Tribology International ∎ (∎∎∎∎) ∎∎∎–∎∎∎

Contents lists available at ScienceDirect

Tribology International

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

Effect of support stiffness and damping on stability characteristics of herringbone-grooved aerodynamic journal bearings mounted on viscoelastic supports

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ARTICLE INFO

Article history:
Received 1 October 2015
Received in revised form
15 December 2015
Accepted 8 January 2016

Keywords:
Herringbone-grooved bearings
Aerodynamic journal bearings
Bearing stability
Viscoelastic supports

ABSTRACT

Journal bearings mounted on viscoelastic supports are often used to improve the stability. However, proper support conditions have not been established especially for herringbone-grooved aerodynamic journal bearings. This study aims to make clear the effect of support stiffness and damping on the stability characteristics of the bearings. For that purpose, the linear perturbation analysis and the nonlinear transient analysis are performed for various viscoelastic support conditions. This investigation reveals that the groove configuration that gives the highest threshold speed of whirl instability changes with the viscoelastic support conditions. This investigation also reveals that it is possible to stabilize the bearings by viscoelastic supports. However, the threshold speed for lower damping support conditions decreases rather than that for rigid support condition.

1. Introduction

Herringbone-grooved aerodynamic journal bearings have been employed in high-speed and lightweight rotational machinery, such as small precision motors and turbo compressors. The use of the bearings is expected to further spread in various applications. Herringbone grooves inscribed on either the rotating or the stationary member pump the lubricating air inward and raise the film stiffness and the threshold speed of the whirl instability. To improve the bearing stability, many researchers showed optimal groove configurations for their objectives. On the other hand, it is also valuable for improving bearing stability to support the bearing sleeve by viscoelastic elements, such as rubber O-rings or air ring.

Early analysis of herringbone-grooved bearings was on the narrow groove theory (NGT). This theory was pioneered by Vohr et al. [1] and applied to investigate the bearings operating under nearly concentric conditions. Then, many authors applied the NGT to journal or thrust bearings with herringbone grooves [2–6]. Hirs [7], Bootsma et al. [8] presented the experimental bearing performance and comparisons to the prediction from NGT analysis. More recently, the finite element method (FEM) was adopted to simulate the performance of the bearings more accurately. Bonneau et al. [9] introduced a FEM analysis and compared to the NGT analysis. They presented the results for the bearing with 4–16 grooves operating with an arbitrary eccentricity. Zirkelback et al. [10] determined the dynamic force coefficients of the grooved journal bearing with plane sleeve using a FEM. Their predictions agree with the test data reported by Hirs [7]. They also conducted a parametric study for the optimum groove geometry. Jang et al. [11] investigated the dynamic behaviors of a hydrodynamic journal bearing by solving the nonlinear equations of motion due to the effect of the rotating or stationary herringbone grooves. Kobayashi [12] discussed the limitation of NGT analysis, comparing to a FEM analysis. According to these publications, the NGT works well for more than ten groove-ridge pairs. For that operating condition, the NGT is widely used even in the latest researches [13–15].

The first result of the investigations about journal bearings with flexible supports was published by Lund [16]. He qualitatively showed that the flexible support of the bearing sleeve can raise the threshold speed at which the loss of stability occurs. Powell and Tempest [17] discussed the influence of rubber properties on the use of O-rings to suppress the whirl instability in the pressurized air bearing. They concluded that rubber O-rings can provide a simple and effective method of whirl stabilization. Kazimierski et al. [18] presented theoretical calculations of the stability threshold of an external pressurized gas bearing system supported by rubber O-rings and compared to the results of experimental

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http://dx.doi.org/10.1016/j.triboint.2016.01.019
0301-679X/

Nomenclature: the nomenclature used in this paper is as follows. The variables with subscripts ˆ mean dimensional quantities while those without ˆ mean non-dimensional quantities.

