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METHODS FOR DETECTING FAULTS AND DAMAGES IN GEAR UNITS

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Abstract

There are many typical damages and faults related to gear unit operation. A crack in the tooth root, which often results in failure of gear unit operation, is the least desirable among them. Methods for detecting faults and damages presented in this article are based on gear units with real damages or faults, which have been formed by means of numerical simulations of real operating conditions; a laboratory test plant has been used. A possible fault or damage can be detected by monitoring vibrations. A fatigue crack in the tooth root is related to significant changes in tooth stiffness; in concern to other faults, changes of other dynamic parameters are more expressed. Amplitudes of time signal vibration are, by frequency analysis, presented primarily as a function of frequencies in a spectrum using procedure for determining the level of non-stationarity of operating conditions. Signal analysis has been performed in relation to a non-stationary signal, by means of Time Frequency Analysis.

INTRODUCTION

Keeping a technical system (gear-unit) in the most suitable working condition is the aim of maintenance; its purpose is to discover, to diagnose, to foresee, to prevent and to eliminate damages. The purpose of modern maintenance, however, is not only to eliminate failures but also to define the stage of a potential danger of a sudden failure of system operation. Diagnostics is interested in the definition of the current condition of the system and the location, shape and reason of the damage formation. The following diagnostic values are used to define incorrect operation, the possibility and the location of damages and the possibility of elimination of those damages: different signals, condition parameters and other indirect signs. The form of damage is identified on the basis of deviations from the undamaged gear system.

CRACK SIMULATION AND FORMATION

The excitation of a gear directly in the meshing area is caused by the following internal sources: the impact at the beginning of meshing, tooth stiffness, resulting in parametrical excitation, geometrical deviations of teeth and deformation of bearings and shafts [1]. A fatigue crack present in the tooth root causes significant changes in tooth stiffness; the dynamic response is different from the one caused by an undamaged tooth. A machine for dynamic tests of mechanical elements was used to form cracks in the tooth roots of pinions of two different gears. The depths of the two cracks were 4.5 mm (Figure 1) and 1.1 mm.

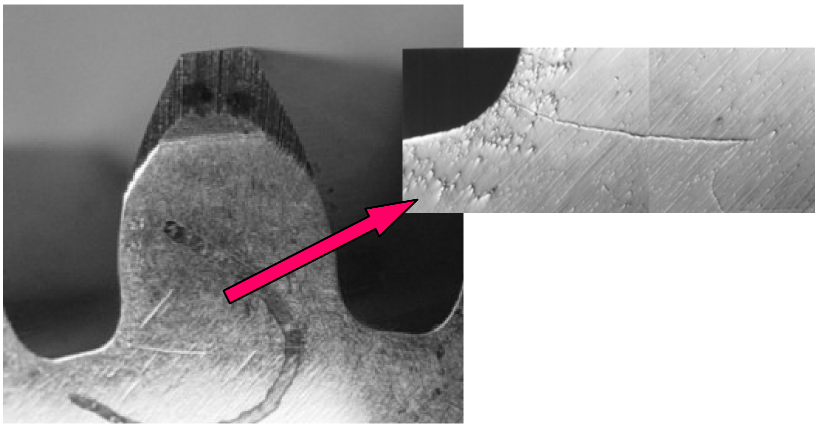


Figure 1: Gear with real fatigue crack in the tooth root

The parameters required for the dynamic test (suitable loading and the number of loading cycles required for the expected crack length) were determined by means of a finite element method (FEM). The purpose of FEM analysis was related to practical production of a real fatigue crack. The model proposed by Glodež, Šraml and Kramberger [2] was used. This model makes it possible to monitor the life of a gear unit regarding the crack size. The results obtained through the FEM analysis make it possible to define the number of cycles necessary for the relevant crack length.
In relation to the fatigue crack initiation the strain-life method in the framework of the MSC/FATIGUE FEM program code was used with the purpose to define the number of stress cycles required for the initiation. Based on gear data and material properties [3] the finite element model was constructed for further numerical calculation of the stress-strain field in a gear tooth root for plain strain conditions. The gear tooth was loaded with normal pulsating (meshing) force which was acting at the outer point of the single tooth contact. The computational analysis was performed at the point where maximum principal stresses occur in a gear tooth root.
The tooth loading equalled the loading obtained by the numerical analysis of the fatigue crack initiation. In Figure 2 the numerically determined crack propagation path in a gear tooth root is presented.

