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Vibration condition monitoring of planetary gearbox under varying external load

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ABSTRACT

The paper shows that for condition monitoring of planetary gearboxes it is important to identify the external varying load condition. In the paper, systematic consideration has been taken of the influence of many factors on the vibration signals generated by a system in which a planetary gearbox is included. These considerations give the basis for vibration signal interpretation, development of the means of condition monitoring, and for the scenario of the degradation of the planetary gearbox. Real measured vibration signals obtained in the industrial environment are processed. The signals are recorded during normal operation of the diagnosed objects, namely planetary gearboxes, which are a part of the driving system used in a bucket wheel excavator, used in lignite mines. It is found that a planetary gearbox in bad condition is more susceptible to load than a gearbox in good condition. The estimated load time traces obtained by a demodulation process of the vibration acceleration signal for a planetary gearbox in good and bad conditions are given. It has been found that the most important factor of the proper planetary gearbox condition is connected with perturbation of arm rotation, where an arm rotation gives rise to a specific vibration signal whose properties are depicted by a short-time Fourier transform (STFT) and Wigner-Ville distribution presented as a time–frequency map. The paper gives evidence that there are two dominant low-frequency causes that influence vibration signal modulation, i.e. the varying load, which comes from the nature of the bucket wheel digging process, and the arm/carrier rotation. These two causes determine the condition of the planetary gearboxes considered. Typical local faults such as cracking or breakage of a gear tooth, or local faults in rolling element bearings, have not been found in the cases considered. In real practice, local faults of planetary gearboxes have not occurred, but heavy destruction of planetary gearboxes have been noticed, which are caused by a prolonged run of a planetary gearbox at the condition of the arm run perturbation. It may be stated that the paper gives a new approach to the condition monitoring of planetary gearboxes. It has been shown that only a root cause analysis based on factors having an influence on the vibration solves the problem of planetary gearbox condition monitoring.

1. Object description and some considerations on design, production technology, operation and change of condition factors considerations

Fig. 1 shows a complex gearbox system for which the first stage is a planetary gearbox with a stationary rim. The planetary gearboxes considered consists of a sun gear $z_1 = 39$ teeth, planetary gear $z_2 = 27$ and rim gear $z_3 = 93$.

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Considering the system given in Fig. 1a and using notation f_{12} as the meshing frequency for a pair of gear wheels marked in Fig. 1 as z_1, z_2, z_3

$$f_{12} = f_{23} = \frac{n_1 z_1 z_3}{60(z_1 + z_3)} = \frac{950 \times 39 \times 93}{60(39 + 93)} = 435.067\,\mathrm{Hz}, \tag{1}$$

where n_1 is the input rotation velocity RPM.

The arm frequency is

$$f_a = \frac{n_1 z_1}{60(z_1 + z_3)} = \frac{950 \times 39}{60(39 + 93)} = 4.67\,\mathrm{Hz}. \tag{2}$$

A planet passing frequency is $f_p = sf_a = 4 \times 4.67 = 18.68\,\mathrm{Hz}$, where s is the number of planets.

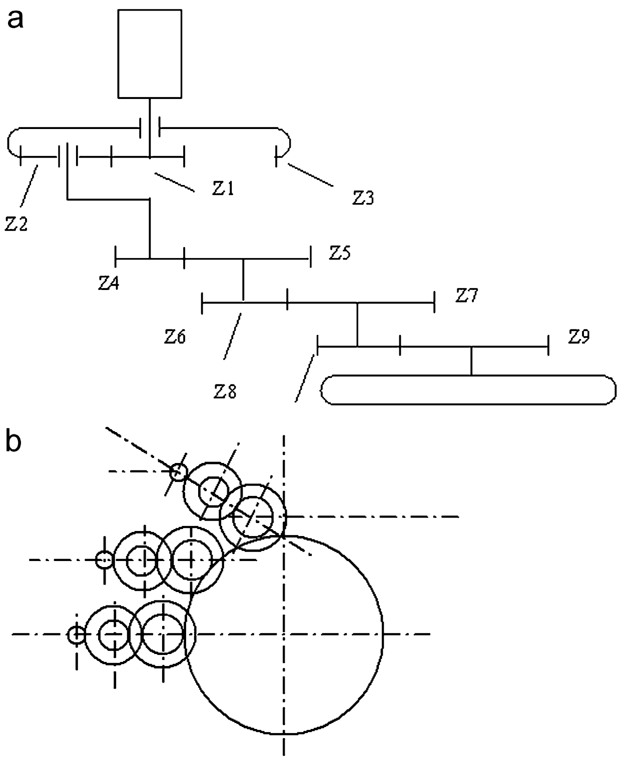

Fig. 1. (a) Part of driving system for a bucket wheel with planetary gearbox (gears: z_1-sun, z_2-planet, z_3-stationary rim, z_4–z_9 three-stage cylindrical gearbox) and (b) three independent drives of bucket wheel.

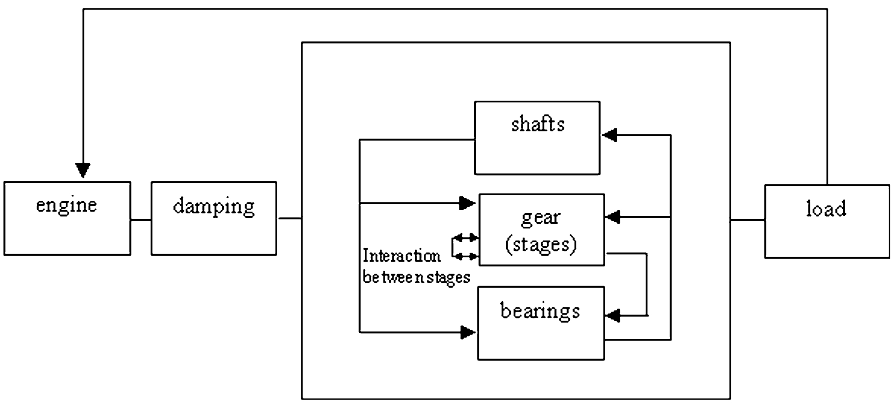

Fig. 2. Interaction between components in a multistage gearbox system and external components to the gearbox such as electric motor/engine, damping coupling and external load.

