Spectral Analysis for the Diagnosis of Defects on Industrial Machines

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ABSTRACT: *The aim of this study is to carry out a new approach for a better organization and prevision in terms of predictive maintenance for production machines in large factories.*
For this purpose, a large number of measurements have been done on industrial machines such as reduction gearings, engines and pumps. We placed particular attention on vibration measurements located in strategic points in order to obtain a precise and accurate spectral analysis [1].

KEYWORDS - *vibration monitoring, vibration, spectral analysis*

I. INTRODUCTION

Recently, vibration monitoring and machine diagnosis have become very important and at the same time key for a serious maintenance program in order to save money. Predictive maintenance regulates these types of applications to determine the effective working condition of a single machine, fixing the safety limits before its final blemishing.
The concept is to intervene in terms of preventive maintenance [2] in order to organize a routine checking program for each working machine. One very effective and inexpensive way to do this is to analyze the irregular vibration conditions by using digital spectrometers.
Given that there are several causes that may determine irregular vibrations and generally induced by structural problems, irregular operation or fatigue wear of a single component, the goal of this research is to determine the exact cause without dismantling the machine and the relative interaction between the analyzed machine and the others that are inside the factory.
More precisely, we want to know if the vibration of the totality of the working machines can influence the measurement of a single analyzed machine changing, of course, the safety limits that were previously adopted.
In figure 1 a schematic view shows the potential induced vibration in a global machine system.

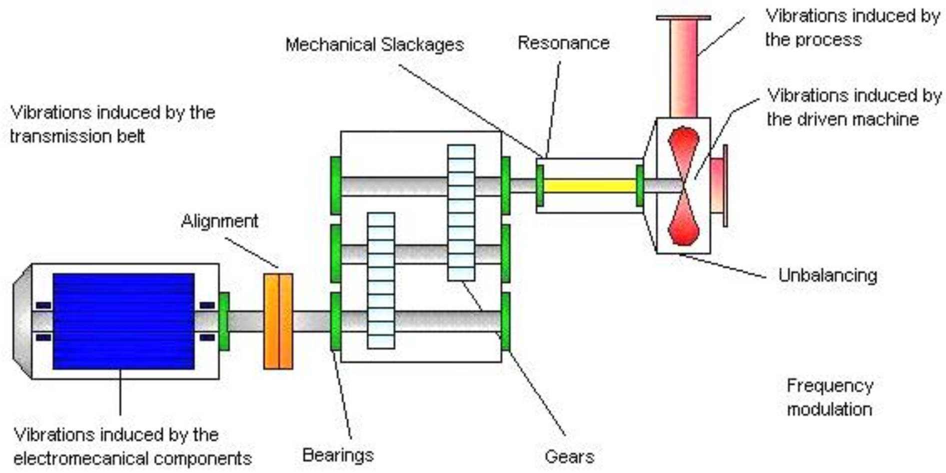


Fig. 1: Vibration induced by each single component

II. VIBRATION CONDITIONS IN INDUSTRIAL MACHINES

The vibration phenomena is characterized by alternate movements with small amplitude and high frequency that are often superimposed to the normal cinematic movement of a single component [3]. The most significant problems caused by the vibration effect are:

- Fatigue fracture;
- Variation of the initial performances;
- Noise production;
- Dangerous conditions for workmen.

As we know, vibrations are phenomena determined by transformation of potential energy in kinetic energy. Disturbing forces on the mechanical system generally cause them and their amplitude depends on the elastic properties of the same system (forced vibrations).

This phenomenon may also occur without the presence of external forces, in particular when the motion or rest conditions of the system are disturbed by certain unstable initial conditions (free vibrations).

Experimentally it is known that in a mechanical system, after a variation of the initial conditions and without external forces, the vibration phenomena trend goes to zero relatively fast.

If elastic and inertia forces were present in the system, the vibration amplitude decay would not be justified because the above-mentioned forces would be absolutely conservative forces. In the reality, dissipative effects are always present transforming at every oscillation cycle a part of the total energy (elastic and kinetic) in thermal and acoustic energy.

