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Numerical analysis on the dynamic contact behavior of hydrodynamic journal bearings during start-up

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ABSTRACT

The dynamic contact behavior of hydrodynamic journal bearings during start-up is investigated. The hydrodynamic oil force and asperity contact force are obtained by solving the mixed lubrication model. The motion of the journal center, the contact time, and the lift-off speed of the bearing are determined. The effects of the relative clearance and acceleration time are discussed. The result shows that the hydrodynamic oil force increases sharply during the early stage of the start-up process, leading to a sharp decrease in the contact force. An increase in the relative clearance of the bearing generates a decrease in both the contact force and the contact time. Also, a high start-up acceleration leads to a sharp decrease in the contact force.

1. Introduction

The oil lubricated plain journal bearings have been widely used in the rotational devices with high speed and heavy load. During start-up, the minimum oil-film thickness that corresponds to the lift-off speed is required to support the rotor. If the speed is lower than the lift-off speed, asperity contact occurs between the rough surfaces of the bearings [1]. Serious wear and local overheating are likely to be generated in the hydrodynamic journal bearings [2–5].

The start-up behavior of hydrodynamic bearings has been investigated previously. Mokhtar et al. [6,7] presented that the wear of the journal bearing was caused by the sliding contact during the early stage of the start-up. Chun et al. [1] used lift-off speed for determining the mixed lubricating status of the journal bearing during start-up. They found that wear of the journal bearing appears in a short period during start-up. The wear volume increased with an increasing surface roughness. Lu et al. [8] tested the Stribeck curve of hydrodynamic bearings during start-up. They found that the frictional coefficient was significantly affected by the contact of rough surfaces, which generates high temperature and seizure of the bearing [9,10]. Harnoy [11] investigated the tribological behavior of journal bearings during start-up. They found that a high start-up acceleration lead to a decrease in the wear of the bearings. Fillon et al. [12] numerically studied the safe start-up condition of journal bearings. They concluded that a larger acceleration time, a higher initial radial bearing clearance and a higher feeding temperature lead to a smaller solid thermal displacement during start-up. In order to fully understand the contact behavior of journal bearings during start-up, it is essential to calculate the dynamic contact force and the hydrodynamic force.

The mixed lubrication model based on the average Reynolds equation has been used for analyzing the contact behavior of hydrodynamic journal bearings [13,14]. Wang et al. [15–17] developed a steady-state mixed lubrication model for the conformal contacts of journal bearings. The effect of surface roughness on the hydrodynamic pressure and the asperity contact was investigated. Rao et al. [18] proposed a one-dimensional numerical model for calculating the lubricant pressure under mixed lubrication. Gu et al. [19,20] built a mixed lubrication model for analyzing the effect of surface texture on the start-up ring/liner system. The asperity contact was considered in the model. Many other investigations on the mixed lubrication regime could be found in the Ref. [21–23]. However, to the author's knowledge, the dynamic contact behavior of hydrodynamic journal bearings during start-up has not been reported previously.

In this paper, the contact behavior of journal bearings during start-up is investigated using a mixed lubrication model. The dynamic oil bearing force and asperity contact force are calculated. The evolution of the transient maximum hydrodynamic pressure is obtained. The contact time and lift-off speed are determined. The effects of both the relative clearance and the acceleration time are presented.