The spectrum of vibration presented by frequencies (1) and their harmonics together with inter-stage modulation components [1] and modulation caused by the arm rotation and passing planet/satellite frequency against the point of sensor attachment give the gearbox vibration spectrum. If distributed faults whose origin may be gear transmission errors/imperfections [1–4] in the vibration spectrum are further taken into consideration, one can see other components. These gear transmission errors depend on design factors [1–4] and can be identified as sideband components in a gearbox vibration spectrum. They occur as a result of amplitude modulation [5,6]. The same effect is produced by distributed faults caused by gear pitting or scuffing [6]. Besides the distributed faults, local faults may occur that originate from a tooth root crack, tooth breakage [7,8], or a spall on a tooth flank. The local faults [11–13] give similar effect in the gear spectrum as distributed faults. There is a possibility of identifying these two types of faults following the procedures given in [8]. The planetary gearbox structure influences the amplitudes of vibration component , which is considered in [9] where summation of the problem is given and where natural frequencies and modal properties of compound planetary gears are considered.

As is pointed out in [10], one should take into consideration the interaction between gearbox components as given in Fig. 2. Fig. 2 shows a developed scheme as compared to the scheme given in [10].

The authors' experience shows that interaction between the internal components of a multistage gearbox system and the external components in the system [1–4,7] (electric motor-coupling—multistage gearbox—varying load) is crucial to the vibration diagnostic evaluation of the multistage gearbox condition. So the classical method of condition monitoring given in [7,8] is not always suitable. The influence of varying load should be stressed for which some aspects are also considered in [14–19]. Considering the external load variation for bucket wheels excavators and keeping in mind the variation in digging ground properties and bucket wheel features generated vibration will be cyclostationary. The described

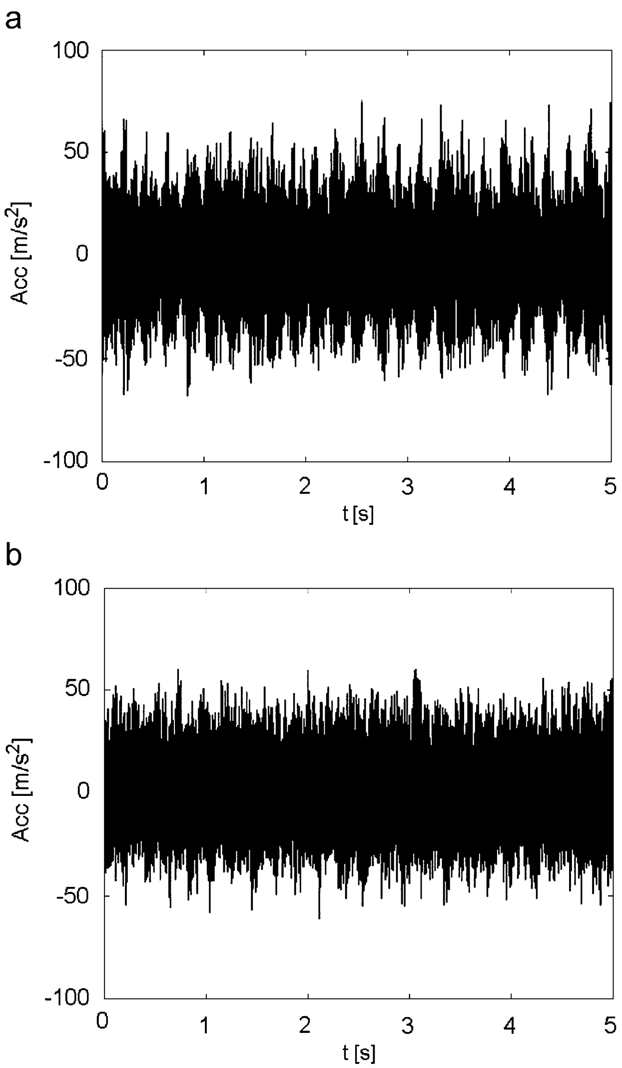

Fig. 3. Acceleration [m/s^2] signal time traces [s]: (a) signal from gearbox before replacement and (b) signal from replaced gearbox.

whole driving system of a bucket wheel excavator consists of three subsystems as shown in Fig. 1b, and such a whole system is used for driving a bucket wheel. The bucket wheel diameter is 17.5 m and has 11 evenly distributed buckets of capacity 4.6 m^3. The rotation speed of the bucket wheel is about 3 RPM, which gives the rotation cycle of 20 s. The gearbox system is monitored by a commercial diagnostic system, which makes it possible to estimate the properties of the vibration signal. The acceleration vibration signal from one sensor attached radial on the stiff flange on each planetary gearbox is received. The commercial vibration monitoring system can estimate vibration parameters such as Peak, RMS of vibration signal, vibration spectrum, envelope (time and spectrum) analysis, tools for spectrum component identification and sideband component identification, trend analysis—long- and short-term trend, shock pulse method—for bearings.

