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Detection of an Imbalance Fault by Vibration Monitoring: Case of a Screw Compressor

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ABSTRACT

The evolution of the means aimed at improving the availability of strategic equipment requires a credible level of maintenance with the development of original monitoring techniques such as vibration diagnostics. The vibration monitoring of strategic equipment through a vibration diagnosis requires mainly the identification of vibrational images of the different types of damage. However, one of the important vibration problems is essentially due to the phenomenon of imbalance. This phenomenon corresponds to an imbalance of the rotor due to the offset between the axis of inertia and the axis of rotation, which causes significant and cyclical vibrations. The aim of this study is to analyze the vibratory behavior of a screw compressor to improve its reliability and consequently its availability. To identify the imbalance fault, two sets of vibration measurements on April and January were fundamentally examined at the compressor level. Fourier transform based on spectral analysis was used to create a vibration detection approach with vibration signals. The comparison of the results obtained with that of the simulation resulting from the model of dynamic behavior of the compressor is conclusive.

1. Introduction

The detection of faults in rotating machines has been extensively studied by [1]-[3] and the performance analysis of industrial machines by modeling has been widely examined by [4] and [5]. The detection systems include experimental methods, the conventional methods one which therefore requires a high level of human expertise [1], [6]-[8] and combined techniques as Statistic Filter (SF) and Hilbert Transform (HT) which are joined with Moving-Peak-Hold method (M-PH) [9] and the unconventional one with an approach of artificial intelligence by the method of neural networks for the prediction of the vibratory behavior of mechanical systems [10] and [11].

For the detection of the imbalance defect, there is many techniques based on spectral analysis. In fact, frequency domain analysis is a technique widely adopted for the study of system vibrations [1], as it

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requires knowledge of the fundamental frequencies of the system [2]. Important to realize that other methods such as the numerical method allow to better identify the optimal location of the measurement points by a finite element code in order to model the vibratory behavior of the systems [12]. Recently, data-based methods have shown great potential in the intelligent diagnosis of defects [13] and [14]. A number of authors have investigated approaches based on artificial neural networks for prediction of misalignment defects and imbalance, this method is carried out in three main steps, initially a measurement step, then the learning step, and thirdly the prediction step [10]. Further, a new imbalanced fault diagnosis framework based on cluster Majority Weighted Minority Oversampling Technique (MWMOTE) and moth-flame optimization (LS-SVM) is proposed by Wei et al. [15] to address the fault diagnosis of rolling bearings under complex conditions. However, the detection of unbalance faults with frequency analysis has several advantages because it is a simple technique to implement, reliable and avoids the use of additional analysis software [1] and [2] adapt to industrial scale.

The commonly used tools to extract a fault indicator are spectral analysis techniques, several authors have used classical analysis techniques such as the Fourier transform [16] and [17], the Hilbert transform [18]-[20] or time-frequency distributions [21]-[23]. These aforementioned methods can identify unbalance faults thereby improving diagnostic performance.

Furthermore, spectral analysis makes it possible to detect anomalies through readings of acceleration or speed for certain frequency ranges in order to follow the state of degradation of an equipment. Similarly, it can also be used to carry out a diagnosis by interpretation of the shape of the vibratory signal [6] and [8]. It consists necessarily of filtering the measured vibration signal on the screw compressor and carrying out a systematic analysis to look for the presence of vibration images of all the faults likely to affect the installation in question. This allows access to the diagnosis successfully, that is to say, to identify precisely the nature of the anomaly and if possible specify its severity. In addition, the envelope spectrum is a tool particularly suitable for monitoring periodic shocks and it also makes it possible to quantify the impulse nature of the signal, hence the analysis of the envelope is best suited for detecting bearing faults [1], [6], [7] and [24].

