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Detecting Mixed Gear Faults Using Scalar and Cyclostationary Indicators in an Industrial Settin

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Abstract: In this paper, an innovative approach is presented to enhance gear fault diagnosis using the cyclostationarity method. The first part of this study focuses on simulating gear signals under various conditions, allowing exploration of signal characteristics in vibration measurements. Spectral analyses and statistical calculations are performed to extract both classical and cyclostationary indicators. In the second part, the cyclostationarity method is applied to signals recorded at the gearbox bearings, clearly revealing the presence of faults. The results from these experiments demonstrate that cyclostationarity indicators can be leveraged to improve the prediction of signal roughness during the production process. This approach thus opens up new possibilities to enhance the reliability of vibration measurements and refine gear fault diagnosis.

Keywords: Scalar indicator, Cyclostationary indicator, Spectral analysis, MID, IMID, Reducer 101 BJT.

Introduction

The control of machines by vibration analysis have been developed considerably due to the evolution of maintenance concepts in order to minimize downtime and revision by adopting the method of conditional preventive maintenance Boulenger, et al (1998), and Pachaud et al (1998). Reducing production costs and increasing the availability of the production tools have indeed become issues that needed to detect and identify all defects at an early stage Heng (2002). The current challenge is not to reveal the importance of using vibration analysis in the field of rotating machines maintenance, but to assimilate the bases and recognize the choice limitations of each technique. It becomes clear that the failures cannot be tolerated because of the complexity of the machine and its criticality in the process Vibro-Meter (1991) and Heng (2005). For efficient conditional preventive maintenance, reliable and accurate measurements of the state of the machine should be made Muller (2005). The best control strategy is to use the most relevant indicators. In addition, misinterpretation of operating conditions often leads to false diagnostic sources, which results in unnecessary repairs and considerable downtime.

Spectral analysis is almost certainly the oldest technique and the most used both. Coming to fill the limits of this technique, several methods have also emerged and generally developed for very specific defects. The "revolution" in the field of vibration diagnostic of machines is probably using periodic methods that allowed the development of more reliable methods based on cyclostationarity (Heng, 2011; Estoque, 2004). And on the separation of cyclical component of the signal (Antoni, 2009; Boustany & Antoni, 2005).

The cyclostationarity formulation presents several advantages for the analysis of such signals over conventional approaches. Firstly it can capture with the same tools a large spectrum of behaviors from simple deterministic periodicity (phenomena of unbalance, misalignment, eccentricity, force inversion, lack of engagement ... etc.) to the non-stationary random (fluid movement, frictional forces, bind ...) (Bonnardot , Randall, & Guillet, 2005; Urbanek, Barszcz, Zimroz, & Antoni, 2012). Then it explicitly incorporates a temporal dimension that lets following the evolution of non-stationarity of the studied systems. This analysis allows to identify the amplitude modulations present in the vibration signals (Urbanek, Barszcz, Sawalhi, & Randall, 2011; Gellermann, 2003). This modulation varies with signal components, these can often provide valuable data. An error occurs when the measurement time is generally insufficient (Antoni, 2007). Several studies have shown that the presence of modulations confirms the existence of defects in rolling bearings and gears (Gardner et al., 1988; Makowski, & Zimroz, 2011; Randall, & Antoni, 2011; Antoni et. Al., 2004).

This paper focuses on the application of two properties of cyclostationarity, commonly, namely spectral correlation and spectral coherence for detecting modulations in intensity distribution of the mechanical signals. First, the proposed method is applied on signal of defective bearing measured on the Machine Faults Simulator (MFS) Spectra Quest. A diagnosis of a turbo-alternator working in real conditions in industrial field based on kinematic study and spectral analysis established. Afterwards, a comparison between the cyclostationarity and spectral analysis performed. Eventually, the influence of the variation of the load on the vibration level change also investigated.

Principle of the Method

The cyclostationarity is a method based mostly on the basis of modulation intensity distribution (MID) for the detection and identification of modulations present in the signal. The MID technique originally designed for defect diagnosis of gears, rolling bearings, and journal bearings. The spectral correlation density is concentrated on the detection of amplitude modulation of sidebands spaced symmetrically in the spectra. It can present values of the modulation indicator on a frequency range of the signal as a function of the carrier frequency f and the modulation frequency α. Several techniques exist such as wavelet and empirical decomposition of modulations that could be used to define the indicator of the modulations presence. Therefore, the proposed method can be customized to specify the properties of different signals to be studied in order to explain the principles of the MID.

A filtered signal in this manner contains, in the idealized case, only the specified component with no additional signals and with a very low noise level Taher F, Fakher C, Mohamed H (2005). In this case, the filtered signal consists of a set of three elements:

$$x_i = x_{\Delta f}\left(t, f - i\alpha\right) \qquad avec \quad i = \left\{-1,0,1\right\} \tag{1}$$

Where xi is the unique value of the signal, $x_{\Delta f}(t, f)$ indicate the filtrated version of x (t) in a side band frequency $\left[f - \frac{\Delta f}{2}, f + \frac{\Delta f}{2} \right]$ with Δf the frequency step.

It may be observed that the symmetric filtration depends on three parameters $f, \alpha, \Delta f$.
The frequency band of the analysis of the MID is calculated as follows:

$$f_{\min} = \frac{\alpha_{\min}}{2} + \frac{\Delta f}{2} \tag{2}$$

and

$$f_{\max} = \frac{f_s}{2} - \frac{\alpha_{\max}}{2} - \frac{\Delta f}{2} \tag{3}$$

Where fs is the sampling frequency of the measured signal.