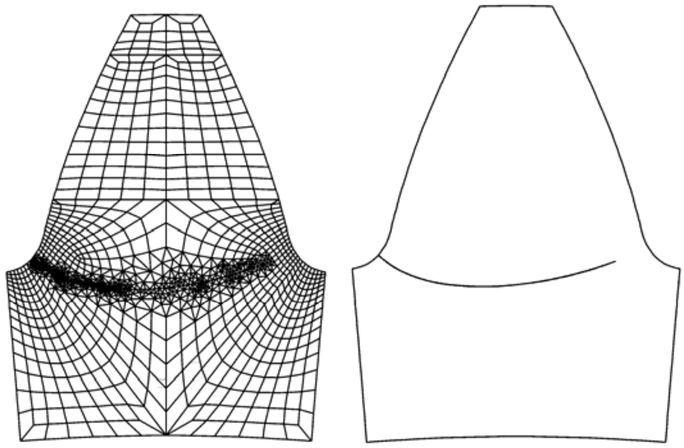
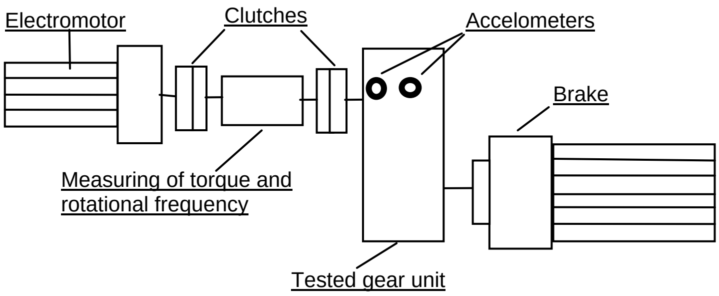


Figure 2: Predicted crack propagation path in a gear tooth root

Based on computational results related to crack initiation and crack propagation period it is possible to acquire the complete service life of gear tooth root.
When comparing Figures 1 and 2, it becomes clear that the direction of crack propagation in the test gear unit is almost identical with the one in the numerical simulation. On the basis of numerical determination of the number of loading cycles required to achieve a suitable crack length, the time required to obtain the relevant results was established successfully.
At the root of one of the teeth of the pinion a real fatigue crack was formed (using an appropriate device for dynamic tests for machine elements) and then the gear was mounted on the test plant. A fatigue load was, therefore, similar to the one that occurs when a tooth is loaded during meshing. For achieving a crack length of 4.5 mm, the total number of loading cycles required was $N_c = 3.2 \cdot 10^7$, and for a crack of 1.1 mm the number of loading cycles was $N_c = 2.9 \cdot 10^7$ at the specific loading of $F = 800$ N/mm.

EXPERIMENTAL PLANTS

Measurements of vibrations of two pairs of spur gear-units were performed. One of the pairs had a fatigue crack and the other one was without it. Tests performed under constant loads and vibrations were measured directly with accelometers, which were fixed on the housings. The tests were carried out in the power test plant. Each gear unit contained a carburised spur gear pair of module 4 mm, the pinion 19 and the wheel respectively having 34 teeth. Further data about the gears is available in [3]. The presented results are related to a nominal pinion torque of 20 Nm and nominal pinion speed of 1200 rpm (20 Hz). In industrial applications, this is a very typical load condition for this type of gear units. Test plant scheme is presented in Figure 3.



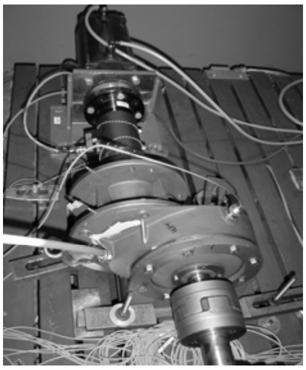


Figure 3: Test plant scheme

Rotational frequency and torque on the input shaft were measured with the Mohilo Steiger measuring system; temperature sensors fixed on the gear unit were used for measuring the temperature of more important parts of the gear. These parameters enabled the control of the load of a gear unit. At the pinion bearing housing (in the middle of the flange), the acceleration vibration signals were measured in the radial direction (20°) and sampled with a sampling rate equal to 12.8 kHz. Two vibration signals were simultaneously measured from the housing of the gear unit. For this purpose two piezoelectric accelerometers, with the power supply of ICP, were used. Borings with the thread had been made so that two accelerometers could be mounted on the housing of the gear-unit. In addition to that, a fine revolution technometer signal indicating the angle position of the shaft at the precision level of 6° was used. The National Instruments acquisition card was used to acquire the signals, which were then saved on a hard disk of a personal computer. The analysis was restricted to the frequencies of up to 6 kHz. Nevertheless it still included the most important meshing harmonics.

