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IMPACT OF SELECTED OPERATIONAL PARAMETERS ON MEASURES OF TECHNICAL CONDITION OF ROLLING BEARINGS IN MEANS OF TRANSPORT BUILT BASED ON ANALYSIS OF VIBRATION SIGNALS

Summary. This article presents an innovative method of diagnostics of rolling bearings used in the bearing nodes of motor vehicles, with the use of a prototype specialist stand. The tests were carried out based on a developed research plan, which included the impact of damage to the bearing and tyre of the vehicle, as well as the vehicle speed. Vibration accelerations were recorded in three measurement axes. Signal spectra were created based on the time courses of the vibration signals and were further analysed. The presented method is aimed at detecting excessive wear of rolling bearings in wheels from its early period.

Keywords: vibration diagnostics, bearing, wear, means of transport

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1. INTRODUCTION

Extensive use of rolling bearings in operating means of transport necessitates the evaluation of their technical condition. This is related to, among others, high costs of removing failures, often caused by damage to rolling bearings. Often, the cost of the rolling bearing is small to the costs associated with their replacement and damage to other parts or the whole components whose correct operation is dependent on the condition of the bearings [3, 4, 7, 9, 10, 14, 17-19]. Figure 1 shows an example of rolling bearing damage.

Fig. 1. Damage to the outer race of a rolling wheel bearing of a vehicle - marked in red

Usually, during normal operation, wear of rolling bearings is a long-term process stretched over the lifetime of a given machine. Generally, bearings undergo fatigue wear (approx. 70 to 80%), much less often rolling bearings undergo abrasive wear (from approx. 20 to 30%) [2, 16].

Often, the life of rolling bearings is shortened due to errors related to incorrect use and maintenance. Common factors that reduce the life of rolling bearings include incorrect axial loads, incorrect assembly, shaft deformation, inadequate shaft and housing tolerances, excessive preload, unsuitable lubricant, foreign bodies (for example, sand), wear corrosive, inadequate selection of bearings for given operating conditions, inadequate rotational speed and faulty housing bore having a major impact on proper or incorrect bearing fit.

Assessment of the technical condition of bearings can be carried out based on specific symptoms accompanying the operation of the bearing hub. Such symptoms include lubricant contamination, noise, increased bearing resistance, vibration and temperature [8].

Given the above symptoms, rolling bearing diagnostics based on the analysis of vibration signals seems to be the most reasonable, due to its non-invasive character - no need to dismantle the bearing. Another advantage is its ability to detect wear at a relatively early stage [15].

This article attempts to address the subject related to the impact of selected operating parameters on measures of technical condition of bearings built based on the analysis of vibration signals.

2. THEORY OF VIBRATION OF ROLLING BEARINGS

Each rolling bearing is a vibration generator. Despite advanced production technology, we are not able to make perfectly smooth surfaces. The production process carried out is characterised to result in a surface roughness at the appropriate level. This level will be conditioned by the applied machines and production technology. During the cooperation of two elements with a given surface roughness, for example, cooperation of the bearing ball with its outer race, vibrations will be generated [1, 11, 13]. During the operation of rolling bearings, regardless of the type of damage, they have the nature of local geometric deformation. Each impulse forcing of the elastic system is characterised such that the system responds dynamically depending on the function of the system's passage through local geometric deformation. The sequence of pulses generated by vibrations can be described by Equation 1 [5, 6, 15]:

$$P(t) = \delta\left(t - \frac{1}{f_{contact}}k\right)S(t)P(t) = \delta\left(t - \frac{1}{f_{contact}}k\right)S(t)$$

$$P(t) = \delta\left(t - \frac{1}{f_{contact}}k\right)S(t) \tag{1}$$

where:
δ – the Dirac puls, $\frac{1}{f_{contact}}k$ - time interval between successive strokes, S(t) –the change in load acting on the damage in time.

Therefore, from the viewpoint of vibration diagnostics, the spectral image that can be obtained after performing the Fourier transform is important. The transform shows the dependence of the signal time variable in the form of a frequency image. It is one of the most commonly used methods of frequency analysis of digital signals. The general form of the total transform is [5, 6, 8]:

$$X(f) = \int_{-\infty}^{+\infty} x(t)e^{-2\pi ift}\,dt \tag{2}$$

where:
t – time,
f – frequency.

