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Fault Detection Method on a Compressor Rotor Using the Phase Variation of the Vibration Signal

G. Zaza*[a], A. D. Hammou[b], A. Benchatti[a], H. Saiah[c]

[a] Laboratoire de Mécanique, Université de Laghouat, Laghouat, Algérie
[b] Laboratoire de Génie des Procèdes, Université de Laghouat, Algérie
[c] Laboratoire de Impédance, Paris, France

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A B S T R A C T

The aim of this work is the application of the phase variation in vibration signal for fault detection on rotating machines in the non stationary cases. The vibration signal from the machine is modulated in amplitude and phase around a carrier frequency. The modulating signal in phase is determined after the Hilbert transform and is used, with the Fast Fourier Transform, to extract the harmonics spectrum in phase. This method is first validated on a simulator of vibration, then it is used for detecting potential faults on a rotor of a centrifugal compressor. In the case study of the centrifugal compressor, the nature of the defect was not revealed by the spectral analysis of amplitude of the non stationary vibration signal, unlike with the proposed method.

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1. INTRODUCTION

Vibration signals measured from a rotating machine are complex multi component signals generated by the different components of this machine. The analysis of shaft vibration signature is the prevailing method for the analysis of the machine dynamics and health monitoring [1]. Most early signal-processing techniques for machine monitoring have utilized the FFT to transform the time domain to the frequency domain, where the machine vibration signals could be analyzed [2]. However, in many practical cases, this tool showed a weakness in the defect detection. The FFT could not diagnose all the defects successfully because this method is unable to analyze nonlinear and non-stationary signals [3]. So in such cases, it is necessary to use another detection method. For example, the works in this domain established by Mobki et al. [4].

A fault in any rotating machinery is likely to introduce amplitude and phase modulations [2]. The phase angles of the vibration signals were generally ignored in fault detection; though the amplitude is widely used. Recently, research has been conducted to improve performance monitoring by exploiting the information contained in the phase angles of vibration signals. The magnitudes of the fluctuations in the phase angles of the rotational frequencies vary as the severity of the faults increases. The phases of the rotational signals, especially those from steady-state rotating machine components that are in good condition, have nearly constant phase rates or nearly zero angular acceleration [2].

This technique is mostly used in fault detection for the reciprocating internal combustion engine and gearbox. Binh and Tuma [5] have derived the phase modulation signal from the instantaneous crankshaft angular acceleration measured signal. They noted that many faults caused by faulty combustion and mechanics in multi cylinder engines can be detected through this method.

Tuma [6] determines the transmission error in a gear using the instantaneous angular variation of the rotational speed. The measurement method is based on the phase demodulation of the impulse signals using the theory of the analytical signals.

*Corresponding Author's Email: *gharbidml@gmail.com* (Z. Gharbi)

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In this study, this technique is derived from the acyclisms extraction method which is based on the angular variation of the rotation speed [7]. Generally used in reciprocating engines, the speed sensor, positioned at the flywheel, registers the rotation speed and the carrier frequency (flywheel teeth frequency). Then, the angular variation caused by the torsional vibration will produce a phase modulation, and create a periodic micro-variation of the rotation speed. Vyroubal [8] proposes a method for the estimation of the relationship between the rotor's radial vibration and the instantaneous angular variation of the rotational speed using optical sensor.

In our case, we use the acceleration sensor where the carrier frequency is defined from the impulsion signals caused by blade pass or mechanical impact. Then, the acceleration signal is filtered around this frequency.The signal obtained is a modulated signal in amplitude and phase. The phase modulation is extracted using the Hilbert transform.

2. PHASE DEMODULATION USING THE HILBERT TRANSFORM

The vibration signal from the machine is filtered around the carrier frequency f_0. We obtain a modulated signal in amplitude and phase $x(t)$ around this frequency:

$$x(t) = A(t)\cos\left(2\pi f_0 t + \varphi(t)\right) \tag{1}$$

where, $A(t)$ and $\varphi(t)$ are amplitude and phase modulations.

