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WIND TURBINE FAULTS MONITORING BY VIBRATION ANALYSIS

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***Abstract**. The growth of wind farm facilities in the last decade, mainly in northeastern Brazil, has boosted the industrial sector and has been generated the need for maintenance in wind turbines. The present work shows a model of rotary machine that resembles the wind turbines, to develop knowledge applicable directly to the predictive maintenance of these machines. This study considers the dynamics and vibration analysis in rotors and the application of unconventional techniques (FullSpectrum, Orbit, SPM and Envelope) to analyze the data obtained from experimental tests to identify components failures. The results show that it is possible to monitor these rotating machines and obtain an accurate diagnosis of their operating conditions.*

***Keywords:** Wind Energy, Rotor Dynamics, FullSpectrum, Orbit, SPM.*

1. INTRODUCTION

A generation of wind power electricity has been expanding in the country in recent years, as can be seen in Fig.1 (ONS, 2017). This fact was observed with the increase in investment in this area between 2009 and 2016 (ABEEOLICA, 2015) and it allowed Brazil to rank in the top ten wind generators in the world, ranked ninth according to Global Wind Energy Council (GWEC, 2016). With this growth comes the expansion of wind farms, the need to maintain their turbines and further monitoring of possible problems in real time.

An important moment of the country's energy grid diversification was in the 2015 drought, that the hydroelectric plants did not have much water resources, resulting in a need for thermoelectric plants and wind farms to supply energy demand in the country (EBC, 2017). In 2016, the current wind farms installed in the Brazilian Northeast are responsible for supplying 30 percent of the energy demand (JONG P., et al., 2017) this shows the importance of wind energy growth in the region.

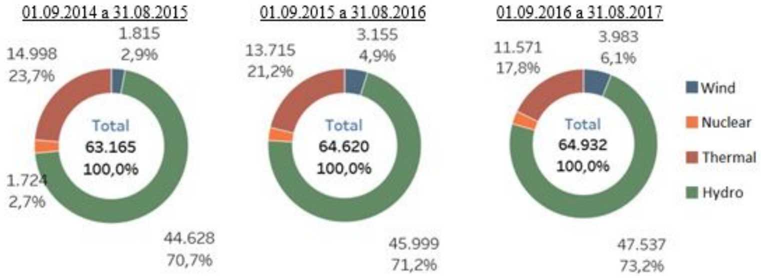


Figure 1. History of Operation - Power Generation about 01.09.2014 to 31.08.2017. Source: ONS.

With the growth of wind turbine installations, new problems and / or opportunities arise, such as the maintenance of wind turbines. Most of the large installations in Brazil are made on the so-called wind farms and they have dozens or hundreds of wind turbines that will need verification, evaluation and, finally, maintenance of their equipment. However, it is known how expensive the time to climb each turbine is to perform a check and therefore one should analyze how best to do so. One way to reduce the time spent with this type of work is to develop a monitoring system capable of collecting relevant data to supervise whether the equipment is operating correctly and, together with that, the choice of appropriately maintenance type.

Maintenance can be divided into the following main types: corrective, preventive, predictive and TPM (Total Productive Maintenance) as described by Viana (VIANA, 2002). Corrective maintenance may occur when the equipment is out of order or the equipment fails during operation. Preventive maintenance makes the change in a planned way from the period or cycle the equipment or component was used. Predictive maintenance, on the other hand, monitors the components of the equipment in order to perform the change only when the equipment shows signs of wear (VIANA, 2002). The purpose of this work is to show that predictive maintenance can be a good option for wind turbines.

