Rolling bearing fault detection and isolation – A didactic study

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Abstract: - The present paper aims to demonstrate why usually when theoretical mathematical models are used to compute the frequencies corresponding to a faulty rolling bearing a deviation is obtained between the computed values and the real frequencies emitted by such a device. A lab rolling bearing test ring has been developed to perform the current studies.

Key-Words: - Fault detection, fault isolation, fault diagnosis, rolling bearing, condition based maintenance.

Notation

$\omega_{i,g,e,r}$ - Angular speed of inner race, cage, outer race and rolling element; $V_{i,g,e,r}$ - Linear speed of inner race, cage, outer race and rolling element; $r_{i,g,e,r}$ - Radius of inner race, cage, outer race and rolling element; $f_{i,g,e,r}$ - Rotation frequency of inner race, cage, outer race and rolling element; d - Diameter of the rolling element; D_i - Diameter of inner race; D_e - Diameter of outer race; D_p - Pitch diameter; θ - Contact angle; N - Number of rolling elements.

1 Introduction

The condition based maintenance philosophy of using vibration information to lower operating costs and increase machinery availability is gaining acceptance throughout industry. Since most of the machinery in a condition maintenance program contains rolling element bearings, it is imperative to establish a suitable condition monitoring procedure to prevent malfunction and breakage during operation [5], [6], [7].

However, the first time someone try to approach the rolling bearing fault detection and diagnosis topic several difficulties arise. Often such difficulties last for years until the degradation process has been well understood and mainly the signal processing methodology, the fault detection procedure and the reason to achieve deviations between the frequencies evaluated through the theoretical model and the frequencies emitted by the rolling bearing.

A vibration signal emitted by a faulty rolling bearing usually include frequencies related to it geometry, fault location and rotation speed of inner and outer races.

This paper presents a didactic study based on lab equipment developed to test and analyse faulty rolling bearings. A comparison of several vibration signals emitted by faulty rolling bearings allows achieving a deeper understanding of how the corresponding fault detection and diagnosis is performed in practice. The paper's aim is to identify the factors responsible for the deviations achieved between the theoretical mathematical models and the frequencies emitted by a faulty rolling bearing. Thus, the paper is organised as follows: Section 2 provides a description of the theoretical mathematical models usually used; Section 3 reviews the fault detection and diagnosis techniques usually applied for rolling bearings; Section 4 presents a case study; Section 5 provides a description of the tests performed together with a results discussion; Section 6 presents some concluding remarks.

2 Theoretical Mathematical Models

A faulty rolling bearing usually emit the following main frequencies:

FTF - (***F**undamental **T**rain **F**requency*) – frequency emitted by a faulty cage; **BPFO** – (***B**all **P**ass **F**requency of the **O**uter Race*) – frequency emitted by a rolling element bearing when it impacts a superficial defect in the outer race; **BPFI** – (***B**all **P**ass **F**requency of the Inner Race*) – frequency emitted by a rolling element bearing when it impacts a superficial defect in the inner race; **BSF** – (***B**all **S**pin **F**requency*) – frequency emitted when a rolling element bearing with a superficial defect impacts the inner or outer races.

The frequency spectrum of a faulty rolling bearing includes not only the inner and outer races rotation frequencies but also the harmonics corresponding to the

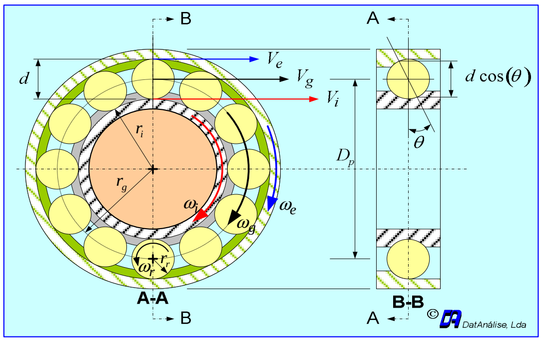

Fig. 1. Rolling bearing geometry

fault frequency and sidebands as a result of the amplitude modulation. Such sidebands are related to the cage frequency rotation or to the inner or outer races frequency rotation.

