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Application of active hydrodynamic bearings to reduce whirl effect on rotor

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Abstract: Conventional hydrodynamic journal bearings applied to rotating machines are susceptible to some undesired conditions, such as oil whirl effects. To avoid this condition, it is proposed to use the new concept of hydrodynamic bearing, proposed by Ramos and Daniel (2021). This new concept considers the own bearing motion, thus making possible to change the average velocity of the lubricating oil film inside the bearing. The rotor dynamic behavior is initially evaluated for the conventional bearing (without bearing motion). In following, the new bearing is simulated to evaluate the differences in the oil whirl effects. The numerical results show that this new bearing can reduce the oil whirl effects, decreasing the 0.5x component of the FFT.

Keywords: Active bearing, Hydrodynamic Bearing, Oil Whirl

INTRODUCTION

The pressure field developed in hydrodynamic bearings allows the support of high loads at high rotational speeds without metal-to-metal contact what reduces the friction on the rotor-bearing system. These advantages presented for the use of hydrodynamic bearings become its applicability very high. However, problems such as overheating of the lubricant film (Linjamaa et al. (2018), Li et al. (2017)), difficulty in adjusting the load capacity without changing the rotor operational conditions (Frene (1997), Urreta et al. (2010)), oil whirl effects (Castro et al. (2008), El-Shafei et al. (2007)) and fluid-induced instability (Pesch and Sawicki (2015), Alves et al. (2019)) make your application carefully studied for each machine design.

This paper investigates the use of a new concept of hydrodynamic journal bearing (Ramos and Daniel (2021)) for reduction of whirl effects on rotors. For this type of bearing, both the shaft and bearing surfaces can rotate, in which the shaft rotational speed is conditioned by the operating condition of the machine and the bearing rotational speed is chosen by the action required to solve or mitigate a undesired condition or effect.

Ramos and Daniel (2021) showed that the bearing rotational motion has a great influence on the dynamic behavior of the rotor-bearing system, thus allowing to act in the bearing load capacity, the oil temperature and to prevent the fluid-induced instability. Since the oil whirl effects are responsible for the fluid-induced instability condition of the bearing (Bently, 2001), this study focused on the whirl effects that can occur under some operational conditions, in order to understand how the new bearing can mitigate these effects and thereby avoid the occurrence of critical situations.

MATHEMATICAL MODEL

Hydrodynamic Journal Bearing

The hydrodynamic journal bearing is widely used in rotating machinery for several applications. Figure 1 shows a schematic representation of the hydrodynamic journal bearing with the main geometric parameters.

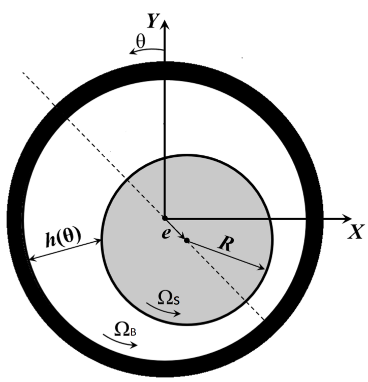

Figure 1: Schematic representation of a cylindrical journal bearing and its parameters

In this paper, the pressure distribution in the bearing is obtained using the $p-\Theta$ lubrication model (Elrod (1981), Profito and Zachariadis (2015)). Thus, the model considers the mass conservation of the oil film in the cavitation region, representing a more complete analysis to the physical phenomena in the bearing. The equation that describes the pressure distribution is indicated in Eq. 1.

$$\frac{\partial}{\partial x}\left[\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial x}\right]+\frac{\partial}{\partial z}\left[\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial z}\right]=\frac{\partial}{\partial x}\left[\rho\Theta h\left(\frac{U_S+U_B}{2}\right)\right]+\rho\Theta\frac{\partial h}{\partial t}+h\frac{\partial(\rho\Theta)}{\partial t} \tag{1}$$

in which h is the oil film thickness, μ is the dynamic viscosity of the fluid, ρ is the fluid density, p is the pressure distribution, (x,z) are the circumferential and axial directions of the bearing domain, U_S is the linear speed on the rotor surface, U_B is the linear speed on the bearing surface, t is the time and Θ is the liquid fraction in the oil film, calculated as $\Theta=\frac{\rho}{\rho_{cav}}$.

