TECHNICAL STATE IDENTIFYING OF A ROLLER BEARING WITH USING OF VIBRODIAGNOSTICS

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Abstract: *The paper addresses the issue of identifying a technical state of rotating parts with the use of vibrodiagnostics. This method has been developing rapidly for recent years and is assumed to be used for predicting failure occurrence. In this paper we focus on monitoring a technical state of a medium truck bearing box where we simulate different field conditions and their dependence on the RMS – root mean square. A measuring device DEWE 50/8 including the DEWEsoft software product of the Dewetron company have been used for the actual measurement. The aim of the work is to calculate the values of root mean square acceleration up to the moment a failure occurs. In the next step we will put the values into a graph and try to determine a gRMS limiting value. In the future this enables us to change the bearing before a failure occurs otherwise it would increase the related cost.*

KEYWORDS: *ROOT MEAN SQUARE, ROOT MEAN SQUARE OF ACCELERATION, VIBRATION, VIBRODIAGNOSTICS.*

1. Introduction

Vibrodiagnostics is one among many methods of technical diagnostics which has been continuously monitoring a technical state of a device by observing the level of a mechanical oscillation in a real time. The mechanical oscillation is the manifestation of a device during its operation. Certain parts become vibration exciters, others, depending on excitation, react specifically.

The vibrodiagnostics is one of the most important methods used in technical diagnostics for identifying a technical state. With the use of vibration diagnostics we are able to detect an incipient failure, locate the place of an incipient failure and predict the length of time during which a device is going to work before a failure occurs or a preventive action is performed. The proper application of vibrodiagnostics can prevent from undesirable damaging machines, thereby saving money spent on a repair and a device in a disabled state.

Experimental vibrodiagnostics measurements have been performed while obtaining primary signals. Primary results of measurement are systematized and its analysis is done. Dependence of work characteristics between rotary systems with bearings of sliding friction and of roll is ascertained. Generalization of research results is done and conclusions are formulated [1][2][3].

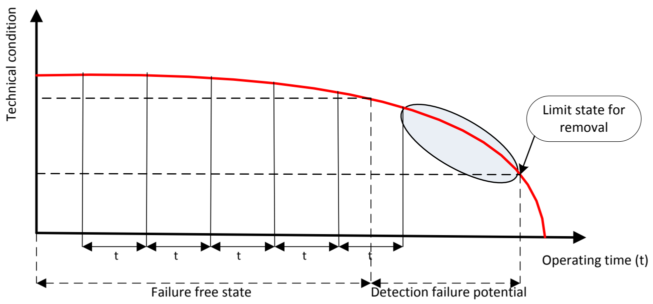


Fig. 1 Dependence of the wear of machine parts on operating time [5]

Two kinds of excitations can produce noise in rotating machinery; excitations due to primary sources, and excitations due to manufacturing defects. It is difficult to act on the primary sources or functional loads as they are specific to a machine. These defects are inherent in both the manufacturing quality and the configuration of the machine. They are caused by the machine's rotating parts (rotors, disks, bearings, gears ...) and create concentrated or distributed loads that produce vibrations and noise. These vibrations depend on the machine itself through the transfer function while the noise is a function of the machine environment [4].

A lot of companies spend a big amount of money on the preventive maintenance of machines and technical equipment despite the fact that at that particular moment there is no need to perform a preventive action. This leads to big financial losses mainly during the machine unavailability. Therefore it is desirable to perform preventive maintenance at the time when it is really necessary (Fig. 1). Determining a preventive action or replacing a part based on a real technical state with the use of a mathematical model describing a machine technical state during its life leads to proactive maintenance. The aim of our article is to describe the technical state of a vehicle bearing box using vibrodiagnostics.

