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SOME PROBLEMS WITH VIBROACOUSTIC METHOD IN RELATION TO THE TRANSMISSION GEARBOX OF A MILITARY HELICOPTER

Summary. The article presents the results of a laboratory gear stage with regard to deepening its consumption. In addition, the study looked at the construction of the transmission gearbox of a military helicopter, according to the basic kinematic results of the analysis of vibroacoustic signals, in order to determine the diagnostic criteria for the assessment of the technical condition of a military helicopter's transmission gearbox.

Keywords: transmission gear, vibroacoustics, gearbox, helicopter

1. INTRODUCTION

The issue of monitoring and diagnosing the technical condition of machines, as well as detecting the damage and wear of their components, is a subject of interest to all personnel performing operational maintenance duties. This is for the following reasons: safety, economy and time. However, the methods and measures used, which take the form of implemented

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control measurement systems, along with the defined current services are not always adequate, such that these machines are often damaged.

This is a problem faced by the aviation engineering services of the Polish Armed Forces, which operate combat helicopters, including their main transmissions. In the main, the only aspect of the transmission gear of a helicopter that is constantly monitored is the value relating to temperature and oil pressure, as well as the rotational speed of the input shaft. Although magnetic plugs in transmission gear trap the products that cause wear and tear in the lube oil, there is no system to monitor its vibrations. Additionally, the evaluation of the main transmission gear's technical condition is carried out periodically, during planned maintenance works. Despite the above-mentioned measures, damage still occurs, which is why the Air Force Technical Institute has attempted to determine the possibilities of using the vibroacoustic method to evaluate the technical condition of the main transmission gear of a combat helicopter under conditions as discussed below.

The literature presents many methods for diagnosing transmissions on the basis of the analysis of the vibroacoustic signal measured with the housing of the gear. The existing methods were classified into four groups: time domain analyses, frequency domain analyses, time-frequency analyses and others [1-4]. The aforementioned methods are based on known symptoms of damage to rolling bearings and gear wheel transmission.

2. GENERAL CHARACTERISTICS OF THE TEST OBJECT

The test object is the main transmission located on board the helicopter, as shown in Figure 1, with a schematic diagram presented in Figure 2.

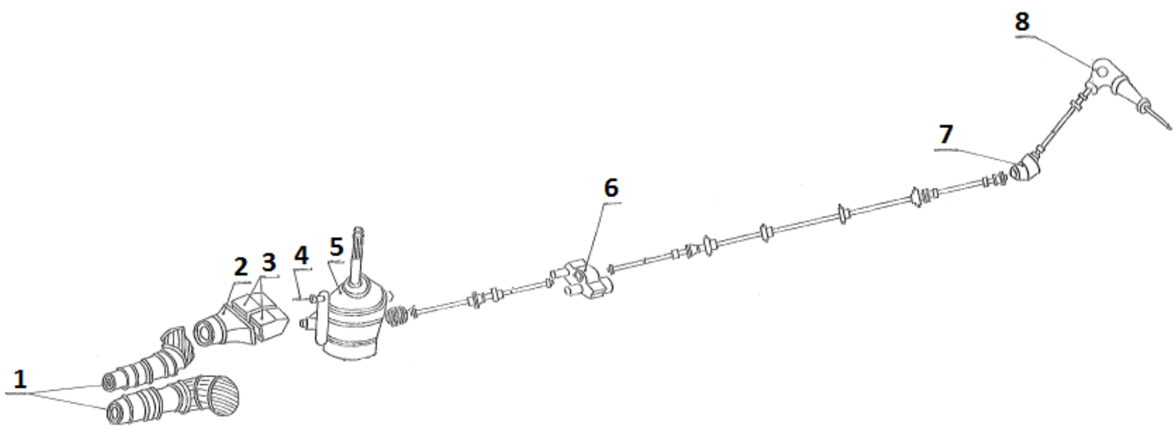


Fig. 1. View of the transmission's location in relation to the basic sub-assemblies of the helicopter
(1 = engines, 2 = fan, 3 = oil cooler, 4 = fan drive shaft, 5 = main transmission gear, 6 = drive box, 7 = intermediate transmission, 8 = rear transmission)