b_b	support damping coefficient $=\hat{C}_r^{\,3}\hat{b}_b/\hat{\mu}\hat{L}\hat{R}^3$
b_i	damping coefficients of air film $=\hat{C}_r\hat{\omega}\hat{b}_i/(\hat{p}_a\hat{D}\hat{L})$ (i=1, 2, 3, 4)
$\hat{C}_r$	radial clearance at nonrotating
$\hat{D}$	shaft diameter $=2\hat{R}$
$\hat{e}$	eccentricity
F_r	shaft load $=W_0$
$\hat{F}_b$	bearing load $=mW_0$
h	film thickness of air film $=\hat{h}/\hat{C}_r$
H	initial groove depth ratio $=(\hat{\delta}+\hat{C}_r)/\hat{C}_r$
k_b	support stiffness coefficients $=\hat{C}_r\hat{k}_b/(\hat{p}_a\hat{D}\hat{L})$
k_i	stiffness coefficient of air film $=\hat{C}_r\hat{k}_i/(\hat{p}_a\hat{D}\hat{L})$ (i=1, 2, 3, 4)
$\hat{L}$	bearing length
$\hat{m}_b$	mass of bearing
$\hat{m}_r$	mass of shaft
m	mass ratio $=\hat{m}_b/\hat{m}_r$
M	dimensionless mass of shaft $=\hat{m}_r\hat{C}_r\hat{\omega}^2/\hat{p}_a\hat{D}\hat{L}=\Omega\Lambda^2/72$
$\hat{p}$	film pressure
$\hat{p}_a$	atmospheric pressure
$\hat{Q}_{mz}$	mass flow ratio in z-direction
$\hat{R}$	radius of shaft
W	film force $=\hat{W}/(\hat{p}_a\hat{D}\hat{L})$
X, Y	coordinates in rotor $=\hat{x}/\hat{C}_r, \hat{y}/\hat{C}_r$
X_b, Y_b	coordinates in bearing $=\hat{x}_b/\hat{C}_r, \hat{y}_b/\hat{C}_r$
z	coordinate in z-direction $=2\hat{z}/\hat{L}$
$\hat{\alpha}$	groove width
α	groove width ratio $=\hat{\alpha}_g/(\hat{\alpha}_g+\hat{\alpha}_r)$
$\hat{\beta}$	groove angle
$\hat{\delta}$	groove depth
ε	eccentricity ratio $=\hat{e}/\hat{C}_r$
ζ_b	damping ratio $=\hat{b}_b/2\sqrt{\hat{m}_b\hat{k}_b}$
θ	coordinate in circumferential direction $=\hat{x}/\hat{R}$
λ	length to diameter ratio $=\hat{L}/\hat{D}$
Λ	bearing number $=6\hat{\mu}\hat{\omega}\hat{R}^2/(\hat{p}_a\hat{C}_r^{\,2})$
$\hat{\mu}$	viscosity of air film
ν	frequency ratio $=\hat{\omega}_p/\hat{\omega}$
ξ, η	coordinates in journal $=\hat{\xi}/\hat{C}_r, \hat{\eta}/\hat{C}_r$
τ	dimensionless time $=\hat{\omega}\hat{t}$
ϕ	attitude angle
$\hat{\omega}_p$	angular velocity of rotation
$\hat{\omega}_p$	angular velocity of whirling
Ω	mass parameter $=\hat{m}_r\hat{p}_a\hat{C}_r^{\,5}/\hat{\mu}^2\hat{L}R^5$

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r	ridge part of herringbone grooves
g	groove part of herringbone grooves
0	equilibrium condition

investigations. Both the experimental and theoretical results showed that the use of rubber O-rings lead to a higher threshold speed. The authors [19] also demonstrated in an experimental investigation that the whirl onset speed of herringbone-grooved aerodynamic journal bearings supported by rubber O-rings can be raised up to about two times higher than that of rigid supported ones. Belforte et al. [20] and Waumans et al. [21] used a flexible supported bearing in their designs of high-speed spindles and verified its effectivity. An important and interesting phenomenon that may appear in flexibly mounted bearings was pointed out by Marsh [22]. He reported the bearings had a second region of stable operation when the parameters of the supports were properly chosen. Czolczynski et al. [23,24] also demonstrated the same phenomenon in a self-acting bearing and an externally pressurized bearing.

The purpose of the present work is to make clear the effect of support stiffness and damping on the stability characteristics of herringbone-grooved aerodynamic journal bearings. For this purpose, the linear perturbation analysis with considering small periodic motion for the rotating shaft is presented to determine the threshold speed of the whirl instability. The non-linear transient analysis is also presented to simulate the behaviors of journal and bearing in a certain operating condition.

2. Theory

2.1. Analytical model

Fig. 1 depicts the journal bearing system analyzed in this research. A rotating shaft is symmetric and symmetrically supported by two identical herringbone-grooved aerodynamic journal bearings at both ends. The shaft is rigid and assumed to be perfectly balanced. In addition, the herringbone-grooved bearings are mounted on viscoelastic supports. It is assumed that the viscoelastic bearing supports are very linear (constant stiffness and damping coefficients). The values in the vertical and horizontal directions are identical. Fig. 2 shows the groove shapes and the coordinates of the journal bearing.

2.2. Lubrication equation

The pressure in a lubricant film can be governed by the lubrication equation of compressible fluid flow based on the NGT. Under the perfect fluid and the isothermal conditions, the dimensionless form of the equation is

$$\frac{\partial}{\partial\theta}\left[\left(\frac{E_1}{E_0}\right)\frac{\partial p^2}{\partial\theta}+\frac{1}{\lambda}\left(\frac{E_2}{E_0}\right)\frac{\partial p^2}{\partial z}-2\left(\frac{E_3}{E_0}\right)\Lambda p+2\Lambda E_4 p\right]$$
$$+\frac{1}{\lambda}\frac{\partial}{\partial z}\left[\left(\frac{E_5}{E_0}\right)\frac{\partial p^2}{\partial\theta}+\frac{1}{\lambda}\left(\frac{E_6}{E_0}\right)\frac{\partial p^2}{\partial z}-2\left(\frac{E_7}{E_0}\right)\Lambda p\right]$$
$$-2\Lambda\left(2\left[\frac{\partial}{\partial\tau}-\frac{\partial\phi}{\partial\tau}\frac{\partial}{\partial\theta}\right]+\frac{\partial}{\partial\theta}\right)pE_8=0 \quad (1)$$

The expressions of E_0–E_8 are given in the Appendix A.

The calculation of the pressure field is performed on half a bearing in the axial direction. The boundary conditions in the axial and circumferential directions are

$$p(\theta,-1)=1$$

$$p(\theta,z)=p(\theta+2\pi,z)$$
$$\left.\frac{\partial p}{\partial\theta}\right|_{\theta}=\left.\frac{\partial p}{\partial\theta}\right|_{\theta+2\pi}$$
$$\alpha(Q_{mz})_g+(1-\alpha)(Q_{mz})_r=0 \ \text{at} \ z=0 \quad (2)$$