ANNALS OF THE ORADEA UNIVERSITY
Fascicle of Management and Technological Engineering
ISSUE #1, MAY 2013, http://www.imtuoradea.ro/auo.fmte/

BEARINGS DIAGNOSIS

Sorin PATER[1], Ion BRATU[2], Georgeta Chindlea[3]
[1]University of Oradea, spater@uoradea.ro
[1]University of Oradea, ibratu@uoradea.ro
[1]University of Oradea, gchindlea@uoradea.ro

Abstract— The cause of vibration, their harmful effects and remedies have also been mentioned for practical utility to control vibrations. A system which has mass and elasticity can start vibrating if it is disturbed. The natural frequencies of a system depend on the degrees of freedom of a system. For a multi-degree of freedom system, there will be several natural frequencies. Mechanical vibration in machines and equipment can occur due to many factors, such as unbalanced inertia; bearing failures in rotating systems.

Keywords— frequency, time period, amplitude, vibration, resonance, eccentricity, harmonics, misalignment, unbalance

I. INTRODUCTION

All mechanical systems vibrate and the amount of vibration generated is related to the vibrating forces through the rotating components. Depending on the complexity of the machine, each rotating component produces a frequency response depending on the magnitude of the vibrating force, direction and phase according to the rotating speed. An overall vibration response from a gearbox is shown in Fig. 1.

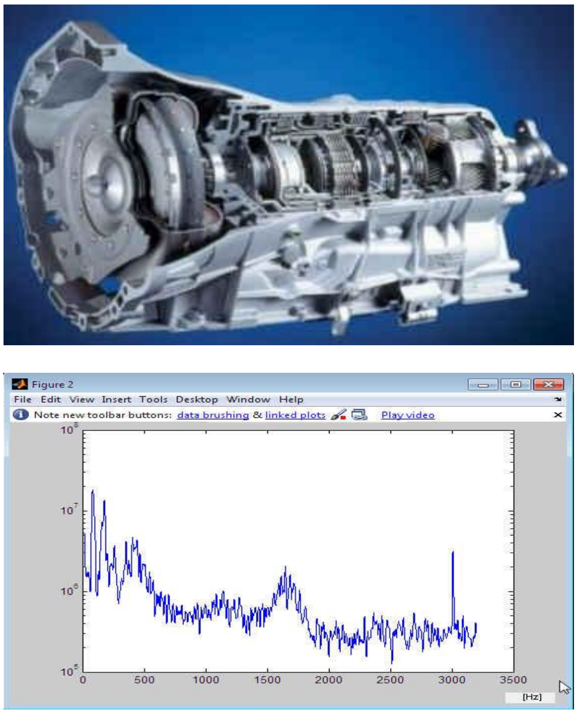

Fig.1. A gearbox vibration

Lumped parameter modeling (LPM) is an efficient means to express the internal dynamics of transmission systems; masses and inertias of key components such as gears, shafts and bearings can be lumped at appropriate locations to construct a model. The advantage of the LPM is that it provides a method to construct an effective dynamic model with relatively small number of degrees-of-freedoms (DOF), which facilitates computationally economical method to study the behavior of gears and bearings in the presence of nonlinearities and geometrical faults.

The characteristic vibration frequency of each individual component can be determined from the known dimensions, number of rotating elements and the rotating speed of the machine. Some of the common causes of machine vibration are:

- Unbalance (static, dynamic unbalance) - due to uneven distribution of masses about a rotor's rotating geometric centre. When the rotating and geometric centers are coincidental, the rotor is described as in a state of balance. Dynamic unbalance is caused by the unbalance caused by the unbalance mass located in a plane off-centre from the bearing mounting planes.

- Misalignment (angular misalignment, parallel misalignment) - produces 1 x rpm frequency component. This can be partly overcome by using self-aligning bearing or flexible couplings.

