Jan Furch
Josef Glos
Jiří Blecha

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IDENTIFYING THE TECHNICAL CONDITION OF ROTATING PARTS BY MEANS OF VIBRODIAGNOSTICS

Summary

The paper addresses the issue of identifying the technical condition of rotating parts with by means of vibrodiagnostics. This method has been developing rapidly in recent years and it is conceived to be used for predicting failure occurrence. This enables us to determine an optimum time to perform machine preventive maintenance. The aim is to determine this point in time for groups of components, such as electric motors, gear boxes, and even combustion engines. In this paper, we focus on monitoring the technical condition of the medium truck bearing box on which we simulate different field conditions. A measuring device, DEWE 50/8, including the DEWEsoft software product of the Dewetron Company, has been used for the actual measurement. Along with the vibrodiagnostics used when determining the technical condition of the bearing box, we have also applied thermography for monitoring thermal changes of individual components. The aim of the research is to determine the moment when a preventive action is to be performed or when to change components before the failure occurs and increases the costs.

Key words: *determining the technical condition, prediction of the technical condition, vibration, vibrodiagnostics, DEWEsoft.*

1. Introduction

Vibrodiagnostics is one of numerous methods of technical diagnostics which has been continuously used for monitoring the technical condition of a device by observing the level of mechanical oscillation in real time. The mechanical oscillation of a device is a phenomenon that occurs during its operation. Certain parts become vibration exciters, others, depending on excitation, react specifically.

The level of mechanical oscillation of a device can be described by the bathtub curve. The bathtub curve describes operating conditions in which a device can be during its life. These conditions correlate with arising vibration strokes which cause oscillations of different amplitudes, velocity, and acceleration.

Therefore, the vibrodiagnostics is one of the most important methods used in technical diagnostics for identifying a technical condition. Using the vibration diagnostics, we are able to detect an incipient failure, locate the place of the incipient failure and predict the length of time during which the device is going to work before the failure occurs or a preventive action is taken. Proper application of vibrodiagnostics can prevent undesirable damage to machines, thereby the money to be spent on a repair procedure is saved and costs of the device which is out of order are avoided.

Bartelmus and Zimroz introduced a new diagnostic feature, which can be used for monitoring the condition of planetary gearboxes in time-variable operating conditions [1]. This diagnostic feature was the sum of 10 amplitudes of spectral components from the spectrum of the gearbox vibration signal. Unfortunately, this vibration diagnostic feature is useful only in a limited range of operating conditions. Bartelmus and Zimroz stated that the recognition of different technical conditions is very difficult or impossible for cases of no load or small load [1, 4].

Experimental vibrodiagnostics measurements have been performed while obtaining primary signals. Primary results of measurement are systematized and analysed. Relation between the operating characteristics of rotary systems with bearings of sliding friction and those with bearings of roll is determined. Generalization of research results is made and conclusions are drawn [3, 12, 13].

Two types of excitations can produce noise in rotating machinery, i.e. 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 rotating parts of the machine (rotors, disks, bearings, gears, etc.) 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 [7].

Gearbox noise often contains a wide range of frequencies within the audible range. Advanced signal processing techniques have been applied to an experimental vibro-acoustic analysis of helicopter gearboxes to identify the vibrations caused by gears, shafts, and bearings [9]. On the other hand, a computational vibro-acoustic analysis of geared systems is sparse in the existing literature [11]. Structure-borne gearbox noise originating from gear vibration is often estimated empirically [8]. These acoustic estimates are specific to the experimental gearbox that was analysed and may not be suitable for other gearboxes because the radiated noise depends strongly on the shape of the housing [6].

