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Analytical model of a tilting pad bearing including turbulence and fluid inertia effects

E.P. Okabe

School of Applied Sciences, University of Campinas - UNICAMP, Rua Pedro Zaccaria, 1300, PO Box 1068, Limeira, Brazil

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ABSTRACT

This study presents the modeling of a tilting pad bearing using the short bearing approach. The model incorporates the effects of turbulence and fluid inertia, which influence the dynamic behavior of hydrodynamic bearings at high rotation speeds.

The hydrodynamic forces of the bearing were calculated analytically, which is much faster than the one performed with the numerical integration. When comparing the runtime of the proposed model with the analytical model that has only the effect of turbulence, it is noticed that the time impact of including the fluid inertia effect is very small (approximately 2%).

The influence of fluid inertia was verified by simulating a symmetrical rotor supported by a pair of tilting pad bearings. The results showed that the fluid inertia generates more hydrodynamic load, but does not significantly alter the bearing stiffness or damping. The proposed model was also compared with experimental tests and with a more complete numerical model. The results showed that the proposed analytical model approaches the results of the numerical model, although it does not consider important effects, such as the temperature variation of the lubricating film.

After this analysis, we can conclude that the presented model can be used in the design optimization of tilting pad bearings. Although it is not a detailed model, its calculation speed is much superior to the other models.

1. Introduction

The increasing demand for energy pushes turbomachinery to be more efficient with a small footprint, increasing the energy density of power plants. In order to decrease the size, the most common solution is to increase the rotation speed of the system. Thus, turbine and generator components are subjected to higher loads at higher frequencies, therefore, they must be redesigned to maintain the same level of performance.

Fixed geometry journal bearings have been adopted for several rotating machines such as car engines and thermo-electrical turbines. Increasing speed causes instabilities such as oil whip, thus, it is necessary to use a more complex bearing, as the tilting pad or active-controlled pressurized fluid bearings.

They are more expensive to build and have more parts to assemble, however, they can push the operation range of rotating machinery further, raising its speed limit.

Although active-controlled bearings are becoming an efficient solution, they depend on backup systems to keep them working during a power outage, which increases the complexity of the system. Therefore, tilting pad bearings are still a good design choice for being reliable and cost effective.

One of the first relevant studies about tilting pad bearings was developed by Lund [1] to calculate their stiffness and damping coefficients. He employed the finite difference method to determine the hydrodynamic pressure over each pad, then the pressure was integrated numerically to determine the load. This load was used to calculate the stiffness and damping of each pad, then these coefficients were summed up to establish the stiffness and damping of the bearing. One of the important conclusions of his study, highlighted by Nicholas [2], was to infer that cross coupling terms of stiffness and damping vanish when pad inertia is zero.

Some effects cannot be neglected in higher rotation speed such as turbulence and fluid inertia. According to Szeri [3], turbulence is an irregular motion of the fluid in which several properties present a random variation regarding of time and position. It would be computationally expensive and unpractical to calculate the impact of this variation on each point of the fluid film in a bearing. To overcome this problem, turbulence is usually accounted by its average effect on fluid properties. In short bearing models, the effect of turbulence is considered by applying a correction factor to the viscosity. This factor basically depends on the Reynolds number and the film thickness, and

E-mail address: eduardo.okabe@fca.unicamp.br.

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Nomenclature

c	shaft damping coefficient
c_r	radial clearance
e	unbalanced mass eccentricity
$\overline{f}_{2y}, \overline{f}_{2z}$	hydrodynamic force without the acceleration components
g	gravitational acceleration (-9.81 m/s^2)
g_x	body force in the axial direction
$\overline{h}$	fluid film thickness
k	shaft stiffness coefficient
m	pad preload
m_1, m_2	disk mass and sum of the journal masses
m_e	unbalanced mass
m_{pad}	pad mass
n	number of pads
p	dimensionless hydrodynamic pressure
r	radial coordinates
$\overline{r}_0$	distance from the pad pivot to the bearing center
t	time
v_x, v_r, v_ϑ	fluid velocities in the axial, radial and circumferential directions
w	averaged axial fluid velocity
$\overline{x}$	axial coordinates
y_1, z_1	displacements of the rotor system's disk
y_2, z_2	displacements of the rotor system's journals
$\overline{y}$, $\overline{z}$	coordinates of the shaft's position
y_{pivot}, z_{pivot}	horizontal and vertical distance from the center of mass of the pad to its pivot
A	pivoting distance
D	bearing diameter
G_x	turbulence coefficient
I_{xx}	pad moment of inertia
L	bearing length
M_{jk}	mass-added coefficients
M_{pad}	moment applied to the pad
R	bearing radius
Re	Reynolds number
$R\varepsilon$	reduced Reynolds number
T_{xr}	fluid shear stress
γ	turbulence constant ($3.67 \cdot 10^{-4}$)
ϑ	angular coordinate of the bearing system
μ	absolute viscosity
μ_c	corrected viscosity for turbulent flows
ρ	fluid density
τ	modified time variable ($\tau = \omega t$)
ϕ	pivoting angle
ψ	pad rotation angle
ω	shaft rotation speed

