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ANALYSIS, EVALUATION AND MONITORING OF THE CHARACTERISTIC FREQUENCIES OF PNEUMATIC DRIVE UNIT AND ITS BEARING THROUGH THEIR CORRESPONDING FREQUENCY SPECTRA AND SPECTRAL DENSITY

ANALIZA, OCENA I MONITOROWANIE CZĘSTOTLIWOŚCI CHARAKTERYSTYCZNYCH PNEUMATYCZNEGO ZESPOŁU NAPĘDOWEGO I WCHODZĄCEGO W JEGO SKŁAD ŁOŻYSKA NA PODSTAWIE WIDM CZĘSTOTLIWOŚCI ORAZ GĘSTOŚCI WIDMOWEJ

This article shows the results of the study of the characteristic frequencies of pneumatic drive equipment and its suspension bearing. The analysis approaches one of the most important requirements of the industrial sector, which seeks to be recognised by the efficiency and performance of its equipment when compared to its coming economic competitors. For data collection and we have followed the ISO 10816 standards, thus using the values of speed in RMS, aiming to reduce the masking of these signals that occurs depending on whether they are high or low frequencies. The study will respond to one of the most important requirements found in the predictive and preventive control of industrial sites. The problem of the predictive systems of maintenance of equipment with bearings lies in the number of monitoring and analysis points that generate a high cost in time and human resources. The aim will be to determine which of all the study frequencies is the most significant and in which position and measurement axis has the biggest impact. To do this, we will analyse the rotation frequency of the blowing machine, the resulting frequency of all the frequencies, the frequency of the impulsion blades and finally the frequency of the bearing. The study would be able to predict when our equipment is going to suffer a failure, reducing the control points and the cost.

***Keywords**: vibration, bearing, diagnostics, vibroacoustics.*

W artykule przedstawiono wyniki badań częstotliwości charakterystycznych napędu pneumatycznego i wchodzącego w jego skład łożyska zawieszenia. Analiza przybliża jedno z najważniejszych wymagań sektora przemysłowego, w którym dąży się do tego by wyróżniać się na tle konkurencji sprawnością i wydajnością urządzeń. Przy zbieraniu danych postępowaliśmy zgodnie ze normą ISO 10816, wykorzystując średnie prędkości kwadratowe, co pozwoliło zmniejszyć maskowanie sygnałów, które występuje w zależności od tego, czy mamy do czynienia z wysokimi czy niskimi częstotliwościami. Badanie stanowi odpowiedź na jeden z najważniejszych wymogów w zakresie kontroli predykcyjnej i prewencyjnej obiektów przemysłowych. Problemem systemów konserwacji predykcyjnej sprzętu, w którego skład wchodzą łożyska jest duża ilość punktów kontrolnych, które generują wysokie koszty jeśli chodzi o czas i zasoby ludzkie. Celem pracy było określenie, które ze wszystkich badanych częstotliwości są najistotniejsze oraz dla których częstotliwości pozycja i oś pomiaru mają największe znaczenie. W tym celu przeanalizowano częstotliwość obrotową analizowanej dmuchawy, częstotliwość wynikową wszystkich częstotliwości, częstotliwość łopatek oraz częstotliwość łożyska.

***Slowa kluczowe**: drgania, łożysko, diagnostyka, wibroakustyka.*

1. Introduction

In an increasingly globalised world, cost reduction is a strategic and competitive advantage. For this reason, a demand has arisen in the industrial sector for automated and computerised procedures to efficiently control and manage systems based on preventive and predictive maintenance, against corrective actions that generate higher costs in terms of time, resources, material and production losses.

Rotating elements, including bearings, are one of the main causes of shutdown or failure in production systems. Almost 40% of all shutdowns are caused by these rotating elements [11], hence the importance of incorporating control and monitoring systems to predict their service life and evaluate their condition. A bearing's behaviour depends on many variables such as lubrication operations [20], the strengths of excessive or external dynamic loads, design errors, pollution, handling defects and transport shocks.