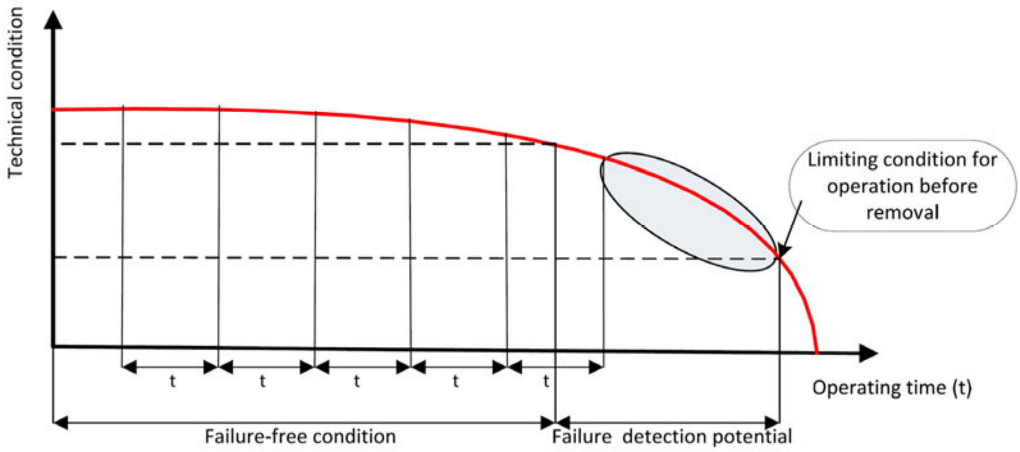


Fig. 1 Dependence of the wear of machine parts on operating time [5]

A lot of companies spend big amounts of money on the preventive maintenance of machines and technical equipment despite the fact that at that particular moment there is no need to take a preventive action. This leads to big financial losses, mainly during the time the machine is unavailable. 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 the real technical condition with the use of a mathematical model describing the machine technical condition during its life leads to proactive maintenance. The aim of our paper is to describe the technical condition of the 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 the technical condition of individual machine components such as shafts, gear boxes, crank mechanisms, cam mechanisms, antifriction bearings, and also by the imbalance of rotating parts, backlash in friction bearings, wear, material fatigue, cracking phenomenon, corrosion, and other parameters affecting the smooth running of the machine. 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 by the rotation about these axes. We can describe them by the amplitude and phase at a certain period of time. Depending on the time variations of values, vibrations are of a periodical, non-periodical, or random character. As for periodical vibrations, the course of vibrodiagnostic values in time repeats. 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 others can be calculated.

The basic way of describing oscillations is to determine their displacement x, velocity v, acceleration a, maximum amplitude X_{max}, the root of mean square X_{RMS}, and the absolute value X_{ave} (Fig. 2).

The measurement of displacement x is convenient for low-frequency events such as measuring backlashes, which may be calculated in the following way:

$$x = X_{\max} \sin \omega t \tag{1}$$

$$\omega = 2\pi f \tag{2}$$

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 the position of a body (particle) changes in time. Velocity is a vector physical value, because it defines both the magnitude and direction of the change. Velocity may be determined as the time derivation of trajectory (displacement) using the equation below:

$$v = \frac{\mathrm{d}x}{\mathrm{d}t} = X_{\max} \omega \cos \omega t\,. \tag{3}$$

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 magnitude and direction of the change. It is possible to calculate the momentary acceleration and the average acceleration. The acceleration may also be determined as the time derivation of velocity using the formula below [10]:

$$a = \frac{\mathrm{d}v}{\mathrm{d}t} = -X_{\max} \omega^2 \sin \omega t = X_{\max} \omega^2 \sin\left(\omega t + \pi\right). \tag{4}$$

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

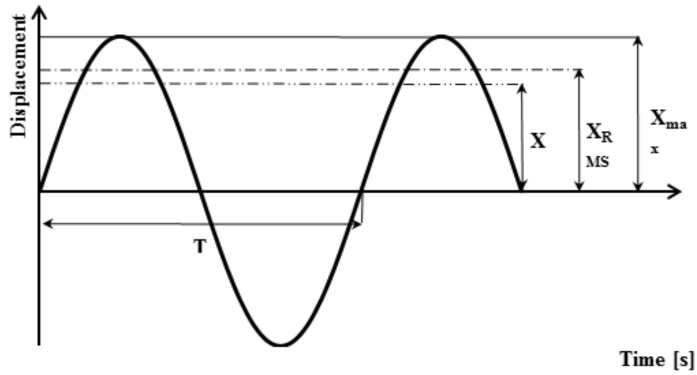


Fig. 2 Harmonic oscillation with the illustration of maximum amplitude X_{max}, the root of mean square X_{RMS}, and the absolute value X_{ave}

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

$$x_{ave} = \frac{1}{T}\int_0^T |x| \mathrm{d}t \tag{5}$$

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

The root of mean square can be calculated by the equation below [10]:

$$x_{RMS} = \sqrt{\frac{1}{T}\int_0^T x^2 \mathrm{d}t} = \frac{1}{\sqrt{2}} X_{\max} . \tag{6}$$