The turbines are equipment that use the same elements of most general rotating machines, such as rolling bearings, gears, shaft and rotor. To determine the maintenance type (for example, corrective, preventive and predictive) with the best cost benefit of these devices, it is necessary to compare the risk and cost of an equipment failure to the cost of maintenance type implemented. To do this comparison it's necessary to know the main features of each maintenance type. Corrective maintenance is also known as "run to failure maintenance". To leave an equipment running until failure often generates collateral damages. For example, a simple bearing change is not done before failure, but catastrophic damage occurs when this bearing failure happens, then the shaft becomes loose and immediately starts to unbalance in the end. released parts. In the other side the preventive maintenance, also called the "calendar maintenance", where the big challenge is to know the exactly life expectancy of the components. Normally the manufactures already given this time but in reality it depends on how the machine is being used. Change a component ahead of time is to lose money, because maybe this component would work for more thousands hours before failure. But, in other hand, if the correct time interval passes, a failure happen and, as in the corrective maintenance, a collateral damage can also occurs. However, a plant operating with a predictive mode of maintenance, through the condition monitoring systems it is possible to evaluate the components condition over the time, create trend graphs and know the exactly time to plan the maintenance repair.

This paper presents a dynamic evaluation of a horizontal rotor in balance, making analogy to horizontal axis wind turbines (HAWT), as shown in Figure 2, and using conventional and non-conventional signal processing techniques to identify faults in its mechanical components. These techniques are currently being used as predictive maintenance tools for monitoring and rotary machines diagnostics (FARIAS, 2016).

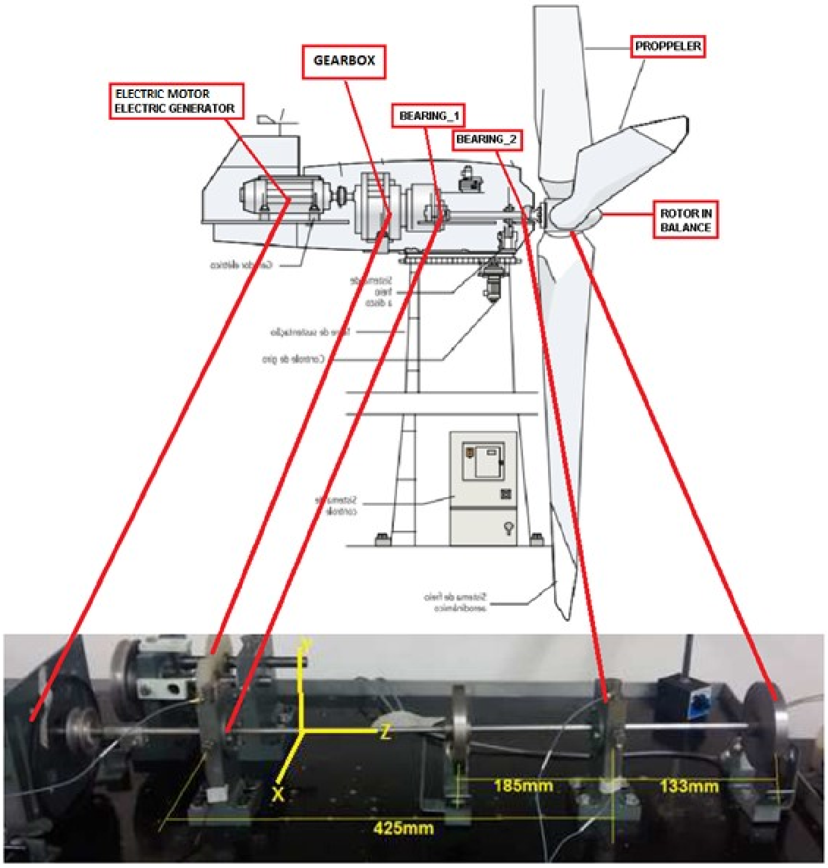


Figure 2. Relation between wind turbine and experimental horizontal rotor rig.

2. EXPERIMENTAL PROCEDURE

It begins the study through of the analysis of a rotor in a balance, where the objective is to identify the vibration modes of this system and to analyze the dynamic effects for a further rotor analysis with three propeller blades analogy to the rotor of a wind turbine. To do this, initially was used a solid disk positioned at the free end of a slender shaft supported by rolling bearings.

The dynamics study allows identifying the system characteristics and the most requested parts, which consequently are more susceptible to failures (MOBIUS, 2012).

Purpose defects (imbalance, bearing damage, and gear damage) are applied to experimental rig elements. Vibration tests are performed under the conditions "with" and "without" defects and the results are compared.

The results are analyzed using different vibration analysis techniques such as FullSpectrum, SPM and orbit, as well as the Campbell diagram.

