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[Extended Abstract]

Stability investigations of an elastic rotor supported by actively deformed journal bearings considering the associated spectral system

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Introduction

Due to the non-linear fluid-solid interaction within journal bearings instability phenomena (often referred to as 'oil-whirl' and/or 'oil-whip') can be observed at higher revolution speeds, which can lead to unwanted oscillations of the corresponding rotor-dynamic system. To improve this behaviour, various methods which are based on the idea of non-circular bearing geometries have been proposed in literature. E.g. in [1] first approaches of a simple two-lobe bearing with an actively controlled change in geometry are investigated in order to suppress the above mentioned instability phenomena. In the present work a more elaborated model of a journal bearing with modifiable geometry is developed. Afterwards, this bearing is implemented in an elastic Jeffcott rotor and the associated spectral system is derived and analysed.

1. Modelling of the Jeffcott rotor in actively deformed journal bearings

1.1 Geometry of the deformed bearing

As depicted in figure 1 the initially circular bearing of inner radius R_0 is deformed by two oscillating vertical forces $F(\tau) = \widehat{F}(1 - \delta_F \cos((\Omega/\omega)\tau))$ with given dimensionless time $\tau = \omega t$. The bearing is modelled as thin, circular beam with middle radius R and Young's modulus E. The rectangular cross-section is characterized by its width $B \ll R$ and its height $A \ll R$. It is assumed that the deformation is not influenced by the fluid pressure at all and that inertia terms can be neglected. Using the classical bending theory for curved beams (cf. [2]), the radial deflection $w(\varphi, \tau)$ from the undeformed state can be calculated.

A normalisation on the initial bearing clearance $C = R_0 - R_W = (R - A/2) - R_W$ leads to:

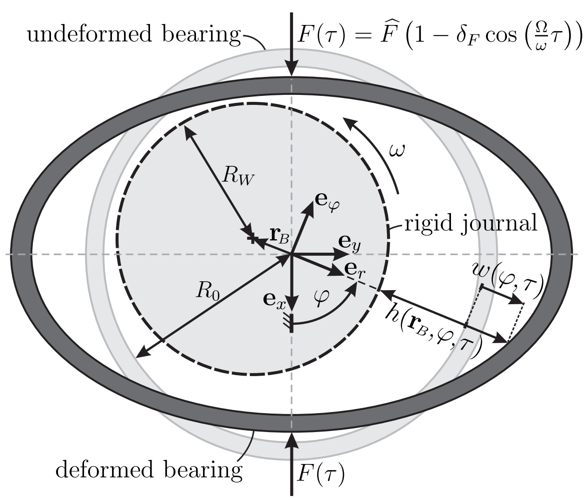

Figure 1. deformed journal bearing

$$W(\varphi, \tau) = \frac{w(\varphi, \tau)}{C} = \frac{3R^3}{CBA^3}\frac{F(\tau)}{E}\begin{cases} \frac{4}{\pi} - \sin\varphi + \left(\varphi - \frac{1}{2}\pi\right)\cos\varphi & 0 \le \varphi < \pi \\ \frac{4}{\pi} + \sin\varphi + \left(\frac{3}{2}\pi - \varphi\right)\cos\varphi & \pi \le \varphi \le 2\pi \end{cases} \tag{1}$$

1.2 Pressure distribution

With the deflection from equation (1) and the depicted kinematic relations in figure 1 the non-dimensional pressure $\Pi(\varphi, \overline{z})$ can be modelled according to the non-dimensional Reynolds equation:

$$\frac{\partial}{\partial\varphi}\left(\frac{\partial\Pi}{\partial\varphi}H^3\right)+\gamma^2\frac{\partial}{\partial\overline{z}}\left(\frac{\partial\Pi}{\partial\overline{z}}H^3\right)=6\frac{\partial H}{\partial\varphi}+12\frac{\partial H}{\partial\tau}\quad\text{with}\quad H=\frac{h}{C}=1+W-X_B\cos\varphi-Y_B\sin\varphi\,,\tag{2}$$

with the boundary conditions $\Pi(\overline{z}=\pm1)=0$, $\Pi(\varphi=0)=\Pi(\varphi=2\pi)$ and $\partial\Pi/\partial\varphi|_{\varphi=0}=\partial\Pi/\partial\varphi|_{\varphi=2\pi}$. The normalised journal coordinates are given by $\mathbf{r}_B/C=X_B\mathbf{e}_x+Y_B\mathbf{e}_y$. Assuming a rather short bearing ($2R_0/B=\gamma\gg1$) the Galerkin approach $\Pi=(1-\overline{z}^2)g(\varphi)$ is used to reduce the Reynolds equation (2) to a one-dimensional problem in $\varphi\in[0,2\pi]$, which is solved by using a finite-difference scheme.

1.3 Bearing forces

With the semi-discrete pressure values $\Pi_i(\overline{z})=(1-\overline{z}^2)g(\varphi_i)$ for $i=1\,..\,N$ the non-dimensional bearing forces f_x and f_y are calculated. After integrating along the axial coordinate $\overline{z}$ the circumferential integration in φ is performed by means of the trapezoidal rule whereby negative pressure values are neglected.

1.4 Equations of motion

Having derived the bearing forces, the equations of motion of the Jeffcott rotor (cf. [3]) are given by:

$$\begin{aligned}\overline{\omega}^2X_R''+\overline{d}_a\overline{\omega}X_R'+\frac{X_R-X_B}{\Gamma}&=f\,,\quad \eta\overline{\omega}^2X_B''+\frac{X_B-X_R}{\Gamma}-\sigma\overline{\omega}f_x=0\,,\\ \overline{\omega}^2Y_R''+\overline{d}_a\overline{\omega}Y_R'+\frac{Y_R-Y_B}{\Gamma}&=0\,,\quad \eta\overline{\omega}^2Y_B''+\frac{Y_B-Y_R}{\Gamma}-\sigma\overline{\omega}f_y=0\,,\end{aligned}\tag{3}$$

with the dimensionless parameters $\overline{d}_a$ for damping, η for the masses allocated at the bearing seats, $\overline{\omega}$ for the revolution speed, Γ for the shaft compliance, σ for the bearing characteristic and f for a vertically acting external load. X_R and Y_R thereby describe the centre coordinates of the rotor and $(.)'=\mathrm{d}/\mathrm{d}\tau(.)$ represents the derivative with respect to the non-dimensional time τ.

1.5 Derivation of the spectral system

As the time-varying bearing deformation (1) enters the equations in (3) as a parameter, the system is exposed to parametric- and self-excitation, which can lead to quasi-periodic behaviour. Therefore, the associated spectral system is derived according to the suggested method of Schilder [4], such that quasi-periodic trajectories can be easily described and analysed.

2. Simulation results

With the previously mentioned spectral system a fast analysis of the dynamic behaviour by means of solution continuation algorithms is possible even for originally quasi-periodic behaviour.
The results reveal a high potential to decrease large rotor amplitudes by selecting an appropriate time-varying deformation function (1), i.e. it is possible to shift the beginning of the above mentioned instability phenomena to even higher revolution speeds.

References

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