- Bent shaft (dominant frequency, 1 x rpm in the axial direction) - This generate high axial vibration with frequency component at 1 x rpm (bend near the shaft centre) and 2 x rpm (bend near the shaft end).

- Looseness (internal assembly, base and structure) - normally cause by improper fit between components. In the frequency spectrum, the response produces many harmonics in the FFT spectrum due to the non-linear response of the loose parts to the forces from the rotor. Looseness often produces sub-harmonics at a fraction of the shaft rpm.

-Gear defects - the frequency spectrum of a gearbox usually shows the 1 x rpm and 2 x rpm together with the gear mesh frequency.

-Bearing defects (inner, outer and rolling element pass frequency) - this produces the frequency components related to the bearing components. The component frequencies can be determined from the known dimensions of the bearing and the shaft speed.

II. VIBRATION ANALYSIS

For a complete and more realistic modeling of the gearbox system, detailed representations of the REBs (rolling element bearings) and the gearbox casing are necessary to capture the interaction amongst the gears, the REBs and the effects of transfer path and dynamics response of the casing. Understanding the interaction between the supporting structure and the rotating components of a transmission system has been one of the most challenging areas of designing more detailed gearbox simulation models. The property of the structure supporting REBs and a shaft has significant influence on the dynamic response of the system.

Fuller representation of the REBs and gearbox casing also improves the accuracy of the effect transmission path that contorts the diagnostic information originated from the faults in gears and REBs. It is desired in many applications of machine health monitoring that the method is minimally intrusive on the machine operation.

This requirement often drives the sensors and/or the transducers to be placed in an easily accessible location on the machine, such as exposed surface of gearbox casing or on the machine skid or on an exposed and readily accessible structural frame which the machine is mounted on. The capability to accurately model and simulate the effect of transmission path allows more realistic and effective means to train the diagnostic algorithms based on the artificial intelligence.

In vibration analysis, the normal approach is to relate the frequency components in the spectrum, cepstrum to specific machine components. Since not all machine components will generate vibration directly related to the fundamental rotating speed, it may be necessary to conduct additional tests to reduce the number of possible causes.

Bearings are essential components in rotating machinery. They provide rotating motion and simultaneously carry heavy load. These bearings are designed to provide up to one million hours of running time, depending on load and speed. Mishandling of bearings which includes improper storage, improper installation, poor lubrication, harsh operating environment, overloads, over speeds and misalignment (70% of bearing failure) often leads to premature failure.

A rolling element bearing comprises of inner and outer races, rolling elements (balls or rollers) and a cage to hold the rolling elements in position.

A machine with a defective bearing can generate at least five frequencies. These frequencies are:

Rotating unit frequency or speed (S)
Fundamental train frequency (FTF)[3]

$$FTF = \frac{1}{2} S \left(1 - \frac{B_d \cos\phi}{P_d}\right) \tag{1}$$

Ball pass frequency of the outer race (BPFO)

$$BPFO = \frac{N_b}{2} S \left(1 - \frac{B_d \cos\phi}{P_d}\right) \tag{2}$$

Ball pass frequency of the inner race (BPFI)

$$BPFI = \frac{N_b}{2} S \left(1 + \frac{B_d \cos\phi}{P_d}\right) \tag{3}$$

Two times ball spin frequency (2 X BSF)

$$BSF = \frac{P_d}{2B_d} S \left(1 - \frac{B_d^2 \cos^2\phi}{P_d^2}\right) \tag{4}$$

Defects in antifriction bearings can occur on the raceways, the rolling elements, the cage, or any combination. Such defects generate unique vibration signals. It is helpful to know the type of bearing installed because different types of bearings can generate different signals depending upon loading, internal clearance, and construction. For ball, cylindrical roller, and other bearings with a zero degree contact angle, defects on raceways can be identified by a narrow banded spectral line at the ball pass frequency of the race on which the defect exists.

