SIGNATURE OF OIL STARVATION FAULTS IN ROTORS SUPPORTED BY HYDRODYNAMIC BEARINGS

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Abstract. *Oil starvation is a critical fault often observed in hydrodynamic bearings caused by insufficient oil supply. The early detection of starvation is important to assure good performance in rotating machines, avoiding sudden shutdowns and reducing maintenance costs. This paper aims to understand the signature of oil starvation faults in the vibrational response of rotors supported by hydrodynamic bearings. A mass-conserving bearing model is used to simulate healthy and starved lubrication conditions. The rotor is modeled by finite element method and its vibrational response is obtained by numerical integration at constant rotational speed, then decomposed in directional components using the Discrete Fourier Transform. Results shown that the oil starvation reduces the forward component while increases the backward component, increasing the anisotropy of the system. The ratio between forward and backward component presented a variation of -49% from a healthy bearing to a severely starved bearing. Furthermore, this variation achieves -65% when the vibration is observed at the outside bearing edge. Even with a noisy vibration signal of 10 dB of signal to noise ratio, the signature of oil starvation is clearly distinguishable in all the lubrication conditions evaluated. Thus, observing the directional response of a rotor is a promising strategy to detect faults of oil starvation in hydrodynamic bearings.*

Keywords: *hydrodynamic bearing, oil starvation, rotordynamic.*

1. INTRODUCTION

Rotors are composed of several components coupled to a rotating shaft, being an essential part of rotating machines such as motors, pumps, turbines, compressors, gearboxes, among others. This type of equipment is found in many industry sectors and has diversified functions such as power transmission for mechanical systems, fluid transport and power generation. In rotating machines under high loads and speeds, hydrodynamic bearings are applied to support the rotors and reduces its rotational friction. This type of bearing has low friction coefficient because the moving shaft surface is completely separated from the bearing surface by a thin oil film. Furthermore, due to the shaft rotation and the squeezing of this oil film, a high pressure field is developed in the bearing, resulting in high load capacity. Therefore, changing bearing conditions can reduce the performance of the machine, observed, for example, by power loss and thermal growth (Vijayaraghavan et al. 1996), excessive vibration (Zhang 2013) and shaft rubbing (Akagaki and Kato, 1992). In critical scenarios, even catastrophic faults of the rotor can occur due to malfunction of the bearings. Thus, it is important to identify bearing faults in order to assure safety and reliability of rotating machines.

There are different techniques to evaluate the bearing conditions such as vibration, oil, ultrasonic, infrared and acoustic analysis. Among these techniques the vibration analysis is widely used in rotating machinery maintenance due to its reliability and accuracy (Jardine et al., 2006; Adams, 2009). From the spectrum of the vibration signal it is possible to observe if the machine is operating in nominal conditions. If there is a fault, for example, in the bearings, an abnormal pattern may be observed in the vibration spectrum. Thus, the knowledge of the faults signatures in the vibration spectrum can be used by a trained operator to detect faults during maintenance routines. Furthermore, these variations in the vibration pattern can be detected by an algorithm to generate automatic diagnosis.

In this context, several studies have been done to understand the influence of bearing faults on the rotor behavior. The most investigated faults in hydrodynamic bearings are the bearing wear, bearing ovalization and the oil film cavitation. Chasalevris et al. (2013) applied continuous wavelet transform (CWT) to a rotor-bearing system response for worn bearings and observed that the wear affects the dynamic of the shaft, introducing sub and superharmonics, especially during resonance, being the 1/2X the more sensitive component. Machado and Cavalca (2015) analyzed the influence of bearing wear on the rotor-bearing behavior from the directional unbalance response. The directional coordinates were used due to its sensitive to anisotropy introduced by the fault. The authors concluded that the backward component increases at the presence of wear in the hydrodynamic bearing. Later, Machado et al. (2019) analyzed the influence of bearing wear on the shaft's orbits, decomposing the displacements into directional harmonics. The authors concluded that the wear has a well-defined signature in the directional spectrum, namely, an increase of the backward -1X component, which becomes more evident for higher wear depth.