These forces can be caused by several effects and are called damping forces. The presence of damping forces is also important in forced vibrations because they limit the vibration amplitude of the whole system especially with regards to the resonance frequencies. In figure 2 a basic linear scheme for mechanical vibration is shown.

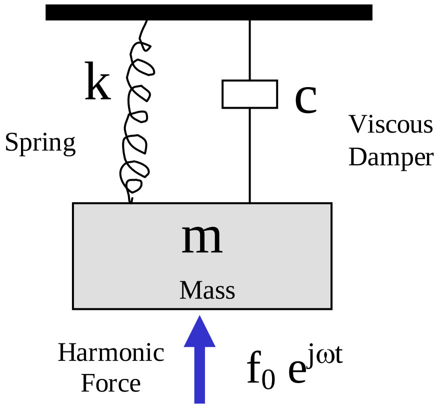


Fig. 2: Scheme of a vibration system

The mathematical model allows studying the mass displacements in relation to the static equilibrium condition:

$$m\frac{d^2x}{dt^2} + c\frac{dx}{dt} + k\,x = f_0\, e^{j\omega\, t} \tag{1}$$

Where x is the vertical mass displacement.

The weight, because of its constant value, affects only the vibration equilibrium condition leaving the amplitude and frequency at their initial values. For this reason it can be not taken into consideration for the dynamic study. The models, which have more than one degree of freedom, are studied by means of systems of ordinary and linear differential equations with constant coefficients having a structure similar to equation (1). The mathematical solution for the linear differential equation is carried out by the addition of two terms:

- The solution at (t) from the associate homogeneous;
- The particular integral xr(t) that is a generic solution of the complete equation.

The solution of the homogeneous equation, which is reformulated considering the dimensionless parameters, describes the free vibrations of the system:

$$x_t(t) = e^{-\omega_n \zeta\, t} \cdot Asen\left(\omega_n \sqrt{1-\zeta^2}\, t + \varphi\right) \tag{2}$$

Where:

$\omega_n=\sqrt{(k/m)}$ [Hz]	- natural pulsation of the system
$c_c = 2\cdot\sqrt{(k\cdot m)}$ [Ns/m]	- critical damping
$\zeta = c/\, c_c$	- damping factor

Machines can be organized into four different groups depending on the vibration type:

- **Alternative machines with rotary and alternative components** (diesel engines and some pumps and compressors): vibrations are measured on the main structure and those with low frequency are more significant;
- **Rotary machines with rigid rotors** (particular types of electric motors, single stage slow pumps): vibrations are measured on the main structure close to the bearings and are connected to the forces generated by the rotor (especially due to the unbalancing);
- **Rotary machines with flexible rotors** (electric generators with steam turbine, pumps and multi-stages compressors): in order to reach the working speed during acceleration, these machines go through different

critical speeds and for this reason, the rotor vibrates with different modes. Vibrations measured on the main structure are no longer connected to the rotor vibrations (vibrations measured on the bearings could be small and at the same time the rotor can vibrate with large amplitudes that can dangerously change working conditions). In these cases, it is convenient to directly measure the shaft vibrations;

- **Rotary machines with quasi-flexible rotors** (low pressure steam turbine, axial compressors): these machines have special rotors whose vibrations, measured close to the bearings, are connected to the shaft vibrations.

The classification of the vibration level depends on the standards adopted for the measurements, the frequency range and other factors. Furthermore, the observation magnitude depends on several factors such as the vibration amplitude, its speed and its acceleration.

- Considering vibrations with frequencies between 10 Hz and 1000 Hz, the vibration speed is the most reliable parameter.
- Considering pure harmonic vibrations, the most utilized parameters are the peak and the average quadratic values related to the vibration level.
- Considering more complicated types of vibrations, the only parameter considered is the average quadratic value particularly for rotation speeds between 600 and 12000 rpm - 10÷200 Hz.