Periodic phase modulations can be written as:

$$\varphi(t) = \sum_{k=1}^{n} \beta_k . \sin(2\pi k f_m t + \phi_k) \tag{2}$$

f_m and ϕ_k are respectively frequency and initial phase of the phase modulation and the index k represents the order of the harmonic.

Applying the Hilbert transform:

$$y(t) = H[x(t)] = x(t) * \frac{1}{\pi t} \cong A(t). \sin\left(2\pi f_0 t + \varphi(t)\right) \tag{3}$$

where, $y(t)$ is the Hilbert transform.
The analytical signal is given by:

$$x_a(t) = x(t) + jy(t) \tag{4}$$

where, $x(t)$ and $jy(t)$ are respectively the real and the imaginary part of the analytic signal.

Which gives:

$$x_a(t) = A(t)\left[\cos\left(2\pi f_0 t + \varphi(t)\right) + j \sin\left(2\pi f_0 t + \varphi(t)\right)\right] \tag{5}$$

When we determine the real part and the imaginary part, we respectively define the amplitude and phase as follows:

$$\begin{cases} |A(t)| = |x_a(t)| \\ \varphi(t) + 2\pi f_0 t = \arg\left(x_a(t)\right) = \arctan\left(\frac{y(t)}{x(t)}\right) \end{cases} \tag{6}$$

The phase modulation signal $\varphi(t)$ is the fluctuation of the phase angle around the linear term $2\pi f_0 t$, which can be considered as a nominal angle of rotation while the argument of the cosine is an actual angle of rotation. The principl value of the argument belongs to either the $(-\pi, \pi\rangle$ or $\langle 0, 2\pi)$ interval. It is also called the wrapped phase, and therefore it is needed to unwrap into [9]:

$$\varphi(t) = \arg\left(x_a(t)\right) + 2\pi n(t) \tag{7}$$

where, $n(t)$ is a sequence of integer numbers.

The harmonics spectrum of $\varphi(t)$ is thus determined by the Fast Fourier Transform.

$$FFT(\varphi(t)) = FFT\left(\arg\left(x_a(t)\right) + 2\pi n(t)\right) \tag{8}$$

3. VALIDATION WITH VIBRATION SIMULATOR

The validation of the phase variation method in fault detection is made via a vibration simulator RS3M1 (Figure 1). It is a rotating machine with mechanical configurations to simulate the effects of defects such as: imbalance, misalignment and looseness.

Recording and acquisition of vibration signal are respectively produced using the piezoelectric accelerometers (Survitec brand sensitivity [100mV/g]) and National Instruments PCI card. The sensors are located at the bearing 4 along the vertical, horizontal, and axial directions. The processing of the recorded signal is taken via the STUDIOVIB360 software (Impedance).

A radial clearance defect is created at the bearing 4 (plain journal bearing). This creates an axial tilting of the cantilevered wheel when reaching the critical frequency of the shaft.

The carrier frequency is determined by an accelerometer placed on the bearing 4. This sensor detects the passing frequency of the four teeth of the wheel by a magnetic field created by a magnet (Figure 1).

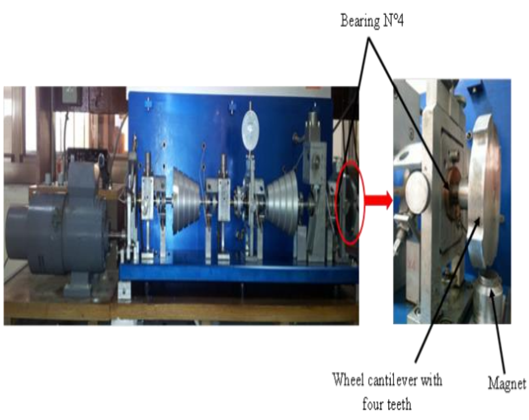

Figure 1. Vibration simulator RS3M1

The transient analysis (phase of start-up, stop and speed variation) is used to determine the critical frequencies of the simulator shaft and real excited modes. At these frequencies, there will be a large amplitude of vibration with a phase shift. The measured critical frequency is 75 Hz, since the amplitude increases rapidly to 9.4 mm/s (Figure 2a) with a 92.92° phase rotation (Figure 2b).

