Appraisal of gear pitting severity by vibration signal analysis

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Abstract. Gears are mechanical components used to transfer power between two machines and to modify the operating speed. When they are operating at high loads and speeds the gear teeth surfaces are subjected to loads, which, even in proper lubrication conditions, are leading to failures like scuffing, scoring, spalling or pitting. As the vibration of a gearbox is carrying the signature of the gear faults, these deficiencies may be detected by measuring and analysing the vibration signal. In this research, experimental investigations were accomplished in order to evaluate the gear pitting severity by means of vibration signal measurement and analysis. Thus, four pinions with different pitting grades, created by artificial means, were incorporated in a single helical gearbox and tested on an open-energy test rig. The collected vibration measurements have provided the information regarding the pitting severity, supplying information for the assessment of the gears failure stage. For processing the signals a relative new technique was used, namely the Sideband Energy Ratio™ (SER).

1. Introduction

Pitting is one of the most typical gear failures, which is mainly caused by repeated Hertzian contact of the mating gear surfaces. The resulted contact stress causes fatigue cracks which are initiated in the tooth surface of the gears. The presence of oil, which is pressed in the cracks, contributes to deepening them and when two cracks meet, a pit is detached from the gear tooth surface. Pitting commonly appears at the tooth dedendum, where both sliding velocity and contact stress are high.

Various theoretical and experimental researches were carried out regarding the pitting mechanism and the factors which are influencing this gear failure type. Thus, Olver [1] pointed out that the gear surface roughness is the main cause for the initiation of pitting. Stantus and his co-operators [2] analysed the rolling contact process, stating that pitting cracks are initiated due to low tooth surface quality and insufficient lubrication. Both Laine et al. [3] and Morales-Espejel et al. [4] revealed that mild wear and pitting damage are in an opposite connection. They reported that some mild wear may contribute to the decrease of pitting at the gear operation start. Winkelmann et al. [5] suggested that super-finishing of gears can definitely exclude pitting failure. Al-Tubi et al. [6] sowed that pitting develops from the pinion dedendum to the addendum, as a consequence of extreme contact pressure and thin oil-film thickness. Further, using a twin disc tester, Oila and Bull [7] found that load is the main motivation for pitting enrolment, whereas slide-to roll ratio (SRR) and speed are responsible for the propagation of this failure. Finally, Höhn et al. [8] reported that pitting may occur even though the transmitted load is under the contact fatigue limit, while Predki et al. [9], performing pitting tests on large gears, concluded that the findings for small gears cannot be automatically transferred to large ones.
The present study involves a recent developed method, i.e. Sideband Energy Ratio\textsuperscript{TM} (SER) [10], which was developed as an early damage detection tool.

2. Sideband Energy Ratio\textsuperscript{TM} (SER) method

SER is a new, patented algorithm, developed for the detection of gear failures. Under normal operating conditions, the gear mesh frequencies and their harmonics are present in the vibration spectrum of a gearbox. As faults develop, amplitude changes are not always detectable, because the mesh frequency is not affected by the fault development. Nevertheless, there is a phenomenon which appears as faults occur in a gear transmission: the vibration signal becomes modulated by the fault. For instance, if a tooth of a pinion is damaged, every time this tooth engages the matting gear, a difference in amplitude is detected. This produces a $1x$ modulation of the mesh frequency at the speed of the pinion shaft. This modulation appears in the frequency domain as $1x$ pinion sidebands around the gear mesh frequency (GMF). Generally, the higher the severity of the defect, the higher the energy of the sidebands.

Sidebands occur in a spectrum at the left and right side of a centre frequency, appearing as a consequence of an amplitude modulation (AM) of that centre frequency. Amplitude modulation is typically used in the technique of communications, where the information is transmitted via a carrier wave. In typical amplitude modulation, the carrier signal, having the frequency $f_c$, may be expressed as:

$$A_m = A(t)\cos(2\pi f_c t)$$  \hspace{1cm} (1)

If the carrier signal is modulated by a lower frequency signal $m(t)$, the amplitude of the signal is:

$$A(t) = A_0\left[1 + m(t)\right]$$  \hspace{1cm} (2)

Assuming that the modulation signal is a single frequency tone, with frequency $f_m$, by substituting this in equation (1) and (2), gives the following:

$$A_m = A_0\left[1 + \beta \cos(2\pi f_m t)\right]\cos(2\pi f_c t)$$  \hspace{1cm} (3)

were $\beta$ is the so called modulation index (for a better signal recovery, in radio communications, $\beta$ is less than 1).