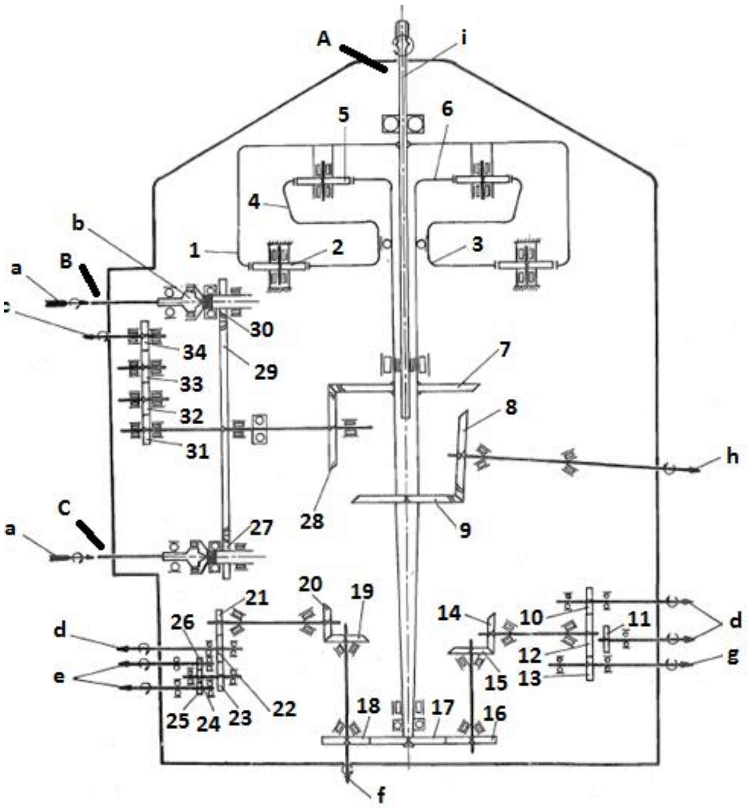


Fig. 2. Kinematics of the main transmission

The design of the transmission and its location on the helicopter causes the occurrence of a series of extortions, which appear in the vibroacoustic signal. First of all, the transmission is powered by two turbine engines, through an idle run clutch (b) and an axle shaft (a). In the engine, apart from a series of mechanical generators coupled with it, the most important elements are the power turbine, generating useful power for the gear, and the turbocharger rotor, which is not coupled kinematically with the engine. Secondly, the transmission of the torque onto the rotor shaft (i), coming from the propulsion turbine, takes places in the course of three stages of the transmission gear. The first stage of the gear transmits rotations from two engines through the idle run clutch and cogwheels 27 and 30 onto cogwheel 29 with helical wheels. The second gear stage consists of two bevel gear wheels 28 and 7 with spiral teeth. The third stage is a closed planetary-differential gear with cogwheels 1, 2, 3, 4, 5 and 6. Sprocket 6, located on one shaft with conical sprocket 7, is a drive wheel (sun wheel) of the transmission gear. Sprockets 5, comprising five pieces, are satellites of the planetary gear. The satellite yoke creates one whole with powered sprocket 1 and is connected with the rotor shaft. Sprockets 2, comprising seven pieces, are transitional wheels, which close the system. The body of these wheels is stationary. Thirdly, the propulsion from the transmission is transferred onto one more series of receiving generators, including the tail propeller and the gearbox generators. Transmission of the drive to the tail propeller (h) is achieved through the first and second stages of the gearbox, which is common for the carrier rotor, and through an additional stage, which increases the rotational speed, consisting of two conical gears, 9 and 8, with spiral teeth. Drives of the main gearbox generators are directed to the front, left and right sides of the gearbox body. To the front side, the fan drive is directed (c) from sprocket 29, through 31, 32, 33 and 340. To the left side, drives of the carrier rotor rotational speed accelerator (e) and the hydraulic pump (d) are directed through sprockets 19 and 20 and cylindrical sprockets 21÷26. To the right side, the drives of two hydraulic pumps (d) and the compressor (g) are directed through sprockets 17 and 16, conical sprockets 15 and 14 and cylindrical sprockets 10÷13. Apart from the assemblies of the above-mentioned toothed kinetic pairs, there are also a number of bearings.

For such complex kinematics, the relations describing characteristic frequencies are presented below:

– rotational frequency of the turbocharger f_{TS} (100% rotations corresponds to 19,500 rpm)

$$f_{TS} = \frac{n_{TS}}{60} \tag{1}$$

– rotational frequency of the turbocharger f_{TN} (100% rotations corresponds to 15,000 rpm)

$$f_{TN} = \frac{n_{TN}}{60} \tag{2}$$

– characteristic frequencies for gears with fixed axes; for sprockets 7 to 34, for example, with a pair of teeth z_{30} and z_{29}, the relation is as follows:

$$f_{29} = f_{30}\frac{z_{30}}{z_{29}} \tag{3}$$

– gearing frequency

$$f_{z29} = f_{29} \cdot z_{29} \tag{4}$$

– frequency of sidebands (similarly for other frequencies)

$$f_{z29-} = f_{z29} - f_{29}; \quad f_{z29+} = f_{z29} + f_{29} \tag{5}$$

– frequency of carrier rotor blade passing (k = blade number, five pieces)

$$f_Ł = f_1 \cdot k \tag{6}$$

Sprockets z_1 to z_6 form a planetary gear, with a layout as shown in Figure 3.

