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TOOTHED WHEELS DIAGNOSIS

Sorin Pater[1], Ion Bratu[1], Nicolae Polojitef Corbu[1]
[1]University of Oradea, spater@uoradea.ro

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Abstract: The initial contact between mating spur gear teeth takes place when the flank of the driver encounters the tip of the driven tooth. This occurs at the point where the addendum circle of the driven gear crosses the line of action. In real functioning conditions, the gearing process has certain deviations versus the ideal conditions. These deviations are provoked both by the execution errors and the other transmission elements of the toothed wheels, and by the assembling errors

The initial contact between mating spur gear teeth takes place when the flank of the driver encounters the tip of the driven tooth. This occurs at the point where the addendum circle of the driven gear crosses the line of action. Then, as the teeth go into mesh, the line of contact will slide up the side of the driving tooth. The tip of the driver will be in contact with the flank of the driven tooth just before contact ends. This is when the addendum circle of the driver intersects the line of action. The period of time between the initial contact and final contact is called the tooth meshing period. In real gear designs, the situation is further complicated, since several gear teeth are in mesh at the same time.

Thus, the load is shared among several teeth allowing them to transmit larger torques. The contact ratio describes the average number of gear teeth in mesh at any instant. In the experimental data presented in this thesis, the gear contact ratio is 2.4. Therefore, a tooth meshing period contains information about the tooth carrying most of the load and approximately 710 of both its neighbors. Large gear contact ratios make single tooth fault diagnostics ambiguous. For diagnostic work, it is more appropriate to talk about localizing a damaged region, than isolating a single damaged tooth.

The tooth meshing period is an essential component in the understanding of gear vibration mechanisms. A pinion with 21 teeth contains 21 tooth meshing periods per shaft revolution Fig. 1, while the companion 70 tooth gear contains 70 tooth meshing periods per shaft revolution Fig. 2.

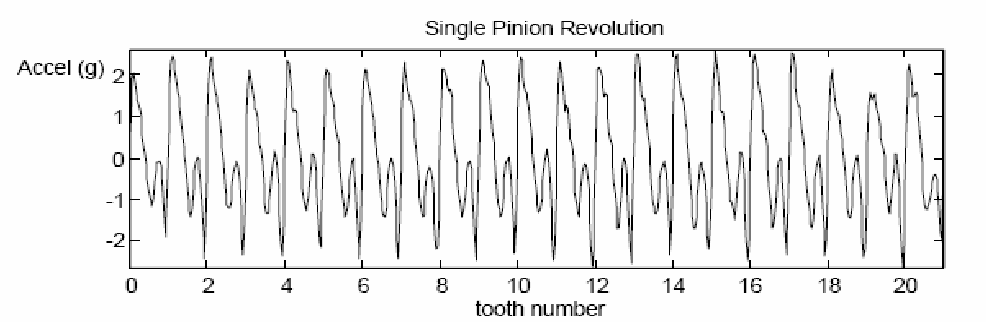

Figure 1 Typical Healthy Pinion Vibration Signal (One Shaft Revolution)

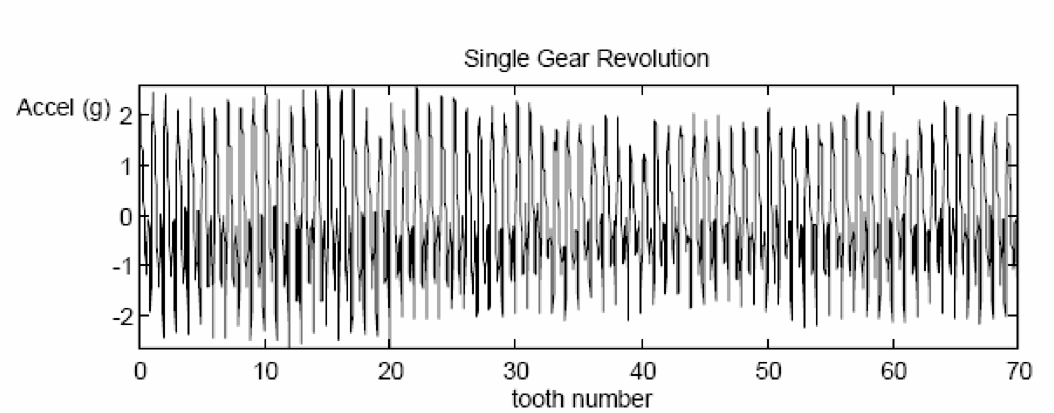

Figure 2 Typical Healthy Gear Vibration Signal (One Shaft Revolution)