Knowing that:

$$\cos(2\pi f_m t)\cos(2\pi f_c t) = \frac{1}{2}\left\{\cos[2\pi(f_c + f_m)t] + \frac{A_0\beta}{2}\cos[2\pi(f_c - f_m)t]\right\}$$  \hspace{1cm} (4)

equation (3), becomes

$$A_m = A_0\cos(2\pi f_c t) + \frac{A_0\beta}{2}\cos[2\pi(f_c + f_m)t] + \frac{A_0\beta}{2}\cos[2\pi(f_c - f_m)t]$$  \hspace{1cm} (5)

As a consequence, the amplitude modulation generates, among the carrier frequency $f_c$, side bands $f_c - f_m$ and $f_c + f_m$ located below and above the carrier frequency.

In real applications, the modulation appears not only for a single tone, but for multiple frequencies, which are causing multiple sidebands in the frequency spectrum. Particularly, at the occurrence of gear tooth pitting, the faulty tooth produces an amplitude modulation of its associated gear mesh frequency (GMF), every time it is engaged. That modulation in amplitude appears at each rotation of the shaft which carries the damaged gear being visible in the frequency spectrum as a series of evenly spaced lines behind and after the central mesh frequency (CMF). These sidebands appear at frequencies of $f_{GM} = nf_c$, where:

$f_{GM}$ is the GMF;

$i$ is an integer, which for the SER method takes values between 1 and 6;

$f_c$ the rotational frequency of the shaft which carries the faulty gear.
Because the wave generated by the meshing gears is generally not a pure sine function, the spectrum shows usually additional harmonics of the GMF with their own sideband families (see figure 1).

![Figure 1. Spectrum of GMF and its harmonics.](image)

SER are calculated by summing up the first six sideband amplitude peaks on each side of CMF and dividing by the amplitude of CMF [11]:

$$SER = \frac{\sum_{n=1}^{6} \text{Sideband Amplitude}_n}{\text{CMF Amplitude}}$$

As it can be observed, SER is responsive to the sideband amplitudes in regard to the CMF. For a healthy gear mesh the amplitude of the sidebands are very low or may be missing, compared to the CMF, arising in a low SER. A gear mesh without failure will typically have a SER value less than 1.0 [11]. The appearance of damages on the gear teeth which are passing through the gear mesh will produce increased sidebands in amplitude and number producing a higher SER value. Commonly, for all meshes, an SER is determined for the fundamental gear mesh frequency (1x GMF), the first harmonic (2x GMF), and the second harmonic (3x GMF).

3. Problem formulation
The aim of this research was to assess the most common gear surface failure, i.e. pitting, by vibration measurement and processing the results by using the SER method. For this objective, an own developed gear test stand with open energy flow was employed, the stand being also used for previous studies [12], [13]. A general view of the stand, showing its main elements, is presented in figure 2.

![Figure 2. General view of the open energy flow gear test stand.](image)
The main components of test stand are an electric motor (1) with variable speed, the one-stage gearbox (2) and a gear pump (3) what was used as a break and which is recirculating the oil from a tank (4). For the variation of the breaking torque the spherical valve (5) was employed. The connections between the motor, gearbox and pump are made by elastic couplings with rubber strips (5) and (6). Technical data of the test stand and its main elements are presented in table 1.

**Table 1.** Technical data of the test stand and its main elements.

| Element    | Technical data                        |
|------------|---------------------------------------|
| Electric motor | Power: \( P = 2.5 \, \text{kW} \)     |
|            | Speed: \( n = 0 \ldots 1500 \, \text{min}^{-1} \) |
| Gearbox    | Centre distance: \( A = 125 \, \text{mm} \) |
|            | Teeth number: \( z_1/z_2 = 17/43 \)   |
|            | Module \( m_n = 4 \, \text{mm} \)     |
|            | Face width \( b = 40 \, \text{mm} \)   |
| Break      | Gear pump type KF 6/400 (producer: KRACHT) |

For simulating different stages of pitting, the gearbox was equipped, in turn, with four pinions having practiced artificial grooves with a diameter of 3 mm and a depth of about 0.5 mm, along the pitch line of each tooth. A centralization of the 4 stages of the investigated gear damage is presented in table 2.

**Table 2.** Stages of gear pitting.

| Pitting condition | Number of grooves |
|-------------------|-------------------|
| C1                | 0                 |
| C2                | 1                 |
| C3                | 3                 |
| C4                | 5                 |

The pinions with the four pitting conditions are depicted in figure 3.

**Figure 3.** Pinions with different pitting condition [11].
The hardware for vibration measurement consisted of a Kistler accelerometer of type 8772A5M10, mounted on the top of the gearbox, a NI (National Instruments) signal acquisition module type NI 9234 and a laptop programmed to run a special application designed in LabView software, for the fine processing of the acquired vibration signal.