In order to interpret the measured values correctly, it is advisable to transform the oscillation time course into a frequency domain, i.e. vibrations are to be replaced by a sequence of their oscillation components. It can be said that the time signal contains the information about when a certain event occurred, but the frequency spectrum contains the information about how often the same event occurs in an observed signal. The procedure during which complex signals are subdivided into their frequency components is called a frequency analysis which uses either selective band-pass filters or more often a fast Fourier transform (FFT). Along with the FFT, also a wavelet, a cosine or the Walsh-Hadamard transform can be used for expressing a signal by orthogonal basis functions. In the paper we have applied the fast Fourier transform; therefore, we introduce the formula expressing its transformation [10]:

$$F(f) = \int_{-\infty}^{\infty} x(t) e^{-j2\pi ft} \mathrm{d}t , \tag{7}$$

where f – frequency, j – imaginary unit, $x(t)$ – continuous signal.

3. Characteristics of vibration sensors

Vibration sensors are used for measuring physical values (displacement, velocity and acceleration) and for transforming them into electric signals which are later used to work with. There is a great variety of different vibration sensors which are used for achieving results of highest possible level of accuracy, such as acceleration, velocity and displacement sensors. The principle of operation of a vibration sensor is the motion of seismic matter of mass m towards an object of mass M whose vibrations are measured. For calculation, the following formula is used

$$ma_h + bv + ky = ma_0 \tag{8}$$

where y - displacement, v - velocity, a_h - acceleration of seismic matter motion, a_o - object acceleration, m - seismic matter mass, k - spring stiffness, b - damping coefficient [2].

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

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

If the measurement is to be accurate, it is necessary to select the right place for measuring and the right way of attaching sensors. Sensors are placed on clean surfaces 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. 3):

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 ºC 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.

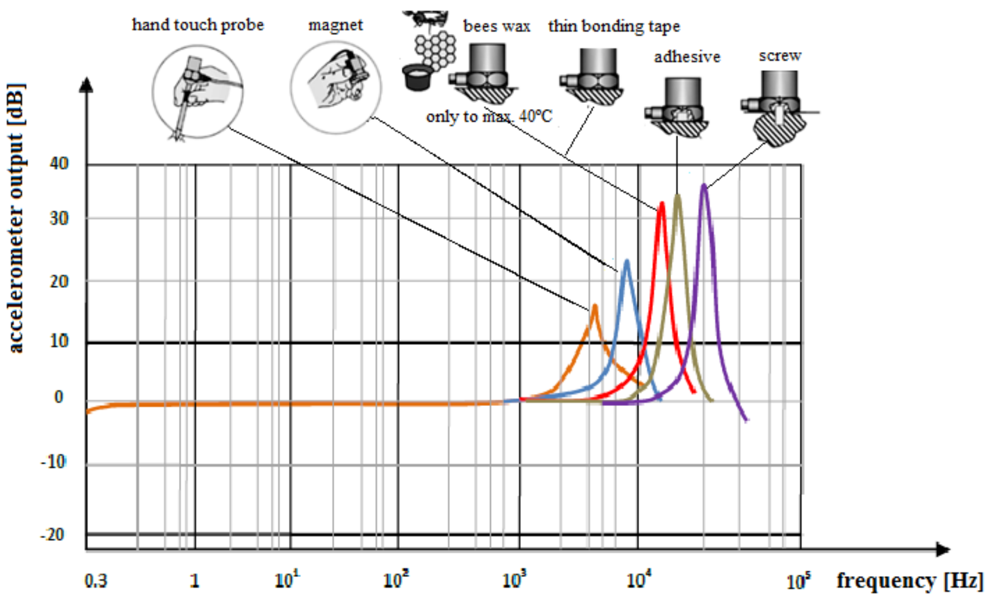


Fig. 3 Ways of attaching measuring sensors and their frequency ranges

4. Ways of damaging antifriction bearings

Due to the surface fatigue of the material of antifriction bearing elements different kinds of damage occur, e.g. surface layer material pitting, abrasion resulting in ply increase, corrosion, grooving, snap ring cavities, and failures. In bearings, the damage of individual elements can be localized on the basis of typical bearing frequencies. As for the kinematic frequencies of impulses, providing that there is a regular rolling motion, the following equations apply [10]:

a) BPFO - Ball Pass Frequency – Outer Race (outer ring defect)