Add to this, several phenomena of instability of the compressor are mainly the cause of their malfunctions and can create significant vibrations such as the imbalance and misalignment of the shafts, etc. As a matter of fact, the presence of small defects can lead to very significant vibration amplifications. Hence, the vibratory behavior of rotating machines and their processing is carried out on the measured signals, in many industrial applications, the vibration is characterized mainly by its frequency, its amplitude and its nature [25]. Eventually, imbalance is a term which characterizes a mass which is not perfectly distributed over a volume of revolution resulting in an imbalance. As the axis of inertia of the rotor no longer merges with the axis of rotation.

Even so, when the rotor is driven in rotation with an imbalanced mass, these results in centrifugal forces which increase the load on the bearings and a vibration appears on the rotor. It should be noted, that this is the specific characteristic of an imbalanced machine. This centrifugal force is thus the excitation force of the imbalance which is given by *Eq. (1)* [17] and [26]:

$$\mathrm{F_c} = \mathrm{m\,r}\,\omega^2. \tag{1}$$

On the other hand, the rotational speed is calculated using *Eq. (2)*:

$$\omega = 2\,\pi\,\mathrm{f}. \tag{2}$$

With:

- F_c : imbalance strength (N);
- m : imbalance mass (kg);
- r : eccentricity (m);
- ω : rotational speed (rad/s);
- f : frequency of rotation (Hz).

The study presented consists of carrying out an analysis of the vibratory behavior of the screw compressor based on the operating parameters and the spectral indicators recorded.

The core objective of this work is to prevent failures in order to ensure optimal availability of the screw compressor, based on experimental tests, with regard to a vibration study.

2. Presentation of the Screw Compressor

Initially, the application system is a screw compressor driven by an electric motor, the two parts are coupled by means of an elastic coupling, the schematic diagram of the screw compressor with the kinematic chain and measuring points is clearly presented in *Fig. 1*.

The measurements were carried out precisely on the motor and the screw compressor, using a mobile accelerometer mounted on the measurement points indicated in *Fig. 1*.

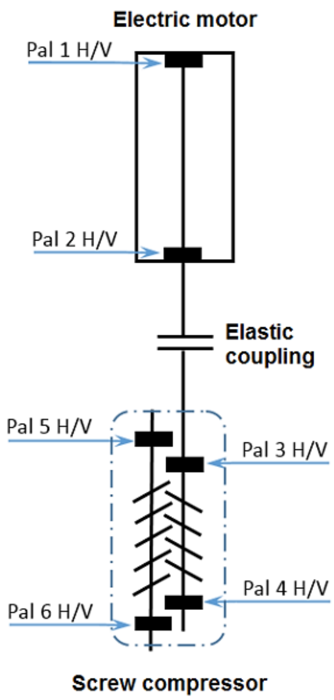

Fig. 1. *Schematic diagram of the screw compressor.*

Characteristics of the screw compressor:

– Electric motor:

  Power: 45 kW;

  Rotational speed: 2940 tr/min;

– Screw compressor:

  Power: 60 CV;

– Rolling:

  Pal1: single ball 6313;

  Pal2: single ball 6313;

  Pal3: cylindrical roller NU 211;

  Pal4: angular contact 7311B;

  Pal5: cylindrical roller NU 211;

  Pal6: angular contact 7311B;

– Coupling:

  Elastic;

– Vibration measurement point:

  H: horizontal;

  V: vertical;

– Fixing:

  Flexible supports;

Five series of measurements are respectively collected over the last two years of monitoring the evolution of the overall level of the six compressor stages are summarized in *Table 1*.

To begin with, the evolution of the overall level allows us to understand the state of the compressor by comparing the values of 'NG' with ISO standard 10816, but it does not allow to identify and discover the real causes behind the failure.

We only used the two series of vibration measurements of January 15 and April 20 in global 'NG' level in *Table 1* to carry out this study.