The conventional procedure of failure diagnosis and analysis of bearings is performed by means of vibration signals, which have a fundamental implication in this type of maintenance. The information provided by vibrations in the frequency domain gives us the keys to determine possible erratic conditions in rotating elements. The most significant ones found in rotating systems are unbalance, misalignment, eccentricity, bent shafts, cavitation in turbo machinery and bearing failures.

The main purpose of vibration monitoring and control is to determine one of the four stages of failure in which the bearing is. It should be noted that bearings do not show any signs of breakage or anomalies until they reach one fifth of their service life [17]. In the first stage, erratic behaviours are generated in elements whose working frequency is mainly higher than 5kHz. At this stage it is not necessary to replace the bearing and although cracks occur, they are not noticeable to the human eye.

The second stage starts when average values are reached. Then, cracks appear, they are now visible to the human eye can also be perceived acoustically due to the generation of small disturbances. The third and fourth stages are the most important ones. Cracks advance, increasing the temperature of elements, causing their rapid advance towards breakage. At this stage, acoustic perception is very severe and easily perceived to the human ear. The fourth stage involves a corrective action, generating important actions caused by the breakage of components and according to the moment of shutdown of the whole equipment until the total breakage [2].

The first research works based on the use of vibration as a preventive maintenance method for rotating equipment were focused on improving the quality of vibration signal detection systems and their processing to improve the analysis of frequency spectra according to the type of machines (turbines, pumps, fans, gearboxes, compressors) and the type of defects (loads, tensions, misalignment, unbalance, lubrication) [6].

The main goal of new working trends is the prediction of failures as a fundamental basis for preventive maintenance [13]. The acoustic analyses for the characterisation and assessment of the vibrations generated are particularly remarkable. The advantage of these techniques is that they are non-intrusive, since they do not disturb the normal operation of mechanical systems, but daily monitoring of equipment, with the consequent need for human and material resources is needed. This factor conditions the efficiency and productivity of many industries subject to tight profit margins [4].

The most important lines of research are the behaviour of bearings, their friction, the internal stresses and how the friction of their own components affects and the dissipation of energy through the roughness. This study also includes the generation of new designs with gas application to reduce the tangential effects of centrifugal forces for rotating equipment. It is important to note that, all the tests have been carried out in the laboratory without being able to study the evolution over the years in real operating conditions [23].

For this reason, this work evaluates the characteristic operating frequencies of a set of bearings in a blower of a real industrial plant, through the spectral analysis of the vibrations generated during 15 years of operation. Most of the studies carried out on neural networks on the behaviour and durability or useful life of rotation systems, including bearings, have been based on experimental work in the laboratory, ending in their breakage [19, 8].

Other experimental work in laboratory using Support Vector Machine (SVM) focuses on the diagnosis of ball bearing failures by processing the signal through the wavelet transform [12].

2. Material and methods

2.1. Equipment under study

The drive equipment under study is a blower that belongs to the production system of a Spanish industrial plant whose main objective is safety, and of course operational and energy efficiency and environmental protection [3]. Safety has led this company to develop a highly specialized equipment control and monitoring program, seeking for durability and fault prediction. To carry out this process, they have instruments and specialized personnel that sample the equipment daily in different positions, studying its variations and trying to predict its failure. That is why they have facilitated the 15-year sampling of a blower, to try to find a relationship or linearity between the different parameters that affect the operation of the bearings.

The equipment under study is a blower designed for pumping fluids. The equipment consists of two different parts, the impulsion system or motor frame size 400 of 3190 kg, with dimensions 1900 x 910 mm, fixed to a steel bench of 2100 x 3900 x 300 mm. At the same time, it serves as support and connection to the system, formed by the drive unit and a fan of 2075 mm in diameter, all of which is connected by a 95 mm and 3900 mm shaft. A coupler drives the fan blades, so the shaft is divided into two sections, the 1900 mm motor part and the 2000 mm fan part. This section is supported by three bearings: SKF6322, FAGNU322 and SKFNU322 [2].

This equipment, like most modern machines operating at high speeds and loads, has a bearing-supported shaft as a mechanical transmission element. Bearings are the first elements to cause system failure. That is why it is so important to have a proper design, an appropriate selection of bearings and a comprehensive testing plan. The vibration signals from bearings are very complex, because they hide and couple together, and are difficult to analyse under normal and extreme operating conditions.