The increased amplitude at the speed 3000 RPM is related to the structure support with a high participation of the connecting assembly. It considered as a critical frequency of the support structure. The literature [10] discussed the influence of the critical frequency of the structure on mechanical deformation using modal analysis.

Figure 3 shows the spectrum [0-500 Hz] of the vibration signal during the passage through the critical speed, recorded by the accelerometer placed at the bearing 4 in the vertical direction (the number of spectral lines equals to 800, the frequency of sampling is 51200Hz). We note in this spectrum with defect (Figure 3) the preponderance of the rotation frequency amplitude (6.3 mm/s) with the harmonics and sub-harmonics. It is the spectral signature of looseness and friction caused by the tilt of the wheel. In the case without defect (Figure 3), the preponderance of the rotation frequency amplitude (1.3 mm/s) with the harmonics and sub-harmonics have very weak amplitudes.

The recorded signal is filtered by a band pass filter around the carrier frequency ($f_0 = 4 \times f_r$, with f_r the shaft rotation frequency). This filter permits only the components of the passage frequency of the magnetic field, while the other components are rejected.

We present the phase modulating signal on one complete revolution for the two cases (healthy-like case and with defect case) in Figure 4, using Equation (7).

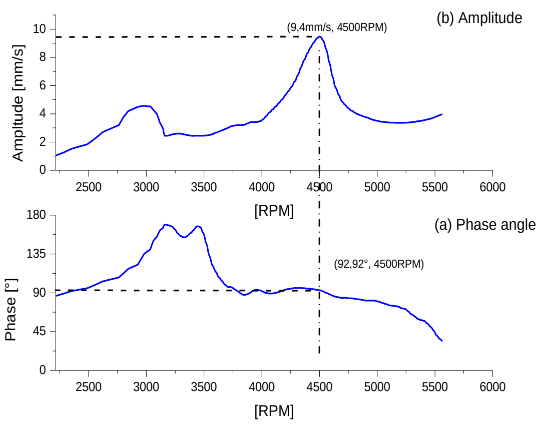

Figure 2. Transient analysis for shaft critical speed.

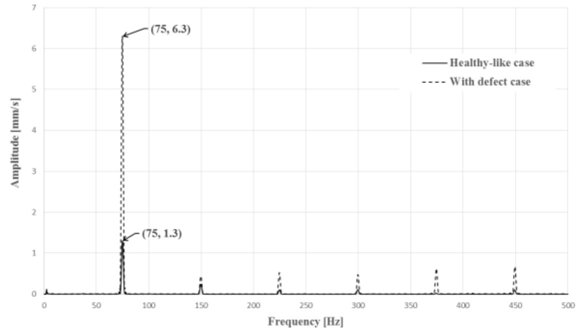

Figure 3. Frequency spectrum [0-500 Hz] of the vibration signal during passage through the critical speed

In Figure 5, analysis of the phase modulating signal spectrum (Equation (8)) shows the predominance of the amplitude of the second order of the rotation speed (H2=1.253°). This represents the spectral signature of the axial tilt of the wheel, as it provides a similar vibration to that of misalignment. The looseness is represented by the third, fourth, and fifth orders. This case study in simulator shows that the method of the phase variation of vibration signal allows the detection of defects related to angular variations of the shaft rotation.

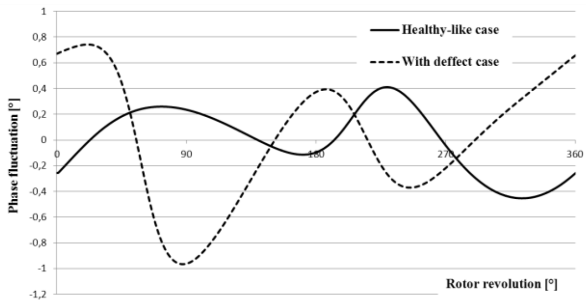

Figure 4. Phasemodulating signal on one complete revolution as a function of thenominal angle of rotation.