For these reasons, the International Standard Organization (ISO) has introduced the ***'vibration severity'*** that is defined as the highest average quadratic vibration value in a frequency range between 10 and 1000 Hz. For measurements of this category, measurement points are predefined for every machine. The measurements are generally triaxial and are in correspondence of the block engine frames. After having recorded the instant vibration speed, its quadratic value can be evaluated using the following equation:

$$v_{rms} = \sqrt{\frac{1}{T}\int_0^T v^2(t)dt} \tag{3}$$

Considering harmonic vibrations with frequency ω_i, the vibration speed can be calculated using the following equation:

$$v_i = \hat{v}_i \cos \omega_i t \tag{4}$$

If the measured vibrations contain the frequencies ω_i (i=1, 2.. ,n), knowing the vibrations (s_i), the speed (v_i) or the acceleration (a_i) for each frequency, than the vibration level (***vibration severity***) can be expressed by:

$$v_{rms} = \sqrt{\frac{1}{2}\left[\left(\frac{\hat{a}_1}{\omega_1}\right)^2 + \left(\frac{\hat{a}_2}{\omega_2}\right)^2 + \ldots + \left(\frac{\hat{a}_n}{\omega_n}\right)^2\right]} = \sqrt{\frac{1}{2}\left[(\hat{s}_1\omega_1)^2 + (\hat{s}_2\omega_2)^2 + \ldots + (\hat{s}_n\omega_n)^2\right]} =$$
$$= \sqrt{\frac{1}{2}\left[\hat{v}_1{}^2 + \hat{v}_2{}^2 + \ldots + \hat{v}_n{}^2\right]} \tag{5}$$

If the instrument directly gives the speed quadratic mean value then the vibration level can be expressed with a reliable approximation by the following relation:

$$v_{rms} = \sqrt{\frac{1}{2}\left[R_{\max}^2 + R_{\min}^2\right]} \tag{6}$$

Where R_{max}, R_{min} are the maximum and the minimum values taken by the instrument.

Considering measurements of this sort, the most difficult part continues to be the interpretation of the values in terms of spectral analysis diagrams that represent the single delicate operation.

The exact diagnosis of the whole sampling process is today the main problem that has to be solved in order to get reliable results. Our study gives a very simple but precise approach for a correct measurement process and the relative interpretation of results.

III. LOCALIZATION OF THE MEASUREMENT POINTS

The choice for the measurement points is the most delicate and difficult operation for these kinds of tests.
The incorrect position of the instrument will carry out generic results that do not correspond to the real condition of the analyzed machine in terms of vibrations.
To obtain reliable results for vibration measurements, the measure must be done in specific points that are directly connected to the main structure and the signal must be resolved in its fundamental components.
Analyzing the entire machine, it is often best to choose the bearings to locate the measurement because of their purpose. In fact they receive the strongest dynamic loads and exciter forces, which is why they are considered the most critical points of the entire machine.
Following the theory, the measurements should always be triaxial, especially when considering rotary machines, but the experimental experience shows that even two measurements are sufficient to obtain reliable values:

- If the axial vibrations values are not very high, the measures are carried out in two different points and are in particular perpendicular to the bearing axis (generally the directions are horizontal and vertical in order to consider the gravity effect in the fluid bearings);
- If the axial vibration values are very high, the directions are radial and axial.

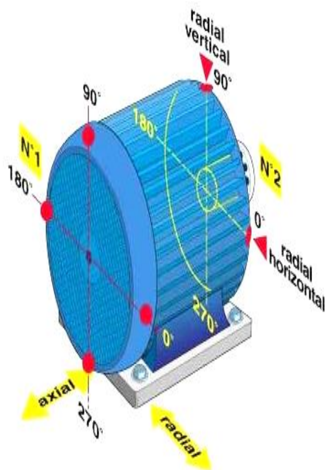


Fig. 3: Location of the measurement points

By analyzing the spectral vibration signal we can find many components in terms of frequency, which can be easily associated to a particular vibration source.
For others frequencies, the analysis can be much more difficult and cannot be related to a known vibration source (very small forces can sometimes determine interesting vibration phenomena because of their frequencies which are very close to those of machine resonance). Moreover, most of the forces are periodic in cyclic machines, but not completely sinusoidal. In this particular case, together with the fundamental frequency, we also find all the other harmonics. Figure 4 shows the principal points for a correct measurement on a generic support. Figure 5 shows a scheme of a real set-up using a digital instrument.
The instrument utilized for our test is the Vibscanner 5.420 and the main characteristics are reported in table 1:

Primary voltage	115/230 V (selectable); 50 - 60 Hz
Secondary voltage	12 V , max 400 mA
Charging duration	Max 8 hours (depends on the charge level)
Env. Protection	IP 20
Operation temp.	0°C to 40°C / 32°F to 104°F
Dimension	86 x 50 x 48 mm / 3.4 x 2.0 x 1.9 inch
Plug Type	3 – pole BINDER plug
Cable Length	1.8 m / 5.9 ft

Tab. 1: Instrument specifications

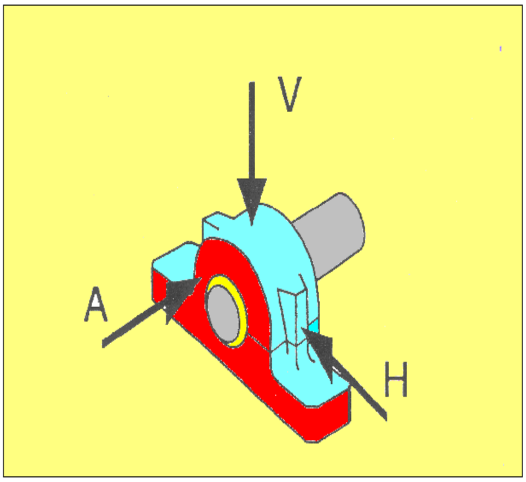


Fig. 4: Location of the measurement points

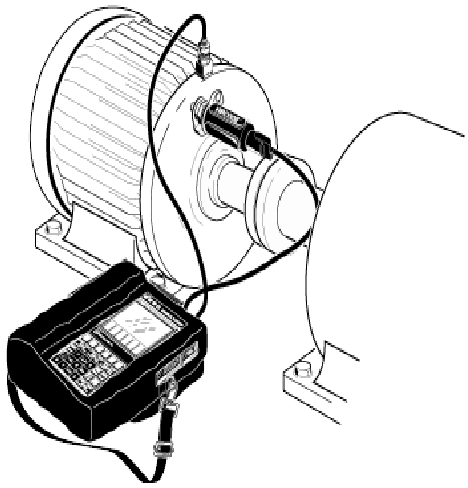


Fig. 5: Probe location on the component

Another very important information that we may need is the bearing type and the number of teeth of the analyzed gears. In particular the teeth number allows us to calculate the meshing frequency that is mandatory to correctly interpret the spectrum analysis.
In fact, by knowing the meshing frequency, it is possible to predict possible damage on the gears as shown in table 2.
Occasionally for old machines whose instruction are lost, the best solution is to take clear pictures during ordinary maintenance in order to store the inside part of the machine with all its components clearly visible as shown in figure 6.