Many studies of bearing vibration analysis have been conducted, trying to respond to this need of the industrial sector by means of mathematical analysis of vibrations, generating models to predict defects, but none have been conclusive [9]. The problem is that the system is in a dynamic state of motion. This energy is partly absorbed by the material itself, so the rigidity of the rolling element is a fundamental variable for the study of faults.

In these models, the vibration signal of a defect is established through the series of pulses. Hitting the moving parts on the coincident surface, generating resonances in the bearing and housing structures, generates these. This knocking is caused by the system being in continuous motion, when it is in stable operation and rotating at constant speeds. These pulses generate periodic disturbances and can thus determine their frequency, depending on their position [18]. The geometry (rolling elements, cage and races) and running speed of bearings precisely determine their characteristic frequencies [15, 9, 1].

Table 1. Nominal characteristics of the bearing supporting the blower shaft under study

Bearing FAGNU322	
Basic dynamic load (C)	415 kN
Basic static load (C0)	475 kN
Fatigue limit load (Pu)	61 kN
Reference speed	3000 r/min
Speed limit	3000 r/min
Calculation factor (kr)	0.15
Geometric characteristics	d = 110 mm; D = 240 mm; B = 50 mm; $d_1 \approx$ 200 mm; F = 143 mm; $r_{1,2}$ min. 3 mm

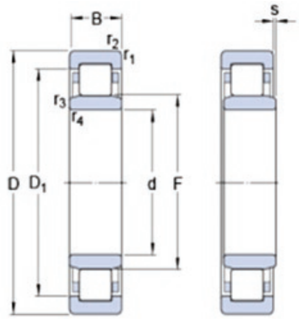

Table 2. Characteristic frequencies of the bearing under study (Hz)

Frequencies	FAGNU322
FTF	508.81
BSF	3938.1
2BSF	7876.1
BPOR	7911.9
BPIR	11452
Speed	1490
Blades Fpa	13410

Each bearing component has a unique breaking frequency and four fundamental frequencies are defined [4], according to the place where the defect occurs. So, if it is generated in the balls we will have the BSF; if it is by the frequency of cage rotation the FTF; if it is by the passage in the inner race, the BPIR and if it is by the outer race BPOR [21].

The equipment under study consists of three different types of bearings. The nominal characteristics of FAGNU322 bearing are shown in Table 1. The fundamental frequencies of the blower shaft bearing are as shown in Table 2. They are compliant with the instructions of ISO 10816.3 [22].

In this study, we have considered not only the characteristic breaking frequencies of the different parts of a bearing, but also the rotation frequency of the system, which in our case has a continuous motion regime without accelerations, generating a stable working motion at a constant angular speed. Another factor to be noted is the characteristic frequency of the driven elements, in this case, the air discharge blades.

2.2. Vibration analysis

Predictive analysis of rotating equipment is possible by transforming the signals caused by its own vibrations. These signals are processed in the frequency domain using the FFT (Fast Fourier Transform) algorithm to relate the shape of the waves generated by vibration to the frequency generating its own vibratory spectrum. Decomposing the vibration, or rather, the wave generated into simple waves according to its characteristic frequency and representing each of them independently, we obtain the representation in the frequency domain of this disturbance; this representation is called spectrum or spectral density.

Signal analysis in the frequency domain creates major impediments to perform a reliable signal analysis. Therefore, the concept of signal energy through the Power Spectral Density (PSD) arises, defined as the amount of energy contained in each characteristic frequency and has the following expression:

$$PSD\left(x(i)\right)=\frac{1}{T}\Sigma\left(x(i)\right)^2\Delta T \qquad (7)$$

where the x (i) sequence for ΔT is the average power of the interval or the sum of the powers of each of the components, independently. The simplest way to estimate this power density is to pass the data from a defined sample window. The defined window is called the periodogram and the advantage of this method is that it can detect hidden frequencies [7].