TIME-FREQUENCY SPECTRUM ANALYSIS

In relation to signals of technical diagnostics, some frequencies occur only in some cases. By classical frequency analysis of such signals, it is not possible to establish the time when particular frequencies appear in the spectrum. Time Frequency Analysis is applied to establish how frequencies of non-stationary signals change with time, and how intense they are [7].

The idea of windowed Fourier Transformation is to divide a time signal into short time intervals first and after that, to perform frequency analysis of each interval separately. A method for eliminating defects of Fourier Transformation is to compare signals with elementary functions, defined in time space and in frequency space.

The Fourier transform of the signal *x(t)* is not adequate for the frequency domain analysis if it is non-stationary. Local observation of the signal is necessary. In relation to that, the signal has to be divided into segments prior to carrying out the Fourier analysis [8]. It is assumed that, within each segment, the signal is stationary. Such a signal (divided into segments) is called a windowed signal.

$$x_w(t) = x(t) \cdot w(t) \tag{1}$$

where *w(t)* is a window function. If the product $t \cdot w(t)$ is also an element $L^2(\Re)$, a function $w(t) \in L^2(\Re)$ is referred to as a window function. In order to take the whole time domain into account, different positions of the window are to be selected, in view of which the windowed signal $x \cdot w(t)$ represents a function of time *t* and window position τ.

$$x_w(t,\tau) = w(t-\tau) \cdot x(t) \tag{2}$$

The result of the Fourier transform for such a signal can be referred to as a windowed Fourier transform, as it is a function of frequency and windows position.

$$X_w(f,\tau) = \int_{-\infty}^{+\infty} x(t) \cdot w(t-\tau) e^{-j2\pi ft}\, dt \tag{3}$$

The selection of the window function *w(t)* is possible in such a way that its Fourier transform *W(f)* is also a window function. The windowed Fourier transform presented in equation (3) is called a short-time Fourier transform. The selection of Gauss function represents an optimal choice of a window, the transform is called Gabor transform. The square of the modulus of the short-time Fourier transform is called a spectrogram in engineering applications. For each position of the window different spectra may be obtained; the total number of these spectra is a function representing a time-frequency distribution.

In engineering applications, the main problems concerning the spectrogram are related to the trade-off between time and frequency resolution. Some modifications for improving the short time Fourier transform have been suggested. In machinery diagnostics detection problems are more important than time-frequency localization. For detection, it is of utmost significance to establish whether the expected non-stationary events, which are the result of a fault, are present in the vibration signal. In addition to that, it may be required to estimate fault advancement. Thus, it is possible to simplify the windowed Fourier transform based methods by using only frequency localisation for detecting non-stationary events. The so called moving window procedure is one of the applications that deserve special attention.

In relation to time frequency analysis, the length of the measured signal is 1 s, frequency sampling 2^{16} samples/s and window length 100 ms. The number of pinion teeth was 19 and of gears 34, the tooth meshing frequency was 380 Hz. For window function, the Gausian function was selected, the transform is called Gabor transform. In Figures 4, no significant periodical changes in the direction of time axis are spotted in the observed frequency range of 1520 Hz (range of the third harmonic of meshing frequency) in consideration to the faultless gear. Similarly, the areas round higher harmonics are without important sidebands.

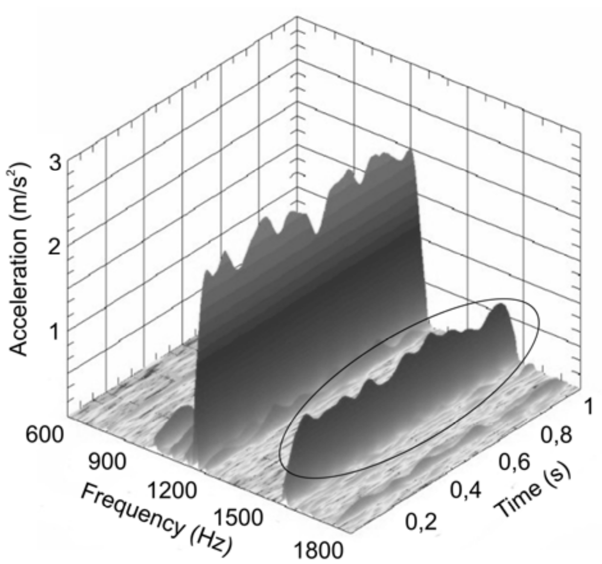


Figure 4: Gabor spectrogram of a faultless gear

In Figures 5 and 6, in the observed frequency range of the third harmonic of tooth meshing (1520 Hz), in the direction of time axis, a regular change in the frequency component can be noted. There are periodical or pulsating changes, with dominating frequency (20 Hz). This is a very significant feature. Twenty peaks of the same frequency emerge individually from the bottom of the spectrum in the direction of the time axis, regularly and at equal intervals.