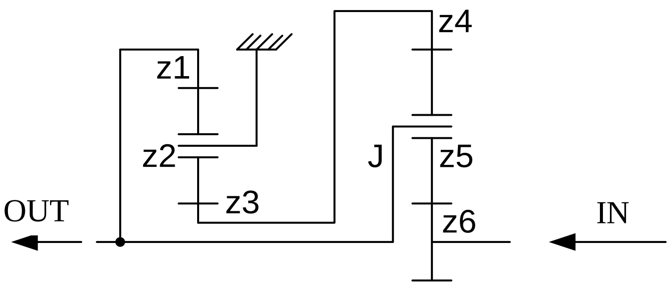


Fig. 3. Planetary gear layout

The planetary gear has *w* degrees of freedom:

$$w = 3n - 2p - k \tag{7}$$

where:

n = number of mobile links,
p = number of rotational links,
k = mesh number.

The rotational frequencies of the planetary gear assembly are determined according to the relations:

$$n_7 = n_6 = n_{28}\,\frac{z_{28}}{z_7} \qquad \rightarrow f_6 = \frac{n_6}{60} \tag{8}$$

$$n_5 = \left[-\frac{z_{28}}{z_7}\left(n_6 - n_J\right)\right] + n_J \qquad \rightarrow f_5 = \frac{n_5}{60} \tag{9}$$

$$n_3 = n_4 = -n_1\,\frac{z_1}{z_3} \qquad \rightarrow f_3 = f_4 = \frac{n_3}{60} \tag{10}$$

$$n_1 = n_6 / \left[1 + \frac{z_4}{z_6}\left(1 + \frac{z_1}{z_3}\right)\right] \qquad \rightarrow f_1 = \frac{n_1}{60} \tag{11}$$

Rotational frequency values for rated operating conditions are summarized in Table 1, which, with the knowledge of the number of gear teeth and appropriate bearing parameters, enabled the determination of the remaining characteristic frequencies associated with symptoms of bearing [6] and transmission gear [7] damage.

Table 1.
Values of the rotational frequencies of the main transmission gear, propulsion unit and the rotor

Designation	Value [Hz]	Designation	Value [Hz]
f_{TS}	325	f_{16}	49.3
f_{TN}	250	f_{15}	49.3
f_L	20	f_{14}	59.6
f_{30}	250	f_{13}	33.5
f_{29}	86.8	f_{12}	59.6
f_{28}	86.8	f_{11}	40.6
f_{27}	250	f_{10}	40.6
f_{26}	39.7	f_9	40.8
f_{25}	39.7	f_8	53.9
f_{24}	39.7	f_7	40.8
f_{23}	39.7	f_6	40.8
f_{22}	40.6	f_5	1.5
f_{21}	59.6	f_4	-8.8*
f_{20}	59.6	f_3	-8.8*
f_{19}	49.3	f_2	-14.6*
f_{18}	49.3	f_1	4
f_{17}	40.8	*opposite direction	

3. LABORATORY TESTS OF A DOUBLE REDUCTION GEAR WHEEL TRANSMISSION

Planned vibration measurements of the helicopter's main transmission were to be carried out without the signal informing about the momentary rotational speed, thus limiting the possibility of applying certain algorithms, e.g., time synchronous averaging. A prepared laboratory bench without the phase marker was designed to validate the applied algorithms for the evaluation of a combat helicopter's technical condition. The expected results concerned the information about the general level of vibrations and the possibility to detect damage to the bearings and the teeth of the transmission.

The gear wheel transmission, as presented in Figure 4, was tested.

