DOI: https://doi.org/10.53052/9788366249974.08

Pavel POLACH[1], Jan RENDL[2], Luboš SMOLÍK[3], Štěpán DYK[4],
Michal HAJŽMAN[5]

# ŁOŻYSKA HYDRODYNAMICZNE PROMIENIOWE ORAZ NIELINIOWE ZJAWISKA DRGAŃ W ŁOŻYSKACH Z PANEWKAMI WAHLIWYMI

**Streszczenie:** Artykuł dotyczy drgań nieliniowych wirników podpartych w promieniowych hydrodynamicznych łożyskach poprzecznych. Szczególną uwagę poświęcono łożyskom poprzecznym wahliwym (TPJB). W artykule rozważono i przedstawiono dwie możliwe niestabilności rozpierania oraz drgań panewek, opracowane w TPJB. Przedstawiono nieliniowe zachowanie się wybranego TPJB działającego w warunkach tych zjawisk.

**Keywords:** łożyska ślizgowe z panewkami wahliwymi, niestabilności nieliniowe, modele obliczeniowe, drgania panewek, rozpieranie panewek

# RADIAL HYDRODYNAMIC JOURNAL BEARINGS AND NONLINEAR VIBRATION PHENOMENA IN TILTING PAD BEARINGS

**Summary:** This paper deals with nonlinear vibrations of rotors supported in the radial hydrodynamic journal bearings. Special attention is devoted to the tilting pad journal bearings (TPJBs). Two possible instabilities, spragging and fluttering, developed in TPJB are considered and introduced in the paper. The nonlinear behaviour of chosen TPJB operating under these phenomena is presented.

**Keywords:** tilting pad journal bearings, nonlinear instabilities, computational models, pad fluttering, pad spragging

[1] Research and Testing Institute Plzen, University of West Bohemia, Czech Republic, polach@vzuplzen.cz
[2] University of West Bohemia, Czech Republic, rendlj@ntis.zcu.cz
[3] University of West Bohemia, Czech Republic, carlist@ntis.zcu.cz
[4] University of West Bohemia, Czech Republic, sdyk@ntis.zcu.cz
[5] University of West Bohemia, Czech Republic, mhajzman@kme.zcu.cz

## 1. Introduction

Radial hydrodynamic journal bearings are widely used in the field of rotational machinery for their low friction behaviour, low wear and vibration-damping capabilities. However, when hydrodynamic journal bearings are used to support a rotor, the whole system becomes a complex dynamic system that may exhibit fluid-induced instabilities. Understanding of behaviour of the journal bearing closely before, during and after the instability origin and growth is the primary motivation for complex research of local and global dynamics of rotor-bearing systems.

Fluid-film bearing related characteristics such as self-excited oscillations, thermo-elastic expansion, and fluid inertia are known to induce nonlinear response states of the rotor system, such as sub-synchronous, quasiperiodic, and even chaotic motion.

There are several common methods for the modelling and dynamical analysis of structural parts of rotating systems. A standard possibility of the dynamical modelling of rotating bodies is based on the finite element method considering one-dimensional Euler-Bernoulli or Timoshenko beams. The models respect the continuous mass of rotating shafts and the possible effects of lumped masses such as discs, gear wheels, etc. The mathematical model is derived in the form of the second-order ordinary differential equation (ODE). Another possibility is the utilization of approaches based on multibody dynamics. Global motion and deformation of each body are described by a system of differential-algebraic equations (DAEs). Hydrodynamic forces acting in radial journal bearings are computed as an integral of the hydrodynamic pressure over the bearing shell. The hydrodynamic pressure is governed by the Reynolds equation, a partial differential equation, which can be derived from the Navier-Stokes equations [1]. The Reynolds equation is used in various modifications (e.g. [2]) and can consider turbulent flow, fluid inertia, fluid compressibility, surface roughness, elastic contact between surfaces, etc.