Changes in the analysed signal in a certain unit of time can take many instantaneous values. For the analysis is not required to observe the change in the instantaneous. Defining the value of the data signals defining characteristic points is made possible by estimated dimensional, for example, the average value, the correlation function of their own, mean-value, effective value.

The RMS (x_{rms}) value is the root of the mean square value [5, 6, 8]:

$$x_{rms} = \sqrt{\psi^2} = \sqrt{\frac{1}{T}\int_0^T x^2(t)\,dt} \tag{3}$$

3. RESEARCH PLAN AND PRESENTATION OF RESULTS

The basic element enabling the conducting of the test procedure for assessing the technical condition of the rolling bearings of the road wheels is a diagnostic device that allows acceleration of the road wheels in a wide range of rotational speed, adjustment of the load on the bearing node and monitoring and recording of vibration signals in three axes X, Y, Z sensors located near the examined bearing hub.

The experimental tests and calculations were carried out following the statistically established plan of a multi-selective experiment, consisting of determining the input quantities and their position concerning the base-mid point. The experimental plans allow determining the influence of individual factors on a normalised scale (x_1, x_2, x_3) on the final value of x_{RMS}.
The design of the experiment for the three input factors on the normalised scale is presented in Table 1 [10, 12]. The values in Table 1: -1,0,1, define the minimum, average and maximum values of a given factor on a standardised scale, which should be referred to as the value in the next step, but on the real scale.

Tab. 1
Research experiment plan
- factors on a normalised scale

Measurement	x_1	x_2	x_3
1	-1	-1	1
2	1	-1	-1
3	-1	1	-1
4	1	1	1
5	-1	0	0
6	1	0	0
7	0	-1	0
8	0	1	0
9	-1	-1	1
10	1	-1	-1
11	-1	1	-1

Then, to create an experiment plan for factors on a real scale, define the input factors x_1, x_2, x_3 and assign their minimum, average and maximum values of input factor. The following is a description of the input factors for the real scale, as the control factors have been selected x_1 - technical condition of the wheel and bearing node:

$$x_1 = \begin{cases} -1\ for \text{ Damaged bearing and tyre} \\ 0\ for \text{ Damaged bearing and new tyre} \\ 1\ for \text{ New bearing and new tyre} \end{cases}$$

Then, as input factor x_2 was defined - the speed of the vehicle wheel:

$$x_2 = \begin{cases} -1\ for 67 \text{ km/h} \\ 0\ for\ 90 \text{ km/h} \\ 1\ for\ 112\ km/h \end{cases}$$

As the input factor x_3 is defined - direction measuring relative to the vehicle's axis of rotation.

$$x_3 = \begin{cases} -1\ for\ \mathrm{X}-\mathrm{axis} \\ 0\ for\ \mathrm{Y}-\mathrm{axis} \\ 1\ for\ Z-axis \end{cases}$$

Table 2 shows the values of the factors in the real scale due to which the research was carried out.

Tab. 2

Research experiment plan – real scale factors

Measurement	x_1	x_2	x_3
1	Damaged bearing and tyre	67 km/h	Z
2	New bearing and new tyre	67 km/h	X
3	Damaged bearing and tyre	112 km/h	X
4	New bearing and new tyre	112 km/h	Z
5	Damaged bearing and tyre	90 km/h	Y
6	New bearing and new tyre	90 km/h	Y
7	Damaged bearing and new tyre	67 km/h	Y
8	Damaged bearing and new tyre	112 km/h	Y
9	Damaged bearing and tyre	90 km/h	X
10	New bearing and new tyre	90 km/h	Z
11	Damaged bearing and tyre	90 km/h	Y

The object of this study was the right wheel hub bearing front passenger vehicle equipped with double ball bearing. On the side, the outer side of the tyre was observed a bulge, which might indicate a damaged braid. Figure 2 presents a prototype stand for assessing the technical condition of the road vehicle wheel bearing hub during the experiment with the vehicle tested. In the created experiment plan, the mathematical model is a second degree polynomial, the formula (4) of the general form of which is [12]:

$$y = b_0 + \sum b_k x_k + \sum b_{kk} x_k^2 + \sum b_k x_k x_j \tag{4}$$

The vehicle speed values proposed in the experiment plan were selected according to the case of the vehicle moving at the speed allowed in the undeveloped area. Signal sampling frequency was 51.2 kHz. The bearing node consisted of a hub, a double-row ball bearing and a bearing pin. Figure 3 shows the mounting of vibration acceleration sensors.

The determined effective x_{rms} values for individual runs related to different directions of measurements X, Y, Z following the experiment plan are presented in Table 3.