its influence increases with speed as demonstrated by Hashimoto et al. [4].

An analytical approximation to turbulent finite bearing was proposed by Zhang et al. [5] based on the infinitely long bearing model. They used the Constantinescu model [6] to include the turbulence effect. To correct the hydrodynamic pressure of the long bearing they multiplied it by a term dependent on the axial coordinate. The results showed that the proposed solution was close to the finite element solution.

As well as the turbulence, the fluid inertia effect is dependent on the rotation speed. Although the mass of the fluid film is considerably smaller than the mass of the bearing components, as the rotation speed increases more energy is required to move the fluid. Reinhardt and Lund [7] developed a first order perturbation solution to determine the effect of the fluid film inertia on journal bearings and concluded that its influence is small on stiffness and damping coefficients, of usually few percent, being smaller than other errors. However, the acceleration coefficients (mass-added) could increase the journal mass by a factor of as much as ten, which can be relevant to small rotors. This effect was confirmed by the experimental results presented by Carter et al. [8] during the analysis of a tilting pad bearing in a "load between pads" configuration. Their results showed that added-mass term in the loaded direction was closer to 60 kg.

Tichy [9] presented an approximate analytical solution for a squeeze film bearing with fluid inertia and viscoelasticity effects. He used a linearized method that accounted periodic vibrations of large amplitude using the short bearing approach. The results demonstrated that fluid inertia and viscoelasticity have a large influence on the damping coefficients at higher speeds.

Hashimoto et al. [10] proposed a nonlinear short bearing model to analyze the effects of the fluid film inertia in super-laminar flow regime. They showed that the model with inertia and turbulence effects presents a more stable behavior that the one without inertia, which indicates that fluid film inertia should be considered to properly model a bearing operating in the turbulent regime.

Capone and Russo [11] also developed a nonlinear cylindrical bearing model including fluid inertia and turbulence effects. They applied this model to support a symmetrical rigid rotor and, therefore, to analyze its stability. They concluded that the combined effect of inertia and turbulence lower the stability limit for balanced rotors. However, they raise the region of the parametrically excited instability and lower the limit of the oil whirl instability.

Malenovský et al. [12] developed a method to incorporate memory effects on journal bearing models. Their method resulted in an extended Reynolds equation that included the fluid film inertia, therefore, generating mass-added term to the dynamical equations. They showed that this approach yields a noticeable difference for mass elements in the tangential direction. An extended Reynolds equation was also formulated by Dousti et al. [13] to include the temporal and convective inertia effects using a short bearing model. A finite length bearing was further developed by Dousti and Fittro [14]. They concluded that the convective inertia increase the load carrying capacity of the bearing, while the added mass related to the temporal inertia decreases the stability of the rotor.

2. Methodology

The film thickness of a tilting bearing can be determined using the geometric parameters shown in Fig. 1. It is a slightly improved version of the fluid film thickness adopted in Okabe and Cavalca [15,16], and it is expressed as:

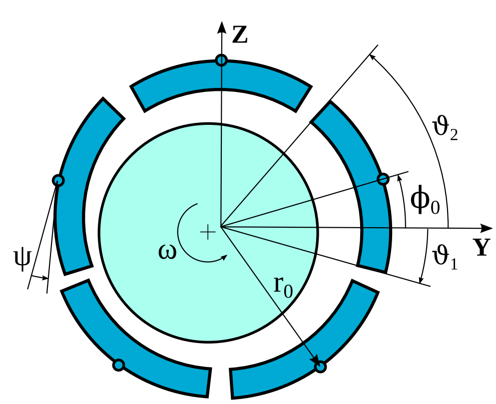

Fig. 1. Geometric parameters of the tilting pad bearing.