There are other methods that improve the accuracy of the analysis, one of which is the Hilber HT (Hilbert Transform) transformation that analyses the signal envelope. This allows obtaining an improved PSD and facilitates the sampling in processes that take place at low frequency, modulating precisely the primary signal

The method followed in this work was to transform two functions s (t) and 1/(лt) into a third one, as expressed below:

$$\bar{x}(t)=\frac{1}{\pi}\int_{-\infty}^{\infty}x(u)\frac{1}{t-u}du \qquad (8)$$

This method can amplify events occurring at low frequencies, through the components that form the signal envelope, but although the system provides improvements, it also has deficiencies since it is very sensitive to noise [14]. The problem was solved with the incorporation of a data treatment process, providing information on the main study variables in signal processing, such as time and frequency [16]. This process is known as Wavelet Transformation (WT) [10].

3. Data acquisition, processing and analysis

The Entek IRD analyser and the OdysseyEmonitor software have been used to obtain vibration data from the blower under study from May 2000 to April 2015. The hardware consists of a 16-channel interface with a nominal voltage of 24 V at 3.6 mA DC, a multiplexer with 4 high-pass filters, an integrator to generate speed-dependent acceleration signals and, similarly, speed-dependent displacement signals. Another important element is the antialiasing filter that eliminates high frequencies, avoiding the appearance of aliasing in the received signal and, finally, the digital analogue converter that captures up to 51.2 kHz with a resolution of 16 bits. The OdysseyEmonitor Software allows you to generate trend graphs of vibration energy levels and their characteristic frequencies.

According to the risk levels of the ISO 10816-3 standard and the critical operating levels, the equipment under study has a flexible shaft, power higher than 300 kW and a speed of 1,500 rpm, so it would have an operating level between 0.18 to 11 RMS (mm/s), with the maximum critical value being 7.1 RMS, where corrective maintenance must be applied.

The equipment consists of two measuring points identified as 3 and 4; at each point, sampling is performed at the three coordinates of the X, Y, and Z space, identified as shown in (Table 3).

An accelerometer with a sensitivity level of 100 mV/g and a translator with contact displacement were used, allowing a frequency range of 10000 Hz. The fixing system will be a magnet and a fast mounting base, which allows reaching faults in a higher frequency range, improving its range and obtaining measurement sensitivity between 0-300 Hz. The monitoring is done with a Hanning window, 3,200 lines, and a speed range of 60,000 and 300,000 Hz for acceleration, obtaining a bandwidth of 28 CPM for speed and 140 CPM for acceleration and a resolution of 18 CPM for speed and 93 CPM for acceleration.

For faults diagnosis, the ISO 10816 standard determines that the evaluation is performed using the spectral level based on speed. This presents greater uniformity at both low and high frequencies.

Table 3. Sampling positions of the FRL104 blower under study

Identification	Number	Site
Sopl. LA H	003	Horizontal
Sopl. LA V	003	Vertical
Sopl. LA A	003	Axial
Sopl. LOA H	004	Horizontal
Sopl. LOA V	004	Vertical
Sopl. LOA A	004	Axial

In the case of acceleration, better resolution levels are obtained at high frequencies and in the case of displacement it is more representative at low frequencies.

One of the problems found when using acceleration or displacement is the skiing effect, caused by the integration of noise, which can hide defects due to the attenuation of high frequencies and the enhancement of components generated at low frequencies. In this way, digital integration is performed on the vibratory spectrum of the vibration and weights the low frequency signals, acting less on the high frequency signals, this is because the speed V(f) is inversely proportional to the frequency f:

$$V(f) = \frac{c_1 \; A(f)}{f} \tag{9}$$

while the displacement D(f) is also affected by the frequency, but in an exponential way:

$$D(f) = \frac{c_2 \; A(f)}{f^2} \tag{10}$$

where A(f) is the acceleration to frequency f and C1 and C2 are constant according to the units of measurement.

This study is compliant with the provisions of the standard and the maximum MSY levels are used, obtaining a total of 617 measurements in each of the six positions, having, therefore, a total of 3,702 measurements. An error of 5% has been considered for the sampling of each characteristic frequency and an error of 2.5% for each position. From the three measuring points, only the horizontal position had continuous monitoring, the rest was done in a reduced the rest was done on a reduced basis over periods of between seven and 15 days. Hence, the measurement value obtained at the same time was about 600 measurements in those 15 years for each of the positions.