Regarding the bearing ovalization fault, Bachschmid et al. (2000) stated that bearing ovalization affects the second harmonic vibration component of rotating machines. Wang et al. (2004a; 2004b) studied the impact of a deformed rod bearing on the orbit of a crankshaft journal. The authors concluded the bearing deformation is maleficent to its dynamic, but few information is presented about the ovalization signature on the journal vibration, expect for a higher horizontal displacement. Later, Silveira and Daniel (2019) studied the influence of ovalized rod bearing on the journal orbit of a slider-crank mechanism. The results showed that the orbit tends to follow the bearing wall shape, i.e. the orbit amplitude decreases in the same direction that the bearing clearance decreases.

The oil film cavitation is another phenomenon that has been widely studied due to its impact on bearing dynamic (Vijayaraghavan and Brewe, 1992; Sawicki and Rao, 2004; Ramos and Daniel, 2020). From the first studies on the cavitation mechanism (Jakobsson and Floberg, 1957; Etsion and Pinkus, 1975), researchers investigated the lubrication condition under insufficient oil supply, a fault known as oil starvation. This fault occurs due to problems in the oil pump and/or in the measurement sensors (flow and/or pressure) and also due to leakage or clogging in the oil distribution lines. This fault is critical for hydrodynamic bearings because the pressure distribution in the oil film is altered and, consequently, the bearing's behavior is modified, impacting the entire rotating system (Bonneau and Frenne, 1983).

Artiles and Heshmat (1985) were the first to study the effect of starvation fault on the dynamic performance of hydrodynamic bearings. From computational simulations of the pressure field in bearings for different starvation conditions, the authors showed that low flowrates reduced the extension of the hydrodynamic pressure region, affecting the stiffness and damping of the oil film. Vincent et al. (1995) showed a drastic reduction in the bearing load capacity under severe starvation conditions. Likewise, Tanaka (2000) showed that the shaft eccentricity can be severely increased under starvation conditions, which increases the risk of rubbing.

Despite the important observations on these works, they do not characterize the vibration inside starved bearings. In fact, the literature about the oil starvation signature on rotor vibration is very sparse. Poddar and Tandon (2020) observed the vibration spectrum of a journal bearing under severe starvation and verified a predominant 1X harmonic. However, this single harmonic does not provide enough information to identify a clear vibration pattern under early starvation stages. Thus, it can be verified that there is still lack of knowledge about the influence of oil starvation on rotors vibration. This paper aims to contribute to the understanding of the oil starvation signature in the vibrational response of rotors supported by hydrodynamic bearings. For this, a rotor-bearing system is modeled using the finite element method, in which the hydrodynamic forces are obtained with a mass-conserving model for different oil flowrates. Then, the simulated displacements inside the bearings are evaluated from the directional spectrum since it provides more information about the journal orbits. The sensibility of directional components to the measurement position and noise levels are also evaluated.

2. METHODS

In this section, the model of the rotating system is presented as well as the technique to analyzed the starvation signature in the rotor vibration. Firstly, the sub-section 2.1 presents an approach used to model the bearings considering the oil flowrate as an input parameter. Then, the finite element model of the rotor-bearing system is presented in section 2.2. Finally, the sub-section 2.3 presents the method of vibration analysis in the frequency domain.

2.1 Bearing model

The hydrodynamic bearing presents a thin oil film that separates the shaft from the bearing wall. Fig. 1 shows the representation of the cylindrical bearing model used in this work, where R is the shaft radius and Ω is the shaft rotational speed. The pressure inside the oil film is obtained as a function of the oil thickness h, described by:

$$h(\alpha) = c_r + e_X \sin\alpha - e_Y \cos\alpha \tag{1}$$

where c_r is the bearing radial clearance, α is the angular position around the shaft, e_X and e_Y are the shaft eccentricity in the horizontal and vertical directions, X and Y, respectively.
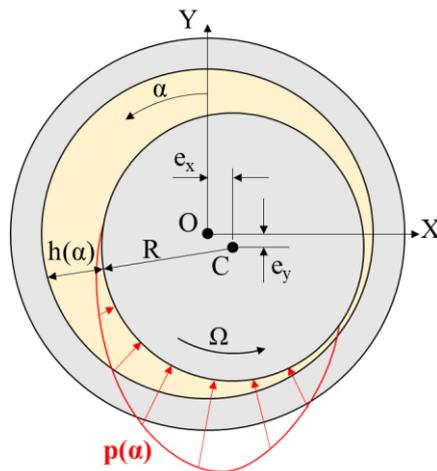
Figure 1 – Cylindrical bearing model.