## 2. Briefly on nonlinear vibrations induced by journal bearings

The hydrodynamic forces acting in journal bearings are inherently nonlinear and can induce several types of nonlinear phenomena [3]. The best-documented nonlinear phenomenon in the journal bearings is oil whirl/whip, sometimes called fluid-induced instability [4]. Oil whirl occurs after the journal surpasses the threshold speed and the acting forces start pushing it from its equilibrium position [4]. This motion occurs when the journal losses its linear stability. The linearly unstable journal would oscillate with an exponentially growing magnitude. In practice, its motion is restricted by the bearing clearance. The hydrodynamic forces increase nonlinearly with decreasing clearance and additional elastic forces can arise if contact between the journal and the shell occurs. Hence, the dynamics beyond the threshold speed cannot be predicted by the linear model [5]. If the oil whirl frequency coincides with a shaft's natural frequency and becomes locked into it, the instability is known as an oil whip (e.g. [6]). Oil whip self-excited vibrations are generally unstable and potentially destructive. However, some systems exhibit stable whip self-excited vibrations. In such a case, increasing the shaft speed leads to further destabilisation and the second mode whirl occurs [6]. Sub-synchronous oscillations can also appear if a flexible rotor passes through the weekly damped critical speed. These oscillations have the character of a stable whip and quickly disappear with the rising speed.

Another nonlinear phenomenon stems from an interaction between the hydrodynamic forces and forces due to rotating unbalance [3]. Barrett et al. [7] were among the first to study this interaction. In 1976, they discovered that the rotating unbalance could suppress vibration due to the oil whirl. Some twenty years later, Brown et al. [8] found that the rotating unbalance could induce chaotic vibration.

Other nonlinear phenomena have connections with thermal effects [9] and friction or contact forces in self-equalising bearings [3]. Non-uniform shear stress in the oil film can heat the journal surface due to hydrodynamic friction. If only a small part of the surface is heated repeatedly, a shaft bends, changing the unbalance distribution. This thermal instability is called the Morton effect [9].

Various bearing designs can suppress mentioned instabilities or undesirable rotor behaviour and effects. The bearing designs are distinguished into two main groups -- bearing with fixed geometry and bearing with adjustable geometry based on the applied load. The typical bearing design for the second group is the titling pad journal bearing. Although it can suppress undesirable phenomena connected mainly with the fixed geometry bearing, this kind of bearing can be a source of another type of instabilities called pad fluttering and spragging [10].

## 3. Tilting pad journal bearings

A typical tilting pad journal bearing (TPJB) is shown in Fig. 1a. The TPJBs are widely used to support large rotors such as turbines or generators. Their main advantage is better resistance to fluid-induced instabilities at high speeds compared to classical journal bearings. Stability is ensured by the pad's tilting (see Fig. 1b, where various pad design is presented), which keeps the radial component of a hydrodynamic force dominant to the tangential component.

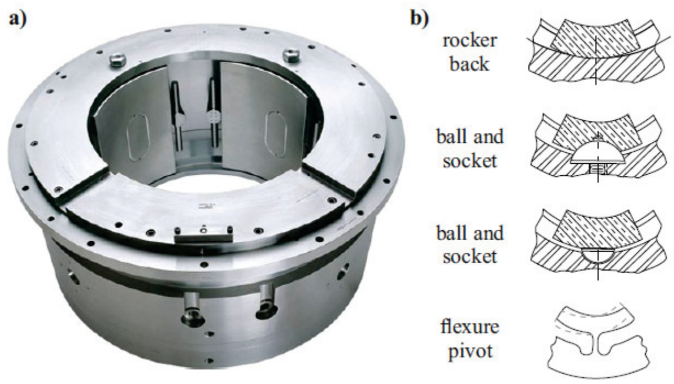


*Figure 1. Example of real TPJB (a) and scheme of the tilting pads (b) – taken from [11]*

The pads are supplied with lubricant oil which fills the bearing gap during the operation. In the ideal case, the bearing gap has a wedge form (see Fig. 2b).