At the end of the filtering process, the calculated spectral correlation could serve as an indicator of the presence of modulation. According to the concept of the sideband filter, and the notations included in the equation 9, the spectral correlation density between components $x_{\Delta f}(t, f+\alpha)$ and $x_{\Delta f}(t, f)$ are given by:

$$SC_x^{\alpha}\left(f+\frac{\alpha}{2}\right)=\lim_{\Delta f \to 0} \lim_{T\to\infty} \frac{1}{T\Delta f}\int_T x_{\Delta f}(t, f)\, x_{\Delta f}^{*}(t, f+\alpha) e^{-j2\pi\alpha t} dt \tag{4}$$

By analogy, additional spectral correlation density can be calculated between components $x_{\Delta f}(t, f-\alpha)$ and $x_{\Delta f}(t, f)$:

$$SC_x^{\alpha}\left(f-\frac{\alpha}{2}\right)=\lim_{\Delta f \to 0} \lim_{T\to\infty} \frac{1}{T\Delta f}\int_T x_{\Delta f}(t, f)\, x_{\Delta f}^{*}(t, f-\alpha) e^{j2\pi\alpha t} dt \tag{5}$$

In order to inspect the relationship between the three spectral components separated by the cyclic frequency α, the multiplication of two spectral correlation density introduced in equations 4 and 5 should be considered.
The suggestion of the frequency *f* and α, for a given Δ*f*, is commonly called the Modulation Intensity Distribution (MID) that can be expressed by:

$$MID_{\Delta f}^{PSC}(f,\alpha)=SC_x^{\alpha}\left(f+\frac{\alpha}{2}\right)SC_x^{\alpha}\left(f-\frac{\alpha}{2}\right)^{*} \tag{6}$$

Where the superscript PSC is the Product of the Spectral Correlation. The degree of cyclostationarity proposed in D' Elia et al. (2011) and Urbanek et al. (2013) represents the ratio of the energy $\alpha \neq 0$ and $\alpha = 0$ for a stationary signal. Its mathematical expression is given by:

$$DSC^{\alpha}=\int\left|R_x^{\alpha}(\alpha)\right|^2 d\alpha \Big/ \left|SC_x^{0}(\alpha)\right|^2 d\alpha \tag{7}$$

In some practical applications of the MID for vibration signals, the multiplication of spectral correlation density as the modulation intensity distribution may not be the most useful measure because of large differences in signal energy in different frequency bands. In this case, the interpretation of the MID plans may be more effective when the absolute value of the spectral correlation density is normalized and varies only between 0 and 1. To this end, the proposal $MID_{\Delta f}^{PSC}$ can easily be extended to the use of coherence spectral density as a modulation intensity distribution. The function of the modulation of the distribution density of spectral coherence is obtained by:

$$MID_{\Delta f}^{PSCoh}(f,\alpha)=\left(\frac{SC_x^{\alpha}(f+\alpha/2)}{\sqrt{SC_x^{0}(f+\alpha/2)SC_x^{0}(f)}}\frac{SC_x^{\alpha}(f-\alpha/2)}{\sqrt{SC_x^{0}(f-\alpha/2)SC_x^{0}(f)}}\right) \tag{8}$$

Integration MID Based on Spectral Correlation

As mentioned above, the MID is a function of the carrier frequency f and the modulation frequency α. However, in some special cases, the user may not be interested in finding the specific range of carrier frequencies, but only in the estimation of the general influence specific modulations of the components on the tested signal. In addition, three-dimensional representations can cause difficulties of interpretation and make automated decisions in the process of monitoring of industrial systems Urbanek J, Barszcz T, and al (2014), and Gardner WA (1986). Based on this observation, it is more convenient to represent the MID not as a surface, but as a curve depending only on the modulation frequency, after integration on a carrier frequency band chosen. In the case of amplitude modulation, such representation could be measured for full modulation.The integration distribution of MID represents the density of spectral correlation, expressed by different sources, is commonly called IMID. This integration will be selected over the entire band carrier frequencies defined by:

$$IMID_{f_1}^{f_2}(\alpha,\Delta f)=\int_{f_1}^{f_2} MID_{\Delta f}(f,\alpha)df \tag{10}$$

Where $MID_{\Delta f}(f;\alpha)$ is a vector calculated in the band carrier frequency from f_1 to f_2. The idea of the IMID.

Case of Bearing of the Machine Faults Simulator MFS Spectra Quest

First, the performances of the cyclostationarity method were tested for the detection of an inner race bearing fault assembled on the Machine Faults Simulator (MFS) Spectra Quest (Figure 1). Table 1 shows the geometrical characteristics of the bearing 6004E and in the Table 2, the defects characteristic frequencies are grouped. The frequency of occurred shocks is usually called ball pass frequency inner (BPFI) and is expressed as follows:

$$BPFI=\frac{nf_r}{2}\left(1+\frac{d}{D_m}\cos\theta\right) \tag{11}$$

with:

$$D_m=\frac{D_1+D_2}{2} \tag{12}$$

Where fr is the rotational frequency in **Hz**, n is the number of bearing balls, **θ** is the contact angle, $\mathbf{D_m}$ indicates the average diameter and d denotes the diameter of a rolling element bearing figure 2.