3. RESULTS AND DISCUSSION

3.1 Rotor dynamic behavior

The rotor analysis begins with simulation results according model shown in Figure 3. The goal of this simulation is to identify the gyroscopic effect in the behavior of system according to position of main disc related with bearing 2.The Campbell diagram (Figure 4) shows the forward modes, reverse modes and the critical speed (FARIAS, 2016), and the gyroscopic effect can be observed trough angle between the curve of forward mode and the reverse mode. The greater the angle between the pairs of curves, the greater the gyroscopic effect. For the case under consideration, the effect is greater in modes 5 and 6 (mode numbers are indicated on the right side of the graph in the direction of each curve). It is observed that for modes 1 and 2 the curves are almost superimposed, so the effect is negligible. Between modes 3 and 4, it is possible to observe a greater distance between the curves, characterizing greater presence of the effect when compared with the first pair.

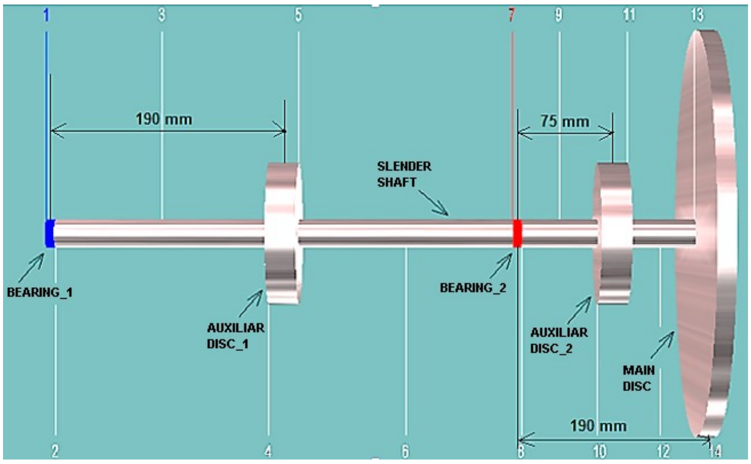


Figure 3. Rotor model.

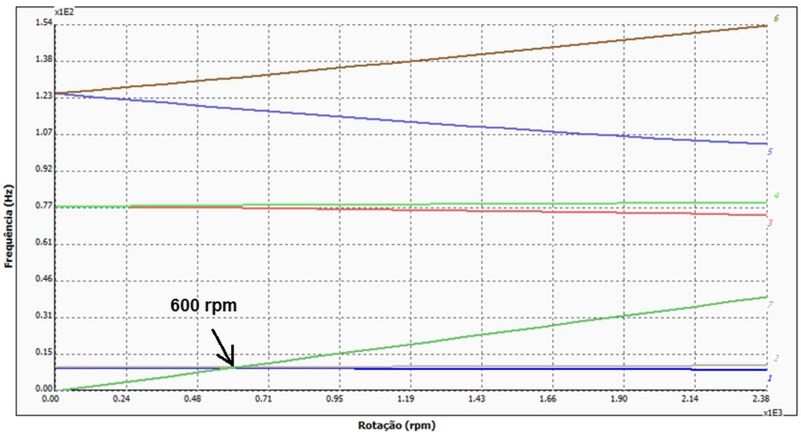


Figure 4. Campbell diagram.

The FullSpectrum is considered as a spectrum of an orbit, that is used in combination with Campbell diagram to evaluate forwards and reverse vibration modes. The Fullspectrum and orbit analysis was done by the matrix of experimental tests results, where the column of matrix represents the different points of measurement and the lines the point of impact excitation. The points of excitation and measure are the discs 1, 2 and main disc. The excitation point has influence on the amplitude of the mode in the spectro, but this can only be clearly observed for the first two modes. Modes 3, 4, 5 and 6 only appeared in some of the tests because they are more difficult to obtain experimentally.

The correspondent orbit matrix shows the trajectory of the center of the main disc (Figure 5).