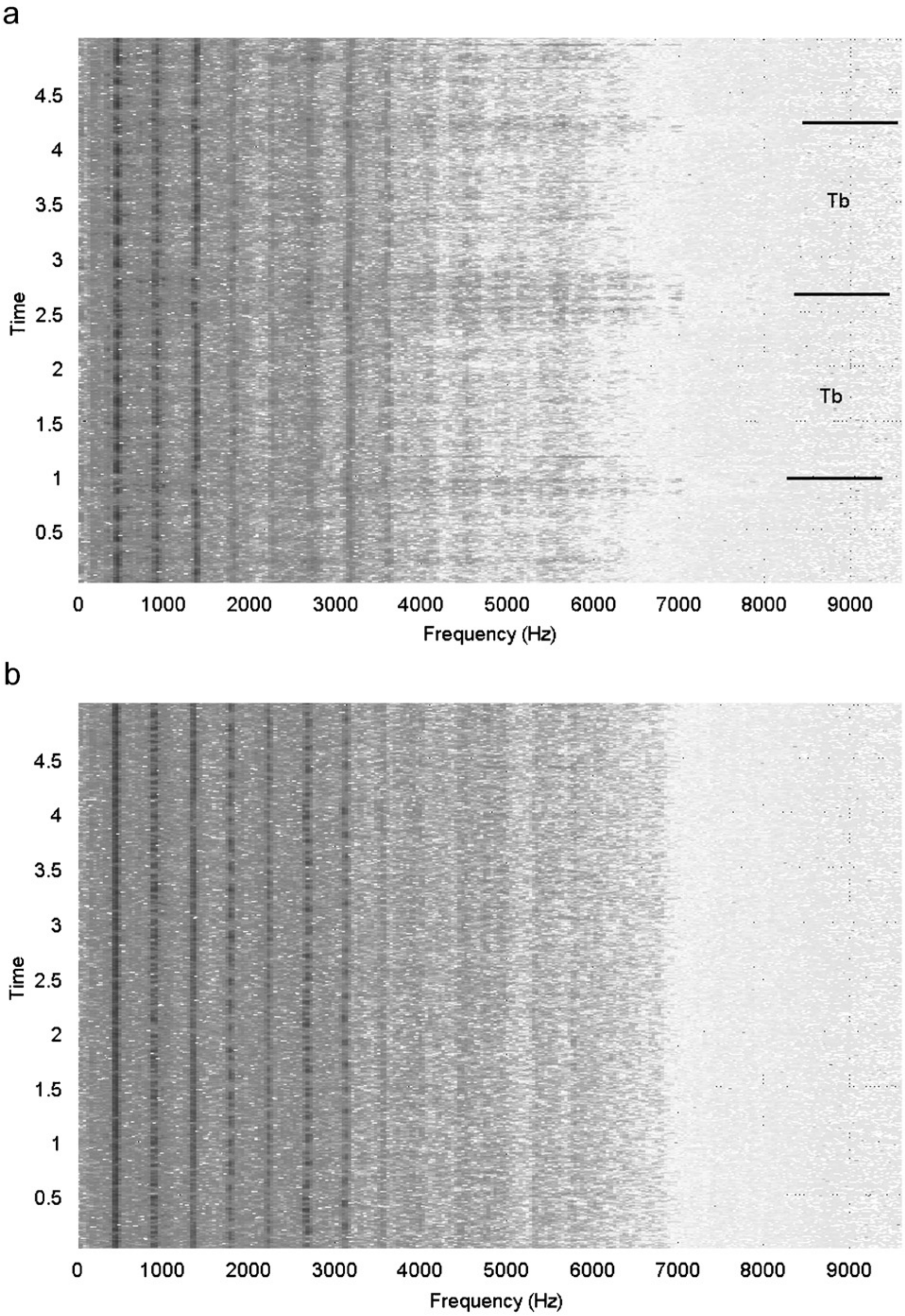

Fig. 4. Time [s]–frequency [Hz] spectrograms: (a) signal from a gearbox before replacement and (b) signal from the replaced gearbox.

The system supplier has adjusted some fixed alarm limit values on which warning signals are activated. In the opinion of the user, the commercial vibration monitoring system gives false alarms. Now the system user in maintenance practice uses subjective planetary gearbox assessment, and the replacement criterion is a fluctuating heavy noise generated by the planetary gearboxes. To examine the condition-monitoring problem of planetary gearboxes, this paper presents a reliable method for objective condition assessment based on vibration signal analysis.

2. Condition monitoring method

The problem of condition monitoring of planetary gearboxes has been reviewed in many publications [20,11,12], . where different diagnostic techniques are discussed and focused on a fatal fault for gearboxes, such as cracking of a tooth. In this paper, different problems are discussed. As was stated, it is important to identify external load variation from the vibration signal received from the gearbox housing. The symptoms of external load variation should be identified from the acceleration signal. The obtained result of external load variation obtained from the acceleration signal is not directly the time trace of the load, but a suitable trace of acceleration, which is proportional to the external load variation. The procedure of load identification is related to the procedures of filtration, enveloping and envelope frequency analysis [21]. The first step in the identification of external load variation is filtration of an original vibration signal with a band-pass filter whose central frequency is the first harmonic of the planetary gearbox mesh frequency (1). For identification of the external load variation, two planetary gearboxes are taken, one before the decision was taken to make a repair and the other when mounted in the system in place of a removed gearbox. Thus we can compare two gearboxes in quite different conditions. The results of earlier measurements suggested that gearboxes in different conditions have