Thus, while the pinion and gear have different rotational periods, they both share the same tooth meshing period. In this example, the tooth meshing period is 0.0015996 seconds. The signal components that are periodic within this time frame are decomposed into a gear mesh frequency of 625.16 Hz and its harmonics Fig. 3. In healthy gearsets, the vibration is dominated by these pure tonal components and the time signal appears very regular. In the next section, we will show that the mesh frequency and its harmonics are due to both the pinion and the gear.

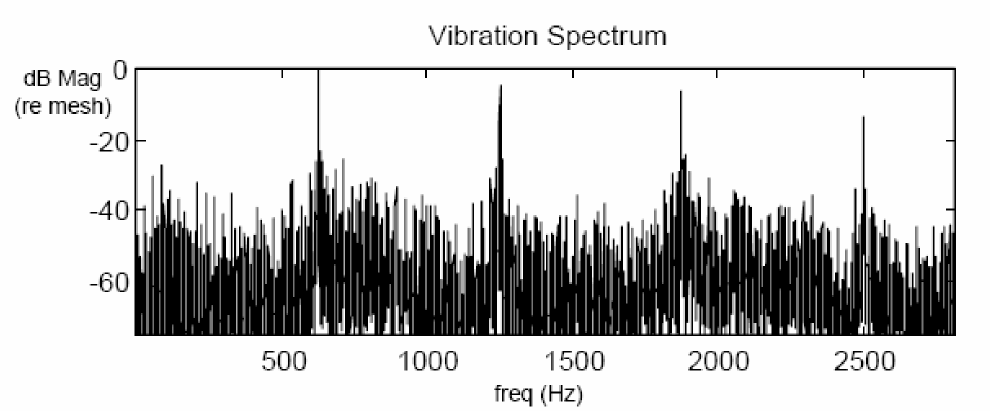

Figure 3 Typical Healthy Gearbox Vibration Spectrum

In real functioning conditions, the gearing process has certain deviations versus the ideal conditions. These deviations are provoked both by the execution errors and the other transmission elements of the toothed wheels, and by the assembling errors.

The dynamic loads that appear in these conditions can be considerable, in comparison with the static forces, and their being token into consideration at the gearing planning is compulsory.

The toothed wheels transmission dynamic is influenced by the following facts:

- the rigidity variation of the gearing due to the variable deformations of the teeth in the process of gearing (the load is transmitted by a different number of teeth).
- the technological errors of the gearing
- the rotation speed, especially in those zones that correspond to the resonance phenomenon

The interior sources are represented by the deviations from the tooth-processing precision, especially the error of the measured step on the basis circle, that lends to the appearance of the periodical percussion between the teeth and creates a short term dynamic load and the profile error that creates a permanent dynamic load, as well as the periodic variation of the gearing rigidity, due to the periodic passing of the load from one tooth to two teeth. These sources are of a great interest for the gearing durability. The vibrations generated by these sources and together with them the dynamic forces and the noise become very strong high, especially when the frequency of the perturbation sources which is always in a relation determined by the gearing revolutions superposes on a frequency of it's own – the resonance phenomenon appears.

The diagrams in fig.1, have been mode in order to diagrams the gearbox: - the diagrams of the signal acquired in time of the power spectrum in frequency and the cepstrum in time for the faultless gear box, considered as reference.

Amplitude values of the cepstrum were obtained up to 0.55m/s^2 and the spacing of the side bands corresponds to the frequencies generated by the bearings, gearing and the belt transmission.

The RMS value, corresponding to the acquired signal is 1.3761m/s^2.