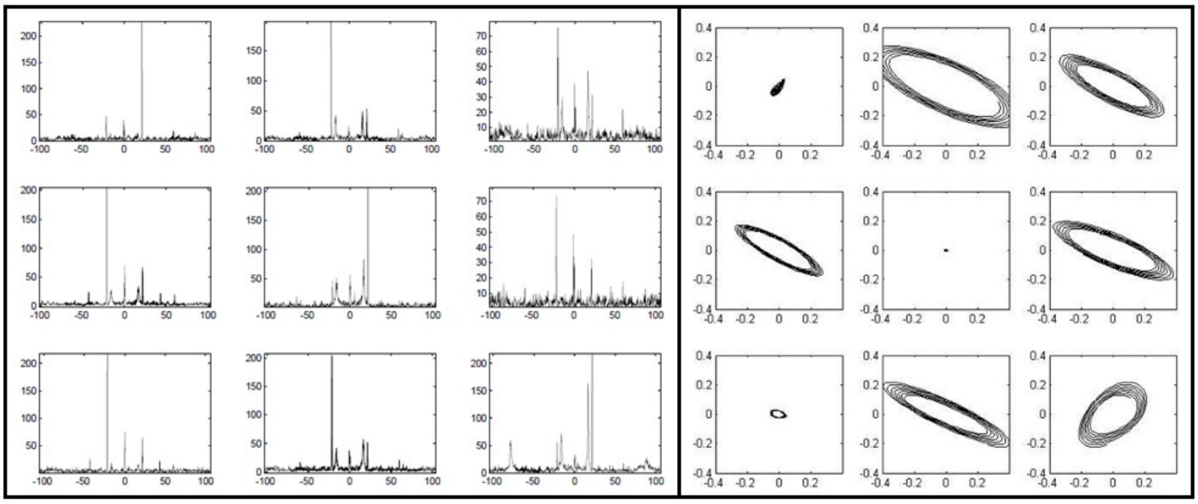


Figure 5. Matrix of Fullspectrum (scale 200 µm x 100 Hz) and Matrix of Orbit - (scale de 0,4 x 0,4 µm). Tests with 1300 rpm.

3.2 Bearing damaged tests:

For the bearing condition evaluation it was used the SPM HD® technique that is a patented further development of the Shock Pulse Method (SPM) used for fast, easy and reliable diagnosis of the operating condition of rolling element bearings. The types of mechanical faults which can occur on a given machine are well known to maintenance personnel. What is needed is a straightforward message that a fault is developing, and an indication of its severity. In that way, according (GWEC, 2016; LEE, 2015 and LI, 2008) the SPM technique compared with other vibration techniques is more efficient to give pre-warning about bearing condition and is able to detect surface and subsurface bearing damages.

Good vibration results was found in tests R1 (perfect bearing) and R2 (damaged bearing), regarding to the bearing condition monitoring.

For these testing of bearing defects, a rigid shaft configuration (smaller gear shaft) and 16 mm internal diameter bearing bearings were used (due to the difficulty of generating defects in small bearings). In order to realize the rotation and torque of the electric motor for this set of bearings and rigid shaft, the pulley and belt system with a gear ratio of 2.47: 1 of the motor for the test shaft was used.

The bearing used was FAG 6203 and the defect was generated in the outer race of the bearing with a disc of 1 mm of abrasive material mounted on a hand drill. The defect was produced until generate a small hole inside of bearing.

Bearing installation (good and bad) was done on the M4 bearing (figure 8a) due to system configuration. Although a device was not designed to generate load on the system, a greater tension was generated in the belt so that the bearing of the M3 bearing suffered a slightly higher load and a residual load was transmitted to the test bearing installed in the support M4. This load generated by the belt tension was observed by motor speed difference of up to 20 RPM between normal and excessive belt tension conditions.

The tests R1 and R2 were performed at two different speeds: 400 RPM and 1237.5 RPM in order to evaluate the difference between the results obtained for each speed, knowing that the impact intensity of the bearing components passing through the defect increases proportionally according to the speed of rotation of the machine. These velocities were measured directly on the test axis through the tachometer instrument.

Figure 6 shows the damaged bearing and Figure 7 shows tests results.