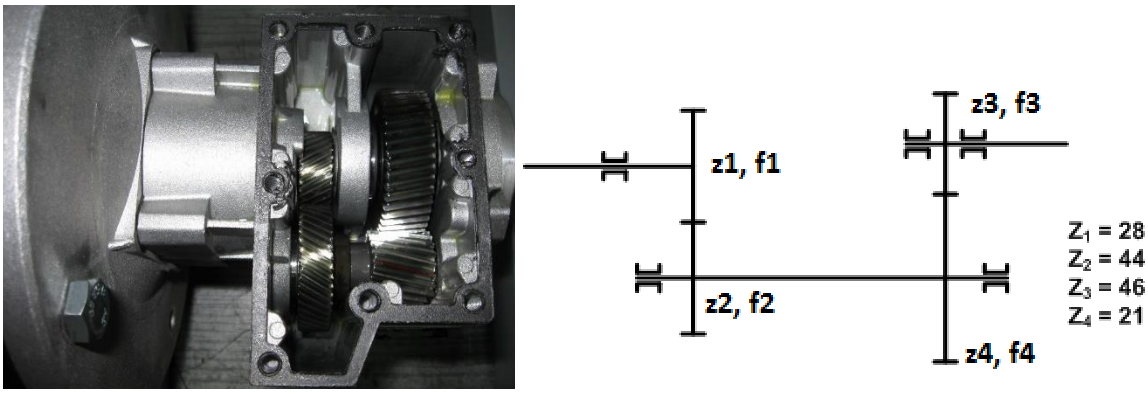
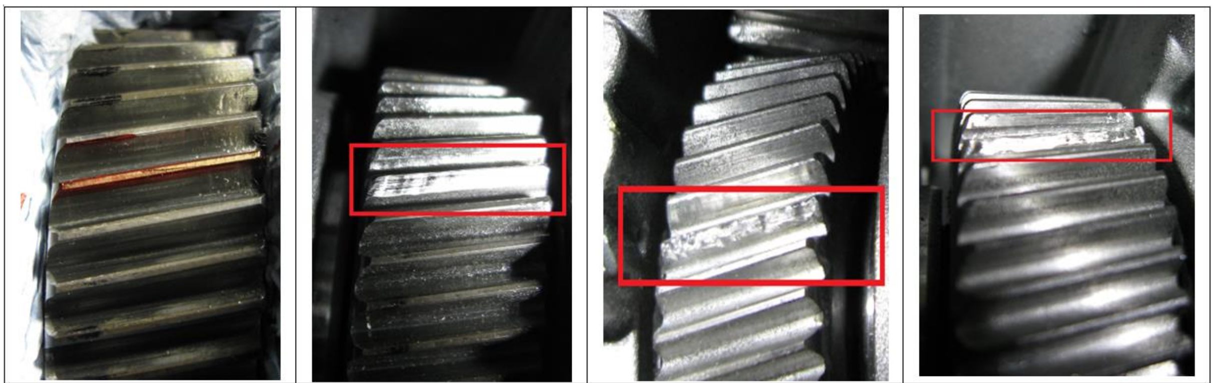


Fig. 4. View of the tested transmission and its kinetic diagram

On implementing, in three subsequent stages, a deepening damage to tooth z_3 (Fig. 5), with stable ranges of the rotational speed, gearbox body vibrations were measured in three, mutually perpendicular, directions and later related to the condition without the damage.



OK – teeth without damage	**dam1: slight grinding of one plane of the tooth**	**dam2: slight grinding of the opposite plane of the tooth**	**dam3: deeper grinding of the tooth planes, with a chipping at a length up to 4 mm**

Fig. 5. View of subsequent damage stages of sprocket z_3

After that, a number of analyses in the time and frequency domains were carried out. In the time domain, in the band 10 Hz÷10 kHz, values were determined for effective vibration speed V, kurtosis K, peak factor CF and signal power P, as presented in Figure 6.