In order to obtain the influence of the thermal effects on the oil film, the $p-\Theta$ lubrication model should be solved along the energy equation. This work adopted the approach used in the work of Nicoletti (1999) for the solution of the energy equation, thus determining the Eq. 2 to calculate the temperature in the bearing.

$$\begin{aligned}\rho c_p\left\{h\frac{\partial T}{\partial t}+\left[-\frac{h^3}{12\mu}\frac{\partial p}{\partial x}+\frac{h}{2}(U_S+U_B)\right]\frac{\partial T}{\partial x}+\left[\frac{\partial h}{\partial t}\right]\frac{T-T_{inj}}{2}+\left[-\frac{h^3}{12\mu}\frac{\partial p}{\partial z}\right]\frac{\partial T}{\partial z}\right\}=\\=k_T h\frac{\partial^2 T}{\partial x^2}+k_T h\frac{\partial^2 T}{\partial z^2}-p\left[\frac{\partial h}{\partial t}\right]+\frac{4}{3}\mu\frac{1}{h}\left[\frac{\partial h}{\partial t}\right]^2+\mu\left\{\left[\frac{h^3}{12\mu^2}\left(\frac{\partial p}{\partial x}\right)^2+\frac{(U_S-U_B)^2}{h}\right]+\left[\frac{h^3}{12\mu^2}\left(\frac{\partial p}{\partial z}\right)^2\right]\right\}\end{aligned} \tag{2}$$

where T is the temperature distribution, T_{inj} is the inlet temperature of the lubricating oil in groove, C_p is the fluid thermal capacity and K_T is its conductivity.

In addition to knowing the operating conditions and geometry of the bearing, it is also necessary to determine the boundary conditions that characterize the solution of the differential equations. The boundary conditions for the modified Reynolds Equation are:

$$\begin{cases}P=P_{amb}\ and\ \Theta=1,\ \ for\ (x,z):\ z=0\ or\ z=L\\P(0,z)=P(2\pi R,z)\ and\ \Theta(0,z)=\Theta(2\pi R,z)\end{cases}$$

Equations 1 and 2 are jointly solved, in order to guarantee the convergence of the pressure and temperature fields simultaneously. After the convergence process, the hydrodynamic forces are obtained as:

$$\begin{aligned}F_X&=\int\int p\sin\theta\ dxdz\\F_Y&=\int\int -p\cos\theta\ dxdz\end{aligned} \tag{3}$$

Rotating System

The rotor model is developed using the Finite Element Method, considering disc and Timoshenko's beam elements with four degrees of freedom per node, being two translational and two angular motions, as proposed by Nelson (1980).

Once obtained the global matrices of the rotating system, the equation of motion of the system can be written as shown in Eq. 4.

$$[M]\{\ddot{q}(t)\}+([C]+\Omega_S[G])\{\dot{q}(t)\}+[K]\{q(t)\}=\{F(q,\dot{q},t)\} \tag{4}$$

in which $[M]$, $[C]$, $[G]$ and $[K]$ are the global mass, damping, gyroscopic and stiffness matrices, respectively; $\{q\}$ is the degrees of freedom vector and $\{F\}$ is the excitation vector, being composed by weight force, hydrodynamic forces of the bearing and unbalance force.

RESULTS

The evaluation of the whirl effects with the new bearing was performed under different bearing rotational speeds. The goal of this analysis is to verify how the rotational motion of the bearing can change the influence of the whirl effects on the vibrational response of the rotor. In order to favor this evaluation, the time response is transformed to frequency domain applying the Fast Fourier Transform (FFT). Firstly, the rotor is simulated with conventional bearing ($\Omega_B=0Hz$)

under an operational condition that occurs whirl effects. In following, numerical simulations are performed for the same conditions, but considering different rotational speeds on the bearing.

The rotor model used in this work is shown in Fig. 2. In this figure, the shaft elements (white), disc element (gray) and bearing elements (green) that make up the rotating system are shown, as well as the nodes (red points) in which the degrees of freedom of the system are described. The material considered for the shaft and disc is steel, whose properties are: Young Modulus $E = 210\ GPa$, density $\rho = 7850\ kg/m^3$ and Poisson ratio $\nu = 0.3$. In addition, the properties of the bearing and lubricating oil are shown in Tab. 1.