Due to the fact that the tests were carried out according to the optimal plan of the experiment, the calculated effective signal values can be presented, depending on the adopted input factors. Figure 4 shows sample results of the effective value of the signal for a damaged bearing and damaged tyre, depending on the set wheel speed and measuring direction. As observed, the highest effective values are obtained for conditions of intermediate value of rotational speed ($x_1 = 0$), that is, 90 km/h speed and measuring direction according to the Z-axis.

Fig. 2. Prototype stand during testing of the vehicle wheel bearing function

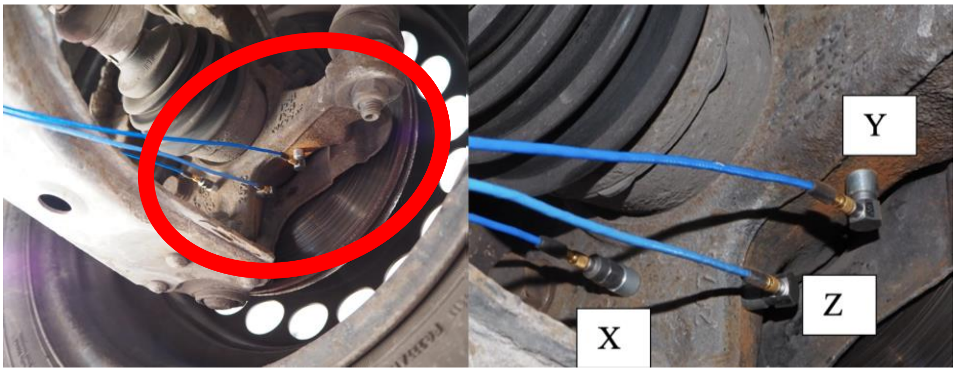
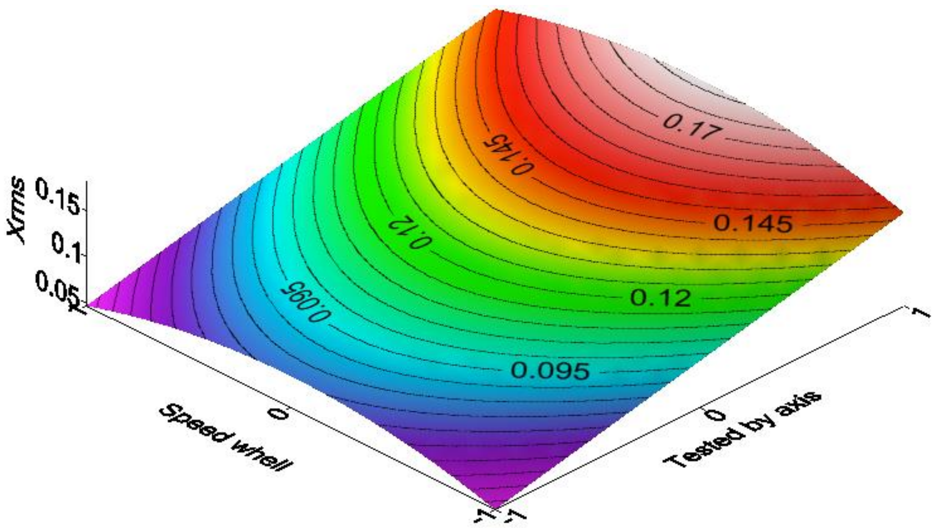


Fig. 3. Locations for mounting vibration acceleration sensors (marked in red) and determining their coordinate axis

Tab. 3
RMS (x_{rms}) results for the tests performed according to the experimental plan

Measurement	x_{rms} [m/s^2]
1	0,0584
2	0,0532
3	0,1429
4	0,0506
5	0,1227
6	0,0895
7	0,0795
8	0,1804
9	0,0925
10	0,0972
11	0,1409



$$X_{rms}(x_2, x_3) = 0{,}05x_1^2 - 0{,}03x_2^2 + 0{,}1315$$

Fig. 4. The results of the effective value of the signal for the case $x_1 = -1$, depending on the rotational speed of the wheel and the tested axle

Thereafter, based on measurements and recorded vibration signals, Fourier transforms were performed to obtain a frequency image of the signal. Figures 5-7 show the waveforms of vibration acceleration for a wheel speed of 112 km/h, various technical conditions and different measuring axes.