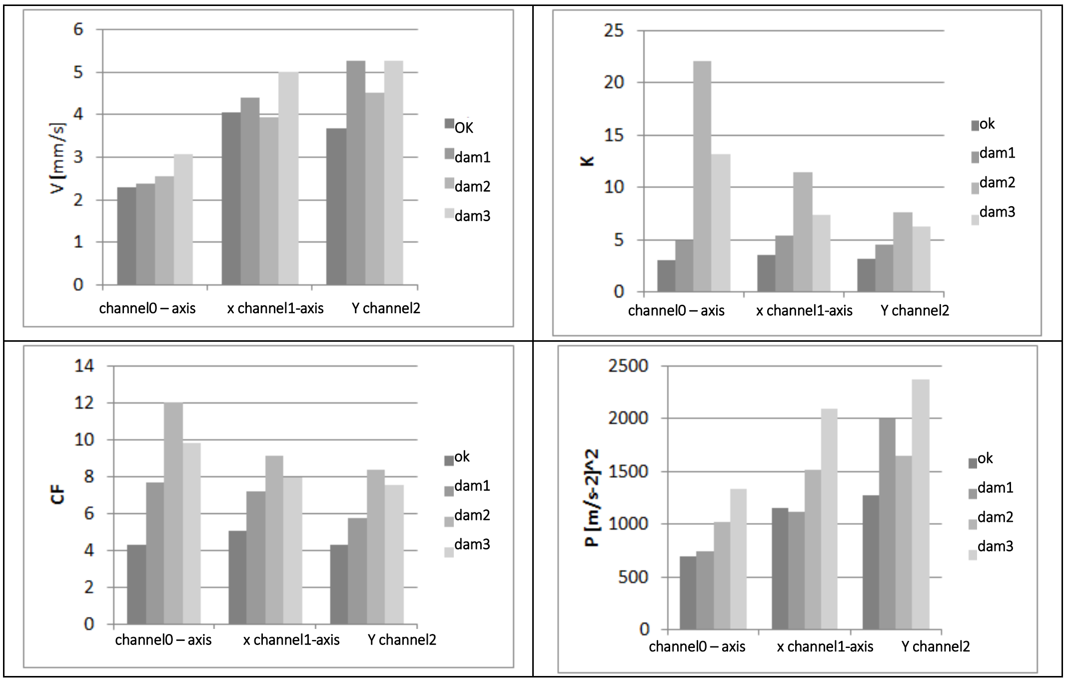


Fig. 6. Results of time domain analyses

In the frequency domain, amplitudes of individual harmonics, resulting from the kinematics of the tested transmission, and a spectrum of the envelope signal were determined (envelope narrowband analysis method with the use of Hilbert's transformation; see Fig. 7).

The increase in transmission damage causes an increase in the analysis parameters in the time domain. Narrowband envelope analysis allowed the determination of modulating frequencies, including, for the condition without damage (marked OK), frequency f_1=67 Hz and the number of its times, while, for damage conditions, bearing information about gearing z_3 with frequency f_3=19Hz and the number of its times. Frequency amplitude f_3 increased along with the damage. However, one inconvenience is the fact that it is a rotational frequency, which also carries information about imbalance. It was assumed that damage to therolling bearings on a lab bench and on an actual, technical object was detectable using this method.