A faulty rolling bearing produces certain frequencies depending on rolling element bearing geometry, which is shown in Fig. 1 and 2, number of rolling elements and shaft speed. The cage angular speed is given as follows:

$$\omega_g = \frac{V_g}{r_g} \tag{1}$$

The outer race angular speed is given by equation (2),

$$\omega_e = \frac{V_e}{r_e} \tag{2}$$

and the inner race angular speed is given as follows,

$$\omega_i = \frac{V_i}{r_i} \tag{3}$$

Considering there is no sliding between the rotating parts, the cage linear speed can be given by,

$$V_g = \frac{V_i + V_e}{2} = \frac{\omega_i r_i + \omega_e r_e}{2} \tag{4}$$

Analysing Fig. 2 the following can be achieved,

$$r_g = \frac{D_p}{2} \tag{5}$$

$$r_i = \frac{D_p}{2} - \frac{d\cos(\theta)}{2} \tag{6}$$

$$r_e = \frac{D_p}{2} + \frac{d\cos(\theta)}{2} \tag{7}$$

$$r_r = \frac{d}{2} \tag{8}$$

Putting all the results together, the cage angular speed can be given as follows,

$$\omega_g = \frac{1}{2}\left[\omega_i\left(1 - \frac{d\cos\theta}{D_p}\right) + \omega_e\left(1 + \frac{d\cos\theta}{D_p}\right)\right] \tag{9}$$

Being $\omega_g = 2\pi f_g$ equation (9) gives de value for FTF, which can be expressed in Hertz (Hz) as follows,

$$FTF = f_g = \frac{1}{2}\left[f_i\left(1 - \frac{d\cos\theta}{D_p}\right) + f_e\left(1 + \frac{d\cos\theta}{D_p}\right)\right] \tag{10}$$

By other words, equation (10) provides the cage rotation frequency which when appears in the frequency

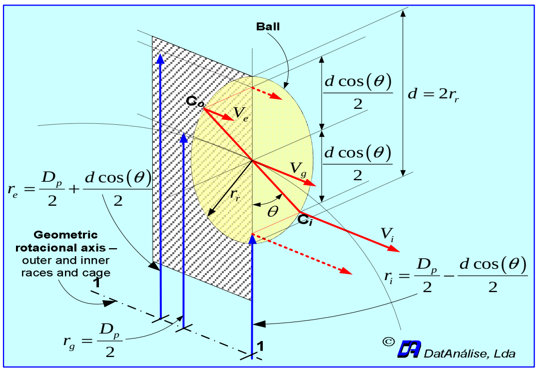

Fig. 2. Rolling bearing geometry – contact angle θ

spectrum is related with the unbalanced of rotating elements – cage and rolling elements due to cage wear or looseness development. Its appearance can also be related to amplitude modulated due to defects in the inner and outer races. The defect frequency called above BPFO can be evaluated multiplying the number of rolling elements by the relative angular speed between cage and outer race. Thus, BPFO can be given as,

$$BPFO = N\left(\omega_g - \omega_e\right) \tag{11}$$

Putting equation (9) into (11) the following is obtained,

$$BPFO = N\left\{\frac{1}{2}\left[f_i\left(1 - \frac{d\cos\theta}{D_p}\right) + f_e\left(1 + \frac{d\cos\theta}{D_p}\right)\right] - f_e\right\} \tag{12}$$

Through mathematical manipulation equation (12) can be written as follows,

$$BPFO = \frac{N}{2}\left(f_i - f_e\right)\left(1 - \frac{d\cos\theta}{D_p}\right) \tag{13}$$

In a similar way the defect frequency quoted above as BPFI can be evaluated multiplying the number of rolling elements by the relative angular speed between cage and inner race, being given as follows,

$$BPFI = N\left(\omega_i - \omega_g\right) \tag{14}$$

Putting equation (9) into (14) and doing some simplifications the following formula is achieved to evaluate BPFI,

$$BPFI = \frac{N}{2}\left(f_i - f_e\right)\left[1 + \left(\frac{d\cos\theta}{D_p}\right)\right] \tag{15}$$

The defect frequency called above as BSF can be defined as the rolling element frequency rotation, ball or roller, by its own centre. The ball or roller angular speed in turn its own centre is given as follows,

$$\omega_r = \frac{V_r}{r_r} \tag{16}$$

Considering that only a pure rotation exists and there is no sliding the ball or roller tangential speed in the contact point with the inner race is given by the following formula,

$$V_r = \left(\omega_i - \omega_g\right) r_i \tag{17}$$

Putting the equation (17) into (16) the following equation is obtained,

$$\omega_r = \frac{\left(\omega_i - \omega_g\right) r_i}{r_r} \tag{18}$$

Putting equations (6), (8) and (9) into equation (18) the following equation is achieve to evaluate the defect frequency BSF,

$$BSF = f_r = \frac{D_p}{2d}(f_i - f_e)\left[1 - \left(\frac{d\cos\theta}{D_p}\right)^2\right] \tag{19}$$

In the case where the rolling elements are rollers is usually the appearance of the frequency 2xBSF since impulses are emitted when the defect contacts the inner and outer races during one roller rotation. When a defect is formed on one of the rolling bearing parts mentioned above, related frequency, its orders, its sidebands, etc. may arise in the spectrum graph.

