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Effects of journal and housing roundness errors on dynamics of journal bearings

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1. Introduction

In real applications, out-of-roundness errors affect profiles of both rotating journal and stationary bearing housing. The overall errors are random but can be decomposed to macroscopic surface waviness and microscopic surface roughness [3]. The errors on the bearing housing are stationary in a coordinate system fixed to the housing [1]. The errors on the journal, however, cause the bearing gap to be time-variant [3, 5]. The errors can vary in individual cross-sections, which further complicates the model [2]. In general, the surface waviness of the journal with the i-th wave number induces journal movement at the frequency which is i-th multiple to the journal rotating speed [3, 5]. On the other hand, the surface waviness of the bearing housing does not induce super-harmonic vibrations. However, it influences the stability, load capacity and friction coefficient of the bearing [1]. Interestingly, state-of-the-art literature does not deal with the interaction of the waviness of both journal and housing surfaces.

2. Methods

Let us assume a rigid journal of effective mass m, which rotates at *constant* angular speed ω and is supported on a journal bearing. If the angular misalignment of the journal is negligible, its motion is described by the equation

$$\begin{bmatrix} m & 0 \\ 0 & m \end{bmatrix} \begin{bmatrix} \ddot{y} \\ \ddot{z} \end{bmatrix} = -\iint\limits_{(s,\,x)} \begin{bmatrix} p\,\cos\left({}^{s}/_{r_s}\right) \\ p\,\sin\left({}^{s}/_{r_s}\right) \end{bmatrix} \mathrm{d}s\,\mathrm{d}x, + \begin{bmatrix} U\,\omega^2\,\cos\left(\omega\,t\right) \\ U\,\omega^2\,\sin\left(\omega\,t\right) \end{bmatrix} + \begin{bmatrix} -m\,g \\ 0 \end{bmatrix},$$

where $y = y(t)$, $z = z(t)$ are lateral displacements and s, x are circumferential and axial coordinates related to the bearing shell of nominal radius r_s. Function $p = p(s,\, x,\, t)$ describes the hydrodynamic (HD) pressure generated in the oil film due to the relative motion of the journal to the shell, U is the static unbalance of the journal, and g is the gravitational acceleration.

An exact form of an equation governing the HD pressure depends on simplifying assumptions. For the laminar flow of an isoviscous incompressible Newtonian fluid in a bearing gap with smooth surfaces, the governing Elrod equation reads [4]

$$\frac{\partial}{\partial s}\left(\frac{\theta\,h^3}{12\,\mu}\frac{\partial p}{\partial s}\right) + \frac{\partial}{\partial x}\left(\frac{\theta\,h^3}{12\,\mu}\frac{\partial p}{\partial x}\right) = \frac{\omega\,r_j}{2}\frac{\partial\left(\theta\,h\right)}{\partial s} + \frac{\partial\left(\theta\,h\right)}{\partial t},$$

where $\theta = \theta(s,\, x,\, t)$ is a fill ratio, which is unknown in areas where $p \leq 0$, and is equal 1 otherwise. Constants μ, r_j are the oil dynamic viscosity and the nominal journal radius, respectively, and $h = h(s,\, t)$ characterizes the bearing gap. The geometry of a profile affected by roundness errors can be described with the use of the Fourier transform [3]. After neglecting terms dependent on the square of eccentricity e^2, the bearing gap can be approximated with

$$h \approx c_r - e\cos\left({}^{s}\!/\!{}_{r_s} - \phi\right) - \sum_{i=2}^{N} \Delta_{s,i} \cos\left[k_{s,i}\left({}^{s}\!/\!{}_{r_s} - \gamma_{s,i}\right)\right] - \sum_{i=2}^{N} \Delta_{s,i} \cos\left[k_{j,i}\left({}^{s}\!/\!{}_{r_s} - \gamma_{j,i} - \omega\, t\right)\right],$$

where c_r is the radial clearance and eccentricity e and attitude angle ϕ are shown in Fig. 1. Parameters $k_{s,i},\ k_{j,i}$ are shell and journal wave numbers, respectively, $\Delta_{s,i},\ \Delta_{j,i}$ are magnitudes of the i-th waves (harmonic components) and $\gamma_{s,i},\ \gamma_{j,i}$ are phase shifts of the i-th wave.

3. Results

Vibration due to the interaction of waviness of both journal and housing surfaces is analysed in this section. In a real system, the Fourier transform of roundness errors describes the bearing gap with a sum of many waves. To simplify the interpretation of the results, only combinations of one wave at the journal profile with one wave at the shell profile are tested. Both waves have the same magnitude – 15 % of the radial clearance – and the phase shifts are neglected for simplicity. The resulting responses of the system with parameters introduced in Table 1 are shown in Figs. 2 and 3. The response is analysed in the interval $\langle 1,\, 3\rangle$ s using the fast Fourier transform (FFT). Hence, the transient response to the initial conditions is omitted.