2. Main characteristics of vibrations

Vibrations occur as a result of rotating or straight-line moving bodies. The course of vibrations is influenced mainly by a technical state of single machine components such as shafts, gear boxes, crank mechanisms, cam mechanisms, antifriction bearings, and also by the imbalances of rotating parts, backlash in friction bearings, wear, material fatigue, occurring cracks, corrosion and other parameters affecting a smooth machine run. The vibration itself is defined then as a dynamic phenomenon when particles or solid bodies move around a zero equilibrium position. They are given by a combination of six movements, namely by a shift in an orthogonal coordinate system x, y, z and a rotation about these axes. We can describe them by amplitude and a phase at a certain period of time. Depending on the time variations of values, the vibrations are of a periodical, non-periodical or random character. As for periodical vibrations, a time course of vibrodiagnostic values repeats. A harmonic vibration which has a sinusoidal waveform is based on these vibrations. For harmonic vibrations we need to set only one determining value and the other ones can be calculated.

The basic way of describing oscillations is to determine their displacement x, velocity v, acceleration a, maximum amplitude X_{max}, a root of mean square X_{RMS}, and a mean absolute value X_{ave} .

The measurement of displacement x is convenient for low-frequency events such as measuring backlashes, etc. which might be calculated the following way [6]

$$x = X_{\max} \sin \omega t \qquad (1)$$

$$\omega = 2\pi f \qquad (2)$$

where: X_{max} - maximum amplitude (maximum displacement); ω - angular frequency; f - frequency (oscillation); t - time.

Velocity can be expressed as the characteristics of motion which informs us about the way of changing the position of a body (particle) in time. Velocity is a vector physical value, because it defines both the change magnitude and its direction. Velocity might be determined as the time derivation of trajectory (displacement) using the equation below

$$v = \frac{dx}{dt} = X_{\max}\omega\cos\omega t \tag{3}$$

Acceleration can be expressed as the characteristics of motion which shows the way the velocity of a body (particle) changes in time. The acceleration is a vector physical value, since it gives both the change magnitude and its direction. It is possible to calculate momentary acceleration and average acceleration. The acceleration might be also determined as the time derivation of velocity using the formula below [6]

$$a = \frac{dv}{dt} = -X_{\max}\omega^2\sin\omega t = X_{\max}\omega^2\sin(\omega t \pm \pi) \tag{4}$$

If the acceleration is in counter-motion, it is called deceleration and has a minus sign.

The International Standards Organization (ISO) which establishes internationally acceptable units for the measurement of machinery vibration suggested velocity – root mean square (RMS) as the standard unit of measurement. This has been agreed on in order to derive criteria that would determine an effective value for the varying function of velocity. The RMS velocity tends to provide the energy content in the vibration signal, whereas the velocity peak corresponds better with the intensity of vibration. Higher RMS velocity is generally more damaging than a similar magnitude of velocity peak [6].

The calculation of the mean value A_{ave} and RMS effective value was based on acceleration and therefore the effective value is marked as *gRMS*. For the calculation we have used the following equations.

The mean absolute value A_{ave} can be expressed as follows

$$A_{ave} = \frac{1}{T}\int_0^T |a| dt \tag{5}$$

where T – a period expressed by the formula $T = \frac{1}{f}$.

The root of mean square can be calculated by the equation below [6]

$$gRMS = \sqrt{\frac{1}{T}\int_0^T a^2 dt} \tag{6}$$

3. Characteristics of vibration sensors for measuring

Vibration sensors are used for measuring physical values (displacement, velocity and acceleration) and their transforming into an electric signal which is later worked with. There is a great variety of different vibration sensors which are used for achieving results as accurate as possible. We have the sensors of acceleration, velocity and displacement. The principle of a vibration sensor function is the motion of seismic matter of mass m towards an object of mass M whose vibrations are measured. For the calculation the formula below is used

$$ma_h + bv + ky = -ma_0 \tag{7}$$

where y – displacement; v – velocity; ah - acceleration of seismic matters motion; a_o - object acceleration; m - seismic matters mass; k - spring stiffness; b - damping coefficient [7].

For the measurement we have used the full equipment of the Dewetron company including evaluation software DEWESOFT. We have used the device DEWE-50-USB2-8 which is an 8-channel measuring system where eight slots might be connected for modules DAQ and PAD.