### Table 3. Relevant frequencies for SER method.

| Frequency                | Symbol | Value [Hz] |
|--------------------------|--------|------------|
| Fundamental GMF         | 1xGMF  | 425        |
| First harmonic of GMF   | 2xGMF  | 850        |
| Second harmonic of GMF  | 3xGMF  | 1275       |
| Pinion rotational frequency | \(f_s\) | 25         |

The test were performed at a motor speed \(n_1 = 1500 \text{ min}^{-1}\). Thus, the relevant frequencies for the SER method are shown in table 3.

### 4. Results and discussion

Figure 4 shows the measurement results concerning the vibration acceleration for the four pinions with different pitting condition. As it can be noticed, the representations indicate only an increase of the vibration amplitude from the condition C1 to C4, being unable to provide the cause for this raise.

As a consequence, it is required to make use of a frequency representation for the vibration spectrum. This can be done by involving the Fast Fourier Transformation (FFT). In this way it becomes visible which frequencies are generating certain acceleration peaks, thus understanding the root cause of the vibration raise. Thereon, figures 5, 6, 7 and 8 display the frequency spectrum for involving the SER technique in the appraisal of the gear pitting severity.
In the spectrum plot shown in figure 5 the first 3 harmonics of the GMF are evidenced as 1xGMF, 2xGMF, and 3xGMF respectively for this pitting condition C1 (new pinion- without faults). The sidebands around the harmonics of the GMF and located inside the blue marked rectangles are visible, but their amplitudes are insignificant related to the CMF amplitude. This is an indication that there are no present damages on the gear teeth. This finding is confirmed by the SER values, which are less than 1.0, typical for healthy gear meshes. Furthermore, one can observe that the SER values increase with the order of the harmonics.

![Figure 6. Spectrum of pinion with pitting condition C2.](image)

Additionally, the spectrum plots of acceleration data for the pinion condition C2 are given in figure 6. The amplitudes of the three harmonics of the GMF are higher compared the C1 condition. Around these peaks about three sidebands became visible. The appearance and spacing of these sidebands in the spectrum indicates that the CMF is being amplitude modulated at the frequency of the pinion shaft. This fact and the SER values (1.78, 2.32 and 2.66 respectively) indicates that some damages have been initiated on the pinion teeth surfaces.

![Figure 7. Spectrum of pinion with pitting condition C3.](image)

Figure 7 depicts the spectrum plots of acceleration data for the pinion condition C3. The acceleration amplitudes of the 3 harmonics of the GMF are substantial higher, for this pitting condition, compared to C1 and C2. Around these peaks are about 6 sidebands visible, but maximum 4 of them have a substantial extent related to the CMF amplitude. As it can be observed, the second
The harmonic of the GMF presents the highest value of the acceleration amplitude. The SER values calculated for the 1x, 2x, and 3x CMF for this gear mesh and sideband spacing are 2.98, 3.35, and 3.81 respectively. These values indicate the presence of gear faults on the pinion shaft.

Figure 8 presents the FFT spectrum obtained for the pinion with pitting condition C4. The frequency range (225–1475 Hz) is the same as in the previous three spectrums. Obviously, the acceleration amplitudes are the highest during this study. The sidebands are very visible, their approximate equal spacing indicating an amplitude modulation at the frequency of the pinion shaft. The SER values (3.42, 4.03, and 5.93 respectively), highlight a significant increase compared with the SER of the previously investigated gear health conditions.

Summarising, this research reveals that the increase in SER is proportionate to gear pitting fault progression which makes the Sideband Energy Ratio™ method a reliable toll in assessing the gear pitting severity.

5. Conclusions

Early detection and diagnosis of gear faults inside gearboxes is a main tool of diminishing unexpected down time of gearboxes. Therefore, in this paper four different stages of gear pitting were investigated using the SER method. The acquisition, analysis and trending of SER has proven that this approach can be an efficient tool for the detection of gear faults and a severity appraisal in different gearbox types.

The results revealed that SER is a reliable technic in tracking the progression of gear pitting failure. Thus, an increase of SER with pitting progression was plainly visible. The SER was found to be less than 1.0 for healthy pinions (condition C1) and more than 1.78 for pinions with different pitting stages (conditions C2, C3 and C4).

6. References

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Acknowledgments
The authors would like to acknowledge the partnership with the gearbox factory Resita Reductoare si Regenerabile S.A. (RRR), where the test stand and the pinions used in the present study were produced.