In order to simulate the oil flowrate influence on the bearing behavior, a mass-conserving oil film model should be considered in this analysis. Thus, the classical formulation of the Reynolds equation with the Swift-Stieber conditions are not suitable for this work's purpose, since it does not ensure mass conservation on the cavitation region on the bearing. Hence, the $p - \theta$ formulation (Ausas et al., 2009) is used instead:

$$\frac{\partial}{\partial x}\left(\frac{h^3}{12\mu}\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial z}\left(\frac{h^3}{12\mu}\frac{\partial p}{\partial z}\right) = \frac{\partial}{\partial x}\left(\frac{\Omega R}{2}h\theta\right) + \frac{\partial(h\theta)}{\partial t} \tag{2}$$

where p is the pressure distribution, μ is the oil dynamic viscosity, x is the circumferential direction, z is the axial direction and θ is the oil fraction distribution.

Solving the Eq. 1 with the finite volume method (FVM), the oil supply flowrate Q_s is included in the flow balance for the volumes at the oil inlet region, as proposed by Oliveira et al. (2021). Then, the pressure and oil fraction distributions are obtained simultaneously using the Gauss-Seidel procedure, in which corrections are done to p and θ in order to enforce the cavitation conditions:

$$\begin{cases} p_i > 0 \rightarrow \theta_i = 1 \\ \theta_i < 1 \rightarrow p_i = 0 \end{cases} \tag{3}$$

where the subscript i refers to the i-th finite volume.

In a static case, for a given rotational speed and oil supply flowrate, the resultant hydrodynamic forces acting on the shaft are obtained by integrating the pressure distribution on the bearing domain, in order to calculated the shaft equilibrium position inside the bearing. However, in a dynamic case, the pressure distribution should be recomputed for each shaft`s position and velocity, in order to obtain the hydrodynamic forces acting in this dynamic condition. Aiming to reduce computational cost, the hydrodynamic forces are linearized by Taylor expansion around the equilibrium position, as proposed by Lund (1987). Thus, the oil film is modeled as equivalent coefficients of stiffness and damping that are the first derivatives of the forces with respect to shaft's position and velocity, respectively.

2.2 Rotating system model

The rotating system is modeled by finite element method (FEM), in which the shaft is discretized into cylindrical Timoshenko beam elements, each one having two nodes, and rigid disk element, each one having a single node. Each node has four degrees of freedom (DOF), being two translational displacements and two angular displacements (lateral vibration). The equation of motion of each element is obtained by the Lagrange method, considering a constant rotational speed. Then, the equations of all disk and beam elements are assembled to yield the global equation of motion of the rotor:

$$M\ddot{q}(t) + (C + \Omega G)\dot{q}(t) + Kq(t) = f(t) \tag{4}$$

where q is the vector of the rotor's DOF, M is the mass matrix, K is the stiffness matrix, G is the gyroscopic matrix and C is the damping matrix, considered as proportional to K in this paper (Machado and Cavalca, 2015). Finally, f is the vector of forces acting on the rotor, including the elements weight, the linearized hydrodynamic forces and the unbalance force, that it is described as presented in Eq. 5:

$$f_{unb}(t) = \begin{Bmatrix} m_\xi \Omega^2 cos(\Omega t) \\ m_\xi \Omega^2 sin(\Omega t) \\ 0 \\ 0 \end{Bmatrix} \tag{5}$$

where m_ξ is the unbalance moment.

