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Short Communication

A method for measuring the hydrodynamic effect on the bearing land

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ABSTRACT

The concept of dynamic pressure ratio (γ) is put forward in this paper to measure the hydrodynamic effect on bearing land. The Reynolds equation and flow continuity equation have been solved using finite difference method, overrelaxation method and underrelaxation method. Taking the four-pocket capillary compensated hydrostatic journal bearing as an example, variations of dynamic pressure ratio(γ) with eccentricity ratio (ε)and rotating speed(N) are studied. The simulation results indicate that, as eccentricity ratio and rotating speed increase, dynamic pressure ratio will increase. With eccentricity ratio lower than 0.5, dynamic pressure ratio is smaller than 5% at three rotating speeds. However, when eccentricity ratio is higher than 0.5, dynamic pressure ratio will rise fast with increasing eccentricity ratio .

1. Introduction

It is well known that oil film pressure on the bearing land is made up of two parts when the hydrostatic bearing operates in hybrid mode, with one part generated by hydrostatic effect and the other part generated by hydrodynamic effect. The latter has a great influence on the static and dynamic characteristics of the bearing.

Ho et al. [1] make a study on pressure distribution of six-pocket hydrostatic bearing, and their research results indicate oil film pressure is generated by both hydrodynamic effect and hydrostatic effect. Jain et al. [2] research dynamic and static characteristics of four-pocket flexible bearing in the hybrid mode of operation, and the bearing is compensated by different flow control devices. They find that direct stiffness coefficients (k_{11} andk_{22}) can be enlarged with the increase of deformation coefficient. The literature [3] holds that the stability of asymmetric slot-entry journal bearings is better than that of symmetric slot-entry journal bearings when the bearing operates in the hybrid mode. Satish et al. [4] analyze the difference between performance of six-pocket journal bearing and that of four-pocket journal bearing for the hybrid mode. They believe that the performance of six-pocket journal bearing is better than that of four-pocket journal bearing from the viewpoint of stability. Singh et al. [5] study dynamic and static performances of a membrane compensated multirecess journal bearing when the bearing having different recess shapes operates in hybrid mode. Sinhasan et al. [6] make a study on dynamic and static performances of an orifice compensated four-pocket journal bearing when such bearing operates in hybrid mode and is lubricated by non-Newtonian fluids. The research [7] shows that hole-entry bearings may be particularly effective when compared with other bearing configurations for good load support and low energy consumption in the two modes of operation. The literature [8] studies the performance of four-pocket orifice compensated journal bearing operating with micropolar lubricant at different rotating speeds. Chaomleffel et al. [9] make the experimental pressure distribution measurements on hybrid journal bearings. The effects of recess depth and supply pressure on pressure distribution in different modes of operation are researched. Artiles et al. [10] investigate a cryogenic hydrostatic journal bearing and discuss the compound effect of hydrodynamic and hydrostatic components on pressure distribution and mass flow. The literature [11,12] presents a general analysis on the static and dynamic behavior of multirecess journal bearings with short sills in the hybrid mode. Jain et al. [13] investigate the static and dynamic performance characteristics for the different values of the journal misalignment parameters in both hydrostatic and hybrid modes of operation. The study suggests that the misalignment significantly affects the performance of the hole-entry journal bearing. Sharma et al. [14] study the performance characteristics of a membrane compensated flexible journal bearing for a wide range of values of the operating load, deformation coefficient and for both hydrostatic and hybrid modes of operation.

A thorough review of available literature concerning the performance of the bearing in hybrid mode clearly reveals that there has not been any information related to the quantitative method for measuring the hydrodynamic effect on the bearing

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Nomenclature

c	radial clearance, mm.
e	eccentricity, mm.
h	fluid film thickness, mm.
p	pressure, N mm^{-2}.
t	time, s.
θ, z	circumferential coordinate, axial coordinate.
D	journal diameter, mm.
r	journal radius, mm.
L	bearing length, mm.
L_1	pocket length, mm.
Q_{ir}	flow of lubricant from the ith pocket ($i=1,2,3,4$), mm^3 s^{-1}
Q_{ic}	flow of lubricant through the ith restrictor, mm^3 s^{-1}.
d_c	diameter of capillary restrictor, mm.
l_c	length of capillary restrictor, mm
d_0	diameter of orifice restrictor, mm.
a_b	bearing land width in axial direction, mm.
a_t	bearing land width in circumferential direction, mm.
c_{ij}	damping coefficients($i,j=x,y$), N s/mm.
k_{ij}	stiffness coefficients, N/mm.
F	fluid-film reaction generated by hybrid effect, N.
F_r	fluid-film reaction generated by hydrodynamic effect, N.
W	external load, N.
X, Y, Z	Cartesian coordinates.
x_J, y_J	coordinates of steady-state equilibrium journal center.
p_{ir}	the ith recess pressure, N mm^{-2}
ε	eccentricity ratio
γ	dynamic pressure ratio
p_s	supply pressure, N mm^{-2}
p_1	pressure generated by hydrostatic effect, N mm^{-2}
N	journal speed, r min^{-1}