For the measurement we used two tri-axial acceleration sensors which are at present basic vibration sensors used for measuring mainly on fixed parts of rotating machines like in this case. We used the sensors KS943B.100 which are placed in a duralumin case, measuring range ± 60 g, measuring sensitivity 100 mV/g, frequency range 0.5 Hz – 22 kHz.

If the measurement is to be accurate, it is necessary to select the right place to measure and the right way of attaching sensors. Sensors are placed on clean surfaces and if possible as close to the place where vibrations occur as possible. If the sensors are attached inappropriately, the measured data may be completely devaluated, or the applicable frequency range of a sensor can be significantly limited. When attaching acceleration meters, the following methods may be used: (Fig. 2):

a) by a contact probe – applicable only for preliminary measurement to about 0.6 kHz,
b) by a magnet – a frequent and quick way of attaching a sensor to ferromagnetic materials, but used only for common operating measurement to a frequency of about 7 kHz,
c) by beeswax or a thin bonding tape – a quick way of anchoring a sensor which is used mainly for laboratory measuring and for smaller sensors. It is applicable only at the temperatures of up to 40 °C and sensor frequency of about 10 kHz,
d) by adhesive – if the surface suits the needs of a sensor and the applicable frequency of the sensor is kept at about 10 kHz,
e) by a screw – it is the most reliable way to anchor objects which decreases only slightly the applicable frequency of a sensor which is about 15 kHz. The surface under the sensor base should be clean and even, the sensor should be in full contact with the surface, the screw hole should be perpendicular to the measured surface and sufficiently deep, and the right screw thread should be used. This way of attaching was used during our measurement.

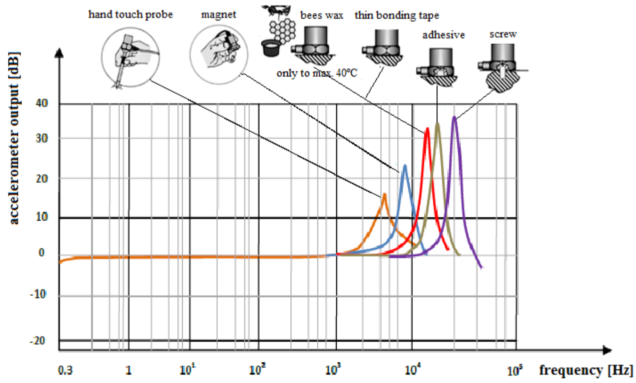


Fig. 2 Ways of attaching measuring sensors and their frequency ranges [8]

4. Description and results of the bearing housing vehicles

The measurement was performed on a testing stand shown in Fig. 3 Testing stand for vibrations measurement. The main parts of the stand are a frame for anchoring measured equipment (1), an asynchronous motor with engine speed control in 10 – 3000 rpm (2), a propeller shaft with a flange (3), a frequency converter MOVITRAC MCLTE B0040-5A3-4-00 (4), measuring equipment DEWE-50-USB2-8 (5), a notebook with software DEWESOFT X2 (6), two pieces of tri-axial acceleration sensors KS943B.100 (7), and measured equipment (8). We have used a bearing box of the PV3S vehicle

The aim of the measurement was to identify a technical state of two conical bearings. Therefore the sensors were placed on a bearing box body directly above the bearings (7). The contact surfaces for attaching acceleration meters had to be prepared so that the meters could be anchored according to the set requirements.

4.1 Determining of the technical condition of bearings based on gRMS

In the bearing box there are conical bearings labelled as FAG 31308A. The bearing 1 is close to an electromotor flange. This bearing was not a subject to any failure during all the period. The bearing 2 is placed in the gear part of the bearing box and it worked without failures up to 122500 km. We disassembled the bearing and performed technical specification and analysis. As part of this experiment we also measured the gRMS values at all three levels. Then we summed all these values and expressed the gRMS dependence on the mileage in graphs 1 and 2. In graph 1 we were observing the bearing 2 where the artificial failure had been made during the mileage 122500 km. The values gRMS were being observed at 1200 and 2200 revolutions.