2.3 Vibration analysis method

The vibration of every node of the rotor can be simulated by solving the Eq. (3) using a numerical integrator. Then, the translational displacements can be represented in the complex plane as a resultant of contra-rotating vectors called forward and backward directional components. These components can be separated in the frequency domain by applying the Fourier Transform to the complex signal. This enables an analysis of the degree of anisotropy of the system from the forward (positive) and backward (negative) spectrum, also called fullspectrum or directional spectrum (Alves et.al, 2021). The spectrum analysis is useful in identifying rotor faults, because these faults can change the vibration by introducing new frequencies or changing the ratio of the amplitudes of the backward and forward modes, which is easily observed in the peaks of the spectrum. In this paper only the rotational frequency (1X) is evaluated, since the rotating system model is linear and reproduces only synchronous vibration due to unbalance force. Thus, the signature of oil starvation fault is evaluated in terms of 1X (forward) and -1X (backward) amplitudes in the directional spectrum. In order to enhance the response sensibility to bearing lubrication conditions, the directional components are extracted from vibrational responses observed near the bearings.

3. NUMERICAL RESULTS

This section presents the analysis of the signature of bearing oil starvation on the directional response. The rotor model used in this analysis is composed of 10 beam elements, 2 identical bearings (same oil supply flowrate Q_s) and a centered disc (Fig.2). Two different geometries of rotors were analyzed, being a flexible rotor and a stiffer rotor, whose dimensions are described in Table 1. Both rotors have the same disk, being the outer diameter of 94.7 mm, an inner diameter of 23.5 mm and axial length of 47 mm assembled at node 6. The material applied to the shaft and the disc is steel with modulus of elasticity (Young's modulus) E = 210 GPa and density ρ = 7800 kg/m³. The simulations were performed for a rotation speed of 30 Hz.
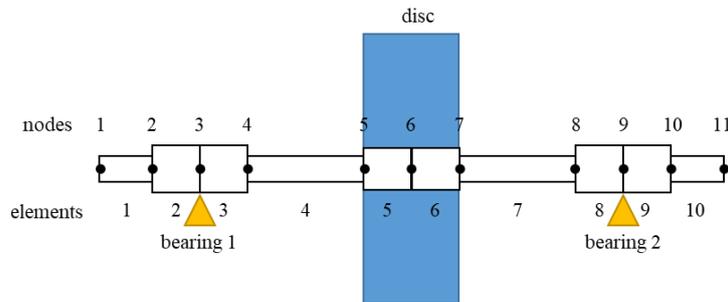

Figure 2 – Finite elements model of the rotating system.

As the rotors are completely symmetric, only the directional response in the bearing 1 is evaluated. Three lubrication conditions were simulated, a healthy condition with Q_s = 24.5 ml/min (equivalent to 0 Pa at the bearing groove), a starved condition with Q_s = 20ml/min and a severe starvation with Q_s = 15 ml/min. The bearing properties are shown in Table 2.

Table 1 – Rotors dimensions.

Elements	Flexible rotor		Stiffer rotor	
	Length (mm)	Diameter (mm)	Length (mm)	Diameter (mm)
1, 10	40.0	12.0	40.0	17.7
2, 3, 8, 9	10.0	31.0	10.0	31.0
4, 7	266.5	12.0	100.5	17.7
5, 6	23.5	23.5	23.5	23.5

Table 2 – Bearing properties.

Property	Value
Diameter	31.0 mm
Width	20.0 mm
Radial clearance	90.0 μm
Groove width	15.5 mm
Groove circumferential length	3.6 mm
Groove position	0°
Lubricant dynamic viscosity	0.05 Pa.s

