Tribology International 119 (2018) 19–26

Contents lists available at ScienceDirect

Tribology International

journal homepage: www.elsevier.com/locate/triboint

Mixed-lubrication analysis of misaligned bearing considering turbulence

Fangrui Lv [a], Chunxiao Jiao [a], Na Ta [a], Zhushi Rao [a,b,*]

[a] Institute of Vibration, Shock and Noise, State Key Laboratory of Mechanical System and Vibration, Shanghai Jiao Tong University, Shanghai 200240, PR China
[b] Collaborative Innovation Center for Advanced Ship and Deep-Sea Exploration, Shanghai 200240, PR China

ARTICLE INFO

Keywords:
Mixed lubrication
Turbulence
Journal bearing
Journal misalignment

ABSTRACT

Some bearings operate in mixed-lubrication regime due to the low rotational speed and the heavy load and misalign due to shaft's gravity. For large bearings, the maximum film thickness is much larger than minimum nominal film thickness and turbulence may locally develop. This paper proposes an approach to analyze misaligned mixed-lubrication bearing considering turbulence. Based on the average flow model proposed by Patir and Cheng and the Ng-Pan turbulence model, a generalized average Reynolds equation is derived. The calculation procedure is established by finite difference method. The results show that turbulence remarkably increases friction coefficient, slightly increases the minimum nominal film thickness, and decreases the transition speed from mixed-lubrication regime to hydrodynamic lubrication regime.

1. Introduction

Bearings are important parts of rotor systems, and their lubrication behavior affects the reliability of the whole system. Some bearings operate under low rotational speed and heavy load conditions, and the minimum liquid film thickness is on the same order of magnitude as roughness. The asperities contact occurs locally as well, and the bearing operates in mixed lubrication regime. Meanwhile, the bearings supporting rotors and the rotor's gravity may misalign the journal bearing. In recent years, many research concerns bearings operating in mixed lubrication regime [1–6] and bearings with journal misalignment [6–9]. Considering bearing deformation, Kraker et al. [10] proposed a mixed elastohydrodynamic-lubrication model based on the average Reynolds equation [11]. To analyze the mixed lubrication bearings with herringbone-groove, Han et al. [12] presented a new shape with herringbone mesh used for finite difference method. Some research proved the influence of journal misalignment on bearing behavior is non-negligible. For example, Litwin et al. [13] analyzed the influence of journal misalignment on water-lubricated bearings and indicated that the load carrying capacity decreases when journal misalignment angle increases. He et al. [6] investigated the effect of journal deflection on a mixed lubrication bearing and concluded journal deflection increases the rotational speed at which mixed lubrication transits to hydrodynamic lubrication. The research above assumed the lubricant flow is laminar. However, for large bearing with high eccentricity ratio, the maximum film thickness is great and turbulence may locally occur.

Several turbulence models, such as those by Constantinescu [14,15], Ng-Pan [16] and Elrod-Ng [17], have been widely applied to turbulent lubrication analysis. Frene [18] used Constantinescu model with local transition concept to investigate a trust bearing in both laminar and turbulent regimes. Bou-Said and Nicolas [19] studied the effect of misalignment on hybrid bearing performance in the laminar and turbulent regimes. Bouard et al. [20] used the Constantinescu, Ng-Pan, and Elrod-Ng models to study the influence of turbulence on the performance of tilting-pad bearing and concluded all the models give similar results. Based on Elrod-Ng model, Braunetiere [21] presented a modified model for flows with low Reynolds number. Shenoy and Pai [22] investigated the effect of misalignment and turbulence on an externally adjustable bearing. Zhang et al. [23] proposed an approximate solution of carrying capacity of sliding bearings with turbulent flow that applies to bearings with high eccentricity ratio and heavy load. Susilowati et al. [24] compared bearing performance in turbulent and laminar regimes by three-dimensional CFD and concluded the pressure has the same trend in both the flow regimes. Mallya et al. [25] investigated tribological performance of a water-lubricated bearing with journal misalignment operating in turbulent regime and concluded that the turbulent lubricant and journal misalignment increase the bearing carrying capacity. The research above involved the analysis of the influence of turbulence on hydrodynamic lubrication bearing. However, the influence analysis of turbulence on mixed-lubrication bearings has not been reported.

So, to analyze the influence of turbulence on misaligned bearing, an approach for misaligned mixed-lubrication analysis considering

* Corresponding author. Institute of Vibration, Shock and Noise, State Key Laboratory of Mechanical System and Vibration, Shanghai Jiao Tong University, Shanghai 200240, PR China.
E-mail address: zsrao@sjtu.edu.cn (Z. Rao).

https://doi.org/10.1016/j.triboint.2017.10.030
Received 4 August 2017; Received in revised form 30 September 2017; Accepted 24 October 2017
Available online 28 October 2017