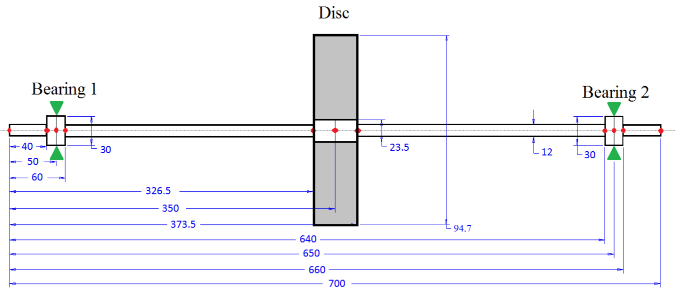

Figure 2: Schematic representation of the rotor considered in the numerical simulations (in mm)

Table 1: Bearing and lubricant properties used for simulations

Parameter	**Unit**	Value
Diameter	mm	30
Length	mm	20
Inlet Pressure	kPa	25 kPa
Inlet Temperature	$^\circ C$	25
Groove length	mm	5
Radial Clearance	μm	90
Coefficient C_μ^A (Viscosity Relation)	-	1.3495
Coefficient C_μ^B (Viscosity Relation)	-	0.7313
Lubricant thermal conductivity	W/m.K	0.132
Lubricant density	Kg/m^3	879
Specific heat of the lubricant	J/Kg.K	1962

For the rotor presented in Fig. 2, Ramos and Daniel (2021) showed that the first natural frequency is $\Omega_S = 22.5Hz$ and the instability threshold is $\Omega_S = 43Hz$. Thus, it was proposed to simulate the rotor at $\Omega_S = 42Hz$ to identify the whirl effects. This same condition was simulated for different bearing rotational speeds. The bearings speeds chosen were: $\Omega_B = -0.2\Omega_S$, $\Omega_B = -0.4\Omega_S$, $\Omega_B = -0.6\Omega_S$ and $\Omega_B = -0.8\Omega_S$. The results obtained are shown in Figs. 3 and 4.

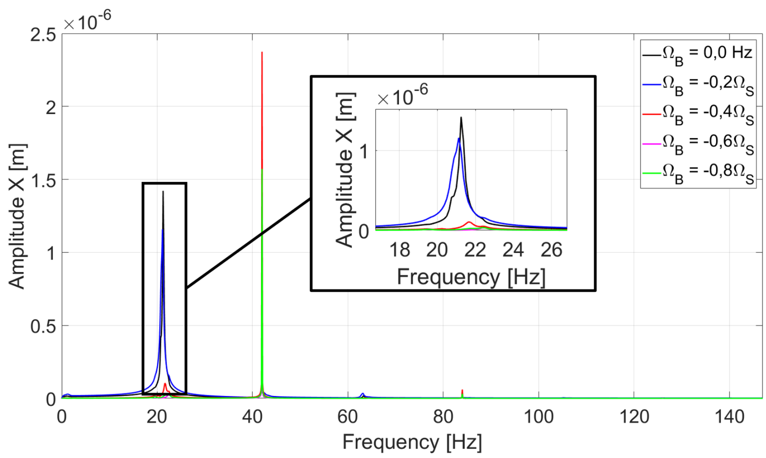

Figure 3: Comparison of whirl effects for different bearing rotational speeds in X direction

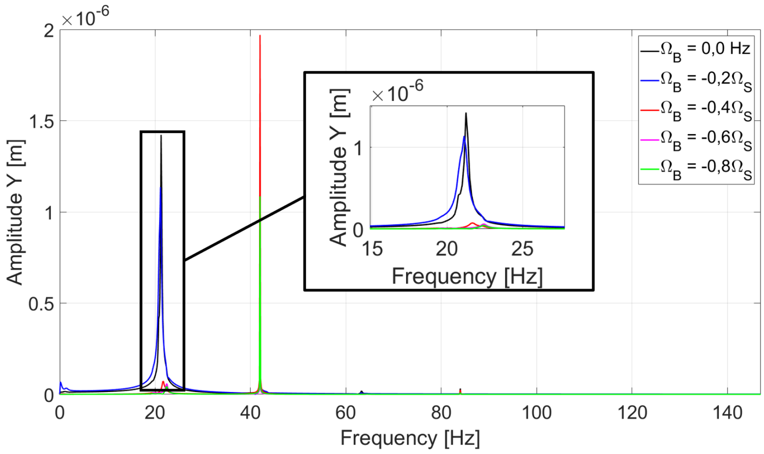

Figure 4: Comparison of whirl effects for different bearing rotational speeds in Y direction

The observed result for the FFT curves of the bearing vibration signal (Figs. 3 and 4), shows that the bearing rotational motion can change the peak amplitude of the 0.5x component of the bearing. It is verified that there are two peaks in the conventional bearing ($\Omega = 0.0Hz$), located at 1x (referring to the rotor unbalance force) and at 0.5x, referring to the bearing's whirl effects. With the rotating motion of the bearing, it is verified that the peak amplitude is reduced, to the point that it no longer occur in the signal.

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