The overall mathematical model presented so far, allowing evaluating the defects frequencies, is general since has being considered not only the inner race rotation speed but also the outer race rotation speed. However, in most of the practical situations the outer race is stationary and the mathematical model presented above can be simplified.

2.1 Simplified mathematical model

Hence, when the outer race is stationary the mathematical model to evaluate the defect frequencies can be written as follows,

$$\begin{aligned}
FTF &= \frac{1}{2}(f_i)\left(1 - \frac{d\cos\theta}{D_p}\right)\\
BPFO &= \frac{N}{2}(f_i)\left(1 - \frac{d\cos\theta}{D_p}\right)\\
BPFI &= \frac{N}{2}(f_i)\left(1 + \frac{d\cos\theta}{D_p}\right)\\
BSF &= \frac{D_p}{2d}(f_i)\left[1 - \left(\frac{d\cos\theta}{D_p}\right)^2\right]
\end{aligned} \tag{20}$$

Comparing the above equations (20) the following relationships between them can be written,

$$\begin{aligned}
BPFO &= N\,(FTF)\\
BPFI &= N\,(f_i - FTF)
\end{aligned} \tag{21}$$

Thus, to evaluate the defect frequencies is compulsory to know the geometric rolling bearing characteristics. When such characteristics are unknown the use of approximate equations is possible, as presented in the following sub section.

2.2 Empirical mathematical model

When the contact angle is unknown and the remaining parameters are known, the mathematical model (20) can be used making $\theta = 0$.

If in extremely cases the number of parameters known not allow the use of the mathematical model (20) alternatively the following equations can be used achieving approximated results,

$$\begin{aligned}
FTF &= 0.4\,f_i\\
BPFO &= 0.4\,N f_i\\
BPFI &= 0.6\,N f_i
\end{aligned} \tag{22}$$

The empirical mathematical model (22) is based on the fact that in a complete rotation of the inner race, about 40% of the rolling elements contact a defect in the outer race and 60% of the rolling elements contact a defect in the inner race [4].

3 Fault Detection/Isolation Techniques

In the past several decades, many different techniques have been developed to condition based maintenance for monitoring and diagnosis rolling element bearings. Some detailed reviews of such techniques can be found in [7], [8]. Among them, vibration and acoustic emission (AE) signals are widely used in condition monitoring of rotating machine. However, from those techniques, visual inspection of time-domain or frequency-domain features of the measured vibration signals has long been performed [1].

3.1 Time-domain / Trend Curves

Time domain methods usually involve indices that are sensitive to impulsive oscillations, such as peak level, Root Mean Square (RMS) value, Crest Factor (peak/RMS) analysis, kurtosis analysis, shock pulse counting, time series averaging method, signal enveloping method and many more. A healthy (no faulty) rolling bearing usually produces a vibration signature with Gaussian distribution [2].

Generally, this type of measurement gives limited information but can be useful when used for trending, where an increasing vibration level is an indicator of a deteriorating rolling bearing condition. Trend analysis involves plotting the vibration level as a function of time and using this to predict when the rolling bearing must be taken out of service for repair. Another way of using the measurement is to compare the levels with published vibration criteria for different types of equipment.

Detection generally uses the most basic form of vibration measurement, where the overall vibration level is measured on a broadband basis in a range for example, 10-1,000Hz or 10-10,000Hz. In rolling bearings where there is little vibration, the vibration signal indicated by the Crest Factor may imply incipient defects, whereas the high energy level given by the RMS level may indicate severe defects. Although broadband vibration measurements may provide a good starting point for fault detection it has limited diagnostic capability and although a fault may be identified it may not give a reliable indication of where the fault is, i.e. rolling bearing deterioration/ damage, unbalance, misalignment, and so on.