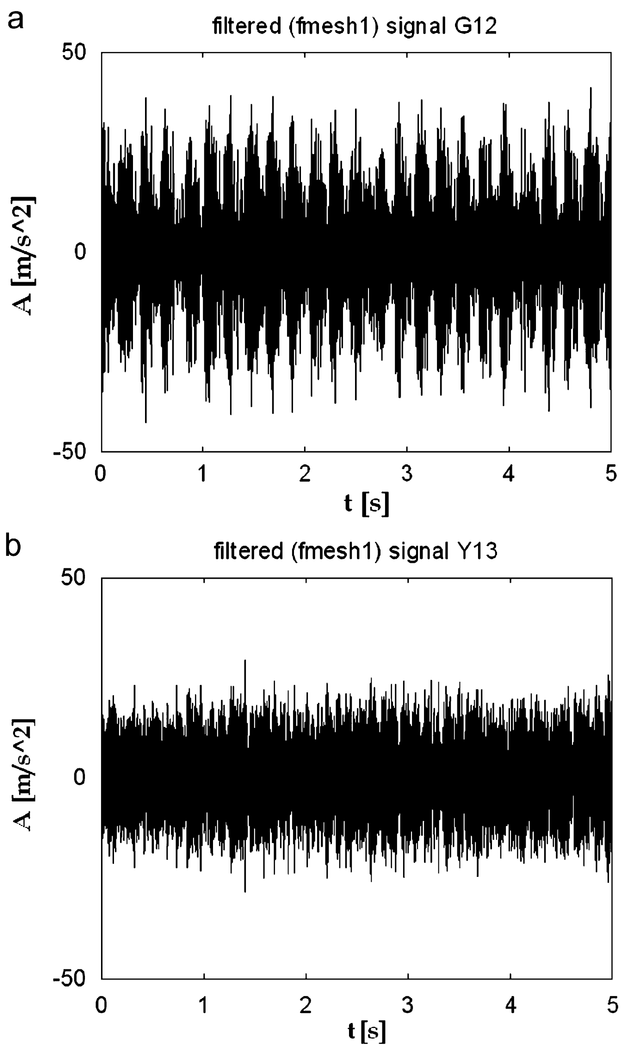

Fig. 5. Time traces [s] for filtered acceleration signal around first meshing harmonic: (a) signal from the gearbox before replacement and (b) signal from the replaced gearbox.

different susceptibilities to external load variation. To investigate the load susceptibility, external load variation was identified.

Figs. 3 and 4 show signal time traces and time–frequency spectrograms for two gearboxes in different conditions. Initially, one can see differences in the signals. Following the procedure of identification, the signals were filtered around the first mesh harmonic and the results are given in the form of time traces in Fig. 5, where in figure (a) a strong amplitude modulation is seen. The next step of the signal analysis is to find the signal envelopes, which are given in Fig. 6.

Fig. 7 shows the spectrum components connected with arm rotation, that is $f_a = 4.67\,\mathrm{Hz}$ (2).

Load variation is related to a bucket period, i.e. 1.8 s, which gives a bucket frequency of 0.55 Hz. Two causes influence the shape of the envelope, namely load variation and improper arm condition, with frequency $f_a = 4.67\,\mathrm{Hz}$. The next step of load identification is separation by filtration of these two sources of vibration. Fig. 7 shows separated signals from the load variation (solid line with stars) and from the arm (solid line).

Fig. 8 shows arm signal envelopes after load signal separation, Fig. 9 envelopes proportional to load variation. Figs. 3–9 show a detailed analysis for two planetary gearboxes with two different conditions. Conclusion can be drawn that there is evidence of a greater susceptibility to load variation when the gearbox is in bad condition. In the planetary gearbox marked G1 after replacement, some susceptibility (Fig. 9b) to the load variation caused by design factors, such as its tooth flexibility, is also seen. This should especially be so for spur gears such as are used in the planetary gearbox considered. For a better identification of arm perturbation, which is seen in Fig. 4, further identification on a perturbation period using Wigner-Ville distribution is given in Fig. 10.

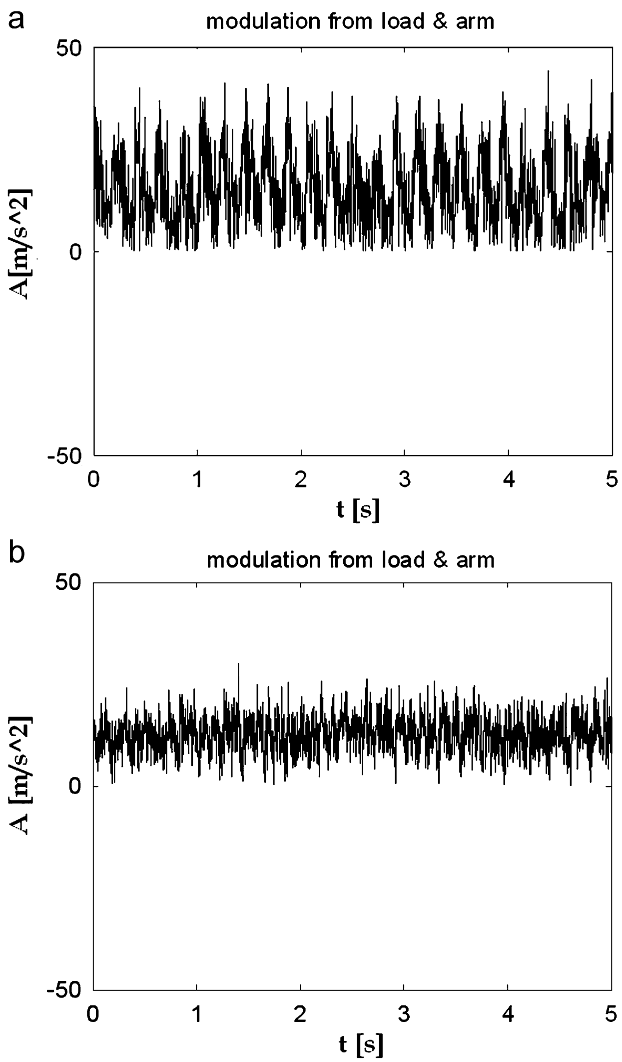

Fig. 6. Signal envelopes from time traces given in Fig. 5(a) signal from the gearbox before replacement and (b) signal from the replaced gearbox.