Table 1. *Measurement of the last two years reference of vibration (NG).*

Measuring point	July 29 RMS (mm/s)	August 21 RMS (mm/s)	February 8 RMS (mm/s)	April 20 RMS (mm/s)	January 15 RMS (mm/s)
Rh Pal1	*2.46*	*2.81*	*2.63*	*2.71*	*9.21*
Rv Pal1	*2.23*	*2.46*	*2.33*	*4.94*	*5.14*
Rh Pal2	*2.1*	*2.62*	*2.82*	*3.1*	*9.01*
Rv Pal2	*3.27*	*3.39*	*3.68*	*4.17*	*5.14*
Rh Pal3	*5.22*	*5.72*	*5.23*	*5.14*	*7.43*
Rv Pal3	*5.03*	*5.23*	*6.59*	*7.85*	*7.85*
Rh Pal4	*6.14*	*6.34*	*6.05*	*6.88*	*6.87*
Rv Pal4	*5.11*	*5.33*	*5.82*	*7.36*	*7.43*
Rh Pal5	*6.21*	*6.4*	*6.01*	*4.36*	*1.94*
Rv Pal5	*5.85*	*6.3*	*6.69*	*8.14*	*6.10*
Rh Pal6	*6.34*	*6.95*	*6.76*	*6.98*	*7.53*
Rv Pal6	*4.65*	*4.75*	*4.56*	*4.85*	*6.01*

3. Vibratory Behavior

3.1. Dynamic Behavior Modeling

The representation of *Fig. 2* is a simplification of the manifestation of the imbalance. Several damages cannot be located precisely as being a mass *(m)* and also fixed at a specific place. Indeed, for example, corrosion, wear or a deposit of impurities are distributed more or less uniformly over the entire surface, which thus has the effect of shifting the center of gravity. Add to this, the imbalance is modeled by a point mass placed at a given distance *(r)* from the axis of rotation *(Oy)* and a given distance *(d)* in accordance to the origin of the reference *(O)* as shown in *Fig. 2* and *Fig. 3*.

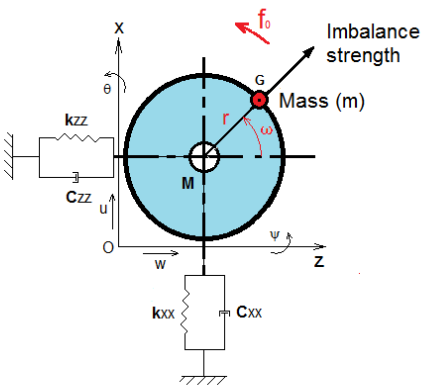

Fig. 2. *Imbalance modeling.*

The fan of *Fig. 2* is modeled by a node having four degrees of freedom: two rotation *(ψ)* and *(θ)* around the axes *(Ox)* and *(Oz)* and two displacements of translation *(u)* and *(w)* along the axes *(Ox)* and *(Oz)*. The disc is considered to be rigid and symmetrical.

The coordinates of the imbalance in the fixed coordinate system *R (Oxyz)* are given by:

$$\overrightarrow{OG}_{|R} = \begin{Bmatrix} u + r\cos\omega t \\ Cte \\ w + r\sin\omega t \end{Bmatrix}. \qquad (3)$$

Hence the speed:

$$\overrightarrow{V_{G/R}} = \frac{d\overrightarrow{OG}}{dt} = \begin{Bmatrix} \dot{u} + r\,\omega \sin \omega t \\ 0 \\ \dot{w} - r\,\omega \cos \omega t \end{Bmatrix}. \tag{4}$$

The kinetic energy of the imbalance:

$$2\,T = m\left\|\overrightarrow{V_{G/R}}\right\|^2. \tag{5}$$

$$2\,T = m(\dot{u}^2 + \dot{w}^2 + \omega^2 r^2 + 2\omega r\,\dot{u}\sin \omega t - 2\omega r\,\dot{w}\cos \omega t). \tag{6}$$