$$f = \frac{n}{2} f_r \left(1 - \frac{BD}{PD}\cos\beta\right), \tag{9}$$

b) BPFI - Ball Passing Frequency – Inner Race (inner ring defect)

$$f = \frac{n}{2} f_r \left(1 + \frac{BD}{PD}\cos\beta\right), \tag{10}$$

c) BSF - Ball Spin Frequency (bearing defect – of a bearing ball or a bearing roller)

$$f = \frac{PD}{2BD} f_r \left(1 - \left(\frac{BD}{PD}\cos\beta\right)^2\right), \tag{11}$$

d) FTF - Fundamental Train Frequency (snap ring defect)

$$f = \frac{1}{2} f_r \left(1 - \frac{BD}{PD}\cos\beta\right), \tag{12}$$

where n is the number of bearing balls or rollers, f_r is the frequency given by the relative revolution (speed) of the inner and the outer ring, BP is the diameter of a bearing ball or roller, PD is the pitch diameter, and β is the contact angle [8].

Bearing wear may be generally divided into four basic stages. In the first stage, initial bearing wear occurs. During the contact of the bearing with the rings, acoustic emission spreads at the frequencies of up to a few MHz. In the second stage of bearing wear, individual elements are damaged. Diagnosing the bearing damage in this stage is performed in a supersonic frequency range from 20 kHz to 60 kHz. For measuring the degree of damage, special acceleration meters with a high value of resonance frequency are used. During the third stage of the damage of bearing elements, conventional acceleration meters may be used. A damaged element when touching another element causes mechanical impact during which kinetic energy is transferred into the bearing body. After the impact, the bearing body is vibrated with its own frequency ranging from 5 kHz to 20 kHz, the vibrations are damped and linger. During the last stage wear occurs, resulting in a critical damage of bearing elements. Vibration spectral elements may be noticed in the area of low frequencies. Moreover, there is an increase in the revolution of spectral components. The wear stages introduced above are described in Fig. 4.

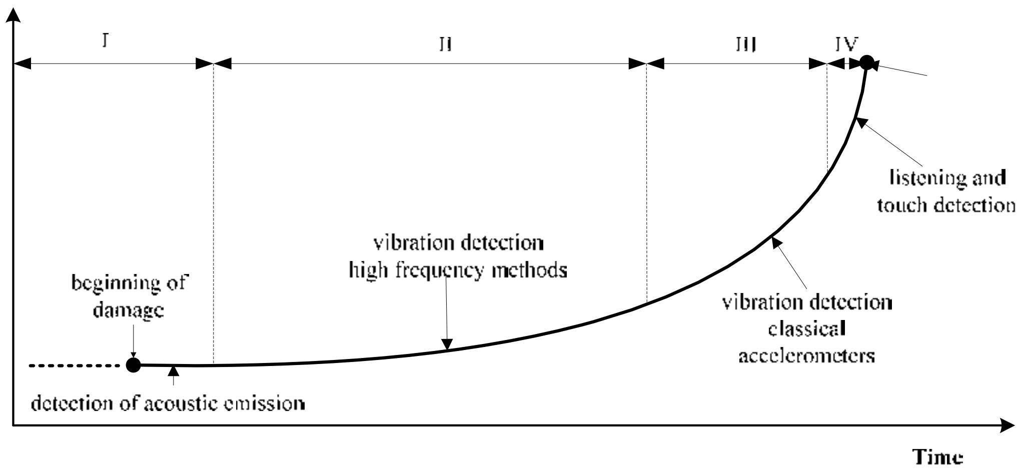


Fig. 4 Course of bearing damage in time and possibility of identification

5. Measurement of vibrations on a vehicle bearing box

The measurement was performed on a test stand shown in Fig. 5. The main parts of the stand are: a frame for anchoring measuring devices (1), an asynchronous motor with engine speed control in 10 – 3000 rpm (2), a propeller shaft with a flange (3), a MOVITRAC MCLTE B0040-5A3-4-00 frequency converter (4), a DEWE-50-USB2-8 measuring device (5), a notebook with DEWESOFT 7.1.0 software (6), two tri-axial KS943B.100 acceleration sensors (7), and a measuring instrument. (8). We used a bearing box of the PV3S vehicle.