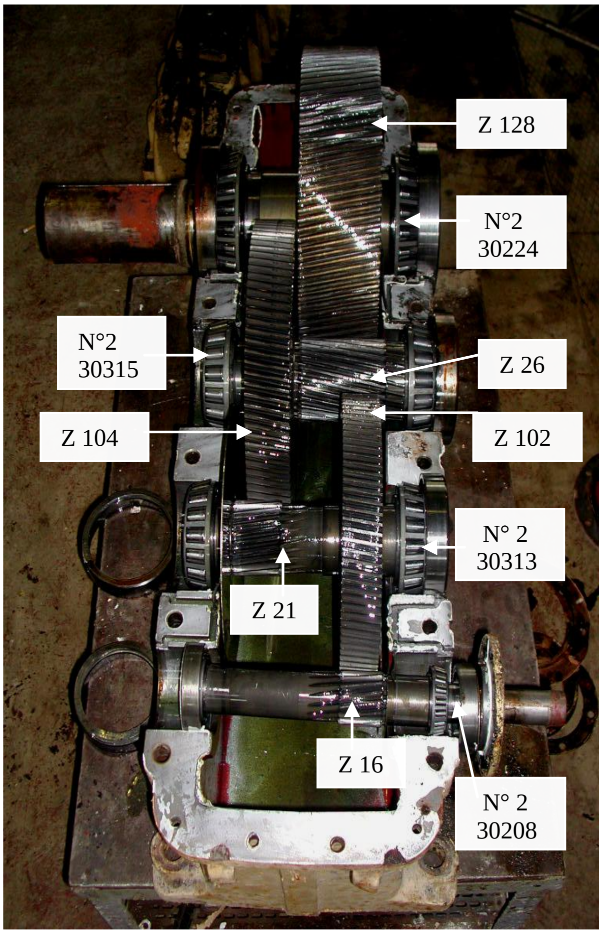


Fig. 6: Internal view of a Flender reduction gearing

REDUCTION GEARING FREQUENCY			
Shaft 1 (1460 rpm) (24,33 Hz)			
Gear 1 Z 16	389,33 Hz	I° Meshing Freq.	
Gear 2 Z 102	778,67 Hz	II° Meshing Freq.	
Shaft 2 (229,019 rpm) (3,8170 Hz)			
Gear 3 Z 21	80,16 Hz	I° Meshing Freq.	
Gear 4 Z 104	160,31 Hz	II° Meshing Freq.	
Shaft 3 (46,24 rpm) (0,7707 Hz)			
Gear 5 Z 26	20,04 Hz	I° Meshing Freq.	
Gear 6 Z 128	40,08 Hz	II° Meshing Freq.	
Shaf4 (46,24 rpm) (0,7707 Hz)			

Tab. 2: Reduction gearing frequency

IV. BASIC SPECTRUM SUSRVEY

A measurement test has been carried out for each machine with special consideration for the period in which the analyzed factory is not working. In particular, we consider the basic spectrum as the one obtained from a single machine while the whole factory is shut down. In this way, for every machine we can obtain the real vibration condition without any interference from other vibration systems. It is important to specify that this approach is not always possible because many factories run 24 hours a day. A particular application was developed in a sugar factory that is operational only three months a year due to the fact that sugar production is active only during the sugar beet season. This condition allowed us to proceed with the measurements whether or not it was during the working period.

After having all the basic spectrums for each machine, the net step was to re-measure everything during the working period while all the machines were on. By then comparing the new spectrum behavior with the previous one, it could be easily seen that the differences were significant and particularly in correspondence with the frequency, speed (or acceleration) and peaks (fig. 7).

The most evident consideration was to verify the irrefutable incidence of external vibrations caused by other working machines on the one taken into consideration.