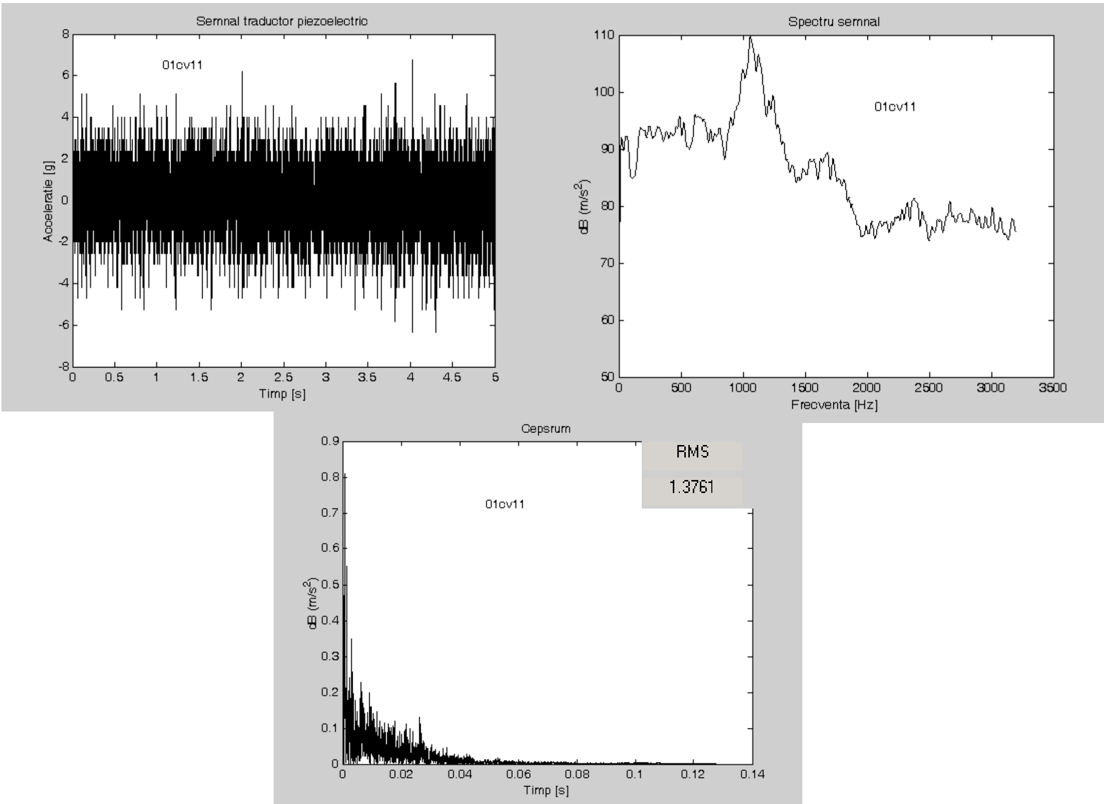

Figura 4. The defect less gear box diagrams.

For the defect gear box diagnosis, the diagrams in fig.4 have been obtained: the diagram of the signal acquired in time, of the power spectrum in frequency and of the cepstrum in time.

There are peaks equally placed on the cepstrum diagram that correspond to the defect in the gearing.

For the approximate determination through calculation of the signal frequency generated by the gearing defect, the following methodology is used:

The gearing frequency is determined through:

$$f_a=f_m/Na \qquad (1)$$

where f_m represents the rotation frequency of the driving wheel and N_a represents the lowest common factor of the teeth number corresponding to the pinion and the toothed wheel.

Mathematically, the rotation frequency f_m can be expressed through the rotation frequency of the pinion and of it's number of teeth, or though the rotation frequency of the driven wheel and it's number of teeth.

$$f_m=f_{rp} \times Z_p=f_{rg} \times Z_g \tag{2}$$

Where:

- f_{rg}=R9/60,represents the rotation frequency of the driven wheel expressed in;
- f_{rp}=Rp/60,represents the pinion rotation frequency expressed in Hz.
- Z_p, represents the pinion number of teeth
- Z_g, represents the teeth number of the toothed wheel.

The gearing frequency f_{tr} , for one tooth of the pinion that comes into gearing with the same tooth of the driver wheel is given by:

$$f_{tr}=f_m \times N_a / Z_g \times Z_p \tag{3}$$

The gearing frequency will be a low one and it can not be easily detected in spectrum, but it can be easily detected in cepstrum.