The bearing in a reference state, i.e. with no surface waviness, is susceptible to a period-doubling, which is apparent as a response at $0.5X$. The period-doubling is magnified by the static unbalance and often precedes the oil-induced instability or is forced by this instability. The shell waviness can suppress the period-doubling with wave numbers $k_s = 2,\, 3,\, 4$. In the case of $k_s = 5$, the bearing is presumably unstable with a dominant response at $0.47X$.

The period-doubling motion is allowed also if the shell is circular ($k_{s,i} = 0$) and the journal is elliptic ($k_j = 2$). The journal waviness with higher wave numbers attenuates the period-doubling motion. However, the journal waviness excites super-harmonic vibrations at the frequency equal to $k_j\,\omega$. The highest magnitude is reached if $k_j = 2$ and decreases with increasing wave number k_j. The journal waviness with $k_j = 5$ induces almost no vibrations.

Some journal and shell waviness combinations tend to magnify the super-harmonic vibrations, with the most prominent being $k_j = 2$ with $k_s = 3$. The shell waviness with odd wave numbers magnifies the induced vibrations more than waviness with even wave numbers.

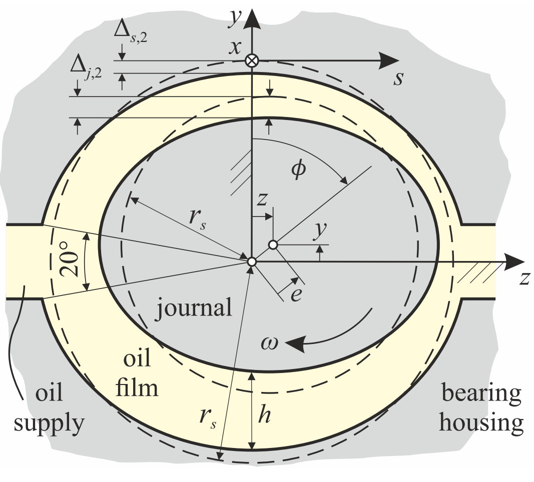

Fig. 1. Scheme of a tested bearing

Table 1. Parameters of the tested bearing

Parameter	*Unit*	*Value*
bearing diameter	[mm]	70
bearing length	[mm]	70
radial (half) clearance	[μm]	79.8
wave magnitude $\Delta_{j,i}$	[% of c_r]	15
wave magnitude $\Delta_{s,i}$	[% of c_r]	15
test speed	[rpm]	6000
static load	[N]	1000
oil viscosity	[mPa s]	12.3
supply pressure	[bar]	1.5
ambient pressure	[bar]	0

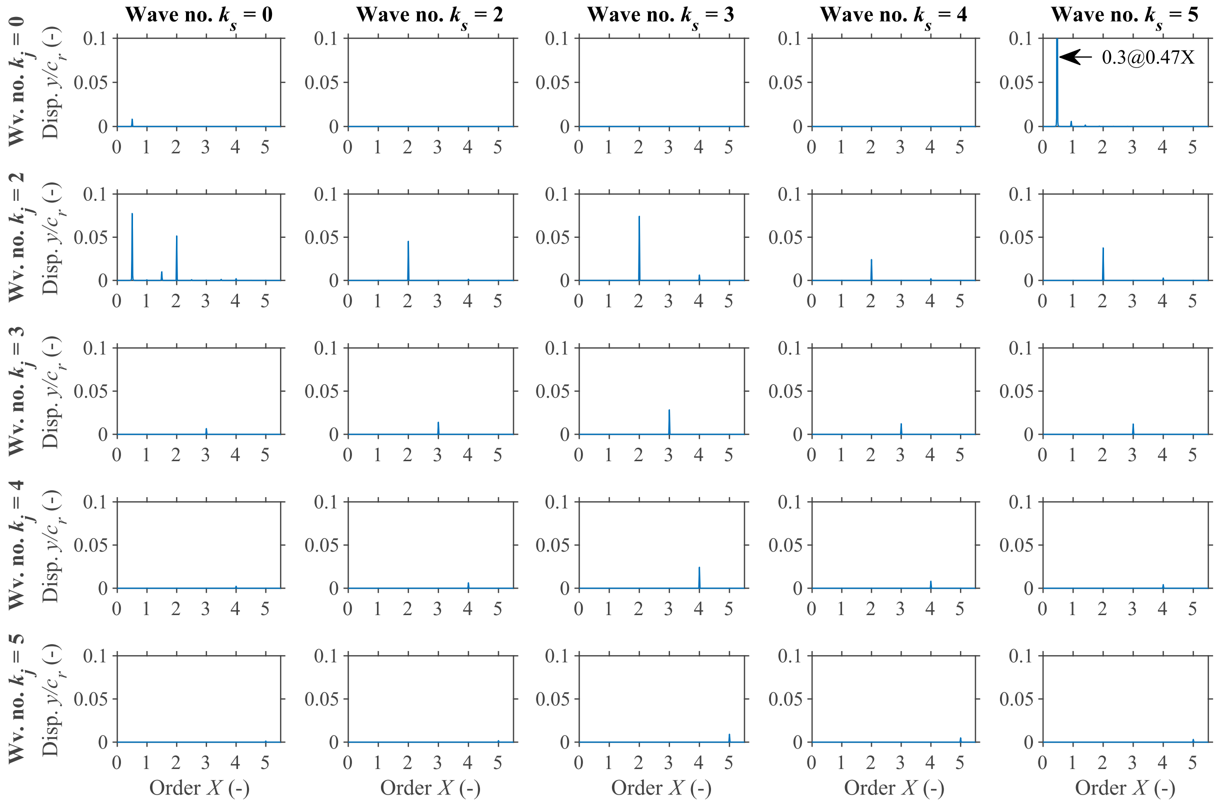

Fig. 2. Frequency spectra of vertical displacements of a perfectly balanced journal