Table 1 Characteristics of ball bearing 6004E

Diameter of the outer ring	42 mm
Diameter of the inner ring	20 mm
Ball diameter	7.14 mm
Number of balls Nb	08

Table 2 Frequencies characteristic of defect BPFI

Rotation frequency	15 Hz
Frequency characteristic of defect BPFI	72.25 Hz

Figure 1. Photograph of simulator MFS Spectra Quest

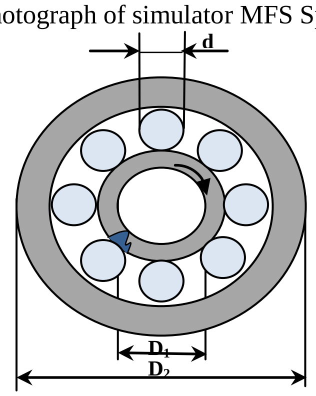

Figure 2. Diagram of a ball bearing with a defect on the inner race

Application of the Cyclostationarity

Shows the result obtained after the application of the MID on the signal measured on the bearing 1. It is clearly noted the appearance of a cyclic frequency at $\alpha = 6.98\times10^{-3}$ *f*s ≈72 Hz and multiple harmonics corresponding to a bearing defect on the inner race (BPFI). This diagnosis was not well clear in the spectral analysis. There is also the occurrence of a carrier frequency at 806 Hz; this value corresponds probably to a resonant frequency of the system see figure 3.a. Shows the result of the use of the integration of the modulation intensity distribution (the IMID) on the same signal. It allows highlighting, by a very clear manner, the cyclic frequency and modulation see figure 3.b.

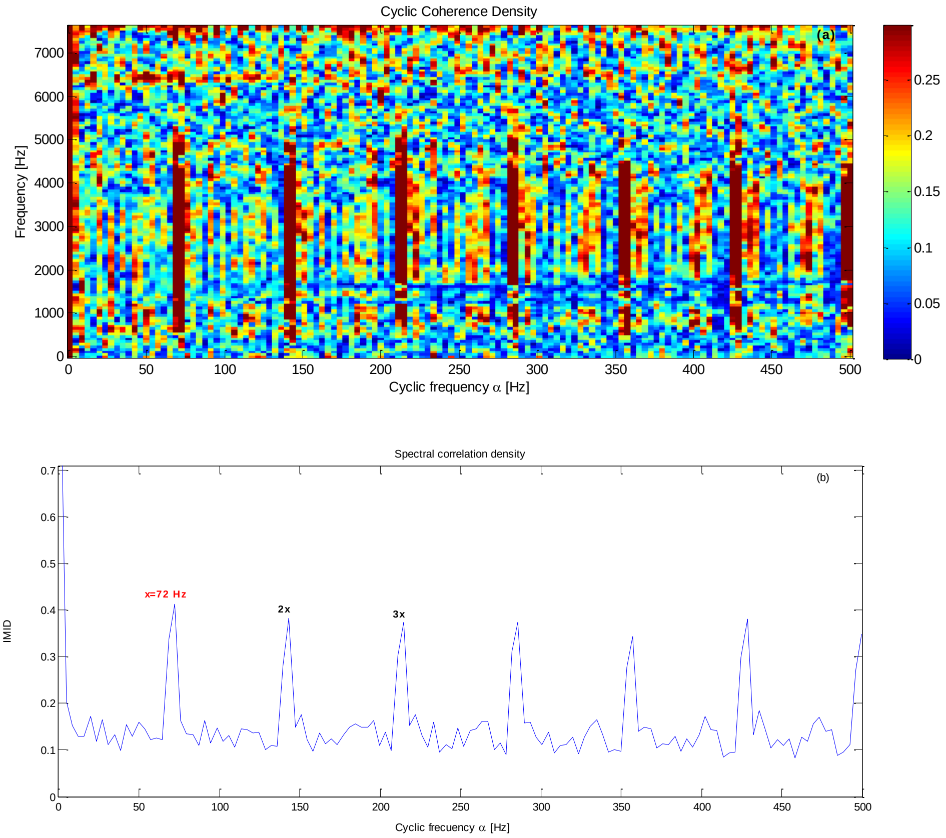

Figure 3. (a) Spectral Correlation (MID) with the corresponding (IMID) (b).

Case of Defects of Turbo-Alternator GZ 1164

Setup Description

The operating principle of the turbo-alternator GZ 1164 comprises introducing superheated vapour at an elevated pressure, commonly called admission vapour, via inlet pipe of the turbine. The latter comprises 16 wheels, each one comprising a number of blades NB (see Table. 3). Each wheel is composed by plurality of eight fixed vanes. The setup comprises a turbine which converts the kinetic energy into mechanical energy that rotates the turbine rotor, a speed reducer in one stage with herringbone teeth, an alternator that rotates at the same output speed of reducer (whatever the load producing electricity), and other accessories figure 4.

Calculation of the Blade Passage Frequency

The passage frequency of the blades of each wheel of the turbine FBP and the frequency of passage of the sets of fixed blades F_{PFBS} can be expressed by the following relationships:

$$F_{BP} = F_{r1} * N_B \tag{12}$$

$$F_{PFBS} = F_{r1} * N_{bW} \tag{13}$$

with N_{bw} = 8 Number of fixed blade in each wheel set.
Fr_1: rotational frequency of the turbine rotor.

