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Effects of oil inlet pressure and inlet position of axially grooved infinitely long journal bearings. Part I: Analytical solutions and static performance

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Abstract

This paper presents analytical expressions for the dynamic pressure and dynamic fluid force in axially grooved long journal bearings with consideration of oil inlet pressure and inlet position. The effects of oil inlet pressure (in the range of $0 \leqslant \bar{p}_i \leqslant 1$, where the dimensionless oil inlet pressure $\bar{p}_i = C^2/(\mu\omega R^2)p_i$) and oil inlet position (in the range of $0 \leqslant \Theta_i \leqslant 90^\circ$) on the static oil film configuration, pressure distribution, and steady state journal position in axially grooved journal bearings are discussed. In this paper, Reynolds–Floberg–Jakobsson boundary conditions are assumed to account for the appropriate starting position of the cavitation, the reformation of oil film at the end of caviation, and the effect of oil inlet pressure and inlet position.

Keywords: Analytical solutions; Oil inlet pressure effect; Oil inlet position effect; Cavitation and reformation of the oil film

1. Introduction

Plain journal bearings utilizing a single axial groove are extensively used in the industry. An axial groove is used to distribute oil over the entire length of the journal bearing to improve lubrication and control the temperature field in the journal bearing [1]. It is well recognized that the location of the oil inlet and the associated oil supply pressure can have a pronounced influence on the bearing performance. However, a review of the existing literature shows that the influence of these parameters on the static and dynamic performance of these bearings in particular with consideration of the appropriate starting position of the cavitation and the reformation of oil film at the end of caviation is lacking.

In most literature, the so-called Reynolds boundary condition is used to define the starting position of the cavitation. A more complete analysis is to include the Floberg–Jakobsson's boundary condition [2,3] to establish the reformation of oil film at the end of caviation. However, as Dowson et al. [4] pointed out "At the rupture boundary the well known Reynolds' condition is applicable to *all but very lightly loaded journal bearing*," cavitation may not exist for lightly loaded or pressured journal bearings. The existence of the cavitation is mainly dependent on the eccentricity ratio ε, oil inlet position Θ_i, and oil inlet pressure $\bar{p}_i$.

Based on the assumption that no cavitation exists in lightly-loaded, long journal bearings, Mori et al. [5] examined how the oil inlet position affects on the static and dynamic performance of these bearings both theoretically and experimentally. They concluded that the oil inlet position has a strong influence on both the journal center locus and the whirl threshold speed.

Solving the Reynolds equations with oil inlet pressure equal to the ambient pressure and neglecting the sub-ambient pressure, Lundholm [6] discussed the effects of oil inlet position on the static and dynamic performance of axially grooved journal bearings. Linearized stiffness and

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Nomenclature

C	radial clearance (m)
f_s	radial component of the fluid force (N)
f_ϕ	tangential component of the fluid force (N)
g	gravitational constant (m/s^2)
h	oil film thickness (m)
L	bearing length (m)
m	rotor mass per bearing (kg)
O_b	center of journal bearing
O_j	center of the journal of rotor
p	fluid pressure in journal bearing (Pa)
$\bar{p}$	dimensionless fluid pressure in journal bearing, $\bar{p} = C^2/(\mu\omega R^2)p$
p_i	oil inlet pressure (Pa)
$\bar{p}_i$	dimensionless oil inlet pressure, $\bar{p}_i = C^2/(\mu\omega R^2)p_i$
R	journal radius (m)
S	Sommerfeld number, $S = \mu RL\omega/(\pi mg)(R/C)^2$
t	time (s)
W	load per bearing (N)
ε	eccentricity ratio
ϕ	attitude angle
μ	lubricant viscosity (Pa s)
ω	running speed of the rotor (rad/s)
$\bar{\omega}$	dimensionless running speed of the rotor, $\bar{\omega} = \omega\sqrt{C/g}$.
ω_{st}	instability threshold speed of the system (rad/s)
$\bar{\omega}_{st}$	dimensionless instability threshold speed of the system, $\bar{\omega}_{st} = \omega_{st}\sqrt{C/g}$.
θ	circumferential coordinate starting from the line of the centers of bearing and journal, $\theta = \Theta - \phi$
θ_s	oil pressure starting position
θ_c	starting position of the cavitation in journal bearings
θ_i	oil inlet position, $\theta_i = \Theta_i - \phi$
Θ	absolute circumferential coordinate starting from the upper load line
Θ_i	oil inlet position in absolute circumferential coordinate system

damping coefficients were used to predict the instability threshold speed. Subsequently, Brindley et al. [7] confirmed the effect of oil inlet position on the instability threshold speed numerically for an infinitely long journal bearing with assumed π-film pressure distribution. Nevertheless, they also concluded that "The influence of supply pressure on the situation is not easy to summarize as decreasing p_f sometimes stabilizes yet in other cases the reverse is ture" [7]. The parameter "p_f" in Ref. [7] represents the oil supply/inlet pressure. Clearly, the effect of oil inlet pressure on the instability threshold speed remains unclear and requires further in-depth research.

Through solving the Reynolds equation with consideration of an appropriate starting position of the cavitation and the reformation of oil film at the end of caviation using perturbation method and finite difference method, Zhang [8] calculated the linearized stiffness and damping coefficients for an infinitely long, flexible journal bearing. Based on the linearized stiffness and damping coefficients, Zhang [8] showed that oil inlet pressure and inlet position significantly affects on the instability threshold speed.

Resently, Costa et al. [9,10] studied the influence of oil inlet pressure and inlet position on the static performance of an axially grooved journal bearing both experimentally and numerically. They pointed out that an axial groove located at a positive angle (say at 30° or 90°) from the upper load line in the direction of shaft rotation can lead to reductions in maximum temperature, peak hydrodynamic pressure, and full-film region.

The nonlinear instability analysis with consideration of an appropriate starting position of the cavitation and the reformation of oil film at the end of cavitation is not available in literature. To conduct the nonlinear instability analysis of an axially grooved long journal bearing, analytical expressions for the fluid force components in the axially grooved journal bearing with consideration of an appropriate starting position of the cavitation and the reformation of oil film at the end of cavitation are needed. They are currently lacking in the open literature.

In the Part I of this paper, analytical expressions for the dynamic pressure and dynamic fluid force in an axially grooved long journal bearing with consideration of oil inlet pressure and inlet position are derived. In the derivations, Reynolds–Floberg–Jakobsson boundary conditions (hereinafter referred to as RFJ boundary conditions) are assumed to account for an appropriate starting position of the cavitation, the reformation of oil film at the end of caviation, and the effect of oil inlet pressure and inlet position. The effects of oil inlet pressure and inlet position on the static oil film configuration, pressure distribution, and steady state journal position in axially grooved journal bearings are discussed. The steady state journal positions predicted based on different types of boundary conditions are also compared and discussed. Based on the derived analytical expressions for the dynamic fluid force components, Part II of this paper will implement an extensive nonlinear analysis of the dynamic performance of a rotor symmetrically supported by two identical axially grooved journal bearings.

2. Analytical pressure distribution

Fig. 1 shows the geometry and system coordinates used in journal bearing. An absolute circumferential coordinate Θ is introduced that starts from the upper load line, i.e $\Theta = \theta + \phi$, where θ is the circumferential coordinate