3.1 Starvation signature on the directional response of the rotor

Hydrodynamic bearings are naturally anisotropic, which justifies the presence of both backward and forward components in the directional spectrum, as presented in Fig. 3a. As a typical spectrum of elliptical orbits, the forward component is generally predominant, indicating a precession in the same sense of the shaft's rotation. However, it is possible to note that the starvation condition increases the backward component while decreases the forward component, tending to equalize both components. This increases the bearing anisotropy and, consequently, the orbit ellipticity as observed in the temporal response shown in Fig. 3b.
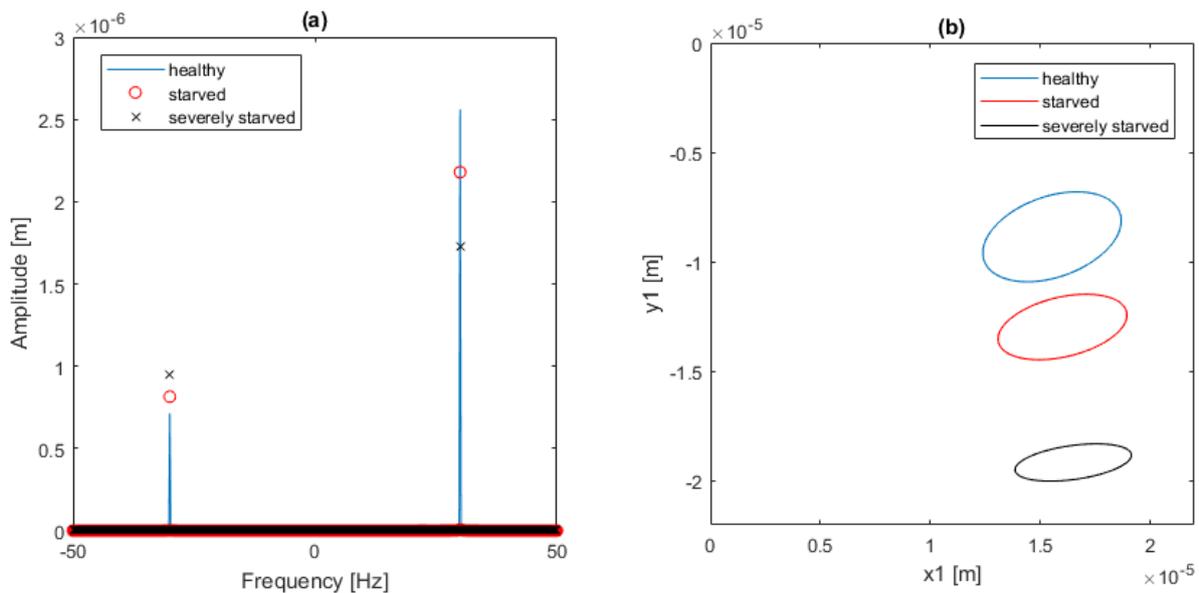

Figure 3 – Signatures of the oil starvation fault in the flexible rotor: a) Directional spectrum; b) Orbits inside the bearing 1.

Comparing different levels of oil starvation, the higher the starvation severity, the higher the reduction of the forward component as well as the increasing of the backward component. Although the forward component seems to be more sensitive than the backward component in terms of absolute amplitude variation, both components presented a relative variation of 32% from the healthy condition to the severe condition, which is clearly significant. Furthermore, the oil starvation signature can be even more emphasized by observing the ratio between forward and backward amplitudes, denoted by fb ratio, instead of evaluating the components separately. The fb ratio is reduced by 49% from the healthy condition to the severe starved condition. Thus, monitoring the fb ratio of the directional response may be a suitable alternative to identify degradation of the oil supply to the bearings.

The same analysis was performed in a rotor with higher stiffness. In this rotor, the forward component is reduced in all lubrication conditions, but the signature of lubrication starvation remains the same observed in the previous rotor, i.e., a reduction of forward amplitude and an increasing of backward amplitude, as presented in Fig 4. However, in the stiffer rotor, the variation of the directional components is not equal anymore. From the healthy condition to the severe starvation, the forward amplitude decreases of 30% while an increasing of 37% is observed in the backward component. It is important to verify that these variations in the forward and backward components due to oil starvation fault results