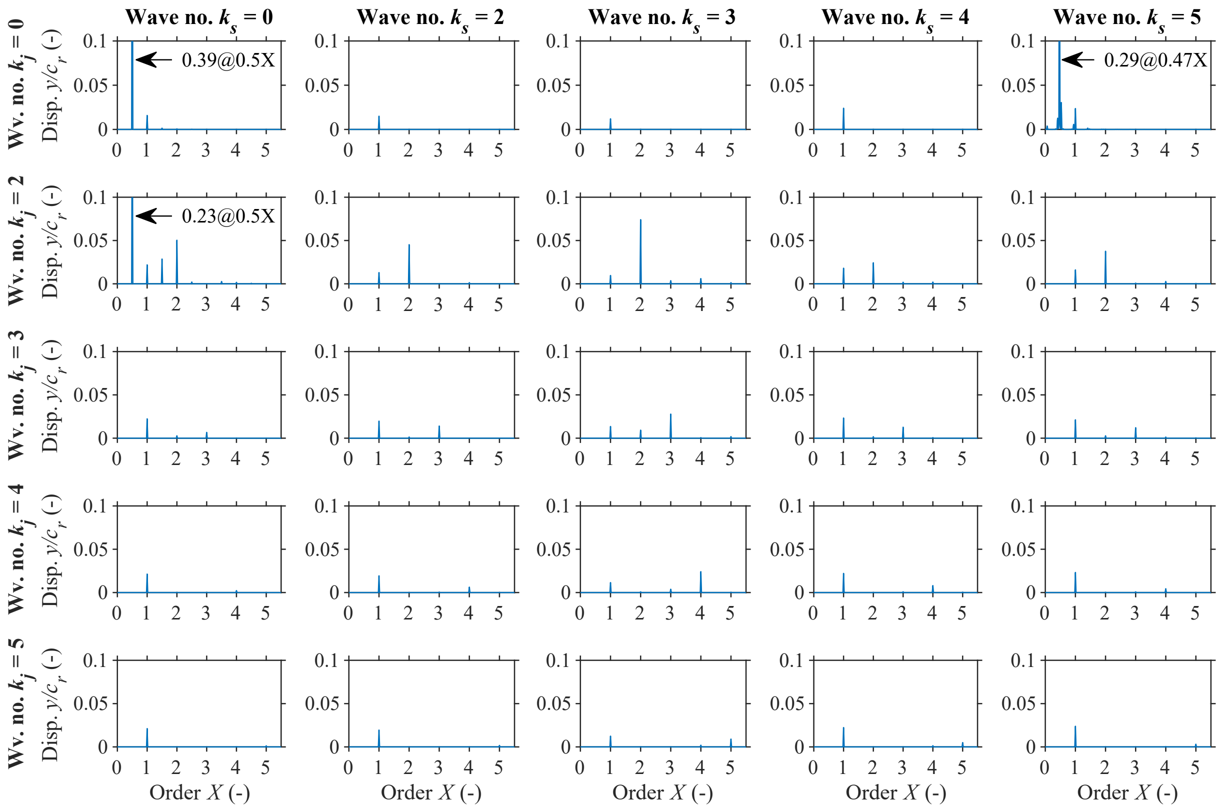

Fig. 3. Frequency spectra of vertical displacements of an unbalanced journal ($U = 217\ \mathrm{g\,mm}$)

The static unbalance has only a negligible influence on the super-harmonic vibrations. However, it affects the speed at which period-doubling bifurcations occur and the speed at which the system loses its stability.

4. Conclusions

This work presents results of simulations for the journal bearing with roundness errors of both journal and shell profiles, which are introduced as surface waviness. Various combinations of one wave at the journal profile and one wave at the shell profile are analysed in the concrete.

It is demonstrated that the waviness of the rotating surface induces super-harmonic vibration at a frequency equal to multiple of wave number k_j and angular speed ω. Although the waviness of the stationary surface does not excite any super-harmonic vibration, waves with odd wave numbers magnify the vibration induced by the rotating surface. The highest magnification has been found for the combination of stationary wave $k_s = 3$ and rotating wave $k_j = 2$. Interestingly, the rotating unbalance affects the super-harmonic vibration due to the surface waviness only negligibly. Furthermore, the waviness of the stationary surface influence the period-doubling bifurcations and stability of the system. However, a more detailed description of these effects would require a more comprehensive study.

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