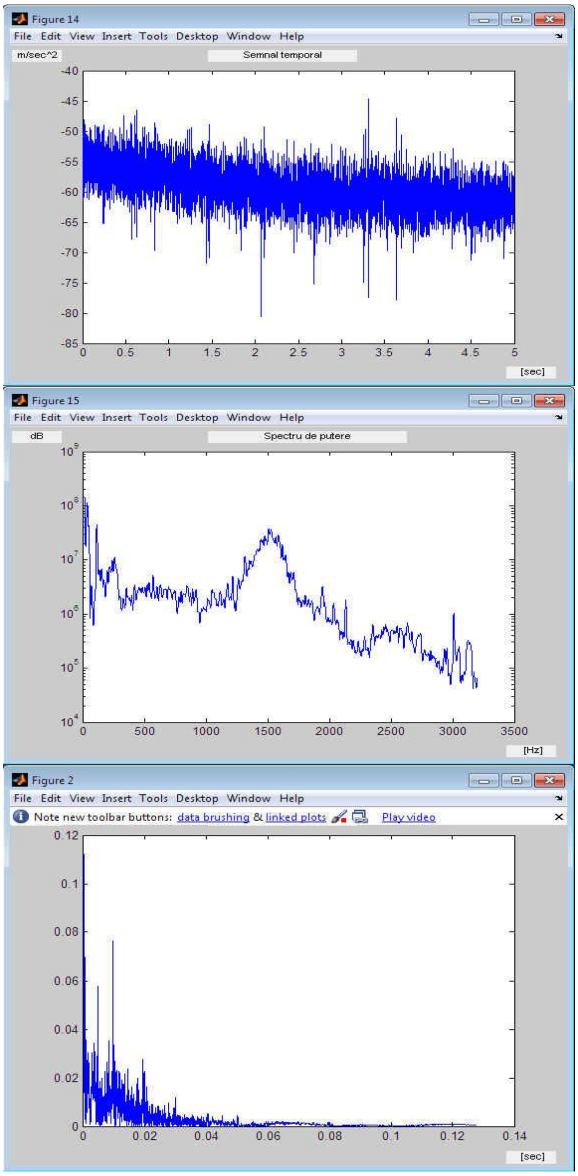

Fig.2. Time Signal, Frequency Spectrum, Cepstrum of a Cracked Outer Race

For spherical and tapered roller and ball bearings that have a contact angle, a defect on the outer race generates the fundamental BPFO and harmonics. The harmonics are generated because a large area of the outer race is in the load zone. In some bearings, 360 degrees of the outer race are in the load zone. The more harmonics generated by a fatigue spall, the larger the spall. Therefore, defect length can be determined by harmonic content for shallow flaking fatigue spalls. Defects on the inner race of ball and cylindrical roller bearings behave similar to outer race defects in that the fundamental BPFI and harmonics are generated and the harmonic content can be used to approximate defect size. Inner race defects in spherical and tapered roller bearings can generate a