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Nomenclature

A_E	nominal contact area
C	radius clearance, $C = R_b$-R_s
d	standard separation of the two surfaces
D_{sum}	surface density of the asperity
e	eccentricity of the rotor in the bearing
E	elasticity modulus
E'	equivalent elasticity modulus, $1/E' = (1-\nu_1^2)/E_1 + (1-\nu_2^2)/E_2$
F_h, F_c	hydrodynamic oil bearing force and asperity contact force, respectively
F_r, F_τ	radius force and tangential force, respectively
h	nominal oil-film thickness
H	Stribeck oil ratio, $H = h/\sigma$
$\overline{H}$	normalized film thickness, $\overline{H} = h/C$
m	the equivalent mass of the rotor
N_j	shape function, $j = 1, 2, 3, 4$
p	hydrodynamic pressure
p_a	ambient oil pressure, $p_a = 0.1$ MPa
$\overline{P}$	individual asperity contact force
P	normalized hydrodynamic pressure, $P = p/p_a$
P_j	normalized hydrodynamic pressure at each node, $j = 1, 2, 3, 4$
P_0	interpolated hydrodynamic pressure over each element domain
R_p	radius of the asperity
R_b	bearing radius
R_s	rotor radius
w	weighting function
t_c	contact time
U	velocity of the rotor surface
V_r, V_τ	the radius and tangential velocities of the journal center, respectively
ν	Poisson's ratio
W	external load
x, y, z	Cartesian coordinate
$\overline{z}$	asperity height
ε	eccentricity ratio, $\varepsilon = e/C$
θ	attitude angle
ϕ, Z	circumferential degree and axial coordinate, $Z = z/B$
δ	relative clearance, $\delta = C/R_s$
σ	standard deviation of roughness distribution, R_q
μ	dynamic viscosity of the oil lubricant
ω	rotational speed of the rotor
ω_τ	angular velocity of the journal center, $\omega_\tau = V_\tau/e$
ϖ	inference of asperities, $\varpi = \overline{z} - d$
δP	change in pressure between consecutive iterations
γ	surface pattern parameter
ξ, η	normalized coordinates of the element, $-1 \leq \xi, \eta \leq 1$
$f(\overline{z})$	Gaussian distribution function of the asperity height
Φ_x, Φ_y	pressure flow factors in x and y direction, respectively
Φ_s, Φ_c	shear flow factor and contact factor, respectively
∇	gradient operator
$\mathop{\mathbf{I}}_{Gauss}$	Gaussian integration

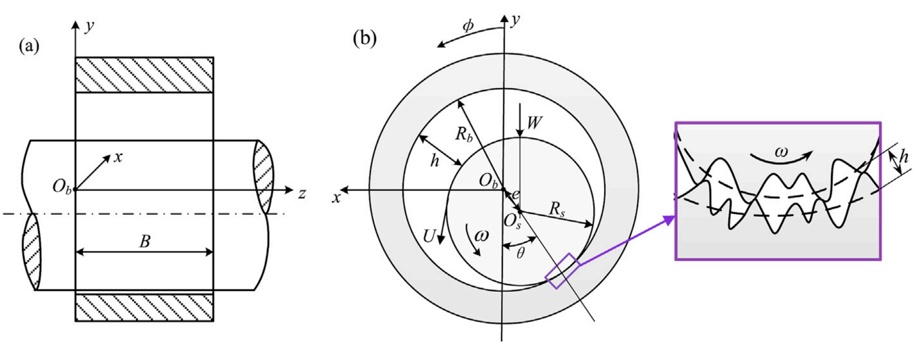

Fig. 1. The schematic of the start-up behavior of the hydrodynamic journal bearing: (a) the geometric of the bearing and (b) the motion of the bearing.

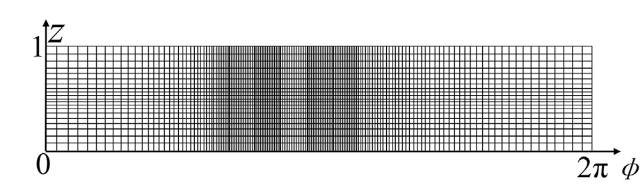

Fig. 2. Meshing of the bearing surface with 3200 elements and 3381 nodes.

2. Models

2.1. Geometry model of a hydrodynamic journal bearing

Fig. 1 shows the schematic of the start-up behavior of a hydrodynamic journal bearing. ϕ is the circumferential angle. R_s is the rotor radius. R_b and B are the bearing radius and length, respectively. The relative clearance is defined as the ratio of radius clearance and the rotor radius, $\delta = C/R_s$. Eccentricity ratio is defined as the ratio of eccentric distance and the radius clearance, $\varepsilon = e/C$. The oil-film thickness is $h = C(1+\varepsilon\cos(\phi\text{-}\theta))$. θ is the attitude angle. The external load W is constant.

Table 1
The parameters for the journal bearing.

Parameters	Symbols	Values	Parameters	Symbols	Values
Rotor radius (mm)	R_s	50	Areal asperity density (asp/m^2)	η	5e11
Bearing length (mm)	B	60	Standard deviation of surface roughness (μm)	σ	1
Density of oil (kg/m^3)	ρ	860	Equivalent modulus (GPa)	$E_{1,2}$	210
Oil viscosity at 20 °C (Pa·s)	μ	0.027	Poisson's ratio	$\nu_{1,2}$	0.29
Asperity radius (μm)	R_p	2	External load (kN)	W	4