When an improved diagnostic capability is required, frequency analysis is normally used, which usually gives a much earlier indication of the development of a fault and also the source of the fault, as will be described in the next sub-section.

3.2 Frequency domain

Frequency analysis plays an important part in the detection and diagnosis of rolling bearing faults. In the time domain the individual contributions to the overall machine vibration are difficult to identify. In the frequency domain they become much easier to identify and can therefore be easily related to individual sources of vibration. Time domain methods only allow rolling bearing fault detection while frequency domain methods allow detecting, isolating and characterizing faults, which is called diagnosis.

Frequency domain analysis or spectral analysis using the Fast Fourier Transform (FFT) is probably the most used technique for rolling bearing fault detection and isolation.

The rolling elements experience some slippage as the rolling elements enter and leave the bearing load zone. As a consequence, the occurrence of the impacts never reproduces exactly at the same position from one cycle to another. Moreover, when the position of the defect is moving with respect to the load distribution of the rolling bearing, the series of impulses is modulated in amplitude. However, the periodicity and the amplitude of the impulses experience a certain degree of randomness. In such case, the signal is not strictly periodic, but can be considered as cyclo-stationary (periodically time-varying statistics). All these make the rolling bearing defects very difficult to detect by conventional FFT-spectrum analysis which assumes that the analyzed signal is strictly periodic. A method of conditioning the signal before the spectrum estimation takes places is needed.

To overcome the modulation problem, several signal envelope demodulation techniques have been introduced. One of the most popular is the high-frequency resonance technique (HFRT), where an envelope detector demodulates the band pass filtered signal and the frequency spectrum is determined by FFT technique [2]. A particular case of this technique is the so called Spike Energy Spectrum where the measured magnitude of the signal is expressed in “gSE” units (acceleration units of Spike Energy) [3].

Spike Energy spectrum and Spike Energy time waveform were developed and used in diagnostic analysis in recent years. Following such technique the vibration signal is measured by an accelerometer and filtered by frequency band pass filters. The purpose of using high pass corner frequencies is to reject low-frequency vibration signals, such as unbalance, misalignment and looseness. Then, the filtered signal passes through a peak-to-peak detector, which not only holds the peak-to-peak amplitude but also applies a carefully selected decay time constant. The decay time constant is directly related to the spectrum maximum frequency (Fmax). The output signal from Spike Energy peak-to-peak detector is a saw-tooth shape signal. This saw-tooth shape signal is further processed to calculate Spike Energy overall magnitude and Spike Energy spectrum by using the FFT.

The peak-to-peak detector in is unique and very sensitive to the defect frequency as compared to other envelope detection or demodulation method. In an envelope detection, the vibration signal is first passed through a high pass (or band pass) filter. The filtered signal is full wave (or half wave) rectified. Then, the rectified signal is passed through a low pass filter to separate the modulation (or defect) frequency from the carrier frequency. The low-pass filtering has an averaging effect on the rectified signal and the peaks are smoothed in the demodulated waveform. In contrast, Spike Energy detection preserves the severity of defects by holding the peak-to-peak amplitude of the impulses. It also enhances the fundamental defect frequency and its harmonics by applying a proper selected decay time constant.

3.3 Other techniques

Normal and damage rolling bearings vibrate when rotate. There has been much research on the vibration measurement of rolling bearings which have damages on the raceway surface, showing that such measurements are useful for rolling bearing fault diagnosis. More recently, some authors have pointed out that the measurement of time intervals of Acoustic Emission (AE) for rolling bearings can also be useful for the same purpose if the AE occurred at normal and damage bearings during operation. Furthermore, as AE refers to the generation of transient elastic waves produced by a sudden redistribution of stress in a material, some authors argue that AE has the advantage to be able to detect the accumulation of microdamage inside components, especially under service conditions [2].

4 Rolling Bearing Test Ring

A lab rolling bearing test ring has been developed, which is shown in Fig. 3. Such a device allows shaft speed rotation changes and changes in the radial load over the rolling bearing. The cage rotational speed is measured through a stroboscopic lamp.

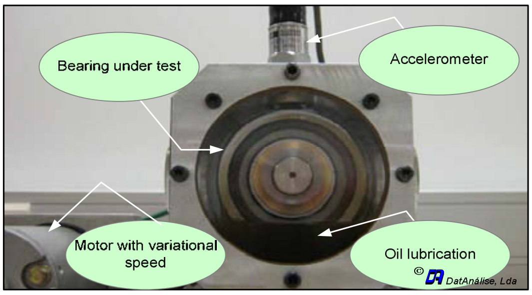

Fig. 3. Rolling bearing test ring

Table 1. Rolling Bearing Parameters.