the same variation of fb ratio observed in the flexible rotor, i.e. a reduction of 49% from the healthy condition to the severe starved condition.
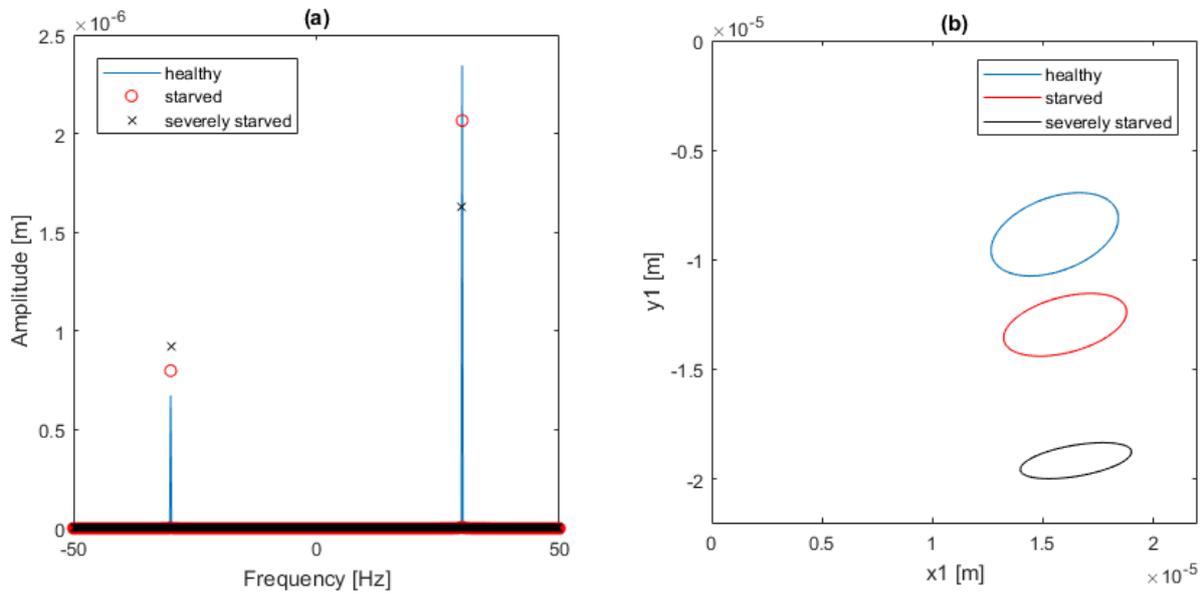

Figure 4 – Signatures of the oil starvation fault in the stiffer rotor: a) Directional spectrum; b) Orbits inside the bearing 1.
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3.2 Influence of sensor position

In many practical situations, vibration signals are measured on the side of the hydrodynamic bearing housing, either due to spatial restrictions inside the housing or the ease of installing the sensors externally to the bearing. For this reason, the influence of the sensor position on oil starvation signature was evaluated. This analysis was performed only for the case with flexible rotor, since there was no significant change in the sensitive of directional response when compared with the stiffer rotor. The results show that, measuring the vibration at the outside edge of the bearing 1 (node 2), the severe starvation yields a pronounced variation in the forward component and the fb ratio, -54% and -65% respectively, while the backward component variation remains around +30%, as seen in Fig.5. Again, the fb ratio shows to be the most sensitivity parameter for detection of oil starvation fault. This result suggests that the directional response is suitable for practical applications where it is not possible to measure the vibrational response directly at the center of the bearings.
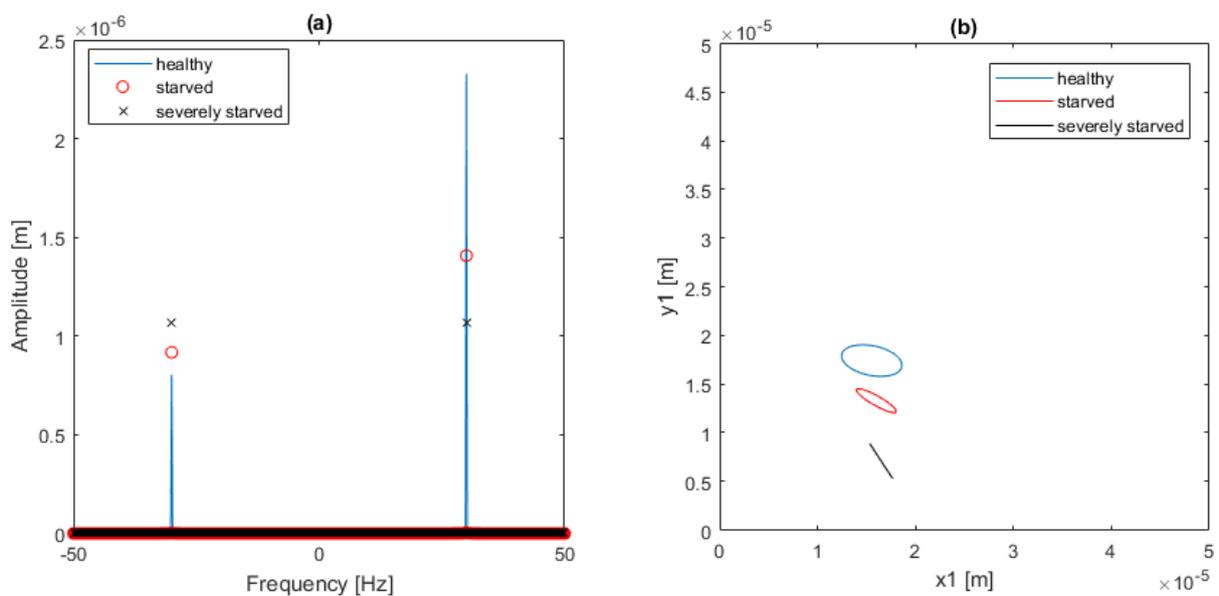

