A Journal Bearing with actively modified geometry for extending the parameter-based stability range of rotor dynamic systems

Long Abstract

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Introduction

Hydrodynamic interactions within journal bearings can lead to unwanted oscillations due to instabilities caused by non-linear effects. In order to extend the range of stability various methods have been proposed in literature (e.g. Chasalevris et al. (2012)).
The present work deals with an actively controlled modification of the bearing clearance in order to stabilize a vertically arranged rotor system operating at high rpm.

1. Rotor model

Assuming a perfectly balanced rotor of mass $2m$ on a rigid shaft rotating with an angular velocity ω, which is supported by two adjustable two-lobe journal bearings. The influence of gravity should be neglected such that the equations of motion expressed in the dimensionless time τ according to (3) read out to be:

$$\mathbf{e}_x : \omega^2 X'' = \omega S\, f_x(X, Y, X', Y') \quad \mathbf{e}_y : \omega^2 Y'' = \omega S\, f_y(X, Y, X', Y') \tag{1}$$

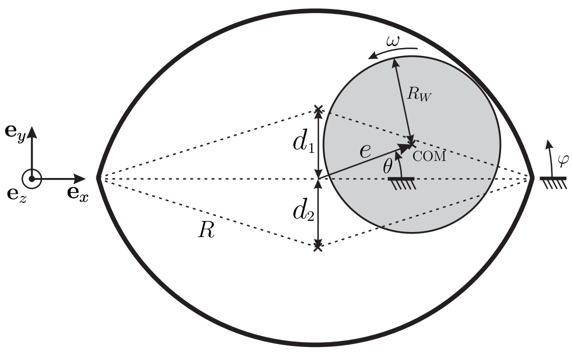

Figure 1. Schematic geometry of the two-lobe bearing

$X = (e/C)\cos\theta$ and $Y = (e/C)\sin\theta$ denote the dimensionless position of the COM of the rotor relative to the bearing's center. The dimensionless forces f_x and f_y represent the bearings' support reactions resulting from integration of the pressure distribution (cf. (6)). The parameter $S = \mu B^3 R_W / 8mC^3$ corresponds to a reduced form of the Sommerfeld-number with μ describing the dynamic viscosity of the fluid, B the bearing's width, R_W the shaft radius, R the lobe radius and $C = R - R_W$ the mean bearing clearance (cf. Figure 1).

2. Pressure Distribution

The governing equation for the pressure distribution within the bearing is given by the dimensionless Reynolds equation

$$\frac{\partial}{\partial\varphi}\left(\frac{\partial\Pi}{\partial\varphi}H^3\right) + \gamma^2\frac{\partial}{\partial\overline{z}}\left(\frac{\partial\Pi}{\partial\overline{z}}H^3\right) = 6\frac{\partial H}{\partial\varphi} + 12\frac{\partial H}{\partial\tau} \tag{2}$$

with the height-function $H = 1 - D\sin\varphi - E\cos(\varphi - \theta)$ and the eccentricities $D = D_1 = d_1/C$ for the upper lobe and $D = D_2 = -d_2/C$ for the lower lobe respectively (cf. Figure 1).
The corresponding partial differential equation is given in a dimensionless form according to

$$E = \frac{e}{C},\ \overline{z} = \frac{2z}{B},\ \gamma = \frac{4R}{B},\ \Pi = \frac{\mu\omega R^2}{C^2}p,\ \tau = \omega t,\ \frac{\mathrm{d}()}{\mathrm{d}\tau} = ()' \tag{3}$$

with p representing the fluid pressure and z the axial coordinate of the bearing.
In order to simplify the PDE the short bearing theory is used, assuming $\gamma >> 1$, the partial derivative $\partial\Pi/\partial\tau$ can be neglected. Integrating the simplified differential equation and averaging the resulting pressure function over the axial coordinate $\overline{z}$ leads to the averaged circumferential pressure distribution

$$\overline{\Pi} = \begin{cases} \overline{\Pi}_{i=1} & ,\varphi \in [0,\pi) \\ \overline{\Pi}_{i=2} & ,\varphi \in [\pi, 2\pi) \end{cases}, \quad \overline{\Pi}_i = \frac{-8E'\cos(\varphi-\theta) + 4E(\kappa - 2\theta')\sin(\varphi-\theta) - 4D_i\cos\varphi - 8D_i'\sin\varphi}{\gamma^2(D_i\sin\varphi + E\cos(\varphi-\theta) - 1)^3}. \tag{4}$$