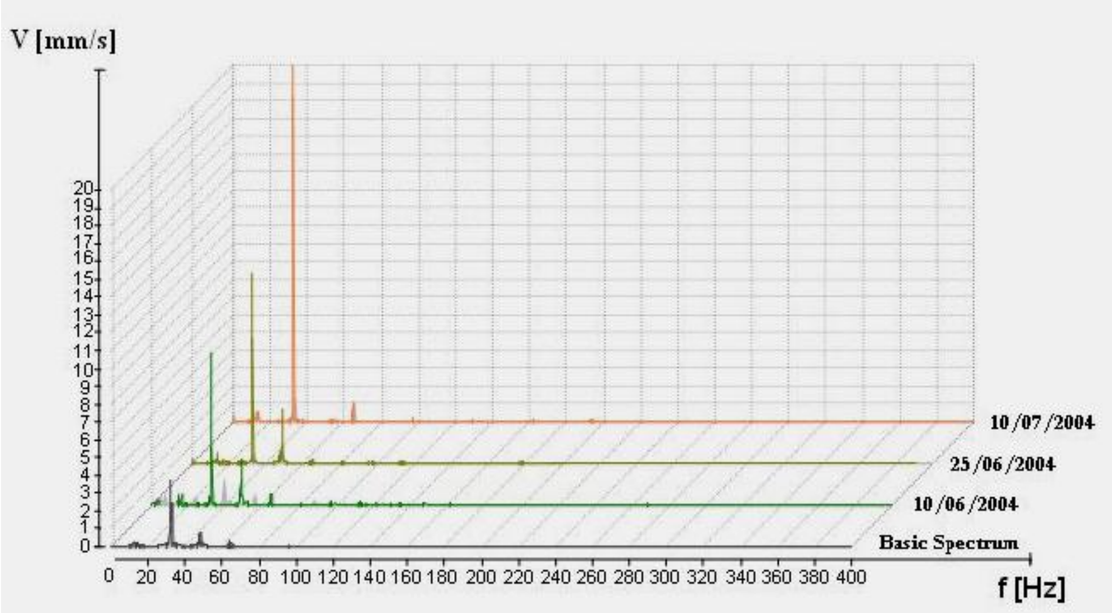


Figure 7: Spectrum diagram

The importance of defining a clear difference between the basic spectrum that correspond to a single machine working and the complete spectrum that instead is obtained when the whole factory is working, is fundamental in obtaining the exact working condition for each analyzed machine. Without these considerations it is very common to obtain errors that may cause an unnecessary maintenance intervention by turning off the machine in order to dismantle it and resulting in enormous economic consequences.

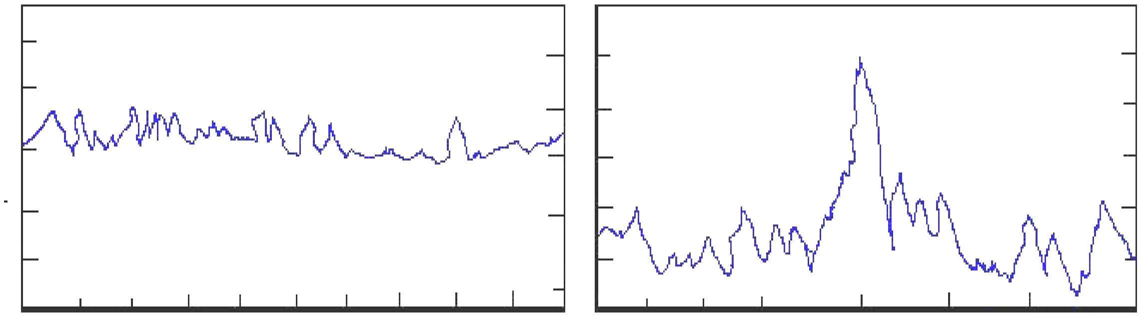
V. SIGNAL PROCESSING

After having located the exact measurement points and of course the appropriate sensors the experiments were conducted with the digital instrument acquiring the time function signal.

The first step, [4] in order to elaborate the obtained signal is the " *weighting* " that comes immediately after the "*anti-aliasing*" filtering performed on the analogic signal. In particular, with this operation all the parts included in the output signal and those that are considered pure noise, are automatically eliminated. Considering, for example the vibrations caused by the unbalancing of a rigid rotor, the high frequency contributions due to the external noise determine a dirty signal that does not correspond to the real conditions.

In order to eliminate this inconvenience, due to the fact that the vibration frequency related to the unbalancing is well known (they are synchronized with the rotor), the most appropriate filter to utilize is the "*pass-band*" type that is balanced on the rotation frequency of the same rotor. Apart from filtering the temporal signal, we performed the ***"Temporal Mean"*** on the vibration signal recorded in a specific period. We examined all the rotary machines, reduction gearings especially all the machines characterized to be working with a periodic regime (rotary and alternative). The temporal behavior analysis carried out in a period allowed us to identify not only the real working condition of the reduction gearing, but also the real location of the damage, even a broken tooth. In figure 8 two different diagrams related to an undamaged and damaged tooth are reported as an example of our approach.