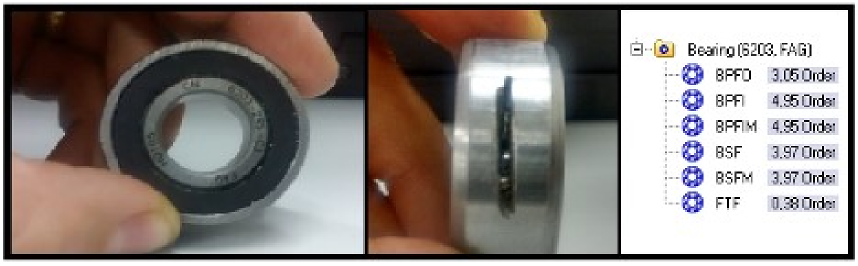


Figure 6. Bearing FAG 6203 with generated defect / defect frequencies.

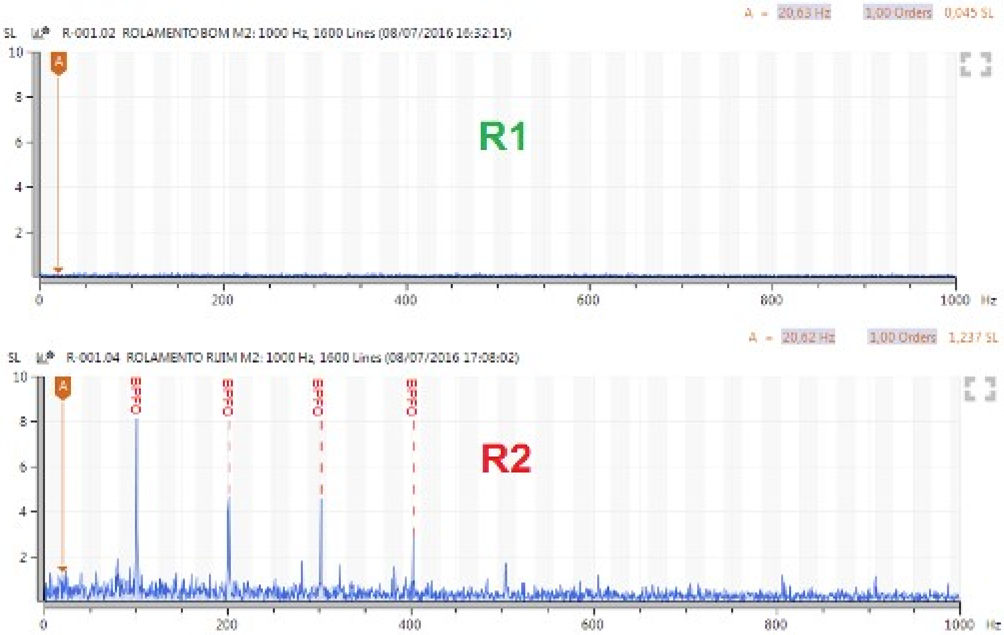


Figure 7. Bearing damage tests R1 and R2 (1237,5 RPM).

The SPM technique stood out as the best to identify this type of defect, presenting very good performance in the tests of 400 RPM and 1237.5 RPM. Where the difference of Shock pulse levels between the tests with good bearing and with damaged bearing are sharply differentiated. In addition, the sensitivity of this technique allows for easy observation of the bearing defect frequencies, as in figure 7, where it was detected at peaks directly related to the bearing defect frequency of Ball Pass Frequency Outer ring (BPFO).