The term $(\omega^2 r^2 / 2)$ is constant and will not occur in the equations, the expression for kinetic energy can be jointly approximated by:

$$T = m\omega r(\dot{u}\sin \omega t - \dot{w}\cos \omega t). \tag{7}$$

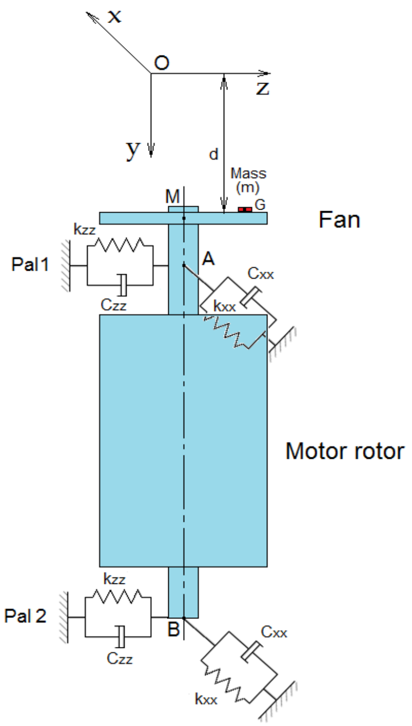

Fig. 3. *Simple model of the motor shaft line.*

In *Eq. (1)*, the term (ω^2) indicates that the force increases with the square of the speed. This explains why compressors at high speed 2940 rpm are very sensitive to the imbalance fault. Therefore, his force (F) rotates at the same frequency as the rotor and it consequently produces sinusoidal type vibrations on the bearings which rotate at the same frequency as the rotation frequency (f).

$$F = F_C \sin(2\pi f t). \tag{8}$$

Additionally, the imbalance defect, perhaps either out of mechanical origin in the form of degradation or deposits of impurities on the rotor or is of thermal origin caused by a modification of the geometry of the rotating parts due to the rise in the temperature. In most cases, this type of fault can be remedied either by cleaning or by balancing. In this modeling, the rotating system consists of a motor rotor, a fan similar to a disc and two bearings *(Pal1)* and (*Pal2)* which are subjected to imbalance forces. However, the behavior equations will be deduced from a formulation of the application of the lagrange equations.

In the case of generalized coordinates, the lagrange equations are written as the following:

$$\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{q}_i}\right) - \frac{\partial T}{\partial q_i} + \frac{\partial U}{\partial q_i} = Q_i. \quad (9)$$

Where *(T)* and *(U)* are the kinetic and potential energies respectively. Correspondingly, the (q_i) are the generalized coordinates of the system and the (Q_i) are the generalized forces. Henceforth, the objective from these lagrange equations of this modeling that we can deduce the equation of motion in order to identify the response *x (t)*.

The Lagrange equation *Eq.(9)* applied to kinetic energy:

$$\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{q}_i}\right) - \frac{\partial T}{\partial q_i} = -m\,\omega^2 r \begin{Bmatrix} \cos\omega t \\ 0 \\ \sin\omega t \end{Bmatrix}. \quad (10)$$

After all, imbalances are the forces of asynchronous excitations. Certainly, the bearings are modeled in *Fig. 3* by considering the shaft as a linear viscoelastic solid, it is about the model of Kelvin-Voigt, characterized by a damping (C_{ii}) in parallel according to *(Ox)* and *(Oz)* and a stiffness (k_{ii}).

3.2. Signal Processing

In the field of signal processing, we are particularly interested in the sinusoids of the *x (t)* response.

The displacement signal *x (t)* of the screw compressor is given by *Eq. (11)* [17]:

Displacement:

$$x(t) = A\sin(2\pi f\,t + \varphi). \quad (11)$$

Velocity:

$$v(t) = 2\pi f A\cos(2\pi f\,t + \varphi). \quad (12)$$

Acceleration:

$$a(t) = -(2\pi f)^2 A\sin(2\pi f\,t + \varphi). \quad (13)$$

With:

- $A \geq 0$: amplitude of the sinusoid in the signal unit;
- φ : phase of the sinusoid in radian;
- f : the frequency of the sinusoid in Hertz.