Table 3 includes the frequency characteristics of the defect of blade passage for a rotational frequency of the turbine rotor of about150 Hz.

Table 3 Characteristic frequencies of blades passage defects

Number of wheel	Number of blades (NP)	Frequency of the defect (FBP)
1	40	6000 Hz
2	48	7200 Hz
3	48	7200 Hz
4	56	8400 Hz
5	64	9600 Hz
6	88	13200 Hz
7	88	13200 Hz
8	88	13200 Hz
9	88	13200 Hz
10	88	13200 Hz
11	88	13200 Hz
12	88	13200 Hz
13	88	13200 Hz
14	88	13200 Hz
15	80	12000 Hz
16	112	16800 Hz

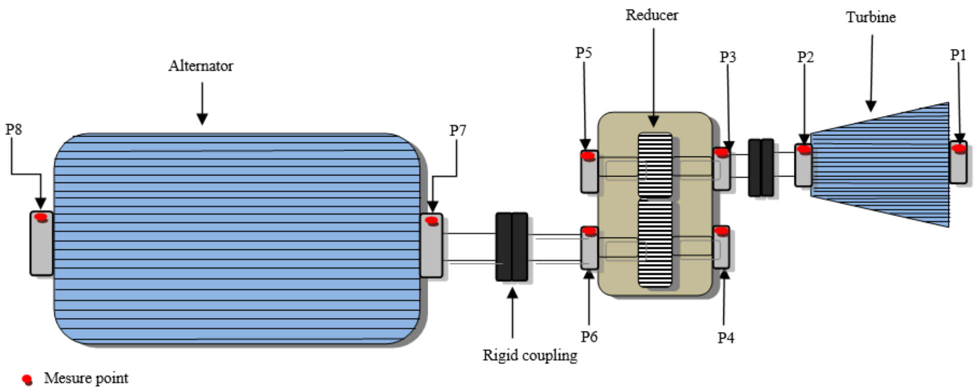

Figure 4. Kinematic diagram of the turbo-alternator GZ1164

Frequency Characteristics of the Reducer

Table 4 shows the technical data of the reducer and its characteristic frequencies. According to the ISO standard 2372 (1974) [34], the studied mechanism is classified in group 3, ie. High power machines (> 300 kW) and mounted on a rigid foundation.

Table 4 Reducer's specifications and its characteristic frequencies

Model / Dimension	GVAB420
nominal power	8300 kW
transmission performance	0.18
input speed (N_1)	9000 RPM say Fr_1=150Hz
output speed (N_2)	1500 RPM say Fr_2=25Hz
Number of pinion's teeth Z_1	41
Number of wheel's teeth Z_2	246
transmission ratio I	6
Meshing frequency $F_m=Fr_1 *Z_1 = Fr_2 *Z_2$	6125 Hz

Equipment Acquisition and Processing of Measurements

Vibration measurements collected on the eight bearings (P1 to P8) of the turbo-alternator in the three directions. Two accelerometers were used; a mono axial accelerometer B&K type 4511-001 and another triaxial one B&K 4524B-001 (Figure 5.a). For the collection and processing of measurements, the analyzer B&K PULSE 16.1 has been used (Figure 5.b).

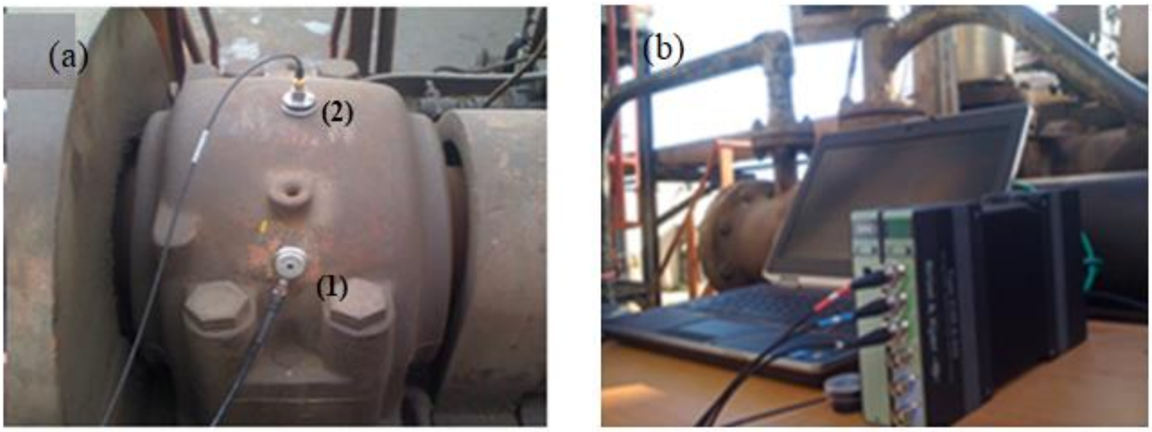

Figure 5. (a) Industrial accelerometer (1) triaxle accelerometer (2), (b) Analyzer pulse 16.1

Data of the Different Sets of Measurements

Because of the importance of the turbo-alternator GZ1164 in the production process of the electrical energy, it requires continuous monitoring. In this study, it was found that this mechanism is simply monitored by periodic tests based on the overall values of the RMS of speed and, time to time, of spectral analysis of speed and of displacement at low and medium frequency respectively [0-200Hz] and [0-1000Hz]. The purpose of the spectral measurements is the detection of any shock into the mechanism, such as shock in gears, shaft friction, bearing wear, etc. Unfortunately, these frequency bands used by the maintenance department of the company do not allow identifying the defects mentioned above, since they are phenomena that mostly occur at high frequency.
Based on RMS overall levels of vibration of measured speed by the company and which show very high levels of vibration up to 14mm / s, we was asked to carry out a global diagnosis of the turbo-alternator. On this basis, it was necessary to launch a series of tests in different frequency bands in order to try to establish a diagnosis of potential defects that cause this increase in the vibration level.