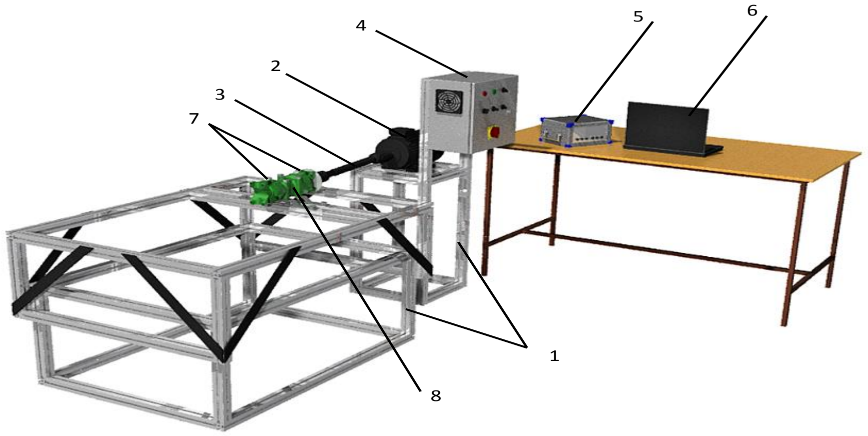
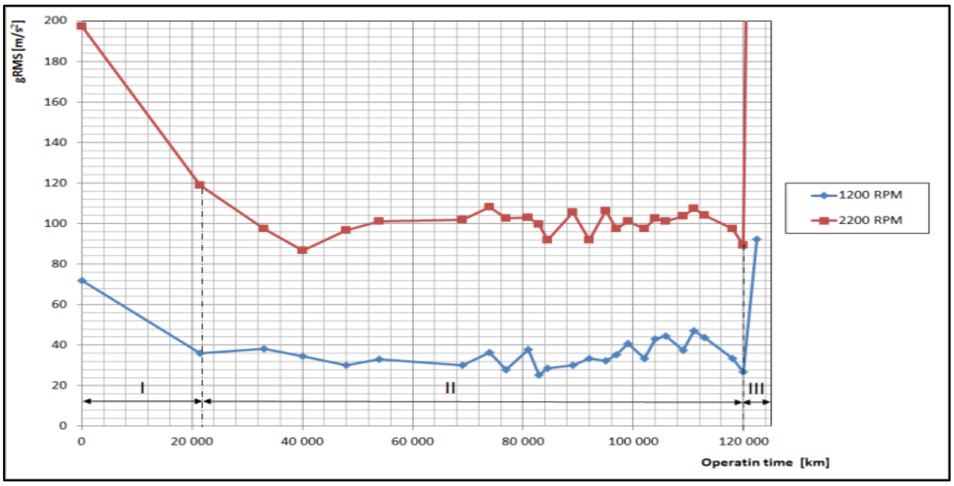


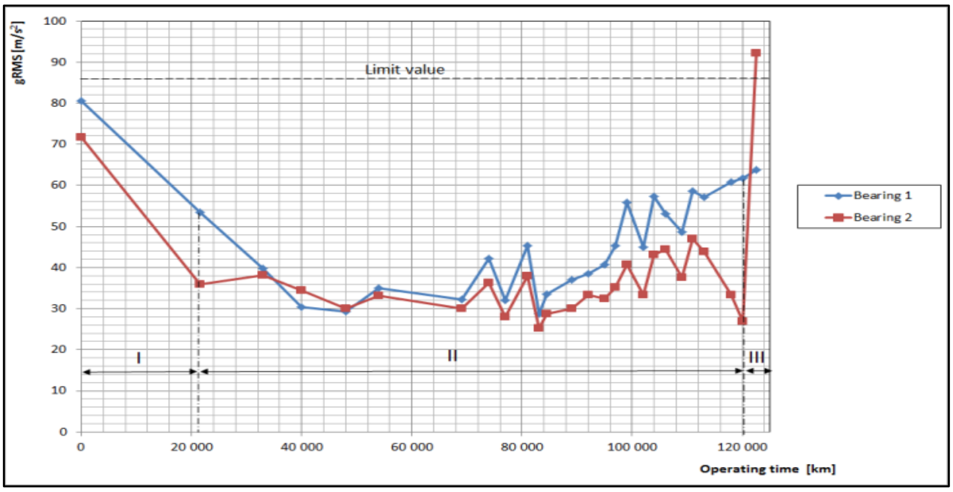
Fig. 3 Testing stand for vibrations measurement [8]