In the following the eccentricity factors D_1 and D_2 are assumed to oscillate harmonically

$$D_1 = -D_2 = \widehat{D}\,(1 + \delta_D\cos(\Omega\tau)), \quad \delta_D << 1. \tag{5}$$

Due to complex trigonometric dependencies the circumferential integration over the positive pressure range $\Omega_p = \left\{\varphi \in [0, 2\pi] : \overline{\Pi}(\varphi) \geq 0\right\}$ is approximated by a Gauß-quadrature method:

$$\int_{\Omega_p} (\gamma^2\overline{\Pi}\cos\varphi)\mathrm{d}\varphi \approx \sum_{i=1}^{n} \gamma^2\alpha_i\overline{\Pi}(\varphi_i)\cos\varphi_i = f_x, \quad \int_{\Omega_p} (\gamma^2\overline{\Pi}\sin\varphi)\mathrm{d}\varphi \approx \sum_{i=1}^{n} \gamma^2\alpha_i\overline{\Pi}(\varphi_i)\sin\varphi_i = f_y. \tag{6}$$

In order to investigate the stability of the obvious equilibrium point $(X_0, Y_0) = (0, 0)$ a linearization of (1) is carried out, resulting in a linear system with time-dependent, periodic stiffness and damping matrices for which Floquet's multipliers can be calculated.
Literature (Dohnal (2007)) reveals that such a behaviour can lead to a stabilization effect.

3. Results

Determining the (in)stability regions by means of a numerical evaluation of the Floquet multipliers of the linearized system for different parameter sets leads to the following stability map (cf. Figure 2).
The boundary line thereby represents the parameter combinations at which the equilibrium point changes its stability property.
In the unstable region the equilibrium point is no longer asymptotically stable such that the original system tends to self-excited oscillations.
By choosing appropriate harmonic excitations for the bearing's clearance the stable region can be shifted to even higher angular velocities allowing an operation of the system at higher rpm.

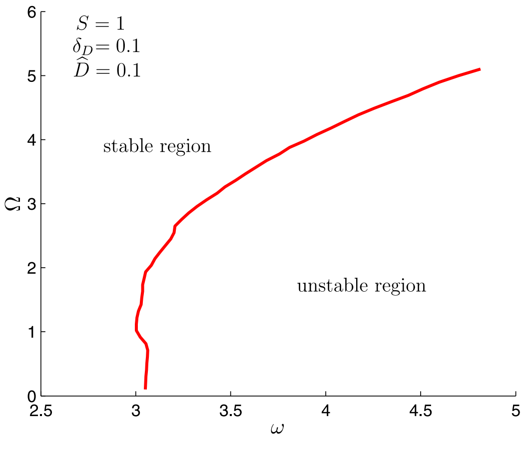

Figure 2. Stability map of the equilibrium point $(X_0, Y_0) = (0, 0)$

Further analysis of the system's dynamic behaviour during special operation states (e.g. start-up, passing through resonance etc.) have been carried out although they aren't presented in this extended abstract.

References

Chasalevris, A. & Dohnal, F. (2012). A Journal Bearing With Variable Geometry for the Reduction of the Maximum Amplitude During Passage Through Resonance. Journal of Vibration and Acoustics, 134(6), 061005.

Dohnal, F. (2007). Suppressing self-excited vibrations by synchronous and time-periodic stiffness and damping variation. Journal of sound and vibration, 306(1), 136-152.