Application of the Cyclostationarity on the Signals Measured on the Reducer's Bearings

Input Bearing of the Reducer

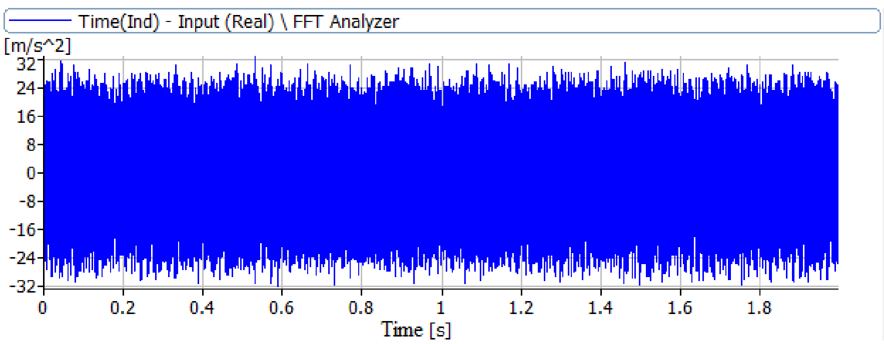

Figure 6. Acceleration signal of the measured on bearing 3 in the frequency band [0-1600 Hz]

The application of the analysis of modulation intensity distribution of the coherence spectral power (with fs=4096 Hz) for each frequency variation $MID_{\Delta f}^{PSCoh}$ of the acceleration signal measured on the bearing 3 figure 6 shows the appearance of two cyclic frequencies: the first $\alpha_2=3\times10^{-3*}f$s ≈12.5 Hz, corresponding to ½ times the rotation frequency of the reducer's output shaft, and the second, with a very important amplitude, at $\alpha_1 = 18.22\times10^{-3*}fs \approx 74.75$ Hz corresponding to ½ times the rotation frequency of the reducer's input shaft, (see figure 7.a). This phenomenon is explained by the presence of an oil whirl defect on the journal bearings see photo 1. We can note also in (figure 7.b) that the use of the integration of the modulation intensity distribution (IMID) allows identifying by a very clear manner the two cyclic frequencies and their modulations.

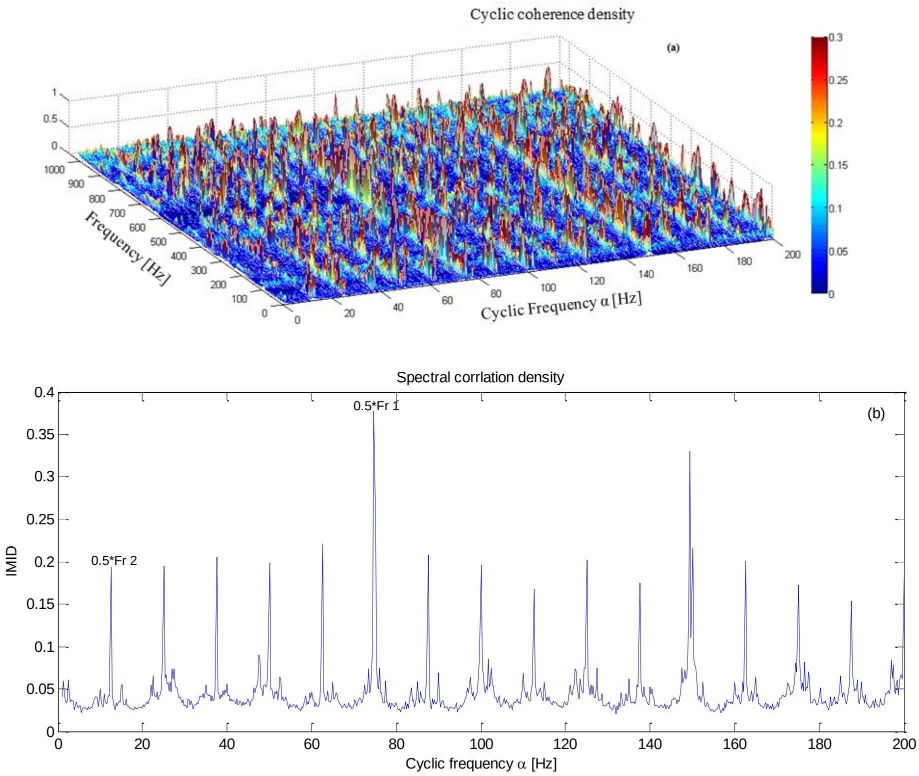

Figure 7. (a) Spectral Correlation (MID) with the corresponding (IMID) (b) of the signal of Fig 6

Output Bearing of the Reducer

The application of the method MID on the signal measured on the output bearing 4 figure 8, gives a spectrum similar to the typological spectrum corresponding to the friction defect in the rotor [35] figure 10. It shows the appearance of a fundamental cyclic frequency $\alpha = 2.4\times10^{-2}$ $fs \approx 12.5$ Hz (with fs= 512 Hz), and its harmonic corresponding to ½ times the rotation frequency of the reducer's output shaft 25 Hz shown on (figure 9. a).
The application of IMID (figure 9. b), allows identifying more clearly the fundamental cyclic frequency α = 12.5 Hz and its harmonic. According to the typological spectrum of the defect (figure 10), the phenomenon presented in (figure 9. b) corresponds to a friction defect in the journal bearing.