In graph 2 there are the values of gRMS for the bearings 1 and 2 during the mileage 122500 km at 1200 RPM. These graphs show that at first there is a period of running-in and positioning bearings against each other with the gRMS value going down. During another period characterized as operating the gRMS becomes more or less constant. In the third period of an object life when the object becomes considerably worn we made, as it has been mentioned before, the artificial bearing failure. This led to a significant increase in gRMS which had been expected.

Based on the measurements we determined in graph 2 the gRMS limiting value 85 ms^{-2} for both bearings at 1200 RPM. The Figure 1 shows that this limiting value will differ depending on RPM.



Graph 1 Damaged bearing 2 – expressing gRMS dependence on mileage

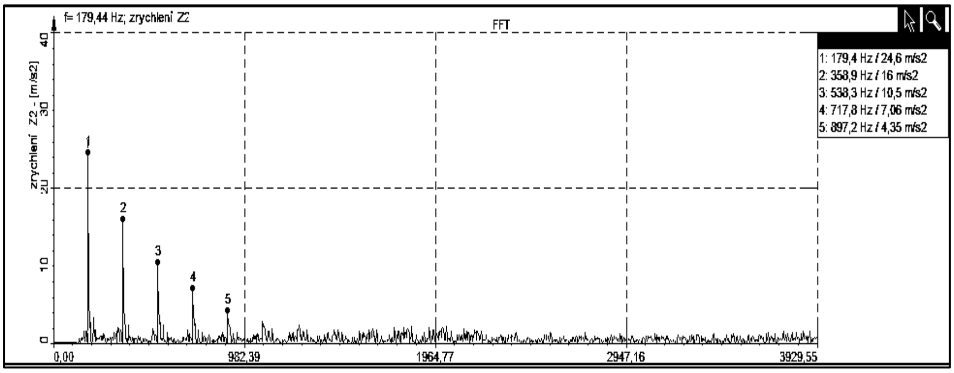


Graph 2 gRMS dependence on mileage at 1200 RPM for the bearings 1 and 2

4.2 Artificially created defect - defect in an outer ring

In order to verify whether it is possible to use frequency values for specific failures calculated in advance, we gradually created defect in the second bearing.

As for the first mechanical damage to the bearing, we chose an outer ring in which a groove parallel with a rotation axis and perpendicular to the route of rolling elements was cut. This ensured that the rollers would be made to roll over the groove.



Graph 3 Frequency spectrum record of the damaged outer ring

For the analysis we selected the measurement 1600 RPM on a shaft and the axis “Z2” record, but also the same can be observed in the other axes. In the right upper corner we can see that these values are really within a given range. Minimum deviations can be caused by the inaccurate measurement of bearing parameters.

Table 1 - The values calculated for bearing fault frequencies labeled FAG 31308A

Designations of fault frequency	f_r frequency 20 [Hz]	f_r frequency 26,66 [Hz]	f_r frequency 36,66 [Hz]
Snap ring defect - FTF	8,42	11,22	15,43
Bearing defect - BPF	107,22	143,72	194,63
Outer race - BPFO	134,64	*179,48*	246,80
Inner race - BPFI	185,36	247,08	339,76

In order to illustrate the frequencies of our interest better, we used a broadband filter where the lower filter limit was six times the amount of rotation frequency which was in our case 160 Hz, and also we applied the envelope. After the application of the filter and envelope we achieved a clear spectrum with the required frequencies.

No visible signs of wear were detected. In order to verify the calculation of failure frequencies, we notched the outer ring of the bearing 2. On the basis of the measurements we managed to verify the calculated value validity Outer race – BPFO, see Table 1.

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