Signal average plot



Undamaged bearing Temp → Damaged bearing Temp →

Fig. 8: Signal-average plots for rolling element vibration

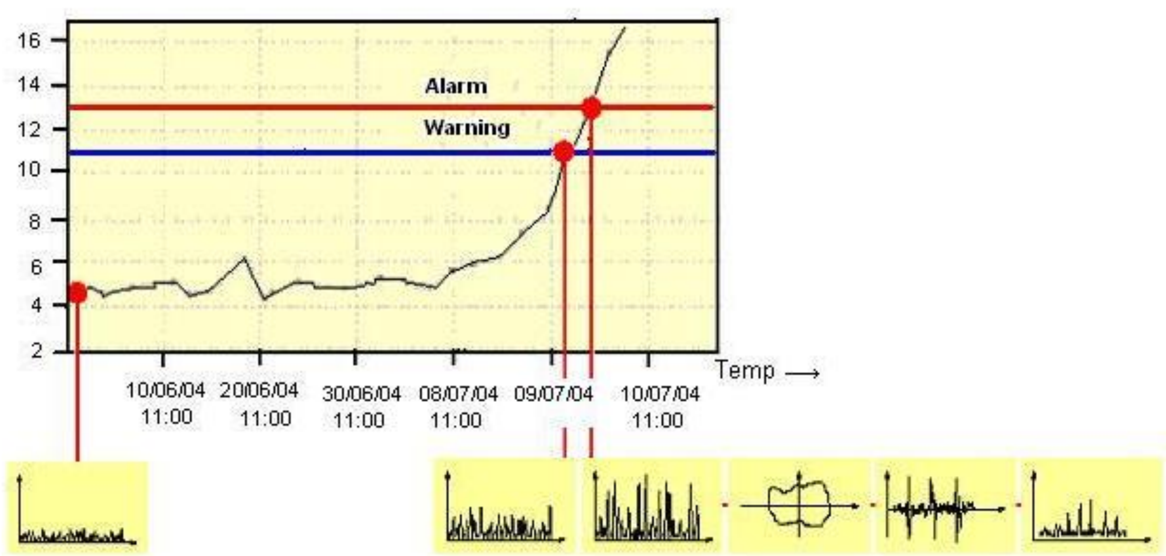
VI. CHARACTERIZATION OF THE SAFETY LEVELS

Finally, in order to better verify the proposed maintenance approach, all the machines that presented high peaks of frequency, speed or acceleration were dismantled (fig. 6) to analyze the component taken into consideration (bearings or gears). In this way, comparing the damage of the real component with the relative peak in the spectrum diagram, a global analysis regarding the safety level for each machine was carried out.

Above all, we could easily obtain a clear situation regarding the real operating condition and predict a possible maintenance intervention.

To obtain reliable monitoring, a precise temporal analysis of the global vibration values was carried out and recorded in a database for every machine.

To complete this operation, all the values were reported on a diagram in order to compare them with the alarm and warning values experimentally obtained shown in figure 9. At the end of the test, the cause that produced the anomalous vibration can easily be found and the relative maintenance intervention can be planned.



Basic Spectrum Warning Alarm Deepened diagnosis Spectral Analysis

Fig. 9: Trending of the operation conditions

VII. CONCLUSIONS

At the end of this research, we can say that the analysis of the basic spectrum and its comparison with the global ones can undoubtedly be considered a reliable tool to help the predictive maintenance and verify the real working condition of the examined machines.
In addition, the choice of the measurement points has been fundamental for a good application of the instrument probe in order to obtain the most real vibration values.
The experienced induced us to select a few of the several organs inside the machine. The bearings and the gears were the best choice for this kind of measurement. Finally, by knowing the number of teeth we were able to easily to carry out the meshing frequency.

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