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

K. Becker[1] and W. Seemann[1]

[1] Institut für Technische Mechanik - Bereich Dynamik/Mechatronik, KIT
Kaiserstraße 10, 76131 Karlsruhe, Germany
E-mail: *kai.becker@kit.edu*, *wolfgang.seemann@kit.edu*

Abstract: In the following a method for an active adjustment of the journal bearing's clearance is presented. It is revealed that the stability range of the equilibrium position of a rotor system can be extended by an appropriate excitation and therefore self-excited vibrations are suppressed, allowing a higher operational rpm of the whole system.

1   Introduction

Hydrodynamic interactions within journal bearings can lead to unwanted oscillations due to instabilities caused by non-linear effects. Depending on the bearing parameters and the angular velocity of the rotor, stable and unstable parameter ranges can be identified. In order to extend the range of stability various methods have been proposed in literature. E.g. in the work of Chasalevris et al. (2012) a variable bearing shell geometry is investigated with a passive geometry adjustment through an appropriate spring-damper mechanism.
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.

2   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:

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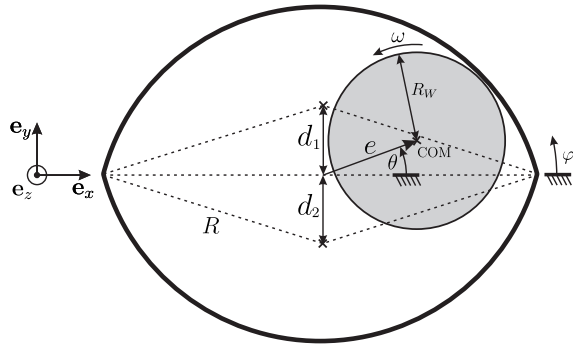

Figure 1: Schematic geometry of the two-lobe bearing

$X = (e/C)\cos\theta$ 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).

3   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

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

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 $\partial\Pi/\partial\varphi$ 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

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

In the following the eccentricity factors D_1 and D_2 are assumed to oscillate harmonically around a given mean value $\widehat{D}$ according to

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

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

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

Taking into account the symmetry of the whole problem the equilibrium point $(X_0, Y_0) = (0,0)$ can easily be read out. In order to investigate its stability a linearization of (1) is carried out, resulting in a linear system with time-dependent, periodic stiffness and damping matrices.

Literature (Dohnal (2007)) reveals that such a behaviour can lead to a stabilization effect. In order to examine the stability of the equilibrium point in dependence of the rotational velocity ω and the excitation parameters Ω and δ_D of the bearing's clearance Floquet's theory is used, i.e. calculating the eigenvalues of the corresponding Monodromy matrix.

4 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, i.e. there are two Floquet multipliers with an exact magnitude of one.

In the unstable region the equilibrium point is no longer asymptotically stable such that the original system tends to self-excited oscillations.

The stability properties can directly be assigned to those of the original system (1) since the stable region stands out to be asymptotically stable.

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.
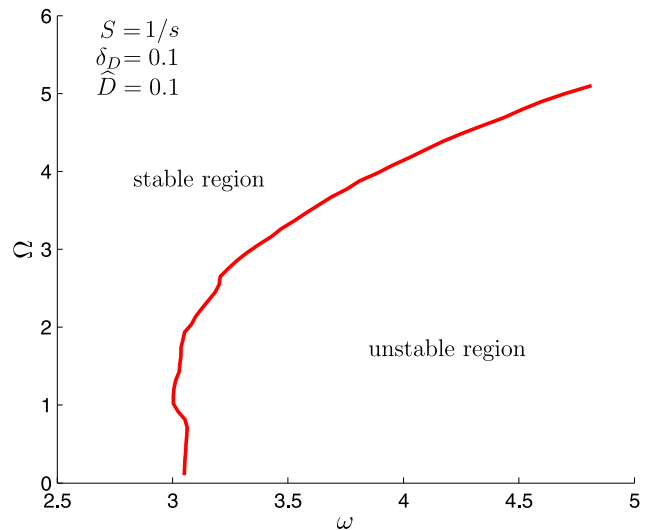

Figure 2: Stability map of the equilibrium point $(X_0, Y_0) = (0,0)$

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.