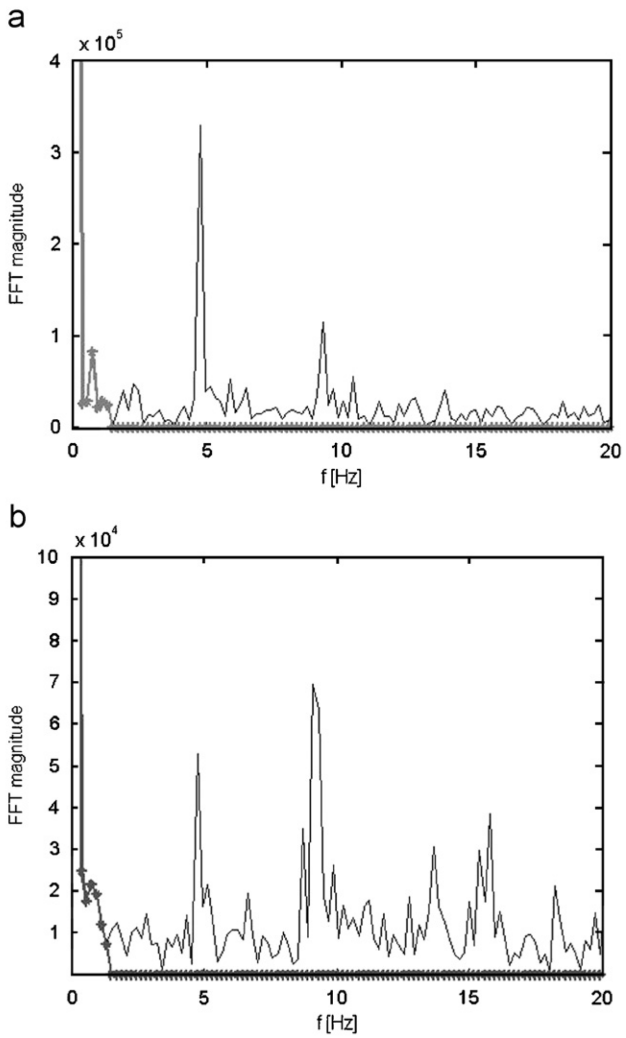

Fig. 7. Spectrum source separation from load (solid line with stars) and arm (solid line): (a) signal from the gearbox before replacement and (b) signal from the replaced gearbox.

2.1. Diagnostics parameters

Besides the above considerations, signal analysis has been made for another two gearboxes that run in the complete bucket drive system Fig. 1. For condition assessment, measures of load susceptibility, as shown in Fig. 9, are chosen to be the mean (L_{mean}) and standard deviation (L_{std}) of the load variation estimated from the acceleration vibration signals. This means that the load is proportional to the estimated values of L_{mean}, and L_{std} of the vibration time course given in Fig. 9 represents its variation. Fig. 11 shows the values of relative changes of gearbox condition parameters.

Besides the above-mentioned load susceptibility measures, which are also measures of planetary gearbox condition, further two condition measures are taken into consideration, these being the RMS values of the signals filtered around the first mesh harmonic $f_{12} \pm 0.5 f_{12}$ (1) and first and second harmonic amplitudes of the arm frequency (2). To evaluate the condition change, relative values of the condition measures are considered. The considered relative values are given in Fig. 11. In the figure, the three planetary gearboxes are marked as G1, G2 and G3 and compose a part of the bucket wheel drive system, Fig. 1. The signals were received from four planetary gearboxes some time before replacement (time t_1) of gearbox G1 and some time after replacement, time marked as t_2. The decision to replace gearbox G1 was made on subjective grounds, not on the basis of measurements from the commercial monitoring system installed on the drive system, because the monitoring system did not fulfil its task. As one can see for the planetary gearbox G1, the RMS value dropped about 14% (Fig. 11a) when comparing the measurements for two times t_1 and t_2 (measurements for two gearboxes marked as G1, one before replacement and one after replacement), and for the PG marked G2 and G3 there were increases of about 9% and 8% of RMS values, respectively (measurements at times t_1 and t_2 for two gearboxes G2 and G3). Similar tendencies are seen for a decrease of the mean values of load parameter for G1, and increases for this parameter for G2 and