In these equations *Eq. (12)* and *Eq. (13)*, the speed increases in proportion to the frequency *(f)*, while the acceleration increases with the square of the frequency, which apply only to sinusoidal vibrations.

In this work, we present the application of the spectral analysis method to the diagnosis of the screw compressor, based on the Fourier transform. We consider that the signal *x (t)* is absolutely integrable at *] -∞, + ∞ [*. The Fourier transform of this signal is equal to [27]:

$$x(w) = F[x(t)] = \int_{-\infty}^{+\infty} x(t)e^{-j\omega t}\,dt. \tag{14}$$

The search for damage, the right monitoring of their evolution and the diagnosis of the state of the compressor are possible. By all means, if one knows the vibratory symptoms associated with each defect likely to affect this compressor, that is to say, if we know the vibrational images induced by these defects. As a result, for the defect of the imbalance subject of this study it clearly manifests itself by [7]:

- An increase in the overall level chosen at low frequencies V_{eff} [10-1000 Hz].
- A clear increase in the amplitude of the fundamental frequency (f_0):

$$f_0 = \frac{N}{60}. \tag{15}$$

With:

- N: rotational speed en tr/min;
- f_0: frequency of rotation en tr/s [Hz].

As illustrated above, the vibrations are the result of forces, which can be of different origins, they are transmitted to the structure via the bearings and accordingly to the foundations via the fixings. The measurements were carried out on the bearings placed on the machine structure, using a quartz 'ICP' accelerometer type sensor with a sensitivity of 100-mV / g. From the sent signals from the sensor, the spectra are designed for the two series of experimental measurements on January 15 and April 20.

In this context, and to theoretically simulate the effect of the imbalance of the screw compressor we used the software (Matlab), the *Fig. 4 (a)* represent the response *x (t)* and the *Fig. 4 (b)* represent the response *v (t)* of the designed imbalance model. The Fourier transform of the speed signal is represented by the *Fig. 4 (c).*

The two time-based signals *x (t)* and *v (t)* obtained by (Matlab) have a frequency $f0 = 49$ *Hz* which corresponds to the speed of rotation of the shaft 2940 rpm acquired during the duration of one second

The Fourier transform allows the passage from the time domain to the frequency domain. The Fourier spectrum of *Fig. 4 (c)* of the vibratory time signal *x (t)* is broken down into frequency components. Thus, this Fourier transformation makes it possible to know the spectral energy or power content present in the localized signal. We can clearly see a high amplitude peak at the frequency 49 Hz which corresponds to the frequency of rotation of the screw compressor.