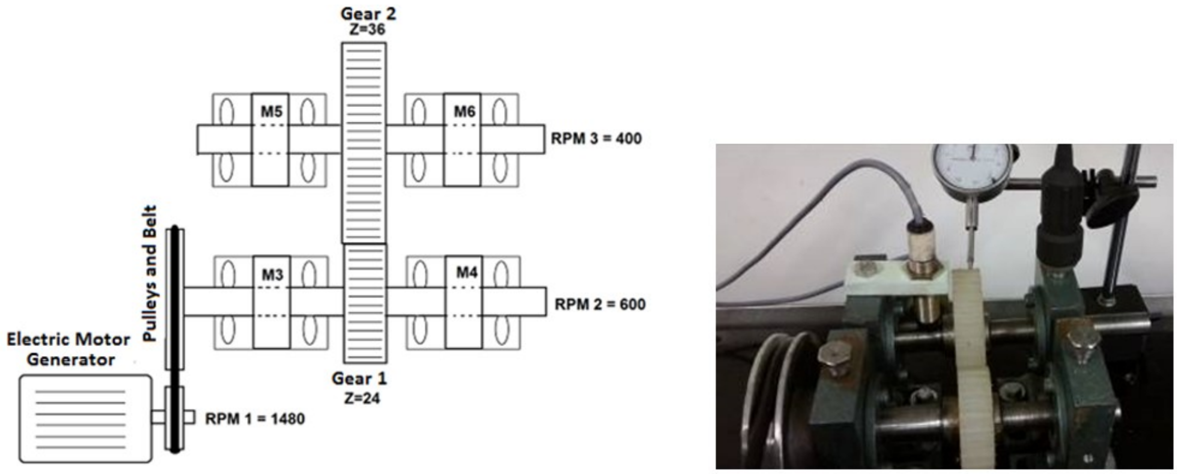
The Envelope technique presented good results between the tests with good bearing and with bearing damaged for the highest rotation. However the spectra of this technique did not allow the identification of frequencies of bearing defects, it is only possible to observe an increase of the background noise, but not specific frequencies of the bearing.

The techniques: Full spectrum, orbit, and phase analysis did not present good results to identify this type of defect. These techniques did not present significant differences between the R1 and R2 tests.

Conventional techniques (global RMS value, frequency spectrum and time domain signal) showed good results, with significant differences between the R1 and R2 tests. However, despite the increase in amplitude of the vibration, it is not possible to distinguish frequencies related to defects in the bearing.

3.3 Gear damaged tests:

Back lash changes and gear tooth damages were done on the gear set to simulated gearbox failure (see Figures 8, 9 and 10), and it was also well detected by vibration analysis (GENTA, 2005).



(a) (b)

Figure 8. Gear damaged test. (a) Gear set and Motor / Generator; (b) Back lash measurements.

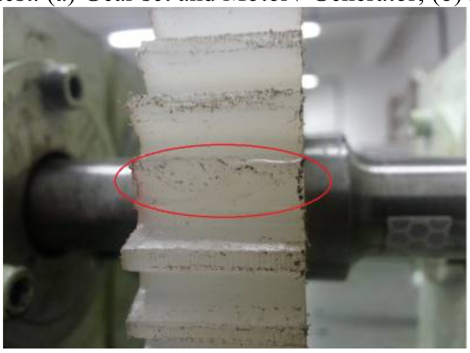


Figure 9. Gear damage – Broken tooth (gear nº2).

In order to perform the test of gear defects, the experimental rig was adjusted to the configuration shown in figure 8, consisting of: electric motor, pulley and belt transmission for shaft 2, Bearing supports M3, M4, M5, M6 and gears 1 and 2 that make the transmission of rotation and torque from shaft 2 to shaft 3. The ratio of speeds between the motor and shaft 2 (pulleys) is i1 = 2.47 and from shaft 2 to shaft 3 (gears) is i2 = 1.5 (according to the number of teeth of each gear Z1 = 24 and Z2 = 36). System rotation speeds during testing are set forth in Figure 8 where the motor was set at 1480 RPM, shaft 2 = 600 RPM and shaft 3 = 400 RPM. These speeds were selected because they are in a good range to capture the signals considering the range of the frequencies involved. Two identical gears (gear nº2) were manufactured, but one was used to add the defect to a tooth.

With the gear 2 tooth broken running in 400 RPM it is necessary to appear a peak related to 6,6 Hz in the results due to the passing of damaged tooth for each revolution of shaft 3. This symptom was clear seen by the conventional techniques as in time signal bellow, where in 1 second is possible to count 6,6 peaks in the graph.