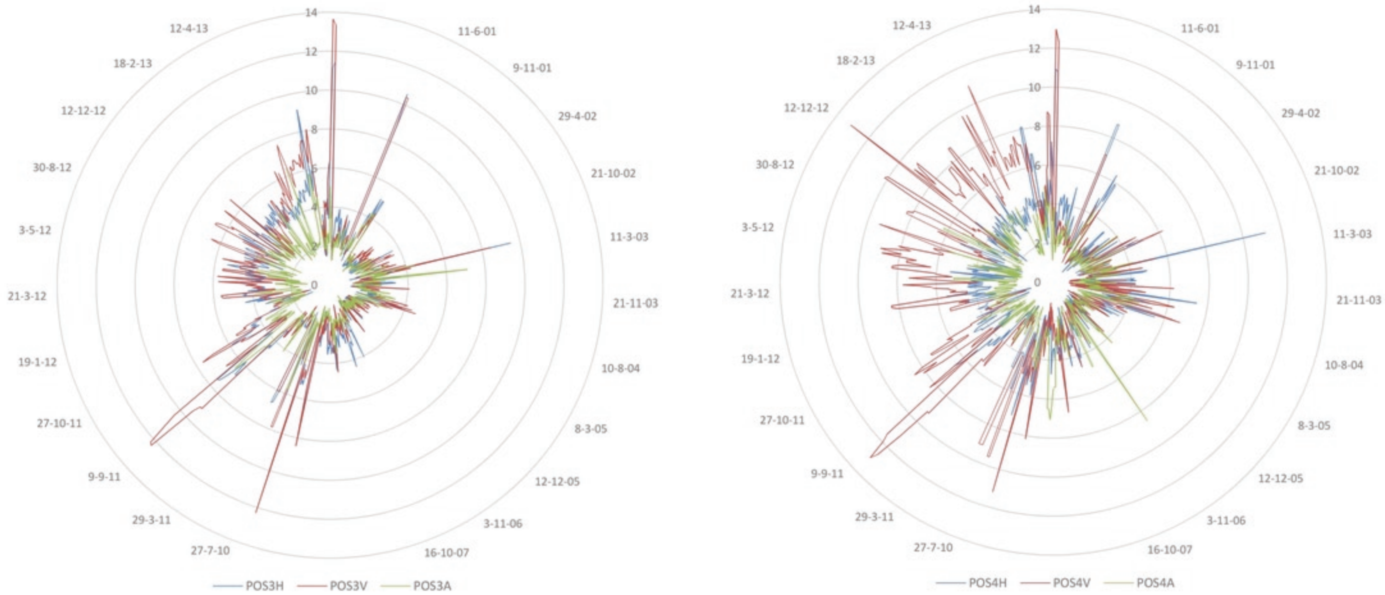

Fig. 1. Graph of Maximum Amplitude results in RMS for positions 3 and 4

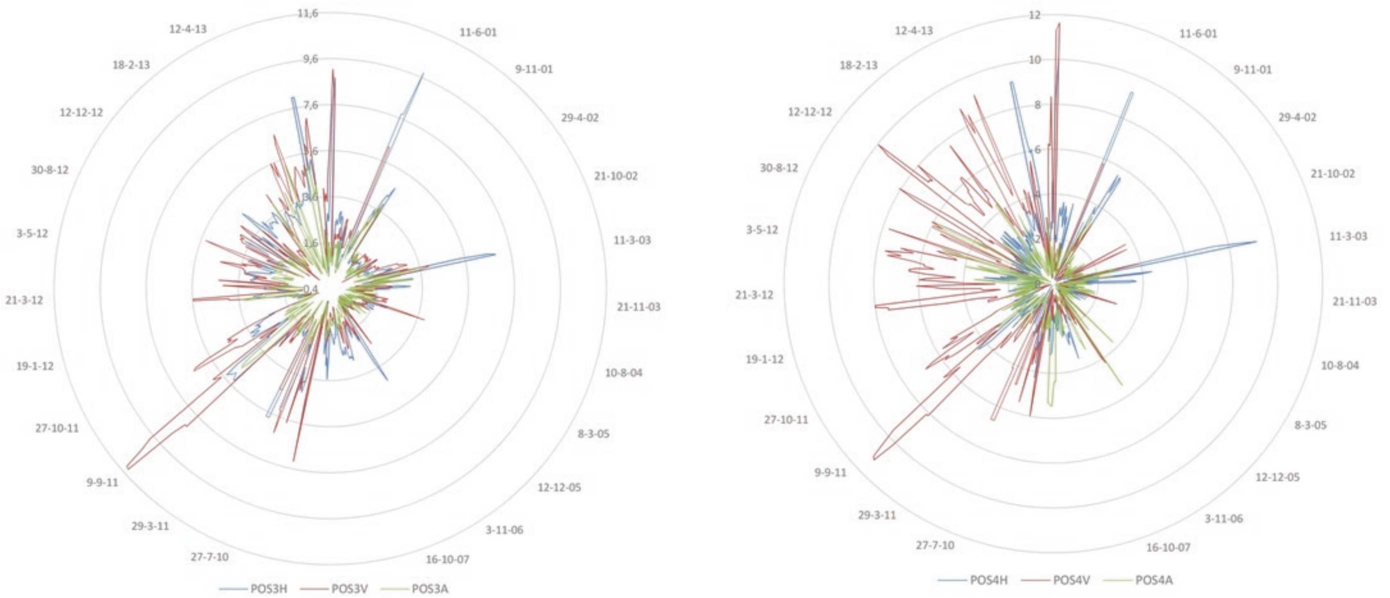

Fig. 2. Graph of SPEED results in RMS for positions 3 and 4

4. Results and discussion

The first study visualizes the maximum amplitude values generated by the digital integration of each spectrum. This analysis is performed in all positions and axes.

The first position of study will be position 3, where the highest average values in the vertical axis (3.46 RMS) are higher than in the horizontal one (3.12 RMS). Finally, the axial axis has an average value in RMS of 2.27. These values indicate that the most incident energy points are in the vertical position.

Another important value is that of peak values. In this case, the maximum value is found in the vertical position with 13.34 RMS, followed by the horizontal one with 11.44 RMS and finally the axial axis with 7.08 RMS. As in the previous case, in position 4 we observe that the maximum value is in the vertical position, followed by the horizontal position, which reaches 84.39% of the value in the vertical axis, and finally the axial with 60.28% of the maximum (4.23 RMS, 3.57 RMS, 2.55 RMS).