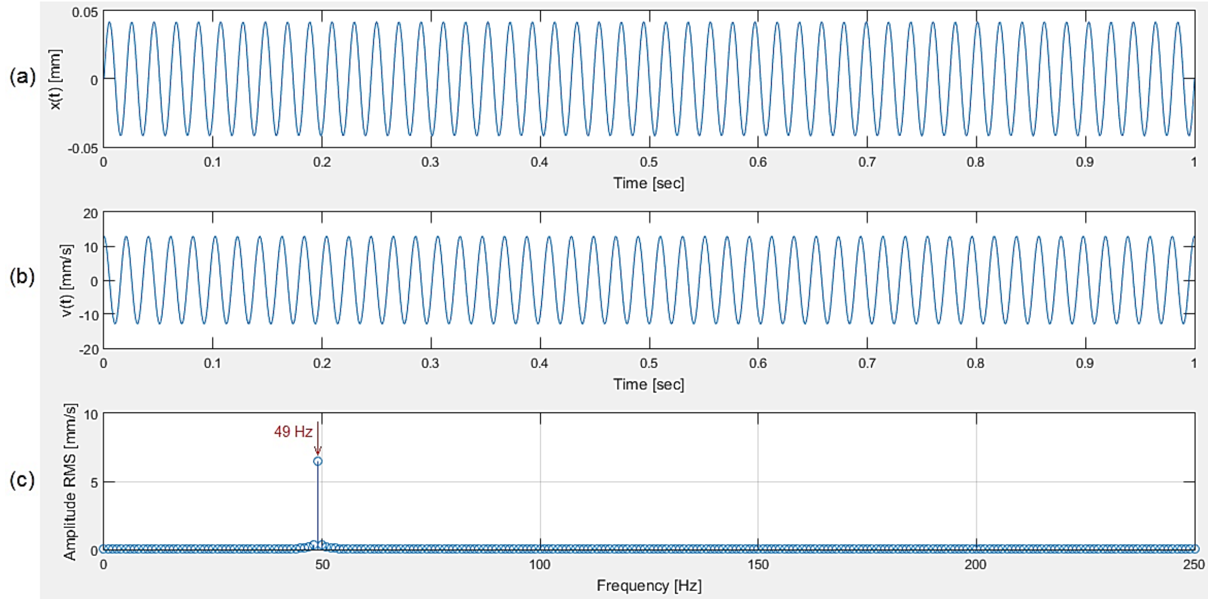

Fig. 4. *Theoretical simulation of the imbalance effect ((a) time curve of displacement, (b) time curve of velocity, and (c) theoretical spectrum of RMS velocity).*

4. Results and Discussion

From two series of experimental measurements of the spectra measured on January 15 and April 20, we can easily distinguish in *Fig. 5 (a)* high amplitude component peak at the frequency 49 Hz.

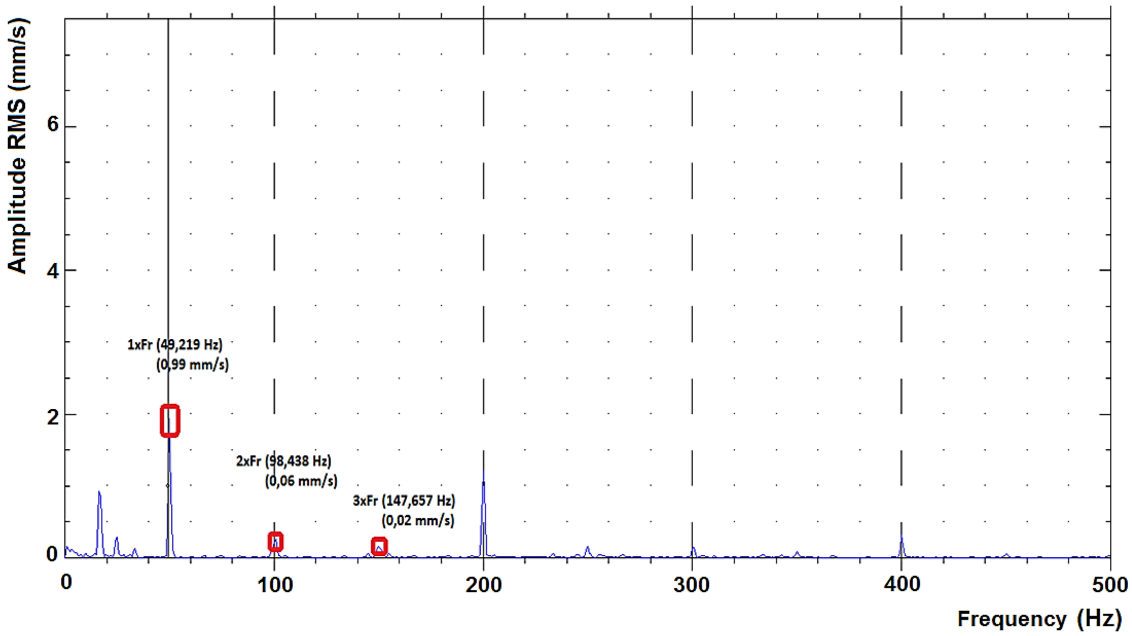

Fig. 5. *Pal 1 H spectrum recorded on April 20.*

As has been noted, the spectrum of *Fig. 5* shows a weak imbalance of 0.99 mm/s at the frequency of rotation $f_0 = 49\ Hz$ with an overall level of vibration 'NG' which is equal to 2.71 mm/s. The frequency signal of an imbalance fault looks like *Fig. 4 (c)* in the simulation.