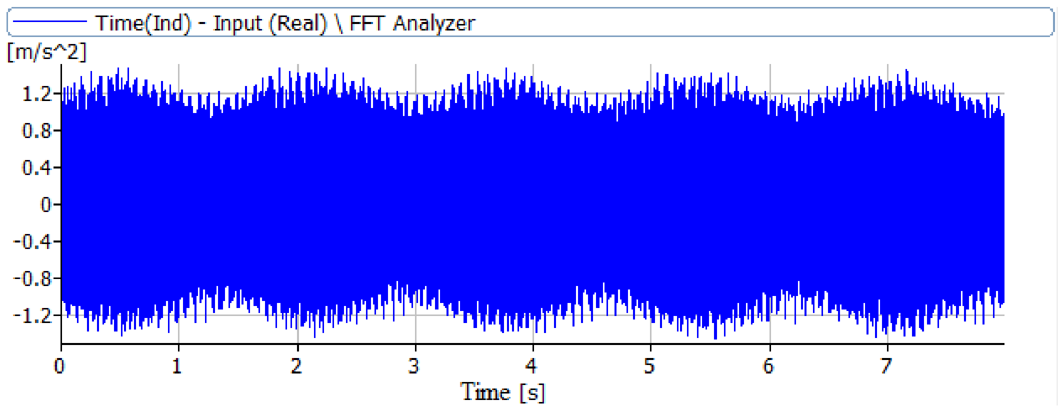

Figure 8. Acceleration signal measured on bearing 4 in the frequency band [0-200 Hz]

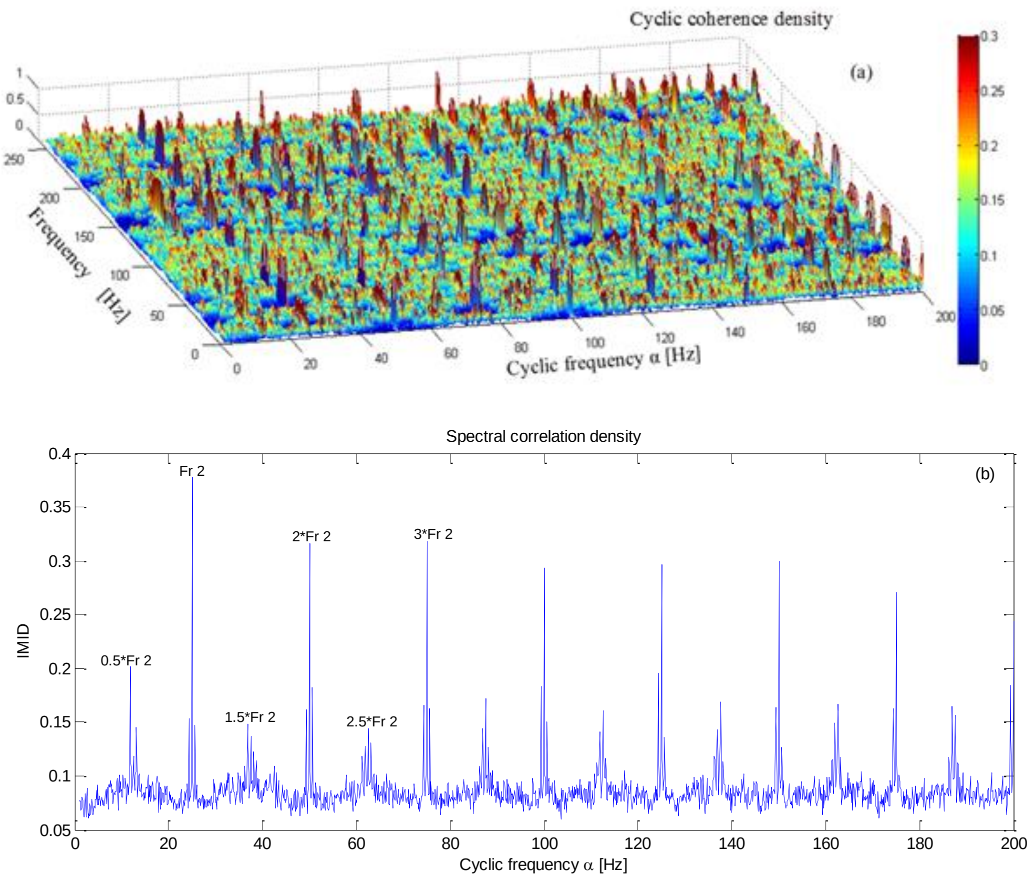

Figure 9. (a) Spectral correlation (MID) with the corresponding (IMID) (b)

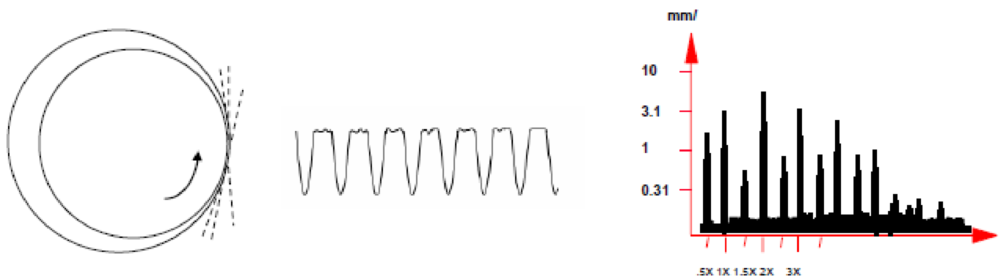

Figure 10. Typological spectrum of oil whirl defect (Brüel & Kjær Vibro) (2005).