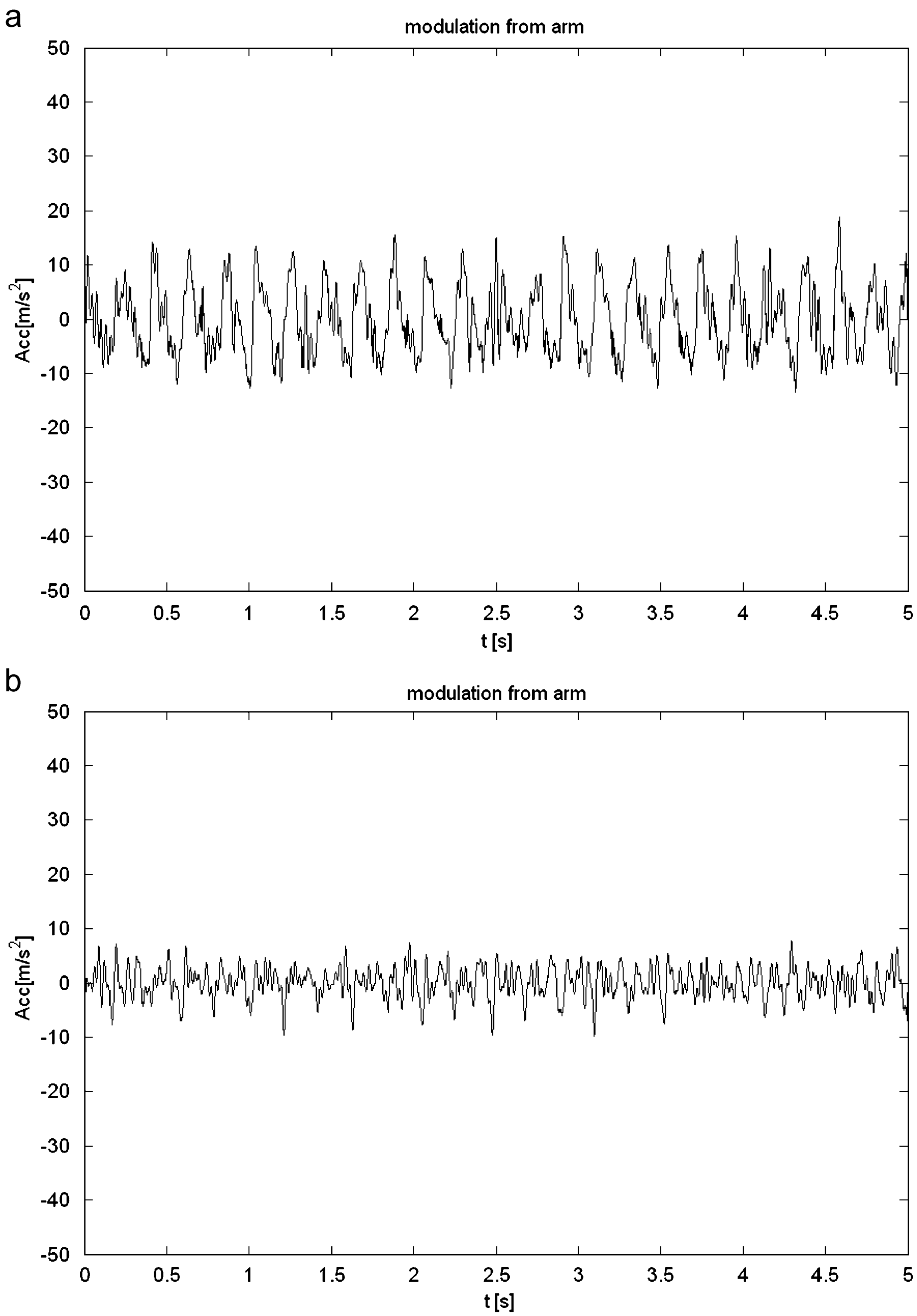

Fig. 8. Arm signal envelopes after load signal separation: (a) signal from the gearbox before replacement and (b) signal from the replaced gearbox.

G3, Fig. 11b. The mean value L_{mean} of the load parameter was estimated from the vibration signals, so the estimated values are proportional to the values of load. The most important indication of load susceptibility is the standard deviation of the load variation where (Fig. 11c) one can see a 50% drop when comparing measurements for t_1 and t_2 for PG marked G1. Important measures of the PG condition are the amplitude value that is connected with arm rotation, as given in Fig. 11d. On the basis of the presented considerations, one can come to the conclusion that for condition evaluation the standard deviation L_{std} of the load variability should be considered. (cf. Fig. 9a with b). Besides this, the changes of the harmonic amplitudes of the arm rotation are also important to consider, Fig. 11d.

It should also be noted that regarding condition, the time–frequency spectrogram and Wigner-Ville distribution as given in Fig. 4a and 10 are useful, where one can see horizontal lines with the period $T_a/2 = 0.105$ s, where T_a is equivalent to the arm rotation period and the same vertical line in Fig. 10a and b. The lines with the period of T_a, $T_a/2$ show perturbation in

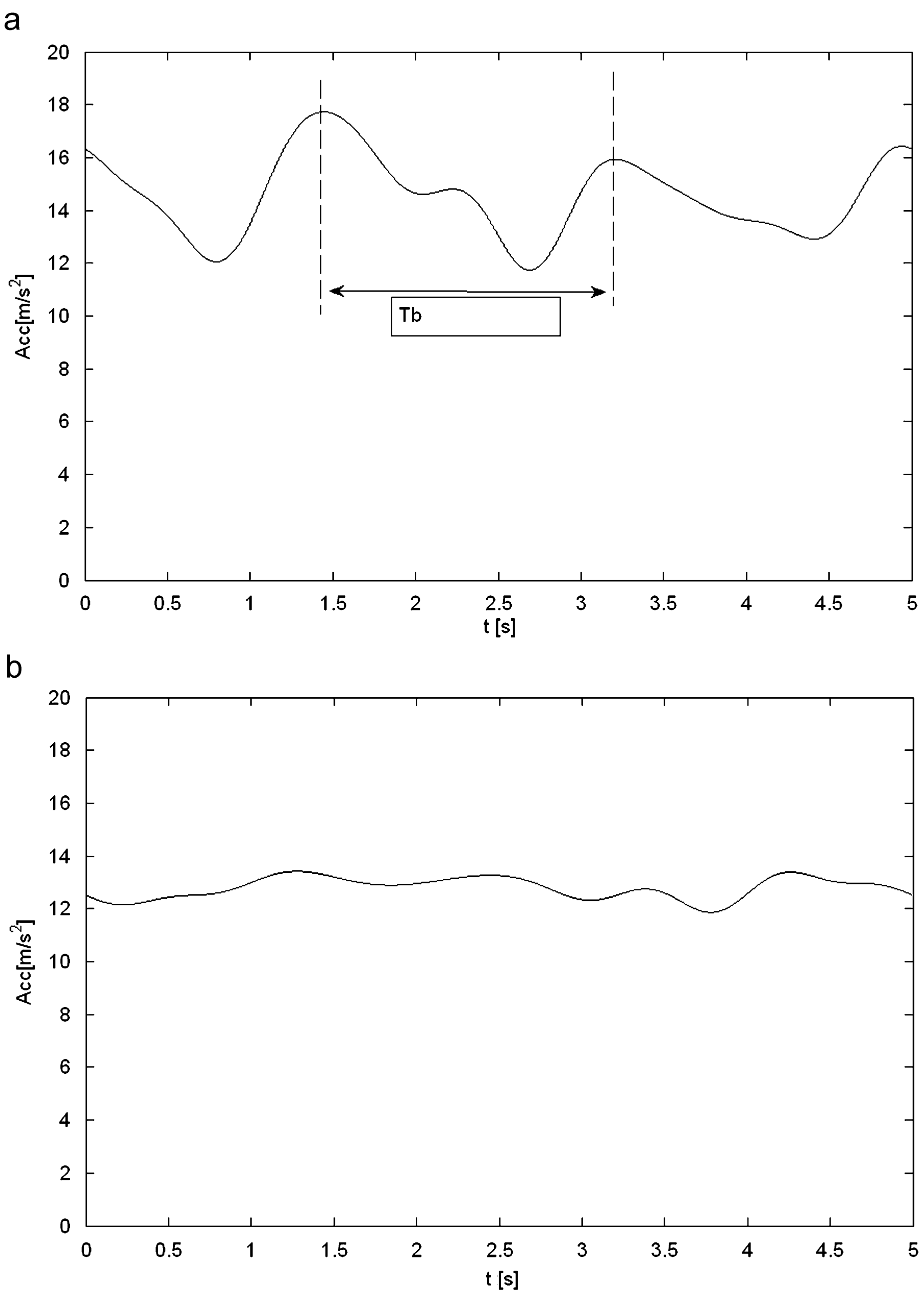

Fig. 9. Envelopes proportional to load variation: (a) signal from the gearbox before replacement and (b) signal from the replaced gearbox.