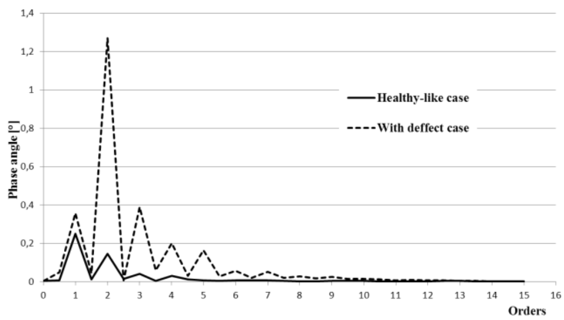

Figure 5. Phase modulating signal spectrum

4. A COMPRESSOR ROTOR FAULT DETECTION

The faults detection method using the phase variation of the vibration signal is applied to a centrifugal compressor rotor (7457 kW, 6500 RPM). This centrifugal compressor has a vibration instability when passing through a certain speed (near 68Hz) when engaged alone (case 1). This vibration instability causes tripping and stops the machine. However, when it is engaged in a series, with a similar compressor, the instability is smaller without stopping (case 2). When the second compressor is engaged alone, no vibration problem is recorded at the already mentioned speed (case 3). These three cases were considered in the analysis.

We present in Figure 6 the compressor rotor and the wheel (cantilevered) with 14 blades. The same instruments previously used for acquiring and processing in the simulator of vibration are used. For all three cases, the accelerometers were placed at the bearing near the compressor wheel. In the analysis, we consider only the corresponding speed instability and just before the time of tripping.

Spectral analysis of the vibration signals of the three cases reveals a strong gas turbulence between 250 Hz and 450 Hz (Figures 7-9). This does not mean an anomaly in this type of compressor. However, in the first case this phenomenon is accentuated by a default. This default is undefined in this case by spectral analysis.

Figure 6. Centrifugal compressor rotor.

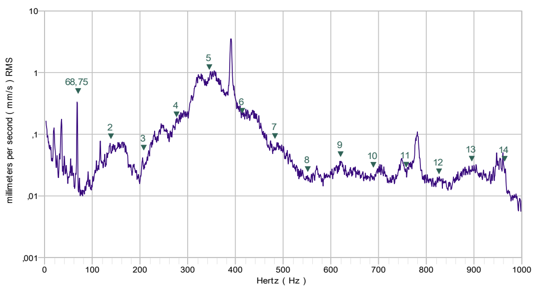

Figure 7. Frequency spectrum [0-1000 Hz] for the first case compressor at speed near 68 Hz

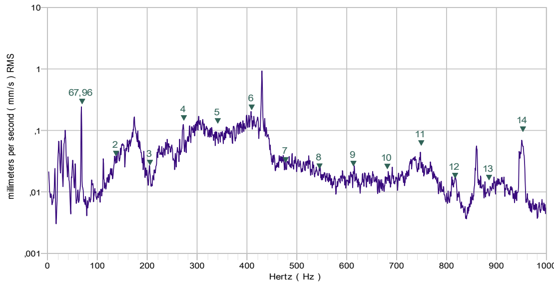

Figure 8. Frequency spectrum [0-1000 Hz] for the second case compressor at speed near 68Hz

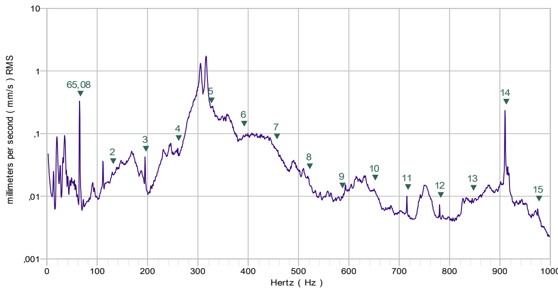

Figure 9. Frequency spectrum [0-1000 Hz] for the third case compressor at speed near 68 Hz

An order analysis was performed using the ORDER STUDIOVIB module. This analysis provides the magnitude and its harmonic decomposition. It is used from start to trigger of the compressor (case 1), so that to keep the tripping speed as long as possible before stopping (~ 300 s).