General characteristics of bearing NU 307 ECP			
Inner race diameter	46,2 mm		
Rolling element diameter	12,0 mm		
Outer race diameter	70,2 mm		
Primitive diameter	58,2 mm		
Nº of rolling elements	12		
Contact angle	0 º		
Inner diameter	35 mm		
Outer diameter	80 mm		
Dynamic load	64400 N	Reference speed (oil)	9500 rpm
Static load	63000 N	Limiting speed	2000 rpm
Radial clearance	25 a 50 μm	EC - Single row cylindrical roller bearing	
Minimum radial load	338 N	P – Injection moulded cage of glass fibre reinforced polyamide	

During the current studies four rolling bearings have been used and the corresponding characteristics can be seen in Table 1. Four single faults have been introduced each one in each of the rolling bearings used to conduct the tests. The four faults considered are the following: superficial defect in inner race, outer race, cage and roller. A data acquisition system has been used to online acquired and store measurement data such as global values, frequency spectrums in mm/s and gSe (envelope analysis), time signals in mm/s and g's (envelope analysis) and the shaft rotation speed (inner race speed). The last mentioned speed has been measured using a laser photoelectric cell.

5 Tests, Results and Discussion

Fig. 4 depicts the envelope spectrum where can be seen the defect frequencies of inner race and its harmonics, as well as the sidebands at the inner race rotation frequency. Such sidebands are due to the fact that the inner race is rotating and, thus, the defect will enter and leave the load zone causing a variation in the rolling element raceway contact force, hence deflections. While in the load zone the amplitudes of the pulses will be highest but then reduce as the defect leaves the load zone resulting in a signal, which is amplitude-modulated at inner race rotational frequency. The time signal depicted in Fig. 5 shows the impacts occurring in the rolling bearing load zone. The time between highest impacts corresponds to one inner race rotation.

The outer race defect can be observed through the envelope spectrum shown in Fig. 6, where can be seen the frequency BPFO and its harmonics.

Fig. 7 has been achieved during test conducted with a rolling bearing with a faulty roller. It can be seen the frequency BSF and its harmonics with side bands at the cage rotation frequency.

When considered isolated a cage defect doesn't occur very often in practice. Usually, the cage defect frequency arises associated with other defects, such as those occurring in the inner and outer races or rollers. That can be observed in the spectrum achieved to the tested rolling bearing with a faulty roller element. However, when a cage defect occurs usually results in the rolling bearing damage due to cage wearing or rupture of the cage material. Such a cage damage mechanism can be observed as an unbalanced of rotating elements (cage, balls or rollers). During the current studies, such a defect has been simulated in a rolling bearing by eliminating one roller. The spectrum depicted in Fig. 8 presents the frequency FTF and several harmonics.

The shaft rotation speed has been measured through a photoelectric cell and the equivalent cage rotation speed has been measured through the stroboscopic lamp. The speed change of the cage rotation as a function of the shaft rotation (inner race) at constant load, allows observing the cage sliding.

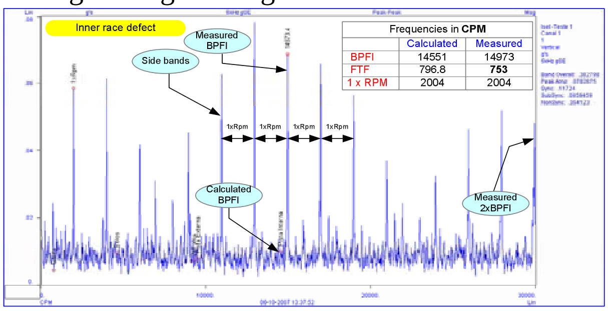

Fig. 4. Envelope spectrum – defect BPFI.

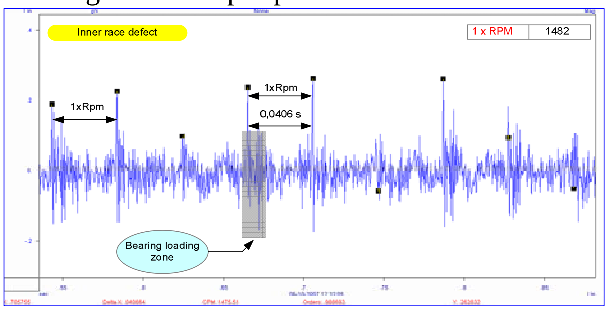

Fig. 5. Time signal in g´s – defect BPFI.