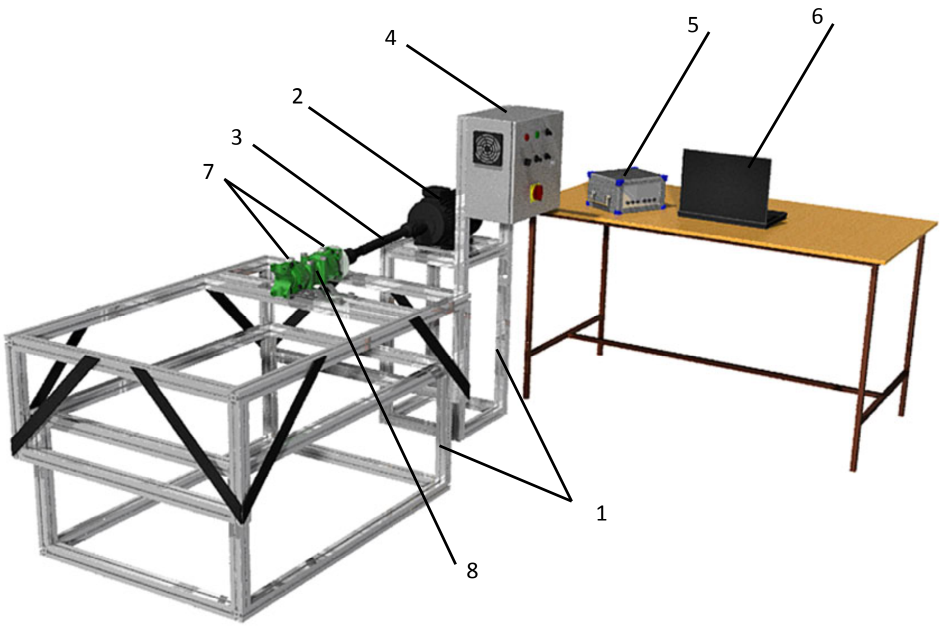


Fig. 5 Test stand for vibration measurement

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

In order to have a better idea and visualisation of temperature distribution during different conditions, we also included thermo-camera Testo 880-3 images.

For the analysis, we selected the measurements during which the vehicle engine revolutions were 1600 rpm with the gear shifted into the third position in the gearbox, and the auxiliary gear shifted into the "road" position in the auxiliary gearbox which corresponds to shaft revolutions of 1219 rpm in the bearing box, see Table 1.

Table 1 Calculated real revolutions in a vehicle bearing box at engine revolutions of 1600 rpm.

Shifted gear in the gear box	Shifted gear in the auxiliary gearbox	Shaft revolutions
I	road	345
	off-road	120
II	road	628
	off-road	238
III	**road**	**1219**
	off-road	425
IV	road	2133
	off-road	744
reverse gear	road	340
	off-road	119

Table 2 Measured values for a 31308A conical bearing in the bearing box

pitch circle	PD	65 mm
contact angle	β	24.374°
the number of rolling elements	N	16
diameter of a rolling element	BD	11.74 mm

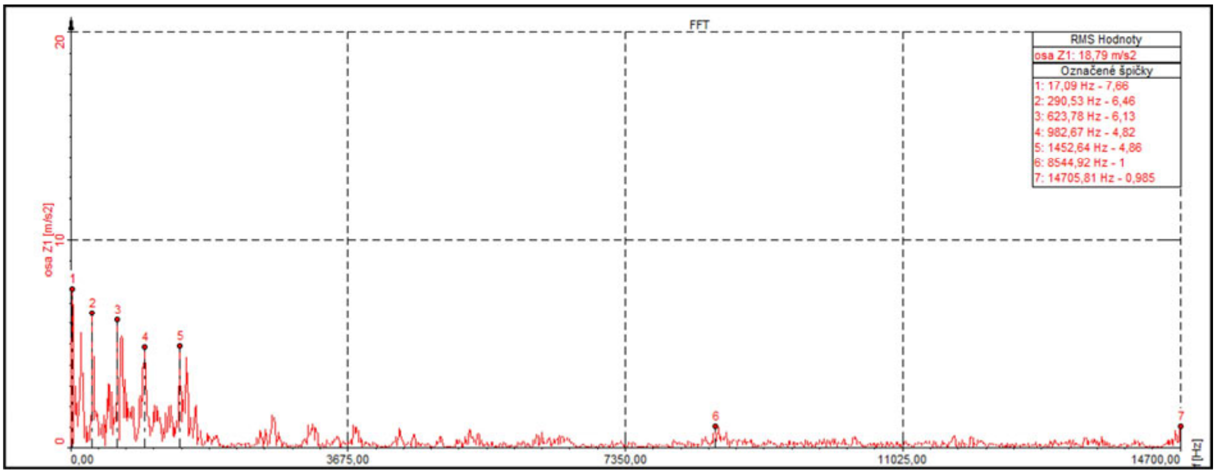
Table 3 Calculated values of typical failure frequencies at the vehicle engine revolutions of 1600 rpm with respective gear positions, see Table 1