Figures10-12, show respectively the contribution of harmonics during the analysis period according to the vertical, horizontal, and axial direction (case 1). In the three directions, we notice the dominance of the 5th and 6th orders, corresponding respectively to the journal bearing (5 pads) and the thrust bearing (6 pads). There is the dominance of the 1st order in the horizontal direction. It is also interesting to note that, in the axial direction, the dominance of the 14th order corresponding to the passage of the blades of the wheel.

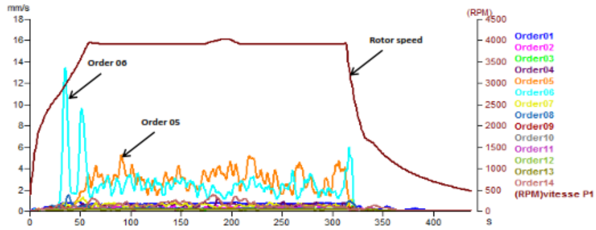

Figure 10. Distribution of orders vs time in the vertical direction (case 1)

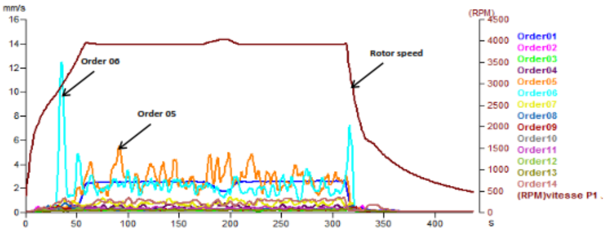

Figure 11. Distribution of orders vs time in the horizontal direction (case 1)

Figure 12. Distribution of orders vs time in the axial direction (case 1)

The latter reveals the existence of a defect in the wheel in the axial direction. Effectively, analysis of order up to the speed of instability in the axial direction of the two other cases reveals elevation of the 14th order in the second case (compressor with instability), while this order is much lower for the third case (safe compressor) (Figures 13 and 14).

The filtering by the band pass filter in this case is around the wheel blades passage frequency ($f_0 = 14 \times f_r$, with f_r the rotor rotation frequency).

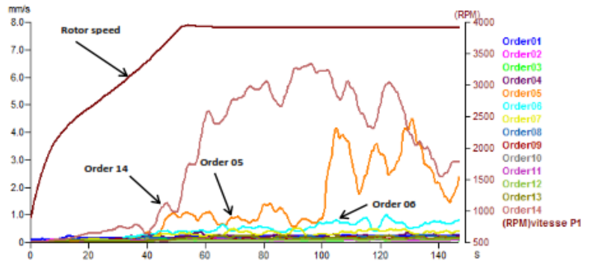

Figure 13. Distribution of orders vs time in the axial direction (case 2)

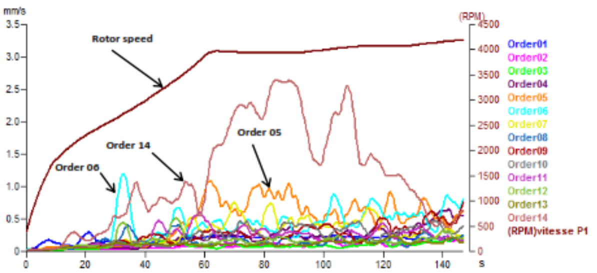

Figure 14. Distribution of orders vs time in the axial direction (case 3)

The filtered signal of the first case (case 1) gives a modulation in amplitude and phase. In Figure 15, the phase modulating signal on one revolution (three cases) using Equation (7) is given. The harmonics spectrums of the phase modulating signals of the three cases respectively are represented in Figure 16 (Equation (8)).

In all three cases, close to the critical speed, we notice the presence of the fifth harmonic corresponding to the frequency of the journal bearing (5 pads). The presence of this frequency reflects a rotor instability caused by insufficient static forces exerted by the pads on the rotor. The dynamic forces, which are the most important than the static forces, create an instability in the bearing, and this is manifested by the presence of sub- harmonics and the fifth harmonic of the rotation frequency. With a signature of misalignment in both cases 1 and 2, this phenomenon is more prominent in the first, when engaged alone, with a higher flow, producing tilting of the wheel (cantilevered). The amplitude of tilting decreases in case 2, compared to case 1, with the presence of harmonics (6, 7, 8, 9, 10, 11, 12 and 14) that reveal a signature of radial clearance with a friction defect at the concerned compressor. This signature characterizes the axial tilting and the radial clearance. This defect does not exist in the third case.