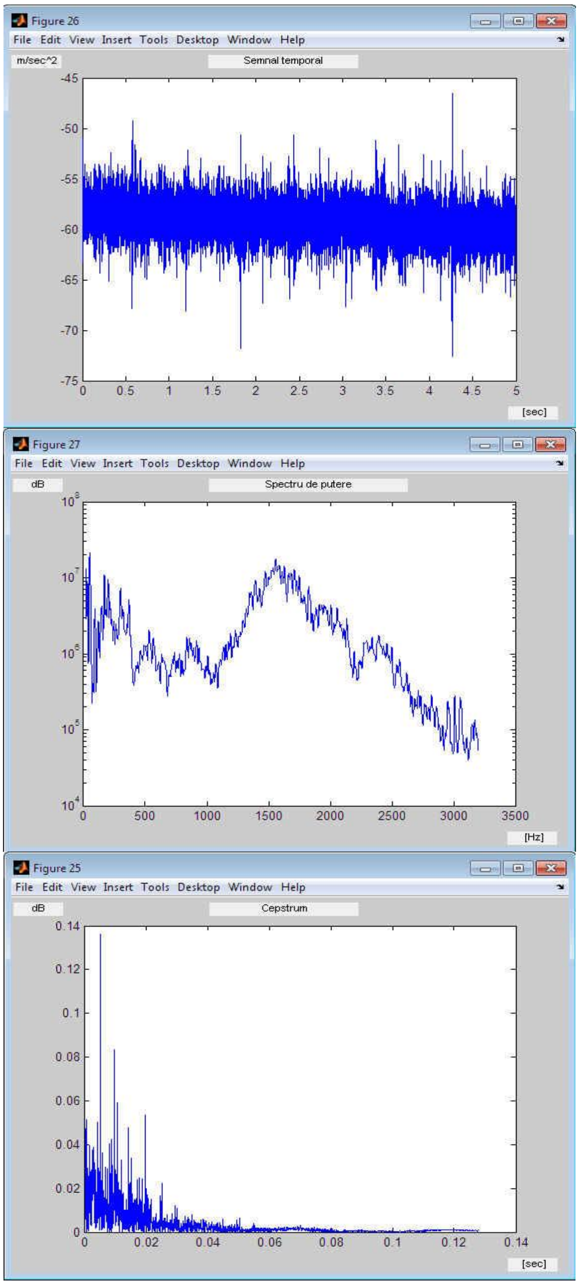

Fig.3. Time Signal, Frequency Spectrum, Cepstrum of a Cracked Inner Race

unique signal because these bearings have clearance opposite the load zone. Depending on the size of the bearing, the clearance varies from about 0.003" to over 0.014".

If the inner race is rotating, the rollers may stop rolling when the defect is out of the load zone. The BPFI is only generated while the defect is in the load zone during each revolution.

Several points should be made concerning the time signal, frequency spectrum, cepstrum in Fig. 3.

