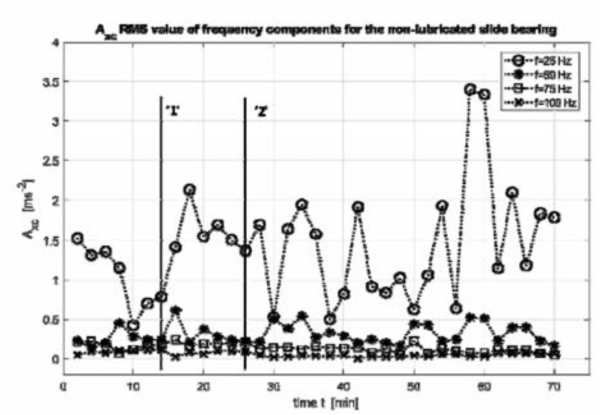

Fig. 17. Values RMS for 24.9 Hz, 49.8 Hz, 74.7 Hz and 99.6 Hz frequency components of the acceleration A_{xc} for the slide bearing without lubrication

Rys. 17. Wartości RMS składowych częstotliwościowych 24,9 Hz, 49,8 Hz, 74,7 Hz i 99,6 Hz przyspieszenia A_{xc} dla łożyska ślizgowego bez smarowania

The decomposition of components of the acceleration signals was made by the Fourier transform. The transformation was performed for each registered 20 second measurement sequence. For the analysis, 4096 samples were used, which, at the sampling rate of 600 Hz, provided a frequency resolution of $\Delta f = 600/4096$ Hz = 0.15 Hz. Measuring points marked on the diagrams are the results of the Fourier transform calculated for periodically performed measurements with a time interval of 5 minutes for a bearing lubricated with SN100 lubricant and 2 minutes for a bearing operating under dry friction conditions.

Analysis of the obtained measurement results of the moment of friction M_{f0} for the lubricated slide bearing (**Fig. 4**) and the slide bearing operating under dry friction conditions (**Fig. 10**) allows noticing slight differences between these values that are within limits of the measurement error. This is due to the low value of friction forces occurring in slide bearings and the structure of the stand and the applied method of friction moment measurement on a slide bearing. This is particularly evident in test results of the total friction moment M_f of the driving unit (**Figs. 5** and **11**). They indicate that the moment of friction on roller bearings is predominant in relation to the moment of friction in a slide bearing. Therefore, the value of the moment of friction M_{f0} of the slide bearing does not seem to be proper for use in the diagnosis of slide bearings as an indicator of the correct operation of a bearing.

The specified results of testing the bearing operating temperature T lubricated by SN100 lubricant (**Fig. 6**) clearly differ from the bearing operating temperature under dry friction conditions (**Fig. 12**). The difference is up to 9°C for a bearing operating at the transverse load F_2 = 32.07 N. The application of temperature measurements in a diagnosis of slide bearings is presented in the authors' own works [**L. 9–13**], where a method has been proposed for analysing the operation of slide bearings using the thermal balance.

The analysis results of recorded signals of the vibration acceleration confirmed the effectiveness of the method proposed by authors for the vibration diagnostic of friction process in slide bearings. It particularly applies to the measurement of an effective value RMS of the vibration acceleration Ax of the bearing sleeve measured in the direction of axis X, perpendicular to the axis of the shaft and to the direction of gravity (the acting direction of the x axis is shown in **Fig. 1**). The comparison of test results (**Figs. 7** and 13) clearly indicates the differences in the effective value RMS of the vibration acceleration A_x for a lubricated bearing compared to a bearing operating under dry friction conditions. The maximum effective value of vibration acceleration for a lubricated bearing was $A_x \cong 1.3$ m/s^2, while for a bearing operating under dry friction conditions $A_x \cong 3.6$ m/s^2. Test results of an effective value RMS of the vibration acceleration A_y and A_z (**Figs. 8** and **9** for a lubricated bearing and **Figs. 14** and **15** for a bearing operating under dry friction conditions) in the direction of axis *y* and axis *z* do not show significant differences.

Let us consider, in turn, the values of frequency components for the vibration acceleration signals. Measurement results of the acceleration effective value A_{xc} for the 24.9 Hz frequency component, as shown in **Figures 16** and **17**, also reveal significant differences in the operation of a lubricated bearing and operating under dry friction conditions. The acceleration effective values A_{xc} for the remaining components of frequency 49.8 Hz, 74.7 Hz, and 99.6 Hz take much lower values than for the component of frequency 24.9 Hz. The signal of frequency 24.9 Hz is closely related to the rotational speed 24.9 rev/s of the driving unit. The analysis of the obtained test results allows to state that the effective values RMS of the vibration acceleration A_x and A_{xc} for a bearing sleeve are parameters that can be applied in the diagnosis of slide bearings as an indicator of the bearing correct operation.

CONCLUSIONS

The article presents a method for assessing the technical condition of slide bearing friction pairs operating at various loads and lubrication conditions. In order to validate the proposed method, an active diagnostic experiment was carried out at a laboratory stand specially constructed for this purpose. This experiment enabled measurements of a lubricated slide bearing and a bearing operating under dry friction conditions at various transverse loads of the bearing. The analysis results of recorded signals of the vibration acceleration confirmed the effectiveness of the method proposed by authors. It particularly applies to the measurement of an effective value RMS of the vibration acceleration A_x of the bearing sleeve, for which significant differences were found between measurements of the lubricated bearing and the bearing operating under dry friction conditions. The amplitude of the vibration acceleration for a lubricated bearing was $A_x \cong 1.3$ m/s^2, while, for a bearing operating under dry friction conditions, $A_x \cong 3.6$ m/s^2. The measurement results of the effective value RMS of the vibration acceleration A_{xc} for the 24.9 Hz frequency component also reveal differences in the operation of a lubricated bearing and a bearing operating under dry friction conditions. A significant difference between these two values suggests that this result can be used in the diagnosis of slide bearings as an indicator of the correct bearing operation. In the remaining measurement directions of the effective value RMS, no significant differences were found.

It is also worth mentioning the next tested parameter, which is the bearing working temperature. For a lubricated bearing, throughout the whole load cycle, the bearing operating temperature did not exceed T = 50°C, while for a bearing operating under dry friction conditions, already at the first three load cycles, the bearing operating temperature exceeded T = 62°C and did not reach a stable value.

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