In Figure 5, the waveforms of the vibration acceleration spectrum for measurements in the X-axis for the states $x_1 = -1$ - marked in red, $x_1 = 0$ - marked in yellow and $x_1 = 1$ - marked in green are in the frequency ranges of 1500 to 5500 Hz to each other. In the frequency range from 500 to 1500 Hz, the values of vibration acceleration are clearly lower due to a decrease in the vibration energy in this range, owing to the mounted new bearing and a new tyre.

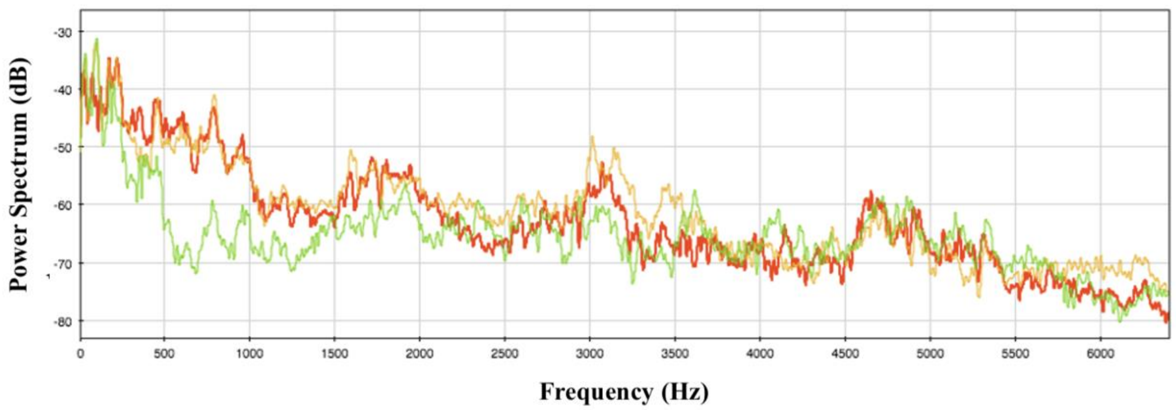


Fig. 5. Vibration acceleration spectrum (frequency) for wheel speed 112 km/h and measurement in the Y-axis, where red - damaged bearing and damaged tyre, yellow - damaged bearing and new tyre, green - new bearing and new tyre

Figure 6 presents the spectrum of vibration accelerations for measurements in the Y-axis for the states $x_1 = -1$ - marked in red, $x_1 = 0$ - marked in yellow and $x_1 = 1$ - marked in green. On the spectrum of vibration acceleration spectra, it can be observed that changing a damaged bearing to a new one and replacing a damaged tyre with a new one causes a decrease in the vibration energy in the frequency range from 500 to 2000 Hz. The remaining frequency range overlaps regardless of the tested technical condition of the bearing node. Installing a new tyre reduces the energy of vibrations generated in the higher frequency range - from 4500 to 5000 Hz.

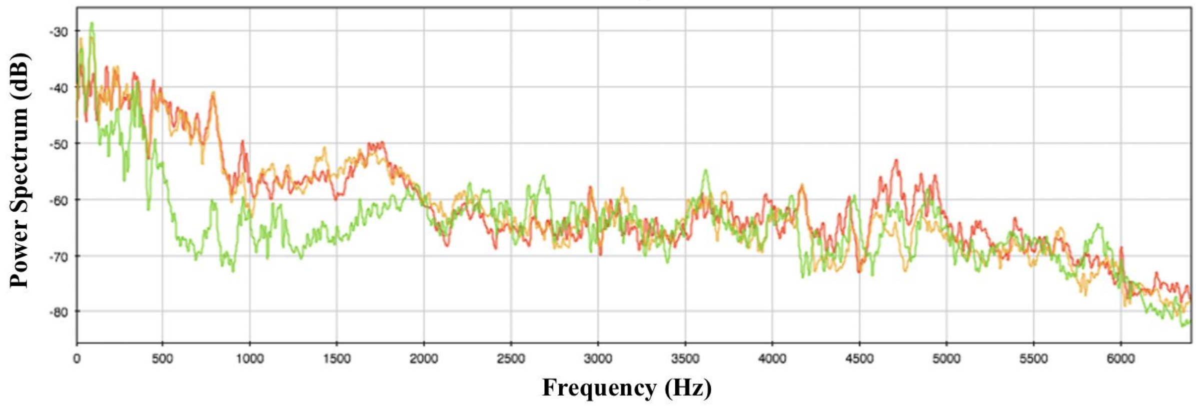


Fig. 6. Vibration acceleration spectrum (frequency) for wheel speed 112 km/h and measurement in the Y-axis, where red - damaged bearing and damaged tyre, yellow - damaged bearing and new tyre, green - new bearing and new tyre