Shifted gear	BPFI	BPFO	BSF	FTF
I	53.51	38.39	15.45	30.91
II	105.83	75.93	30.56	61.12
III	**189.28**	**135.80**	**54.66**	**109.32**
IV	331.24	237.65	95.65	191.30
reverse	52.75	37.84	15.23	30.46

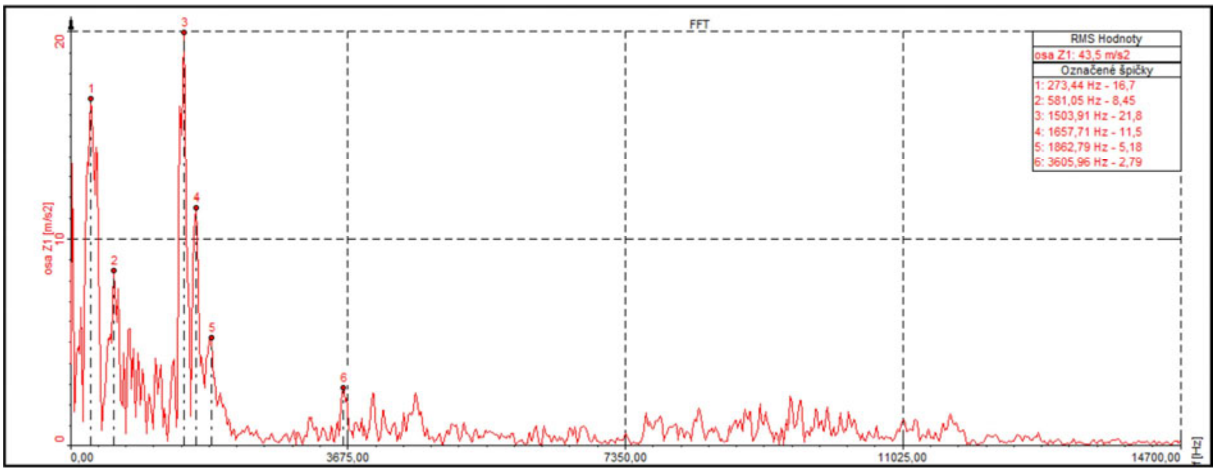
Before performing measurements on the bearing box, we measured the vibrations of the frame itself (Graph 1) on which we performed an experiment to eliminate these frequencies.

The measurements were performed in different modes, namely:

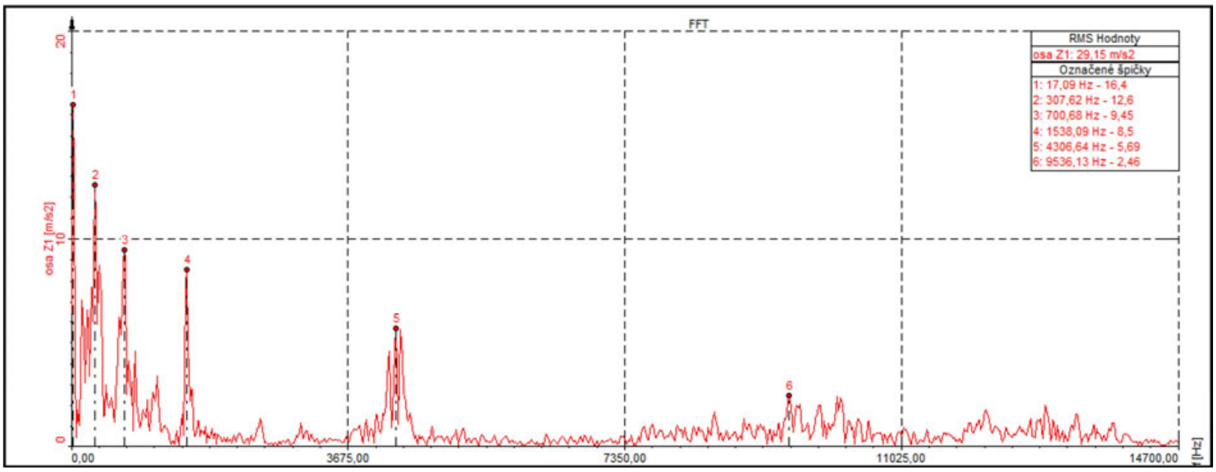
1) According to the way of attaching the sensors during the field conditions of the bearing box:
   a) with a magnet made of precious minerals – Graph 2,
   b) with screws – Graph 3,
2) According to the set technical condition of the bearing box:
   a) bearing with a set running clearance of 0.1 mm and lubricated – Graph 3,
   b) bearing with a set clearance of 0.3 mm and not lubricated – Graph 4.