In the case of peak values, we found a repetition of the sequence in the previous position and the maximum value is in the vertical axis with 13.12 RMS, followed by the horizontal 11.15 RMS and the last the axial with 8.58 RMS. In position 4 values are much more intense than in position 3. And regarding the axes, they are higher in the vertical axis compared to the rest of positions (horizontal and axial), as shown in (Fig. 1).

After the previous study, we analyse the values of the characteristic frequency of the machine (SPEED), which is at a frequency of 1490 Hz. The first position of study is position 3 and its three axes. For this position the values obtained were: the horizontal axis reaches an average value of 2.17 RMS; in the vertical one the value is 2.48 and in axial, 1.26 RMS.

The results according to the peak values are: 9.82 RMS in the horizontal axis; 11.38 RMS in the vertical axis and 5.32 RMS in the axial axis. It is verified that the vertical axis is again the most sensitive to the rotation frequency of the equipment. After analysing the SPEED frequency in position 3, it is studied in position 4. The study shows the following values: 1.61 in the horizontal axis; 2.61 in the vertical one and 1.34 RMS in the axial. Therefore, the previous results are reproduced again, obtaining the highest energy level in the vertical axis.

According to the peak values, the results are: 10.04 RMS in the horizontal axis; 11.64 RMS in the vertical axis and 5.76 RMS in the axial axis. Once again the vertical axis appears as the most sensitive to the rotation frequency of the equipment, as shown in (Fig. 2).