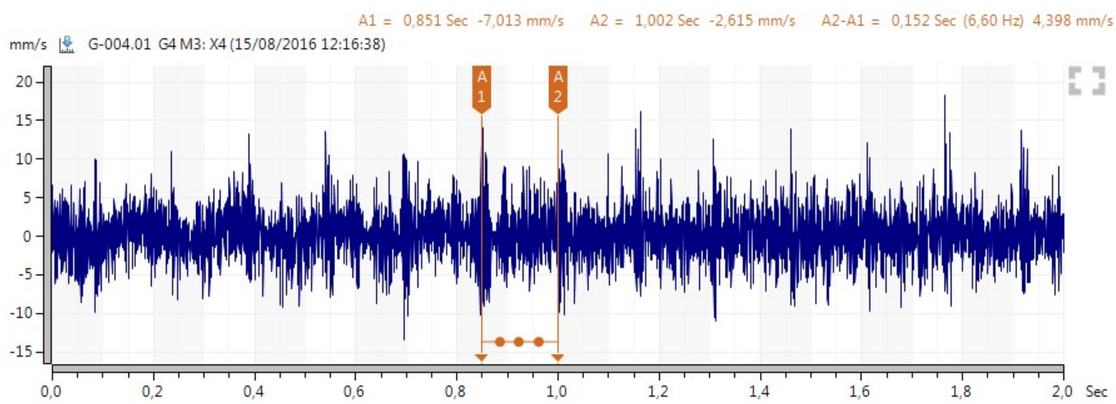


Figure 10 – Gear nº2 tooth broken test – Time signal graph – Conventional technique.

By the envelope technique it was possible also to identify an eccentricity of gear nº1 related to the frequency of 10 Hz (each revolution of gear Nº1) and the gear mesh frequency of 240 Hz related to the shaft speed x the number of gear teeth.

The periodic monitoring of gear mesh frequency in the same operational condition (RPM, load, temperature, etc.) is powerful tool to evaluate gear wear and damage. Trend graphs can be generated to help in the decision of correct maintenance time.

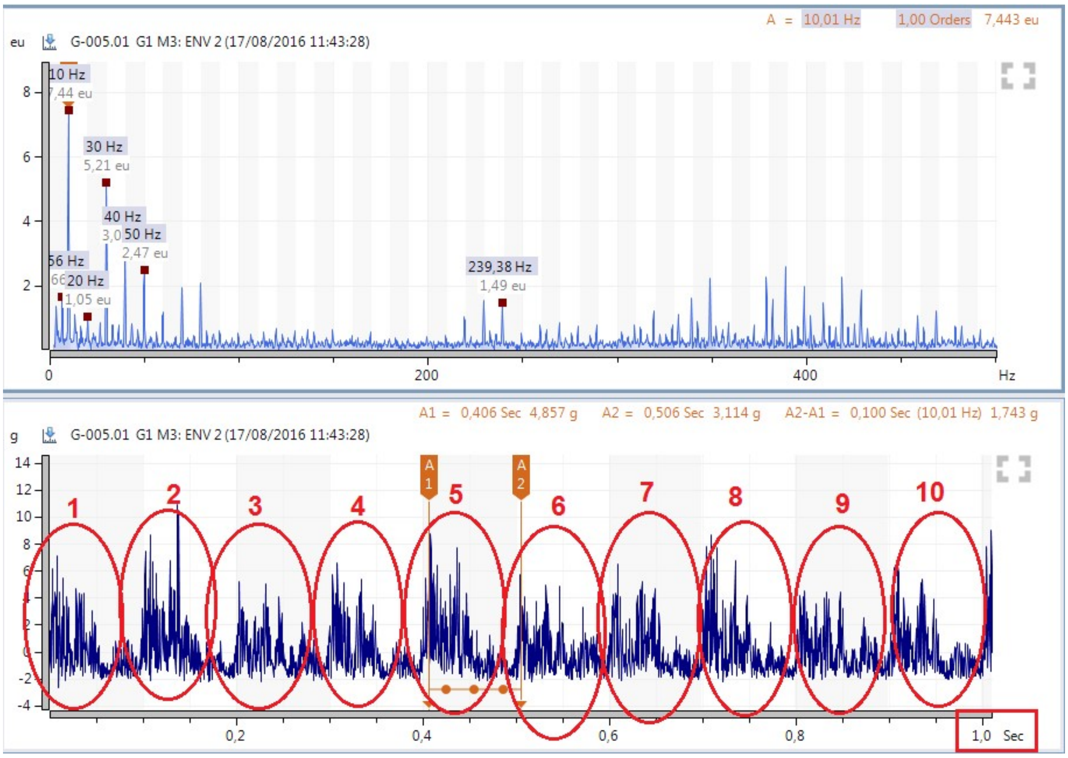


Figure 11. Envelope technique – Gear nº1 eccentricity and gear mesh detection.