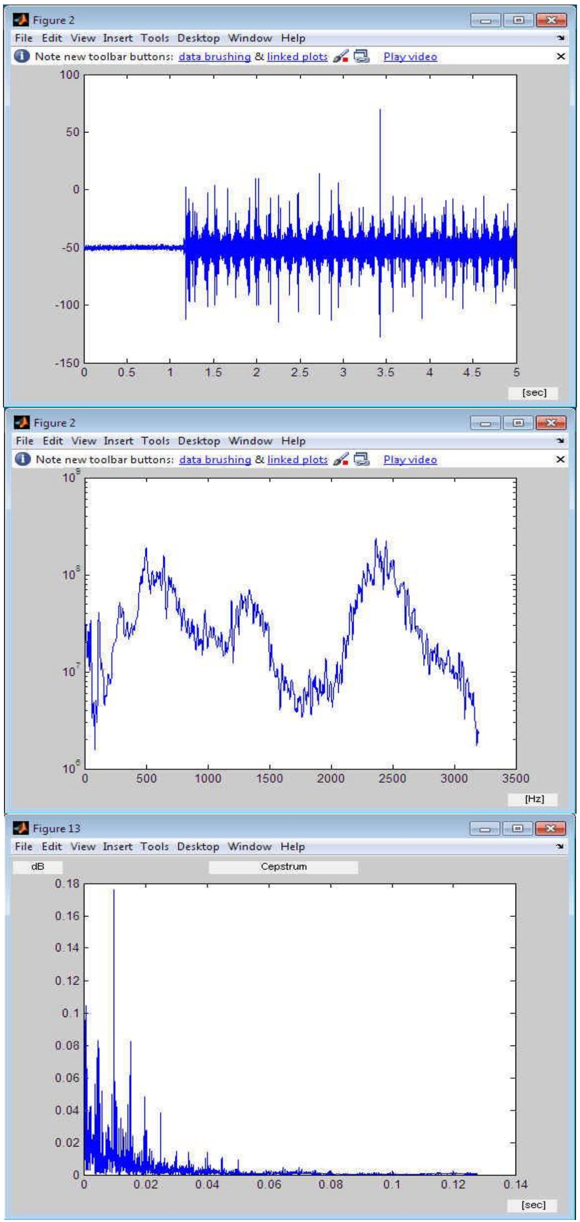

Fig.4. Time Signal, Frequency Spectrum, Cepstrum of a Defective Balls

-The fundamental train frequency is present with the second and third harmonics. This indicates looseness.

-The unit speed is present and identifies residual imbalance, loading, looseness, etc.

-The spectral line at 1570 Hz is the measured BPFO. The sideband to the left at 1250 Hz is the difference frequency between BPFO and motor speed. This

sideband indicates the defect is large enough to permit movement of the shaft and approximates defect size.
-The spectral line at 1850 Hz is the second harmonic of BPFO. In this case, the second harmonic is probably caused by fragment denting and/or frosting and is not related to defect size.
-The spectral line at 2992 Hz is 5 X BPFO and is also caused by fragment denting and/or frosting.
In difficult conditions (Fig. 4), the **FTF** can excite one or more natural frequencies.

When this occurs, the natural frequencies are modulated by the source of excitation, the FTF in this case. The band of frequencies around 2.378 kHz is an excited natural frequency. The difference frequency between the spectral lines equals the FTF. Balls in a bearing may never generate BSF or 2 x BSF because balls roll in one direction and spin in the other direction. This action virtually prohibits a defect on a ball from hitting anything with the repeatability required to generate one or two times BSF.

After a defect has begun, it will get larger, and the spectral bandwidth will get wider until the spectrum is modulated with the speed of the rotating unit. The ball pass frequency and the ball pass frequency plus or minus the unit speed may be generated. Modulation can continue until the ball pass frequency is no longer apparent. (In some cases the amplitudes of the sum and difference frequencies are equal to or exceed those of the ball pass frequency.) The spectrum then becomes a series of frequency peaks whose difference frequency is equal to the unit speed. These phenomena occur when a growing fatigue spall is present on the race.

From the analysis it is observed that the amplitude of the cepstrum in the three cases analyzed is 0.18 dB for defects of the rolling element, inner ring of 0.14 dB and 0.08 dB for the outer ring.

III. Conclusion

Except for defects that occur in bearing components during manufacturing, the cage is usually the last component to fail. The typical failure sequence is as follows: defects form on the races, the balls, and then finally the cage. A severely damaged cage can cause constant frequency shifts that are observable with the use of a real-time analyzer. When the cage is broken in enough places to allow the balls or rollers to bunch up, wide shifts in frequencies accompanied by loud noises can occur. When these signs are present, bearing seizure is imminent.

A single defect in a bearing can be identified by the frequency it generates. When several defects are present, some or all of them may be identified from the basic frequency, but sum and difference frequencies are almost always present in the spectra.

Analysis of complex spectra can be difficult. One approach is to first identify any basic frequencies. Multiples of the basic frequencies - lx, 2x, etc. - must then be identified. Finally, any remaining frequency peaks are identified as combinations of the basic frequencies already identified.

In rare cases when one or more rollers are missing from a bearing, the FTF can be generated. The problem occurs as a puise at the FTF. The frequency spectra contain a series of harmonics of the FTF. The amplitude of the first harmonic is quite low, the second, third, and fourth harmonics are higher in amplitude as determined by the puise.

Sometimes, attempts to lubricate sealed or shielded bearings can cause the seal or shield to deflect inward. If the cage touches the seal or shield, the FTF and /or two times FTF plus harmonics can be generated.

Excessive clearance in an antifriction bearing can cause the generation of a discrete frequency at the FTF and/or modulations of the FTF at rotating speed and harmonics.

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