The spectrum of *Fig. 6* of the date January 15 shows a marked increase at the amplitude of the fundamental frequency of 6.88 mm/s with an increase in the overall level 'NG' of vibration which is equal to 9.21 mm/s.

We compare in *Fig. 7* the evolution of the two frequency signals that of *Fig. 5* and that of *Fig. 6* measured at the point (Pal1H). We can see the presence of a major defect. The results obtained from the spectral analysis show that the amplitude at the frequency of rotation $f_0 = 49\ Hz$ of the reference signal in *Fig. 5* has increased by 7 times compared to that of *Fig. 6*, which implies an effect of imbalance. At the same time, when we are in a case of imbalance, there is always a large peak which corresponds to the frequency $(1 \times f_0)$ in the spectrum. In addition, we can see that only this peak is visible and relevant. An imbalance will therefore induce, in a radial plane, a vibration, the spectrum of which has an amplitude component predominant at the frequency of rotation of the rotor.

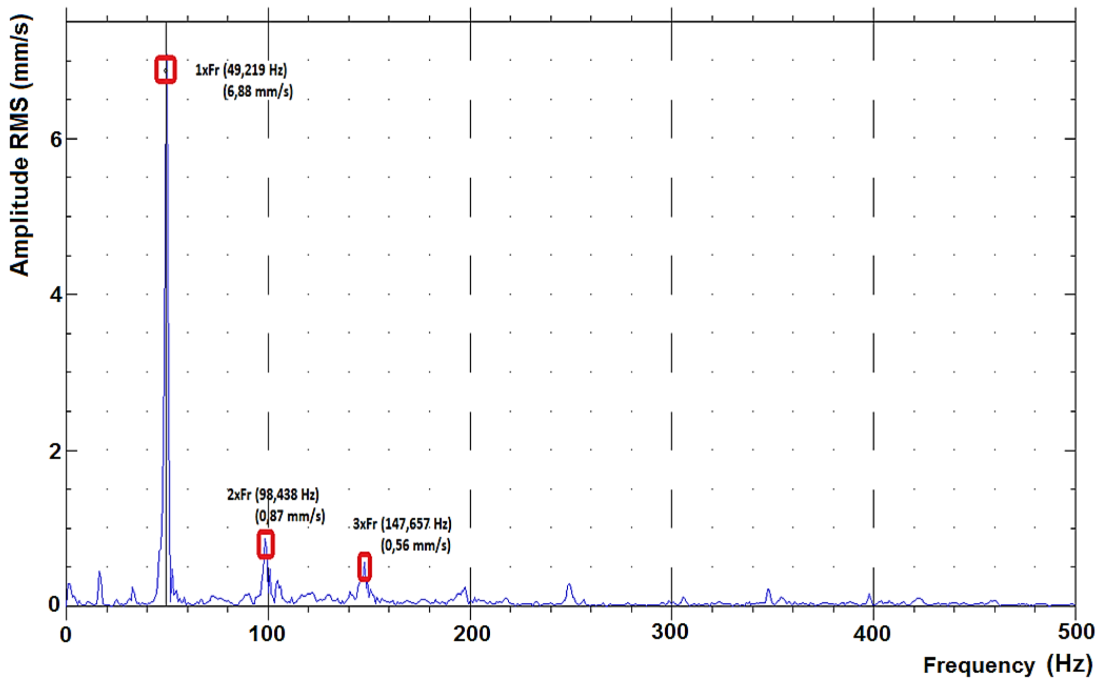

Fig. 6. *Pal 1 H spectrum recorded on January 15.*

All things considered, for a given equilibrium class, the overall level of induced vibration is compared with the thresholds set in standard ISO 10816 [7]. This makes it possible to judge the acceptability of the imbalance and the need or not to proceed with a balancing. In this case, the screw compressor motor rotates at 2940 rpm and has a power of 45 kW with a fixing made with flexible supports, the ISO 10816 standard imposes a danger threshold of 7.1 mm/s while the measurement carried out greatly exceeds this value 9.21 mm/s. This problem is corrected by an on-site balancing after dismantling and maintenance in the workshop.