In conclusion, we can understand that near the critical speed, compressor bearing (near the wheel) undergoes very important dynamic forces amplified by the cantilevered mounting of the compressor wheel.

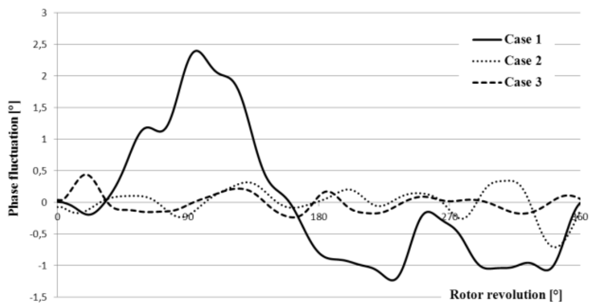

Figure 15. Phase modulating signal on one complete revolution as a function of the nominal angle of rotation (three cases)

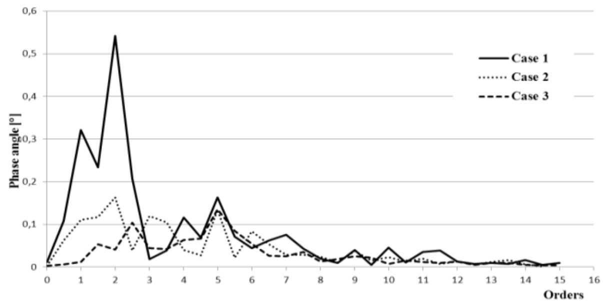

Figure 16. Phase modulating signal spectrum (three cases)

This exceptional stress causes premature wear of the bearing pads and creates the radial clearance problem causing the axial tilting.

5. CONCLUSION

In the present work, we used two vibration analysis methods on a compressor having a defect causing instability at a certain speed; spectral analysis method and the phase variation of the modulated signal method. The first method could not reveal the nature of the defect, unlike the second method. It was verified with a vibration simulator where we generated a similar defect to that of the compressor. This simple method is based on the phase demodulation using the Hilbert transform of the vibration signal. In fact, failure on a rotating machine generates a phase fluctuation around the carrier frequency phase, then many operational faults of rotary machines originate or manifest in non-uniformity of rotational speed. As found in the literature, the spectral analysis of the amplitude of vibration cannot be applied in all cases (especially for non-linear and non-stationary cases). In such cases for analysis, the Hilbert transform and the phase variation of the vibration signal is a powerful method for non-linear and non-stationary vibrations analysis.

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G. Zaza[a], A. D. Hammou[b], A. Benchatti[a], H. Saiah[c]

[a] Laboratoire de Mécanique, Université de Laghouat, Laghouat, Algérie
[b] Laboratoire de Génie des Procèdes, Université de Laghouat, Algérie
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چکیده

هدف از این کار استفاده از تغییر فاز در سیگنال ارتعاش برای تشخیص خطا در ماشین های دوار در موارد غیر ثابت است. سیگنال ارتعاش از دستگاه در دامنه و فاز در اطراف یک فرکانس حامل مدوله می شود. سیگنال مدولاسیون در فاز بعد از تبدیل هیلبرت تعیین می شود و با تبدیل سریع فوریه برای استخراج طیف هارمونیک در فاز استفاده می شود. این روش ابتدا بر روی یک شبیه ساز ارتعاش تأیید شده است و سپس برای تشخیص گسل های بالقوه روی روتور یک کمپرسور سانتریفوژ استفاده می شود. در مطالعه موردی کمپرسور سانتریفوژ، ماهیت نقص با تجزیه طیفی دامنه سیگنال ارتعاش غیر ثابت، بر خلاف روش پیشنهادی مشخص نشد.

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