Next, we study the frequency of the blades. In position 3 the highest RMS value is reached in the axial axis (3.39 RMS), and in the other two axes, horizontal and vertical, the values are 0.18 RMS and 0.13 RMS, respectively. In the case of peak values, the highest value is also found in the axial position with 3.35 RMS; 1.21 RMS in the horizontal axis and 0.97 RMS in the vertical one. Unlike all previous studies where the most relevant value was found in the vertical axis, in this case is in the axial axis where this frequency affects the most. This is due to the effect of impulsion generated by the profile of the blade. The graph also shows that the sequence obtained in position 3 is repeated in position 4, obtaining the highest value in the axial axis with 0.42 RMS, as shown in (Fig. 3).

In the axial axis we obtain an average value of 0.39 RMS for position 4 and 0.42 RMS for position 3. This is the first frequency whose highest value is reached in position 3. Peak results credit the previous result, with 3.35 RMS at position 3 and 1.94 RMS at position 4.

After analysing frequencies and the SPEED of the blade, we carried out the analysis of the frequencies of the FAGNU322 bearing. The most significant variables of bearings are the frequencies generated by the cage of balls defined as FTF, those caused by the balls (BSF), followed by the ones of its second harmonic (2BSF), those generated by the balls in the outer track (BPOR), and the one generated in the inner track (BPIR). After analysing all the significant variables of the FAGNU322 bearing and their influence in each measured axis and position, the results in terms of RMS are shown in the table below, as shown in (Table 4). The positions where the maximum values are generated have been: vertical position 4 for FTF, and 2BSF. The other two, BPOR and BPIR, are in axial axis 3 and in horizontal axis 4, respectively.

Frequencies generate their greatest influence on the vertical axis, followed by the axial one, and being the horizontal axis the least important.

After studying the variables of the FAGNU322 bearing, we compare them with the results of the other variables under study, the rotation frequency of the machine called SPEED and the frequency of the blades. The table of results in terms of RMS is shown in table 5. We can conclude from the results shown in the table that the most important frequency is that caused by the machine itself, much higher than those of the blades and the cage of the bearing or FTF.

When checking the peak values, again we take SPEED with 11.644 RMS as reference, but in this case the second one is not the one generated by the blades but by the frequency FTF FAGNU322, and in the third place that of the blades, as shown in (Fig. 4).

5. Conclusions

The study determines that the most important maximum amplitude values appear in position 4, and that the vertical axis is the most sensitive to the mechanical actions of the equipment. The above result is also valid for specific frequencies, such as the one generated by the rotation frequency of the equipment (SPEED). The analysis of the frequency of the blades shows us that the most determining position

Table 4. Summary of FAGNU322 bearing frequencies in terms of RMS.

VARIABLE Frecuency Position	**FTF** Cage Defects Pos4V	**BSF** Ball Defects Pos4V	**BPOR** Outer Race Defects Pos3A	**BPIR** Inner Race Defects Pos4H	**2 BSF** Second Harmonic, Ball Defects Pos4V
TOTAL SUM	217,471	44,126	38,223	22,819	64,616
AVERAGE	0,355	0,072	0,062	0,037	0,105
PEAK VALUE	10,056	1,392	0,545	0,388	0,659

Table 5. Summary of SPEED, ALABES and FTF frequencies in terms of RMS.

VARIABLE	SPEED Pos4V	FTF Pos4V	ALABES Pos3A
TOTAL SUM	1606,085	217,471	245,778
AVERAGE	2,612	0,355	0,399
PEAK VALUE	11,644	10,056	3,354