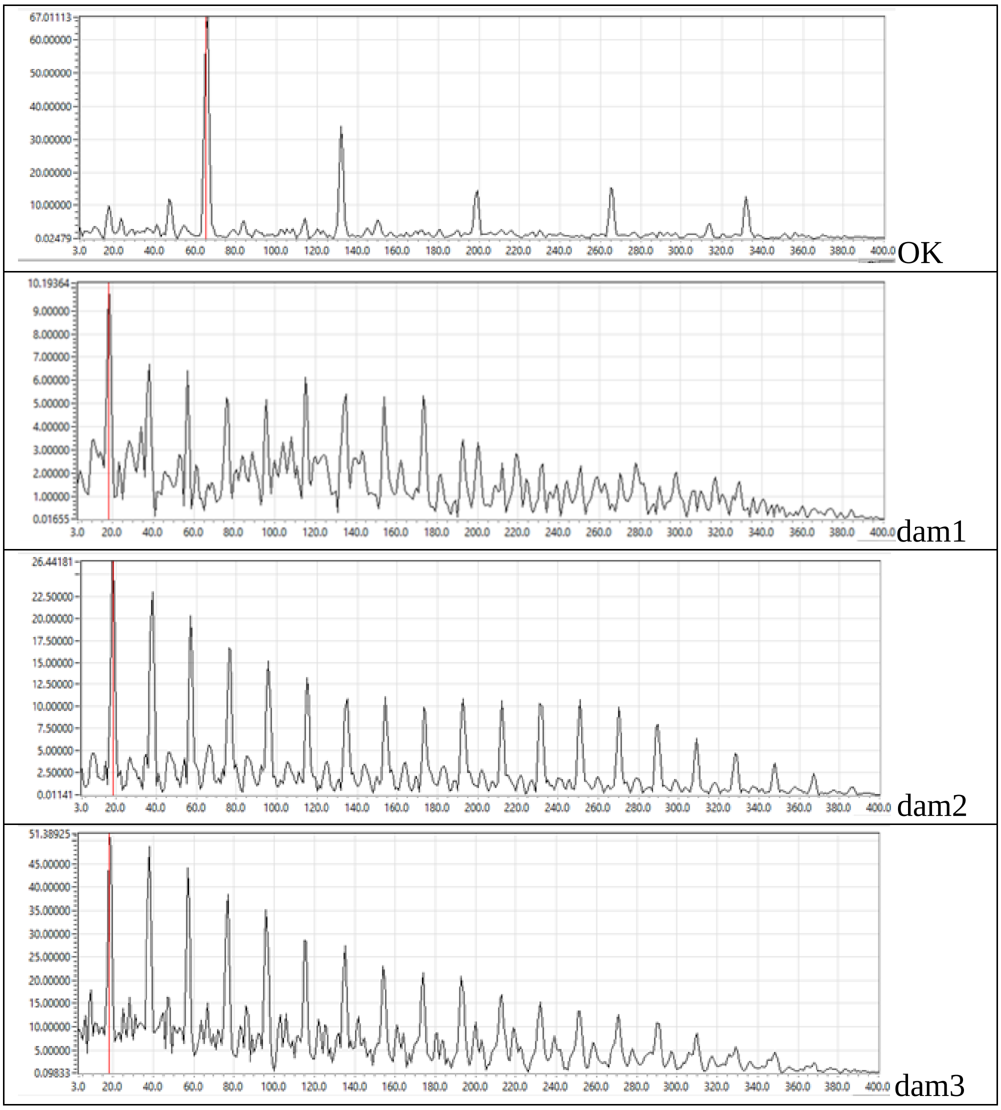


Fig. 7. Frequency domain analyses results: envelope spectrum

4. TESTS OF THE MAIN TRANSMISSION GEAR OF THE HELICOPTER

The measurements of the main transmission body vibrations were taken during the on-ground test of the helicopter, at set and temporary ranges, which changed the speed of the propulsion units to the maximum acceptable (hovering of the helicopter). A sample profile, determined on the basis of parameters recorded by the on-board recorder, including rotational speed values of the left (NSL) and right (NSP) engine turbochargers and the rotor (NR), is presented in Figure 8.

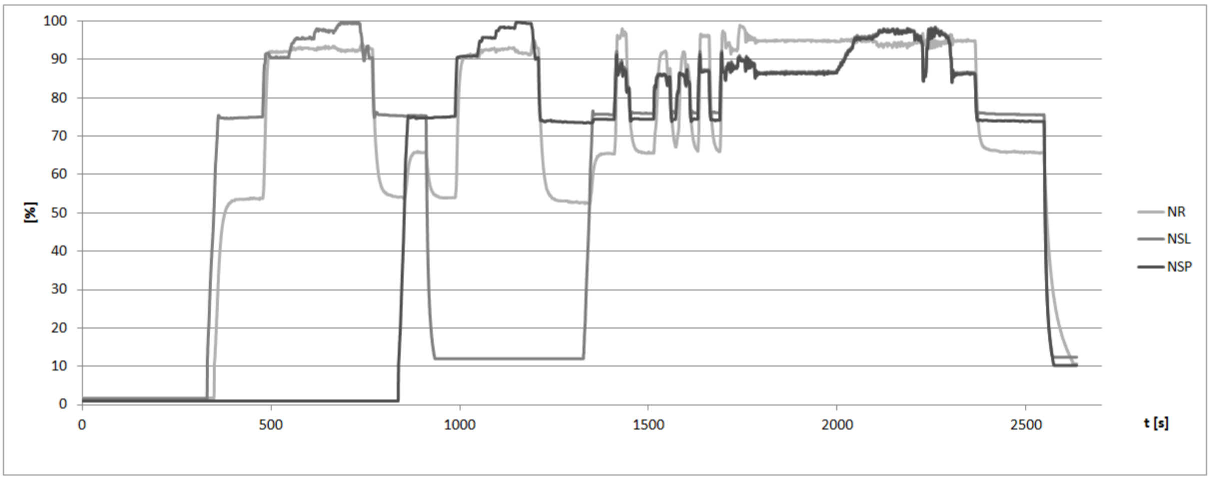


Fig. 8. Profile of the on-ground helicopter test

Due to structural reasons (non-magnetic body, no process openings, no planes with the possibility to use adhesive sensors) and safety reasons, there was a limitation in terms of the possibility to install sensors, which can measure temporary vibrations of the transmission's body. For this reason, accelerometers were fixed (Fig. 2) at measurement points: A (bonding method) - three remaining, perpendicular directions, marked WNX, WNY and WNZ; and B, C (with the permanent magnet) - vertical direction marked with LSZ and PSZ. Information about the current rotational speed was initially determined from the test sheets and data indexing, corrected with values from Figure 8 and determined by the vibration signal spectrum. As in the scope of laboratory tests, a number of analyses in the time and frequency domains were carried out. In the time domain, i.e., in the band 45÷1,590 Hz, 25÷10 kHz, 190÷340 Hz, 2 Hz÷10 kHz (see Fig. 9), the effective vibration speed V values and its maximum values were determined for the whole sample.