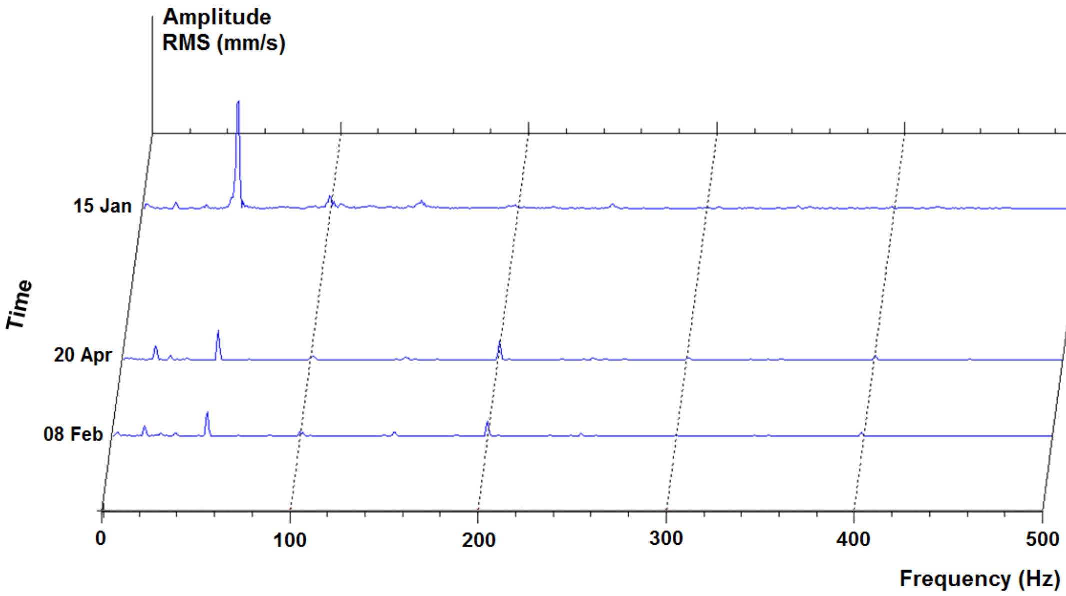

Fig. 7. *Comparison of the spectra of the Pal 1 H point of April 20 and January 15*.

As has been noted, that spectral analysis based on the Fourier transform present a technique that is simple to implement for industrial applications. On the other hand, if a precise description of the vibratory behavior is required, consequently theoretical analyzes of the dynamic behavior by additional analysis software are preferred. In reality, the choice of the characteristics and parameters of the system has a direct impact on the type of fault identification, since the estimation of the imbalance fault is not simple in the time domain. However, the identification of the frequency domain makes it possible to directly and effectively identify this defect.

5. Conclusion

In brief, monitoring the vibrations of this screw compressor plays a pivotal role in improving the reliability of this system. Nevertheless, mastering the vibration problems addressed in practice on this type of equipment increases their performance. To recap what has been analyzed so far in this article, the modeling of the vibrations of a screw compressor was examined, the detection and analysis of the vibrations were carried out using spectral analysis technique, to develop specific spectra using signal-processing techniques. This proposed approach was based on the modeling of the defects affecting the screw compressor, it make it possible to precisely identify the imbalance fault and to specify that it is very important in order to develop an accurate, correction of this problem.

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