Important Remark

The diagnosis presented below is based on the signals measured on March 23th 2022. Department maintenance FERTIAL. A year later, February 04th 2023, a revision was made to the reducer and confirmed the presence of significant wear in the two inputs and output journal bearings, see photo 1, which is confirmed by the present analysis.

Analysis of the Reducer at High Frequency

The treatment of the signals measured on the reducer's bearings at high frequency bands by the spectral analysis allowed highlighting the presence of wear on a large number of teeth of the wheels. We wanted to confirm this result by applying the cyclostationarity. The application of the MID on a signal measured in the frequency band [0-25600Hz] on the bearing 3 of the reducer, see figure 11, shows the appearance of two cyclic frequencies: the first for $\alpha_1=3.81\times10^{-4*}fs\approx25$ Hz corresponding to the rotation frequency of the output shaft and the second one for $\alpha_2=2.28\times10^{-3*}fs \approx150$ Hz (with $fs=65536$ Hz) with very high amplitude corresponding to the rotation frequency of the input shaft. The modulation of these two cyclical frequencies explains the presence of generalized tooth wear on both wheels, (see figure 12. a) and the photo 2. The application of IMID allowed to highlight, by a very clear and visible way, the both cyclic frequencies and its modulations, (see fig 12.b).

There is also the appearance of a carrier frequency at 1196 Hz, which corresponds to the passage frequency of fixed blades sets and the fifth and sixth harmonics at 5980 Hz and 7176 Hz respectively. These two latter frequencies correspond to the passage frequencies of wheel blades 1 and 2 of the turbine, (see figure 12. a).

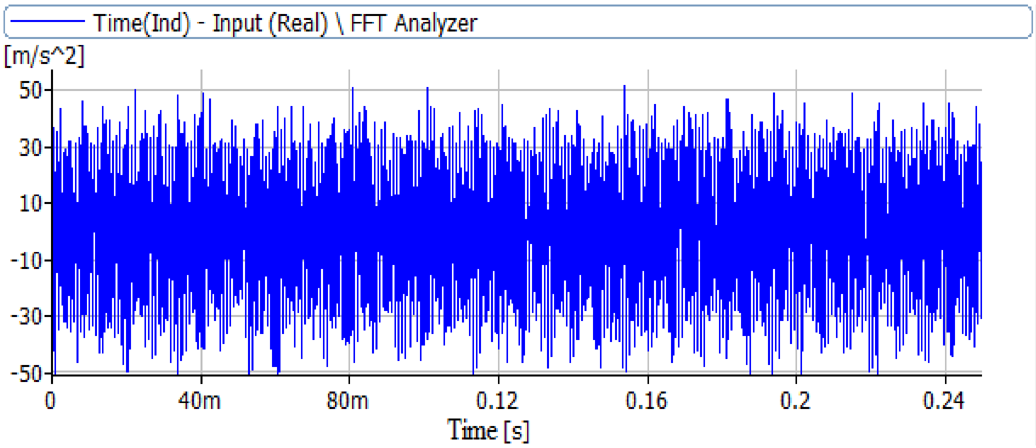

Figure 11. Acceleration of signal measured on the bearing 3 in the frequency band [0-25600 Hz]