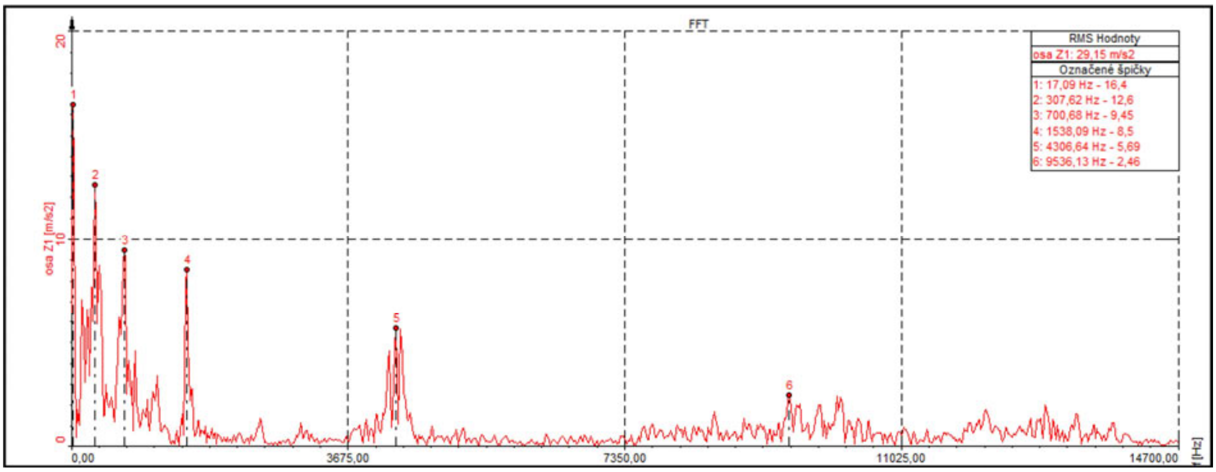
Graph 1 Frame vibration frequency spectrum at 1219 rpm of an electric motor



Graph 2 Vibration frequency spectrum at 1219 rpm during field conditions (lubricated) – the sensor fastened with a magnet



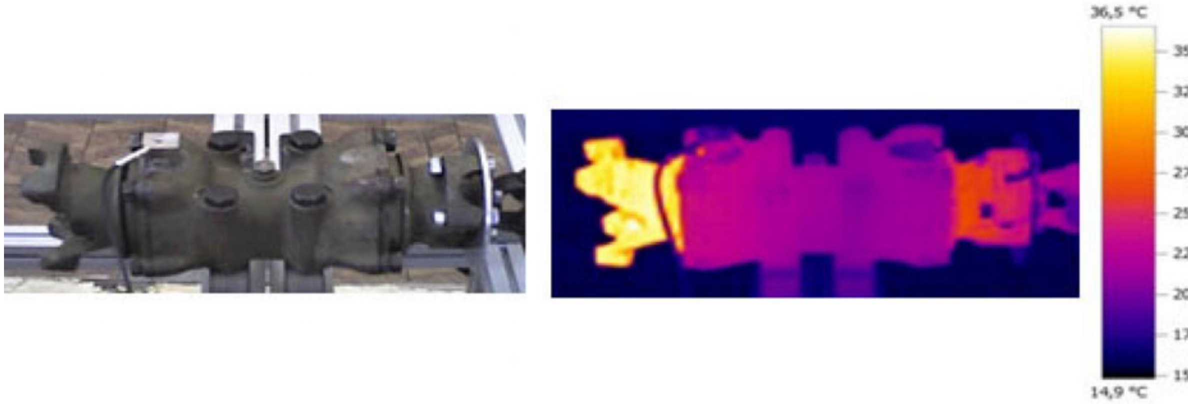
Graph 3 Vibration frequency spectrum at 1219 rpm during field conditions (lubricated) – the sensor fastened with a screw