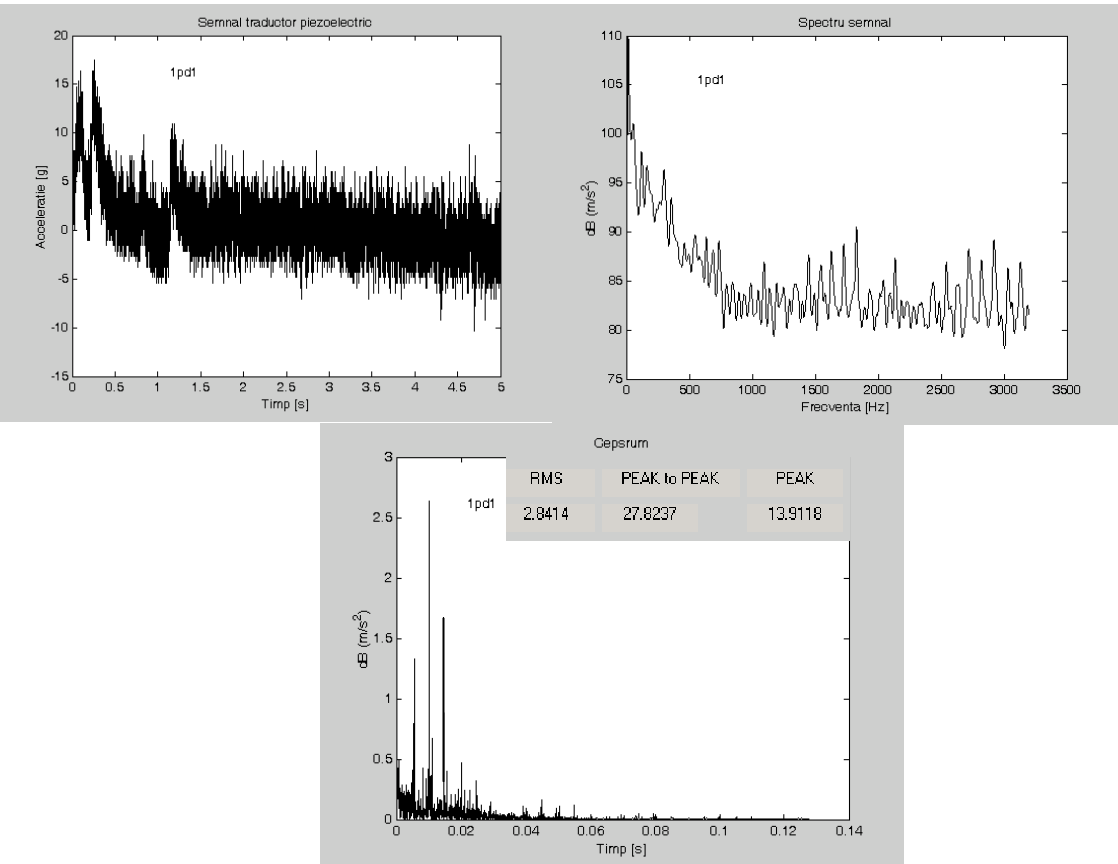

Figura 5. The defect induced by the toothed wheels

For this gear box analyzed in the time domain, the gearing frequency has the value given in the following table:

Type of box	Speed step	Number of wheel teeth	Number of pinion teeth	Na	Rotation frequency	Gearing frequency
365	1	42	11	1	899,97	50,94
	2	38	17	1	899,98	49,09
	3	34	23	1	899,98	47,36
	4	33	34	1	899,96	40,29
	5	31	36	1	899,93	40,29

This obtained values indicates the appearance of frequency peaks on the cepstrum diagrams at an interval of

$$\Delta\, t_{cp} = 1 / f_{tr} = 1 / 50,94 = 0,0196 \text{ [s]}.$$

Analyzing the experimental obtained diagrams, we can notice spacing between two peaks in the cepstrum diagram, close to the calculated value of
Δt_{mp}=0,0192 [s], so the measured frequency will be $f_m=1/\Delta\, t_{mp}$=52.083 [Hz].

These peaks and their spacing can be used to discover the gearing defects. In comparison with the defect less gear box, the amplitude in cepstrum is approximately twice two folded.

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