arm rotation, which gives vibration impacts. There is another period (Fig. 4a) connected with the bucked period 1.8 s, which separates groups of horizontal lines with the period $T_a/2$, which corresponds to the second harmonic of the arm rotation frequency (2). The occurrence of the perturbation of arm rotation can be interpreted as the influence of the increased backlash in rolling element bearings caused by the severe environment (dustiness) in open-cast mines.

3. Conclusions

The problems of planetary gearbox condition evaluation described in the paper represent quite new issues not considered before in the literature. The results of varying load evaluation show that the load trace is quite different from those considered in the literature [14–19]. It can be stated that identification of the varying external load is crucial to the evaluation of the planetary gearbox condition. It is expected that a proper consideration of factors influencing the vibration

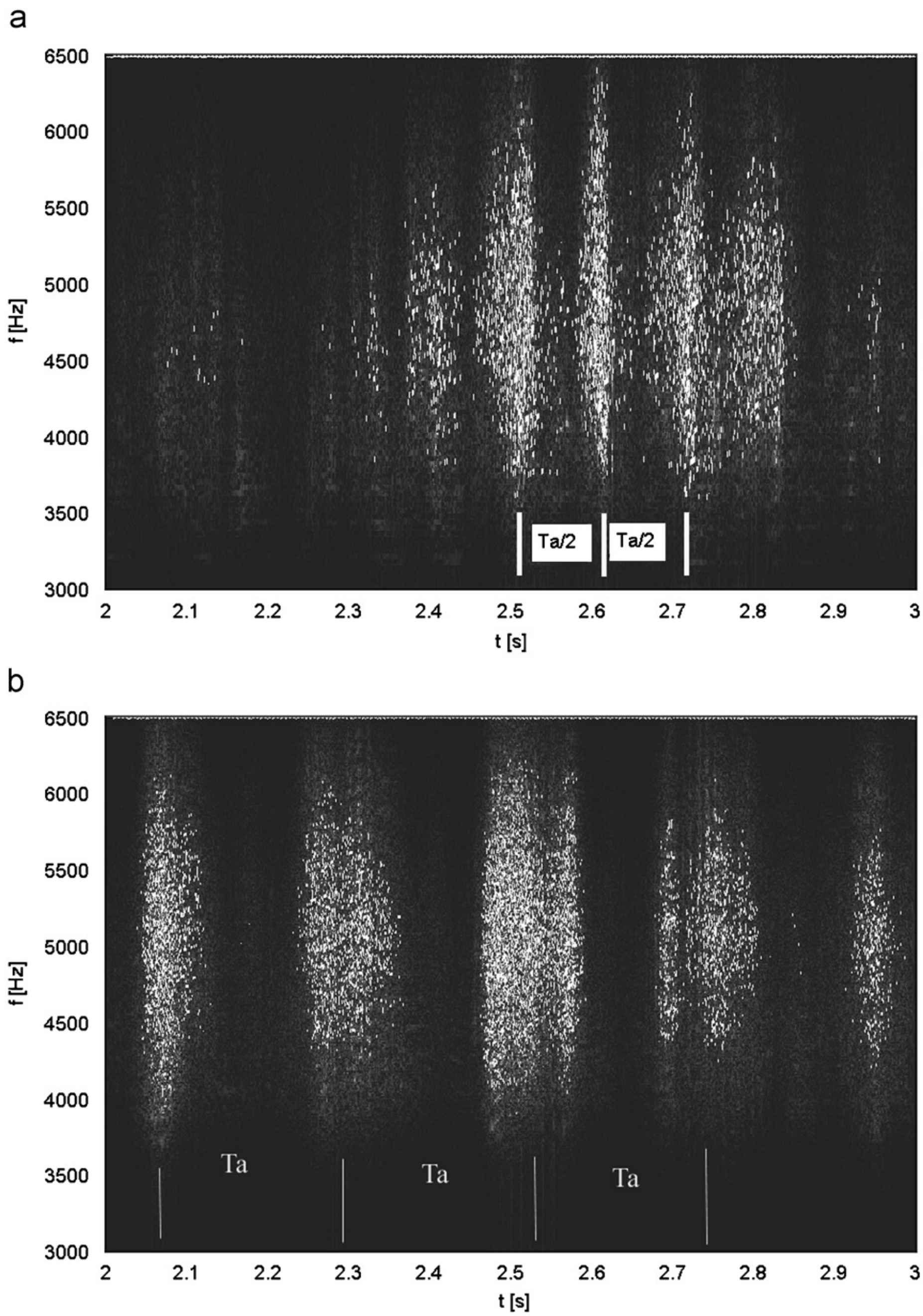

Fig. 10. Evolution of Wigner-Ville distribution with change in condition of the arm of the planetary gearbox: (a) occurrence of distinct perturbations at second harmonic of arm rotation frequency (PG-G1) condition before replacement and (b) occurrence of distinct perturbations at arm rotation frequency (PG-G3).