Figure 5 – Signatures of the oil starvation fault in the flexible rotor: a) Directional spectrum; b) Orbits at outer endface of the bearing 1.

According to Fig. 5, the forward and backward components of the vibrational response on the outside edge of the bearing (node 2) become practically equals (fb ratio ≈ 1), indicating an imminent inversion of the sense of rotor precession and resulting in the straight-line motion observed in Fig.5b. However, no node of the rotor was observed with backward precession (fb ratio < 1), which could characterize a dangerous condition of operation due to shaft torsion.

On the other hand, measuring the vibration at the inside edge of the bearing (node 4), the sensibility of the directional response to the oil starvation fault decreases. According to Fig. 6, the forward component presents a variation of only - 7% and the backward component presents a variation of +24%, resulting a variation of -7% in the fb ratio.
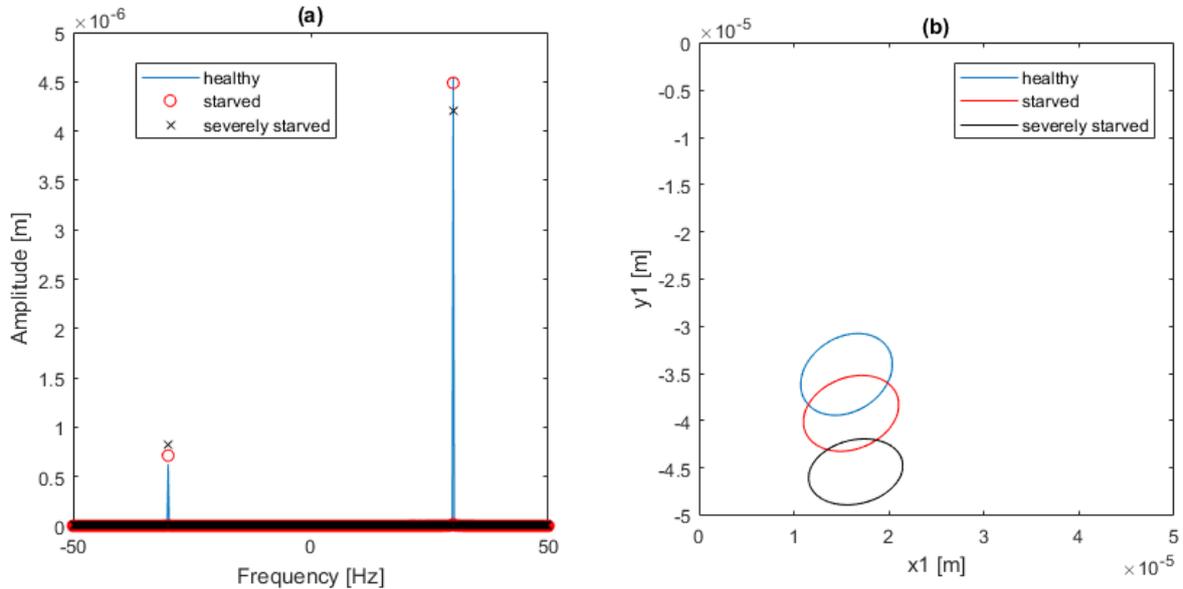
Figure 6 – a) Directional spectrum and b) orbits at bearing inner endface.