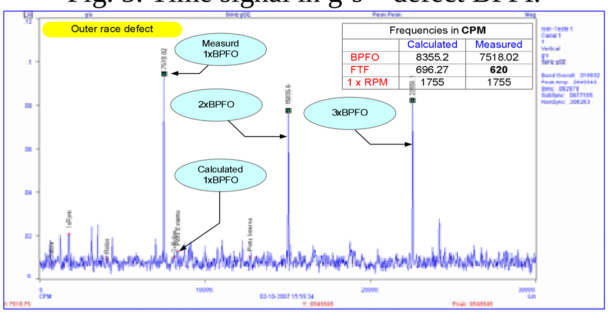

Fig. 6. Envelope spectrum – defect BPFO.

The defect frequencies are computed as a function of the cage rotation speed. The mathematical model expressed by equations (10), (13), (15) and (19) are expressed as a function of the rolling bearing geometric parameters and not show that fact. Therefore, the mentioned fact is hidden by the model, since the cage rotation speed doesn't appear explicitly in equations. A detail analysis of the spectrums previous presented allows to conclude that exists always a deviation between the defect frequency computed and measured. During the current studies has been observed that the factor responsible for that deviation is the cage rotation speed. However, the mathematical models previous present has been derived considering there is no sliding between the elements of the rolling bearing.

Consider the graph shown in Fig. 4. The defect frequency computed (14551 CPM – cycles per minute) is lower than the measured frequency (14973 CPM). The inner race rotation frequency is in this case 2004 CPM, while the cage rotation frequency is 753 CPM. Equation (15) not easily allows justifying the deviation. However, if equation (14) is applied the results are obvious, justifying the deviation and confirming the defect frequency corresponding to the fault mentioned above.

Therefore, the current studies allow concluding that in the presence of sliding, the defect frequency for a fault in the inner race of a rolling bearing is always higher than the computed corresponding frequency. Furthermore, the defect frequency for a fault in the outer race of a rolling bearing, when sliding is considered, is always lower than the corresponding computed frequency. In a similar way, the defect frequency for a fault in a roller of a rolling bearing, considering sliding, is always lower than the computed corresponding frequency.

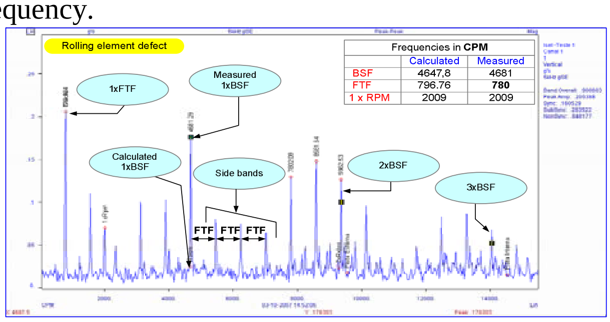

Fig. 7. Envelope spectrum – defect BSF.

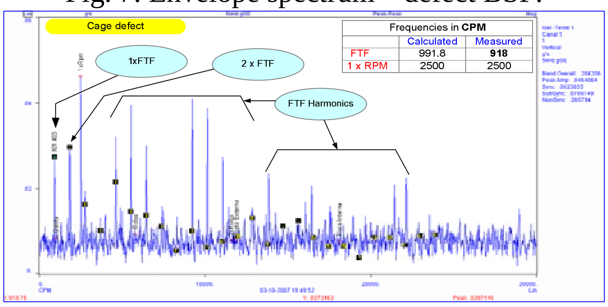

Fig. 8. Envelope spectrum – defect FTF.

5 Conclusions

The tests performed during the current studies show the influence of the cage rotation speed in the emitted frequencies of a faulty rolling bearing. Such influence can not be seen by analysing the mathematical model that allows computing the quoted defect frequencies. To simulate changes in the cage speed, a didactic approach has been followed in order to obtain an intentional exaggerated sliding, to demonstrate its effect in the spectral analysis. The use of rolling bearings with cylindrical roller elements, in tests conducted during the current studies, eliminate the influence of other geometric parameters, such as the contact angle. The influence of that parameter in fault diagnosis of faulty rolling bearings is under study and will be soon pointed out.

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