Nomenclature

A	area of bearing extended surface
c	radial clearance $\psi = c/r$
f	friction coefficient
f_{asp}	surfaces contact friction coefficient
$F_{asp\xi}$	asperity contact force in horizontal direction
$F_{asp\eta}$	asperity contact force in vertical direction
F_f	total friction force
F_{foil}	friction force arising from lubricant shearing
F_{fasp}	asperity contact friction force
$F_{oil\xi}$	hydrodynamic force in horizontal direction
$F_{oil\eta}$	hydrodynamic force in vertical direction
h	(nominal) film thickness
h'	film thickness ratio
h_T	local film thickness
$\bar{h}_T$	average film thickness
$h_{\min}$	minimum film thickness
$h_{\max}$	maximum film thickness
k_x, k_z, k_τ	turbulence coefficients
L	bearing length
$p(\bar{p})$	(mean) hydrodynamic pressure
p_{asp}	equivalent asperities contact pressure
r	journal radius
R_L	local Reynolds number
R_c	critical Reynolds number
z	axial coordinate
δ_1	surface roughness of surface 1
δ_2	surface roughness of surface 2
ε_0	eccentricity ratio at bearing axial midplane (referred as to eccentricity ratio)
ε_z	eccentricity ratio at each axial section
θ_0	attitude angle at the bearing axial midplane (referred as to attitude angle)
θ_z	attitude angle at each axial section
μ	dynamic viscosity of lubricant
$\bar{\tau}_{yx}$	mean shear stress in x-direction
φ	bearing circumferential angle
$\phi_x, \phi_z, \phi_{sx}$	flow factors
$\phi_f, \phi_{fsx}, \phi_{fpx}$	shear stress factors
γ	misalignment angle

turbulence is proposed, and the numerical procedure is established. The influence of turbulence on a misaligned mixed-lubrication bearing is analyzed.

2. Modeling

2.1. Geometry of misaligned bearing

Gravity acting on the shaft often bends bearing's journal along the vertical direction. Fig. 1 shows the geometry of a bearing with journal misalignment in the vertical direction.

The misalignment angle γ is usually small, so $\tan\gamma \approx \gamma$. According to Fig. 1, eccentricity ratio at each axial section can be written as:

$$\varepsilon_z = \sqrt{\frac{z^2}{c^2}\gamma^2 + \frac{2z}{c}\varepsilon_0\gamma\cos\theta_0 + \varepsilon_0^2} \tag{1}$$

where z is axial coordinate; c is radial clearance; ε_0 is eccentricity ratio at the bearing axial midplane. θ_0 is attitude angle at the bearing axial midplane. In this paper, the eccentricity ratio at bearing axial midplane is hereinafter referred to as eccentricity ratio, and the attitude angle at bearing axial midplane is referred to as attitude angle.

The attitude angle at each axial section is:

$$\begin{cases} \theta_z = \arctan\left(\dfrac{\varepsilon_0\sin\theta_0}{\frac{z}{c}\gamma + \varepsilon_0\cos\theta_0}\right) & \left(\frac{z}{c}\gamma + \varepsilon_0\cos\theta_0\right) > 0 \\ \theta_z = \arctan\left(\dfrac{\varepsilon_0\sin\theta_0}{\frac{z}{c}\gamma + \varepsilon_0\cos\theta_0}\right) + \pi & \left(\frac{z}{c}\gamma + \varepsilon_0\cos\theta_0\right) < 0 \end{cases} \tag{2}$$

Asperities may contact at the area around the minimum film thickness. So, as Fig. 1 shows, the distance between the mean levels of the two surfaces is referred as to nominal film thickness h, and the local film thickness h_T is defined as:

$$h_T = h + \delta_1 + \delta_2 \tag{3}$$

The maximum film thickness $h_{\max}$ and minimum nominal film thickness $h_{\min}$ are shown in Fig. 1.

The nominal film thickness of a vertical misaligned bearing is:

$$h = c[1 + \varepsilon_z\cos(\varphi - \theta_z)] \tag{4}$$

where φ is circumferential angle.

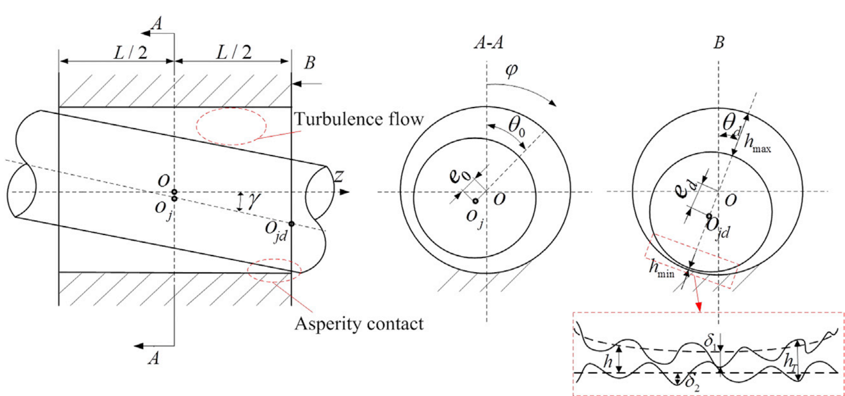

Fig. 1. Geometry model of the misaligned journal bearing.