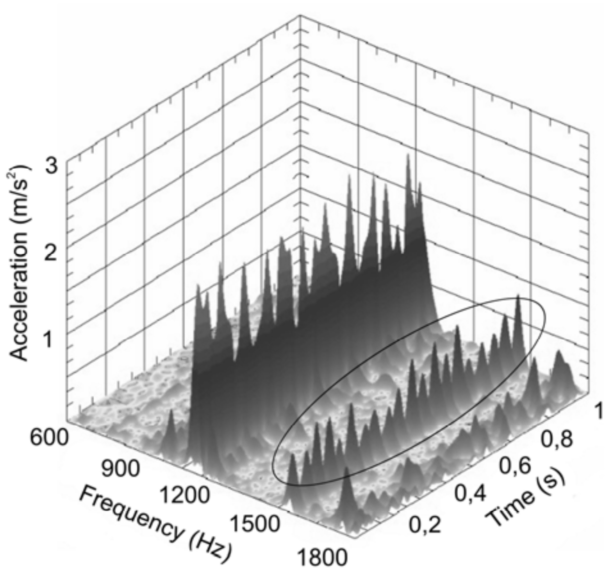


Figure 5: Gabor spectrogram of a gear with a longer (4.5 mm) fatigue crack in the tooth root

They define 20 Hz (1200 min^{-1}) of the rotational frequency of the shaft, where a gear with a crack in the root of one of its teeth is located. The peak can be noted in the spectrum only once during each rotation – when the tooth pair with a crack is engaged.

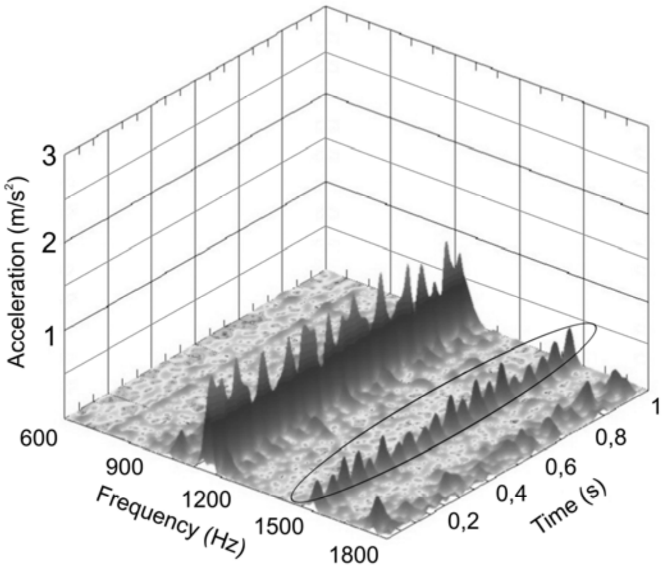


Figure 6: Gabor spectrogram of a gear with a shorter (1,1mm) fatigue crack in the tooth root

In relation to identifying the changes in the spectrogram correctly the application of a suitable time window and its length are of utmost importance. It is evident from Figure 4 that the intensity of 3rd harmonics through the whole time window is semiconstant. This is an important difference in comparison to Figures 5 and 6 – where it is evident that some frequencies appear only occasionally, i.e., periodically pulsating along the whole time axis. It is possible to determine the source by defining the frequency of appearance.

CONCLUSION

It is possible to plan experimental crack production well as it is easier to obtain the required lengths by using numerical methods for determining the life cycle of gear-units with a crack in the tooth root. Afterwards gear-units with cracks are built into gears, vibrations are measured and typical spectrogram patterns are sought for; on the basis of these spectrograms the condition of the gear can be established. A typical spectrogram pattern of a time frequency analysis related to the presence of a crack in a tooth root is presented. By searching for typical frequency components in spectrograms and for their changes in time, the condition of a gear can be determined in a much more reliable way than with classical spectral analysis.

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