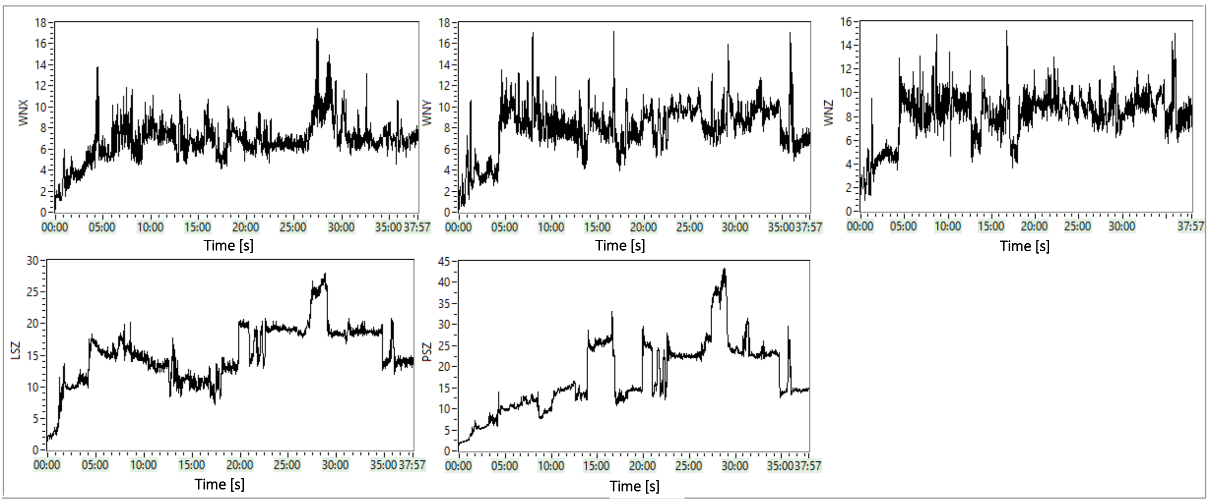


Fig. 9. Course of changes of the effective vibration speed in the band 2 Hz÷10 kHz

Peak vibration speed values at particular points for individual bands are presented in Figure 10. These values are related to the assumed limit values. Points B (vertical axis marked LSZ) and C (vertical axis marked PSZ) are defined by the producer for the propulsion unit (being 45 mm/s → warning value and 60 mm/s → limit value). Point A (directions WNX, WNY and WNZ) is defined by the standard [8]. Only in the band 2 Hz÷10 kHz, at point A, was the assumed limit value found to be exceeded.

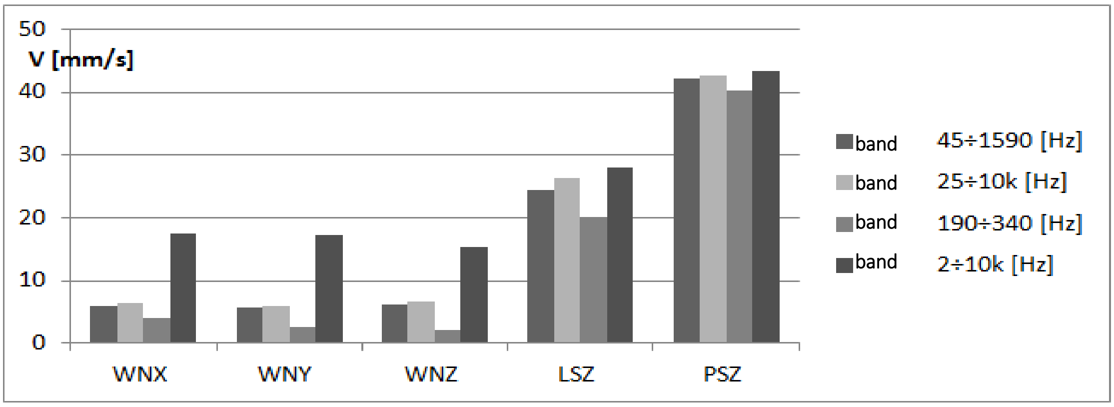


Fig. 10. Peak vibration speed values at particular points for individual bands

In the time domain were determined the kurtosis K value (Fig. 11), the CF peak factor and signal power P, within stable ranges of the operation of the left or right engine and two at the same time.