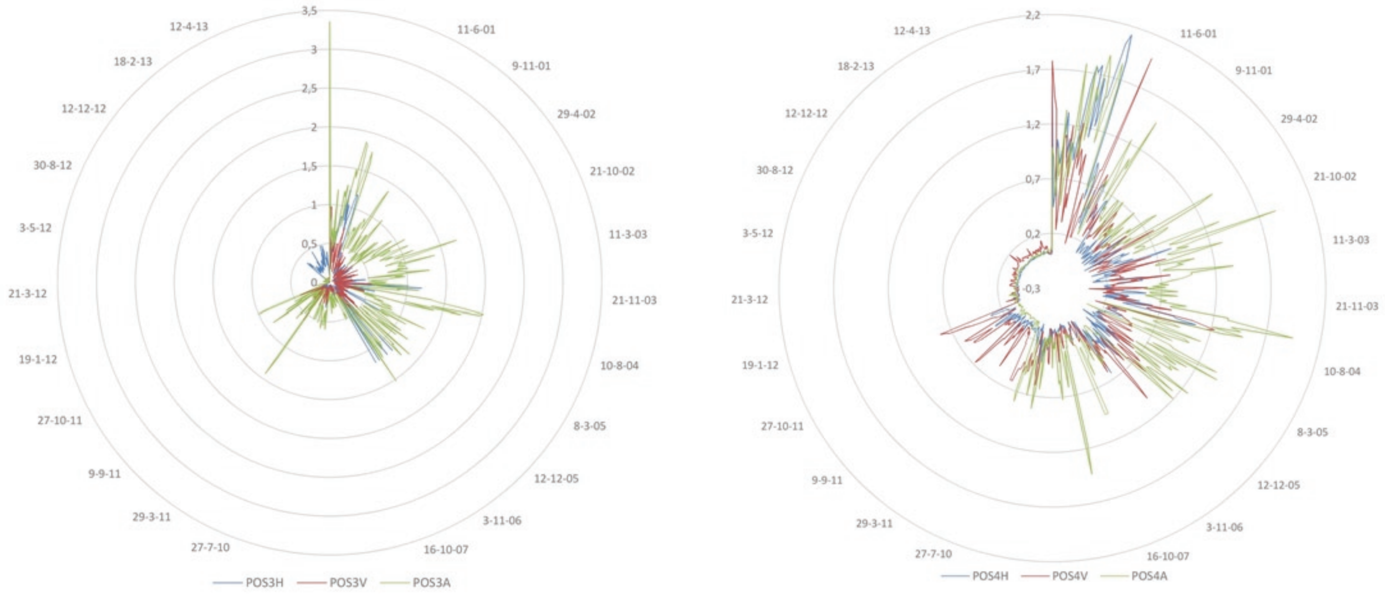

Fig. 3. Graph of the frequency results of blades in RMS for positions 3 and 4

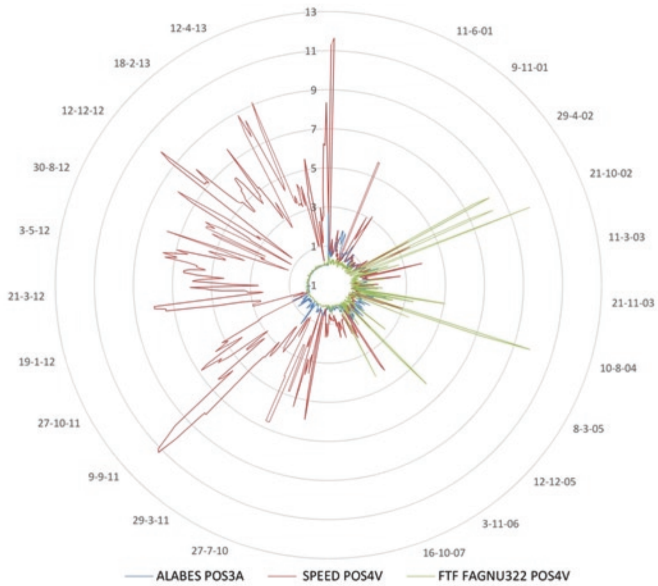

Fig. 4. Comparison of SPEED, ALABES and FTF frequencies FAGNU6322

is position 3 on the axial axis. This is due to the thrust action generated by the blades and the fluid on the axis itself. On the FAGNU6322 bearing, the study determines that the FTF frequency is the most important and reaches its highest value in the vertical axis position 4. The second most important frequency is the one generated by the 2BSF, with average and peak values much lower than the frequency of the FTF. This indicates that the most determining frequency in the operation of the equipment is SPEED in vertical position 4, as well as axial position 3 by the action of the blades in the background. The final conclusion would be that continuously evaluating position 4 on the vertical axis we would be able to predict when our equipment is going to suffer a failure, reducing the control points by 5.

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