3.3 Influence of sensor noise

In this section, the starvation signature in the fb ratio was evaluated when the vibration data are contaminated with white noise with 10 and 30 dB of Signal-to Noise-Ratio (SNR). As considered in the previous section, this evaluation was also performed only for the case with the flexible rotor. Moreover, the vibration was evaluated at the outside edge of the bearing (node 2). The white noise was added to the vibration signal with the *awgn* function from the Communication Toolbox of MATLAB. For each lubrication condition (heathy, starved and severely starved) and SNR value (10 and 30 db), 1000 samples were generated and the variation in the fb ratio was computed with respect to fb ratio value at the healthy condition without noise. As can be observed in Fig. 7, the variation in the directional response caused by the noise is insignificant compared to the variation due to a deterioration in the lubrication condition of the bearings. Thus, even with high level of noise, it is possible to clearly distinguish an oil starvation fault on bearing from a healthy condition using the directional response of the rotor.
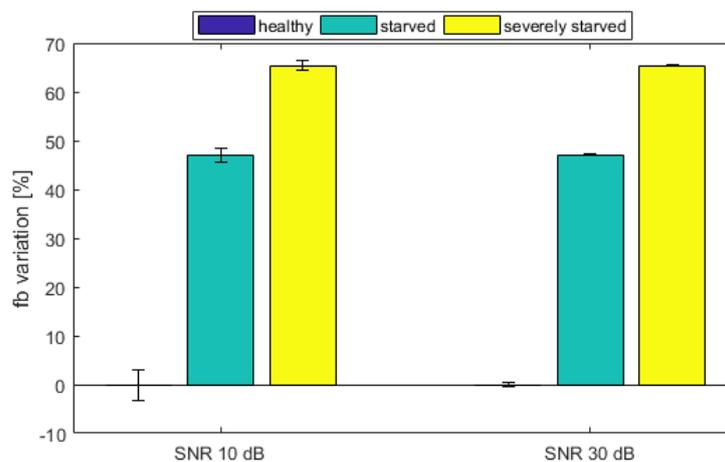
Figure 7 – Influence of noise on the fb variation.

4. CONCLUSIONS

The oil starvation condition was simulated in hydrodynamic bearings supporting a symmetric rotor. The signature of starved and severely starved lubrication was extracted from the rotor vibrational response in comparison with that in healthy lubrication. For this, the rotor response was analyzed in the frequency domain in terms of forward and backward synchronous amplitudes. The results showed that both amplitudes are sensitive to bearing oil supply flowrate variation. As the oil supply flowrate decreases, the forward amplitude increases and the backward decreases, augmenting the orbit ellipticity. The higher the reduction of flowrate, the higher is the variation of amplitudes. This signature is clearly highlighted in the ratio between the amplitudes of forward and backward components (namely fb ratio), being it a promising reference parameter to monitor oil starvation faults. The signature is even more expressive when measuring the rotor vibration at the bearing outer endface, favoring the oil starvation detection in practical applications where is not possible to measure the vibration at the bearing center. In addition, noisy measurements did not significantly interfere the oil starvation signature, allowing clear distinction of healthy and starved bearing conditions.

Finally, it is important to highlight that the results obtained in this paper show how the oil starvation fault in hydrodynamic bearings affects the dynamic response of the rotor, which directly contributes to the development of monitoring and identification techniques for this type of failure in rotating machines.

5. ACKNOWLEDGEMENTS

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