Graph 4 Vibration frequency spectrum at 1219 rpm at the clearance of 0.3 mm and with a non-lubricated bearing – the sensor fastened with a screw

1) During the first measurement the sensors were fastened with a magnet and screws. We identified the values of bearing box vibrations and put them into Graphs 2 and 3. The graphs show that the way of fastening sensors influences the accuracy of the measurement. It does not correspond with the conditions specified in the standard [ČSN ISO 5348]. The standard says that fastening a sensor with a magnet does not influence the accuracy of the measurement in the frequency range of 0-7 kHz. In the frequency range of 0-2.5 kHz, when a sensor is fastened with a magnet, greater acceleration amplitude occurs, and in the range from 2.5 kHz to 7 kHz the frequencies occur in the given range. We assume that the situation is caused by a poor connection between the sensor and the measured object. That is the reason why the vibrations occur there. The place where the sensor is supposed to be fastened was rubbed and cleaned. It is advisable then to perform measuring when the sensor is fixed firmly with a screw.
2) During the second measuring, we set two technical conditions of a bearing which can occur during operation. First, we measured bearing box vibrations during an ideal operating condition which can be characterized by a bearing clearance of 0.1 mm and lubrication with grease (Graph 3). Afterwards we simulated the condition that can occur because of improper maintenance which can lead to the increase in bearing clearance to 0.3 mm and then to the operation of a non-lubricated bearing. When evaluating the result we should also take into account the vibrations of the frame itself, see Graph 1. As we can see, there are frequencies ranged 0 – 2 kHz on the frame with the maximum amplitude of 7.66 ms^{-2}. When evaluating Graphs 3 and 4, we might therefore attribute the vibrations caused in the range of 0 – 2 kHz to the vibrations of the frame itself. The presumed bearing frequency is about 4 kHz which can be noticed both while observing vibrations under field conditions and at the increased clearance of 0.3 mm of a non-lubricated bearing. When measuring the bearing of 0.3 mm clearance, the amplitude is 5.69 ms^{-2} in the given range as compared with the amplitude of 3.86 ms^{-2} (Graph 4) when measuring the bearing under favourable operating conditions. This leads to the conclusion that the bigger the ply, the more significant the wear is. As for the non-lubricated bearing of 0.3 mm clearance, we can observe that from 7.5 to 14 kHz the amplitude increases, which might be caused by the fact that the bearing was not lubricated by grease.

Along with the measurement of vibrations, we also monitored the thermal dependency of the bearing box. The thermo-camera image in Fig. 6 shows that during operation both bearings are heated slightly, but the heat is transferred more into the input shaft which might be caused by the faster heating of the inner ring. The measured temperatures by no means jeopardize the properties of the bearings which are designed for the temperature of 180 °C. If the temperature increased, it would be the lubricant which would degrade first and only then the bearing.



a) Bearing box of a vehicle

b) Thermo-camera image Testo 880-3

Fig. 6 Bearing box under operating conditions

6. Conclusion

The aim of this article is to introduce vibrodiagnostics of engineering systems and to use it for antifriction bearings. We have focused on monitoring the technical condition of a vehicle bearing box which is used for transferring the angular momentum to the third axle of the vehicle. Along with monitoring the vibrations, we have also measured the temperature of individual bearing box components.

Our observations bring us to the conclusion that the frequency range from 0 to 2 kHz is caused by the vibrations of the frame itself. The frequency of the antifriction bearing is about 4 kHz with different amplitudes depending on the fact whether the bearing is operated in favourable technical conditions or whether its clearance increases to 0.3 mm and the bearing is not lubricated. Unfavourable technical conditions are reflected in the rate of the bearing wear.

The final conclusion drawn from this experiment is that using magnets for fastening sensors is not advisable during the measurement. We recommend therefore fastening a sensor by a screw.

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prof. Eng. Jan Furch, Ph.D.
Eng. Josef Glos, Ph.D.
Bc. Jiri Blecha
University of Defense Brno
Faculty of Military Technology
Kounicova Str. 65
662 10 Brno, Czech Republic