diagnostic signal, together with the proper signal analysis methods, will lead to the proper choice of condition criteria for planetary gearboxes used in bucket wheel drive systems. It is expected that such design factors, as the flexibility of spur teeth used in the considered planetary gearbox, should cause amplitude modulation under the fluctuating load. It has been found that for a planetary gearbox in good condition, a varying load modulates a vibration signal measured on the gearbox housing. A time trace of the varying load obtained by a demodulation process is given in Fig. 9b. But it is also stated that a planetary gearbox in bad condition is more load-susceptible than a gearbox in good condition. The time trace obtained by a demodulation process for a gearbox in bad condition is given in Fig. 9a. It can be further stated that the most important factor in proper planetary gearbox condition evaluation is related to perturbation of the arm rotation, which cause rising a

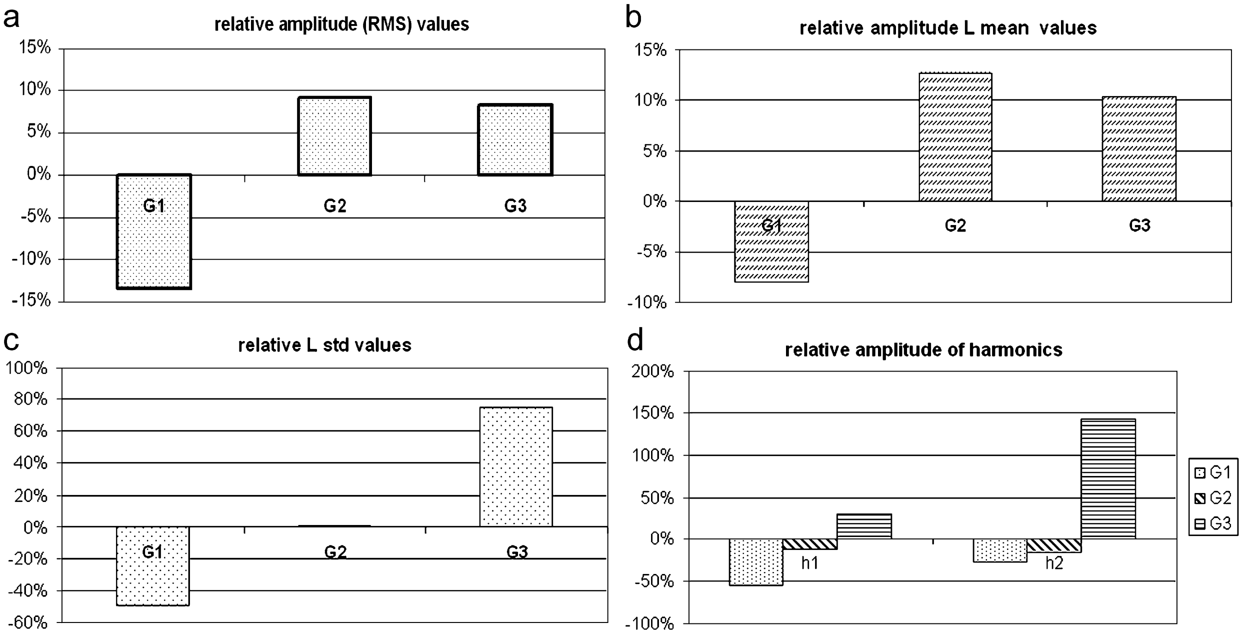

Fig. 11. Relative values of parameters representing condition change for gearboxes marked G1, G2, G3, for G1 are considered two gearboxes before replacement and after replacement: (a) change of RMS for filtered signals, values being compared for times t_1 (before replacement of G1) and t_2 (after replacement of G1), (b) change of relative parameter values of mean (L_{mean}) of the load variation, estimated on vibration for times t_1 and t_2, (c) change of relative values of standard deviation (L_{std}) of the load variation estimated on vibration and (d) signal amplitude change of first (h1) and second (h2) harmonic of arm frequency rotation for times t_1 and t_2, relative values for gearboxes G2 and G3 are also given for times t_1 and t_2.

specific vibration signal given in Figs. 4a and 10, depicted by STFT and Wigner-Ville distribution presented as time–frequency maps. The specific vibration signal as shown in Fig. 4a is characterised by two forms of perturbations: the first related to the arm rotation period and the second related to the bucket rotation period. The first kind of perturbation shows horizontal lines in STFT with the time period $T_a/2$ or T_a of arm rotation. The second kind of perturbation shows perturbations with the period equivalent to the bucked digging period T_b, which can also be noticed in STFT, Fig. 4a. These perturbations are not seen in Fig. 4b. The paper gives evidence that there are two dominant low-frequency causes that influence the vibration signal modulation, namely the varying load, which comes from the bucket wheel digging process, and the arm rotation. For an arm rotation in good condition, one may expect sinusoidal modulation of the vibration signal at the arm rotation frequency (2). For an arm rotation in bad condition, one may expect a periodic perturbation with the period of arm rotation or its second harmonic. The expected scenario of changes in conditions suggested in the literature, such as faults in rolling element bearings, or a tooth cracking or breakage, has not occurred in the cases considered. The suggested diagnostic method for planetary gearbox condition evaluation is limited to values given in Fig. 11, including demodulation techniques, and by separation of the main causes of change in the planetary gearbox condition given by the load susceptibility, and the arm perturbed rotation depicted in time–frequency spectrograms. It may be concluded that an objective method for determining planetary gearbox condition has been presented in Section 2, which may be used instead of the subjective method currently used because the commercial diagnostic system did not fulfil its tasks. The objective diagnostic method is quite different from the method used for condition assessment by the commercial system. It should also be stated that the subjective method of condition assessment was carried out to avoid complete destruction of a planetary gearbox. Using the commercial diagnostic system, this aim of avoiding planetary gearbox destruction has not been achieved, but many fault warnings have been noticed. The proposed diagnostic procedure and a means of condition assessment should avoid such confusing situations. It is expected that the occurrence of the above-mentioned local faults would be recognized using the signal filtration and time–frequency spectrograms. In actual practice, local faults of the planetary gearboxes have not occurred, but cases of severe destruction of planetary gearboxes have been noticed, which are caused by prolonged running of the gearbox under conditions of the arm perturbed rotation.

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