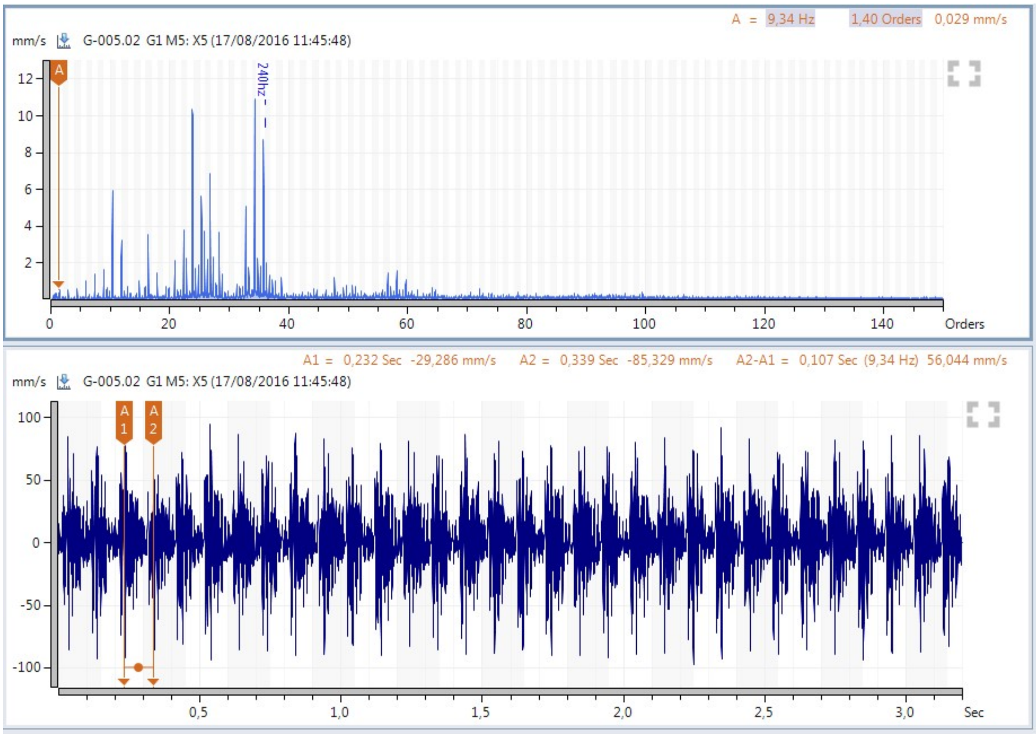


Figure 12. Conventional technique – Gear nº1 eccentricity and gear mesh detection

4. CONCLUSIONS

Regarding to the horizontal rotor dynamic behavior, it can be concluded that the tests have expected results and can help to evaluate the gyroscopic effect and resonance conditions.

The defect tests done shown that vibration analysis is a great tool to evaluate the machine condition, detecting fault and defect prognostic before machine failure. Known that mechanisms tested on the experimental rig are also present in a real wind turbine, it can be conclude that using a predictive maintenance mode with vibration analysis can give a optimum reliability to this kind of industrial plant. Online vibration monitoring systems can be installed in the wind turbines due to the difficulties of access to do periodic inspections manually. As a way to upgrade this research, it is planned in future evaluations perform Structural Health Monitoring (SHM) with more emphasis, to case of machines that run almost without stopping and damages due to fatigue must be consider.

It is considered that the horizontal rotor of the experimental rig has similar characteristics of a real rotating machine and it be used for research and development in this field of mechanical engineering.

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