Greek symbols

θ_1	wrap angle of oil recess
θ_2	wrap angle of bearing land in circumferential direction.
θ_3	wrap angle of circumferential return oil chute
η	dynamic viscosity of lubricant, N s m^{-2}
φ	attitude angle
ρ	density of lubricant, kg mm^{-3}
ω_J	journal rotational speed, rad s^{-1}
O_J, O_b	journal center, bearing center

Non-dimensional parameters

G_1	L_1/L
G_2	$4(\theta_1+2\theta_2)/(2\pi)$
G_3	$\theta_1/(\theta_1+2\theta_2)$
$\overline{C}_{ij}$	$c_{ij}(\omega_J c/p_s L r)$
$\overline{K}_{ij}$	$k_{ij}(c/p_s L r)$
$\overline{C}_{sR}$	design parameter of capillary restrictor, $3\pi d_c^4/(32c^3 l_c)$
$\overline{C}_{sRo}$	design parameter of orifice restrictor, $3\sqrt{2}\pi d_0^2 C_d \eta/(c^3\sqrt{\rho p_s})$
C_d	discharge coefficient of orifice restrictor
$\overline{F}$	$F/p_s L r$
$\overline{F}_r$	$F_r/p_s L r$
$\overline{W}$	$W/p_s L r$
$\overline{X}, \overline{Y}$	$(x,y)/c$
$\dot{\overline{X}}, \dot{\overline{Y}}$	$(\dot{x},\dot{y})/c\omega_J$
$\overline{Z}$	z/L
H	h/c
$\overline{Q}$	$(12\eta/p_s c^3)Q$
$\overline{Q}_{ic}$	$(12\eta/p_s c^3)Q_{ic}$
$\overline{Q}_{ir}$	$(12\eta/p_s c^3)Q_{ir}$
Λ	speed parameter, $(6\eta r^2/p_s c^2)\omega_J$
τ	$\omega_J t$
$\overline{P}_{ir}$	p_{ir}/p_s
$\overline{P}$	p/p_s

Subscripts and superscripts

s	supply
—	corresponding non-dimensional parameter
c	capillary
r	recess
J	journal
b	bearing

land. However, the hydrodynamic effect on bearing land can exert significant influence on the performance of the bearing, such as increasing the bearing capacity and improving its stability. Thus, the present research aims at establishing an evaluation indicator (or a quantitative method) for measuring the hydrodynamic effect. In this paper, a concept of dynamic pressure ratio (γ) is proposed and its calculation method is presented. The purpose of the paper is to provide a reference for the designers and the experimenters of the bearing.

2. Analysis

2.1. Fluid film thickness

As shown in Fig. 1(b), O_b is the bearing center, $O_J(x_J, y_J)$ is the journal center, θ is the circumferential coordinate, e is the eccentricity, and ϕ is the attitude angle. By using geometric knowledge the film thickness of point A can be expressed as

$$h = c + e\cos(\theta - \phi) \tag{1}$$

According to the geometrical relationship, there exist $\sin\phi = x_J/e$ and $\cos\phi = y_J/e$, and H is dimensionless oil film thickness, $H = h/c$, i.e.

$$H = 1 + \overline{X}_J \sin\theta + \overline{Y}_J \cos\theta \tag{2}$$

Where dimensionless coordinates of the journal center can be written as

$$(\overline{X}_J, \overline{Y}_J) = \left(\frac{x_J}{c}, \frac{y_J}{c}\right) \tag{3}$$

2.2. Reynolds equation

The generalized Reynolds equation governing the laminar flow of an isoviscous incompressible lubricant in the clearance space of a journal and bearing in non-dimensional form is expressed as

$$\frac{1}{r^2}\frac{\partial}{\partial\theta}\left[h^3\frac{\partial p}{\partial\theta}\right] + \frac{\partial}{\partial z}\left[h^3\frac{\partial p}{\partial z}\right] = 6\omega_J\eta\frac{\partial h}{\partial\theta} + 12\eta\frac{\partial h}{\partial t}. \tag{4}$$

The non-dimensional form of the Reynolds equation can be gained by substituting dimensionless fluid film thickness ($H = h/c$),