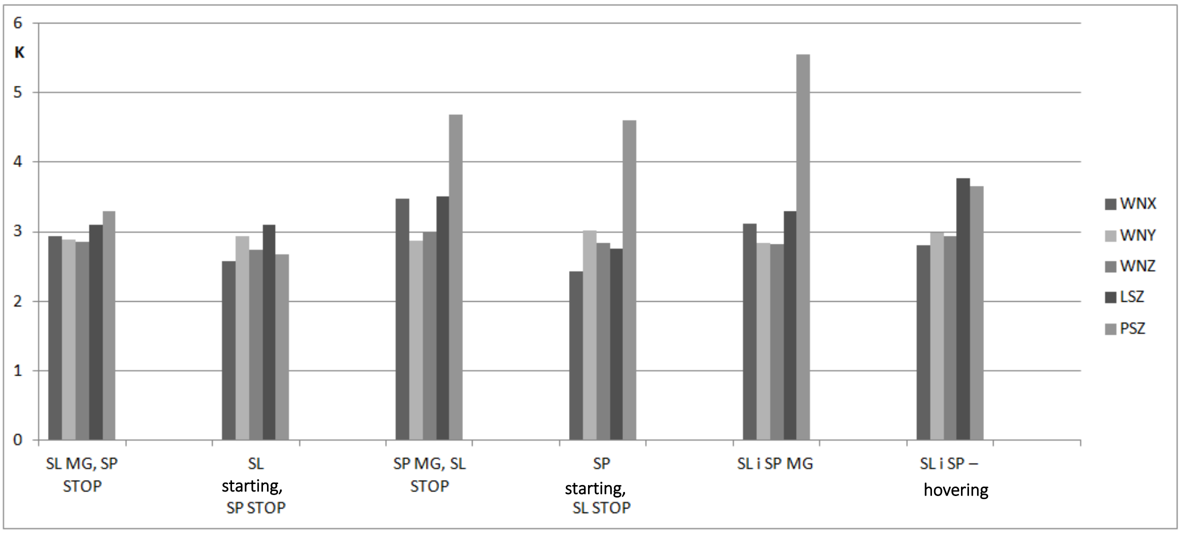


Fig. 11. Kurtosis value for individual directions in stable ranges of the rotational speed

The above analyses and a number of others from the scope of the time domain, due to the lack of explicit levels determining the acceptable limit values, were treated as informational and a reference for comparison in the case of extending the population of studied objects.

In order to detect damage to the bearings and transmission wheels, envelope spectrum analysis was used (narrowband envelope analysis method). Frequencies were found in this spectrum, which correspond to the kinematic calculations, part of which were not linked to the identified kinetics of the studied transmission gear (Fig. 12).

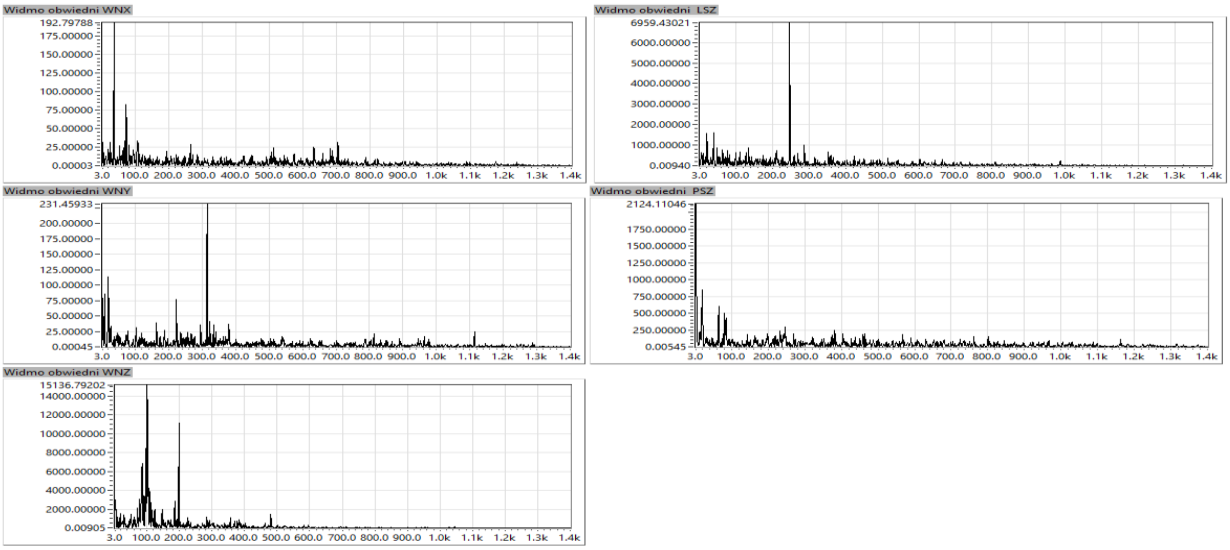


Fig.12. Envelope spectrum (left engine starting, right engine off)

5. CONCLUSION

The use of vibroacoustic testing to assess the technical condition of the transmission is an issue widely discussed in the literature. A number of methods to process a measured signal, with different efficiency, has been developed. There are also certain requirements, which need to be satisfied, e.g., correct parameters describing the tested kinematics, allowing for the determination of characteristic frequencies and the appropriate number of sensors located in the correct places. Due to operational limitations and a given technical object, it was impossible to use the phase marker, because, having the reference signal, it was possible to use other damage detection algorithms.

The results presented above regarding studies associated with the diagnostics of the main transmission of a helicopter might indicate a potential for monitoring its sub-assemblies with the vibroacoustic method. However, we need to expand the population of objects and verify the parameters describing the kinematics, thereby enabling characteristic frequencies to be determined.

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