Figure 7 presents the spectrum of vibration acceleration for measurements in the Z-axis for the states $x_1 = -1$ - marked in red, $x_1 = 0$ - marked in yellow and $x_1 = 1$ - marked in green. The tests performed in the Z-axis correspond to the lateral displacements of the bearing node and the road wheel. On the spectrum of vibration acceleration spectra, it can be observed that changing a damaged bearing to a new one and replacing a damaged tyre with a new one causes a significant reduction in the vibration energy in the frequency range from 50 to 2250 Hz. The frequency range from 5000 to 6500 Hz represents a significant decrease in vibration energy for the considered case of a damaged bearing and a new tyre. This is due to levelling the lateral displacement of one wheel that occurred in the case of a damaged tyre.

4. CONCLUSION

Regardless of the technical condition, each rolling bearing generates vibrations. The level of vibration generated can be enhanced, for example, by a damaged tyre mounted on the vehicle or, by inappropriate tyre pressure. The diagnostic methods used to date - the sense of hearing of the diagnostician, are subjective and should not be used to assess the technical condition. The vehicle wheel hubs are one of the most important elements responsible for the safety of passengers in the vehicle and its surroundings.

The application of the experiment plan made it possible to carry out this planned research and create a dependence of the influence of factors x_1, x_2 and x_3 on the x_{rms} value. This is an extremely valuable feature of planning experiments. The use of the developed test method for diagnosing the technical condition of rolling bearings of road wheels allows for the preparation of an appropriate testing algorithm, considering the nature of the work of a given bearing node and its parameters.

Because of the measurements, specific signal waveforms were obtained, which were then subjected to time-frequency analysis, which allowed identifying components with a characteristic frequency corresponding to the damage to the outer race. As can be done easily in Figures 5-7, change of tuners x_1, x_2, x_3 introducing power changes in the frequency range 500-1500 Hz. Based on the conducted tests, the following conclusions can be stated:

- testing wheel bearing hubs using vibroacoustic tests allows to state the technical condition of the tested hub bearing,
- it is possible to determine the technical condition of cooperating elements, for example, technical condition of a tyre,
- the prototype stand enables the simulation of bearing conditions similar to real conditions, corresponding to urban traffic,
- the stand ensures that the measurement conditions are repeated.

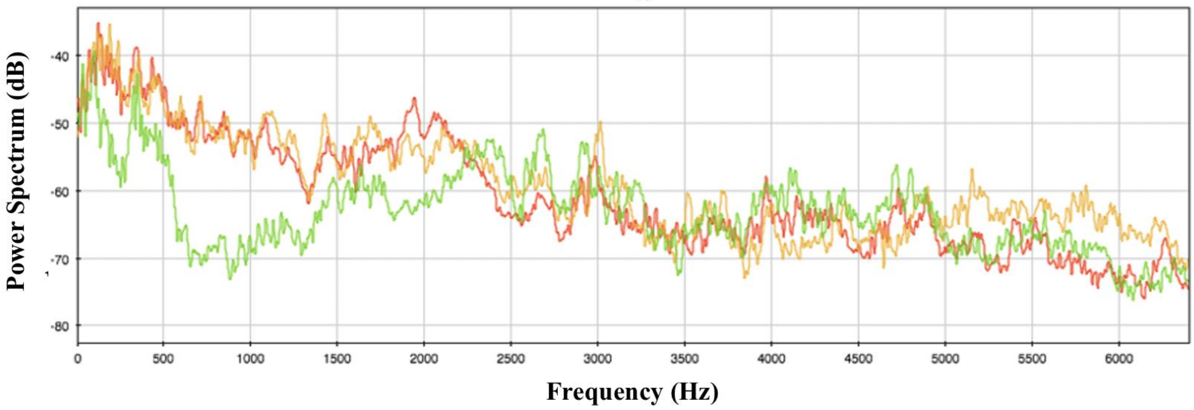


Fig. 7. Vibration acceleration signal waveforms and vibration acceleration spectrum (frequency) for wheel speed 112 km/h and measurement in the Z-axis, where red - damaged bearing and damaged tyre, yellow - damaged bearing and new tyre, green - new bearing and new tyre

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