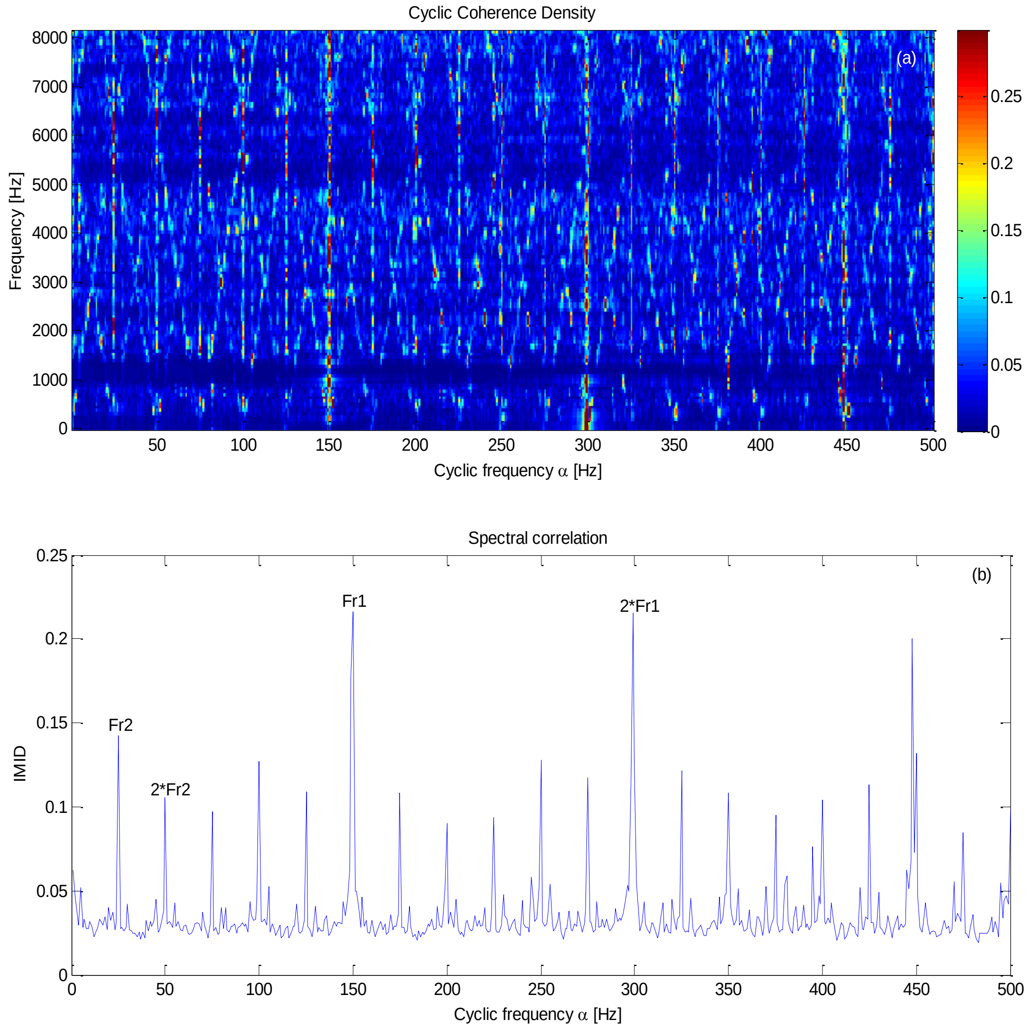

Figure 12. (a) Spectral Correlation (MID) with the corresponding (IMID) (b) of the signal of Fig 14

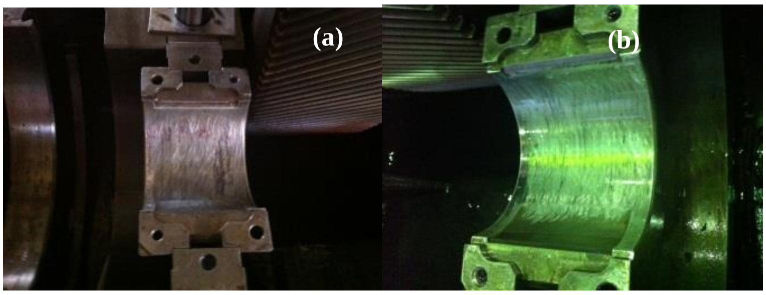

Photo.1 (a) wear in the bearing 3 (b) wear in the bearing 4

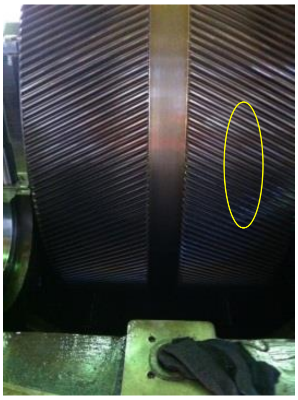

Photo.2 Wear of the gear’s teeth

Conclusion

In this paper, it has been shown that the diagnosis of a turbo-machine working in real conditions by the conventional methods of signal processing has shown limitations in the identification of defects which occur at low frequency. To remedy this problem, we used the method of cyclostationarity that has great power to track modulations present in the non-stationary signals. Actually, spectral analysis allowed identifying wear defects on the teeth of the two reducer’s wheels by the appearance of a large number of peaks corresponding to two rotation frequencies of the input and the output shafts in addition to the meshing frequency and its harmonics. It also showed the clear appearance of the blade passage frequency F_{BP} and the passage frequency of fixed sets’ blades F_{PEFA}. Unfortunately, spectral analysis can’t locate the defects in the journal bearings.

The use of cyclostationarity approach through its two indicators (MID and IMID) has highlighted all existing modulations in measured signals whether in low or high frequency. Both indicators allowed identifying the major problem responsible of the vibration level rise in the reducer, which is the presence of the oil whirl and the shaft friction in the journal bearings. The results we found were confirmed by the Maintenance Department of the company almost a year after that the measurements were carried out.

Recommendations

This work was carried out with the material resources of the Laboratory of Mechanics and Structures, University of 8 May 1945 Guelma, with the support of the Algerian ministry of higher education and scientific research, the delegated ministry for scientific research (via PRFU project code : **A14N01EP230220220002**. The authors would also like to thank the company’s maintenance crew for their assistance.

Scientific Ethics Declaration

I, Mr. Tarek Kebabsa, corresponding author of the paper “Detecting Mixed Gear Faults Using Scalar and Cyclostationary Indicators in an Industrial Settin”, submitted for publication to the in EPSTEM Journal, certify that we have no potential conflict of interest for the mentioned article.Moreover, I certify that the paper follows the ethical rules of good scientific practice mentioned in the “Ethical Responsibilities of Authors” of the journal. Herewith, I confirm, on behalf of all authors, that the information provided is accurate.

Acknowledgements or Notes

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