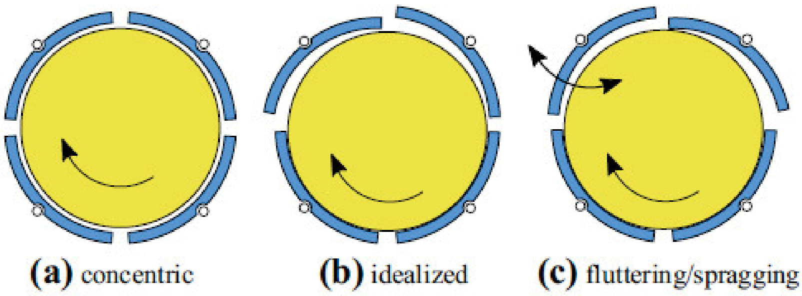


*Figure 2. Possible scenarios of TPJB behavior (taken from [12]). Figure (c) depicts both fluttering (left-upper pad) and spragging (right-upper pad)*

The self-aligning nature of the TPJB also imposes some undesirable phenomena. Lower pads are loaded mainly by rotor mass and tend to tilt to achieve a convergent bearing gap from the leading edge. However, upper pads are loaded only by their mass and can move freely in the bearing gap [12, 13]. In this case of the unloaded pad, the leading edge can come close to the journal and form the divergent bearing gap (see Fig. 2c), due to operating conditions dependent on the preload, pad moment of inertia, volume of supplied oil, circumferential journal speed and pivot position. The pressure generated close to the pad's leading edge then forces the journal out of its current position, leading to a realignment of the TPJB. This phenomenon is called spragging [10, 12]. Spragging is characteristic by spontaneous changes of the journal and/or pad positions due to rapid changes in the pressure field.

Continuous, often unstable vibration of the pad is termed pad fluttering [10, 14]. Fluttering is caused by repeated formation and reformation of the pressure field cause fluctuation and/or contact between the bearing parts, and vibrations at subsynchronous frequency occur). Pad fluttering affects the level of machine vibrations and shortens the pad's life because the fluttering pad repeatedly hits the journal [12, 15]. A proper design of the TPJB can suppress pad fluttering effectively. The most common practices are pad preloading, shortening pad arcs, or the introduction of pockets or reliefs at the pad's edges [12, 16]. More recent works also stress the importance of sufficient supply flow [17].

## 4. Computational modelling of the tilting pad journal bearings

Computational modelling of tilting pad journal bearing is a complex problem involving the modelling of hydrodynamic lubrication and sufficient description of the pads' motion [10]. The planar lateral motion of the pad without axial misalignment is supposed. Hence, the pad motion is described by one degree of freedom (tilting around the pivot necessary for proper tilting pad journal bearing operation). The additional degree of freedom (radial) can be added to consider the radial pivot flexibility under the loading transferred through the oil film.

A load of the individual pad is mainly given by pad preload and can be considered different for separate pads. The spragging or fluttering phenomena can occur in case of improperly chosen preload, mainly for upper pads. Then, the bearing clearance can disappear and the modelling of contact between the solid parts based on Hertz contact theory must be taken into account. All bearing parts are supposed to be rigid.

The pressure distribution in the bearing gap is given by the Reynolds equation. The laminar or turbulent flow is distinguished based on the Reynolds number, which involves the bearing dimensions and operating conditions. Developed hydrodynamic pressure acts on the bearing parts by the hydrodynamic force determined by pressure integration over the pad surface. It is a nonlinear force coupling. A slight change of resultant force sets the pad into motion. Pressure distribution is influenced by the relative position of the pads and the journal. Another aspect of the bearing modelling is the thermodynamics in the lubricant flow, which affects the dynamic viscosity of the lubricant and cavitation.

For the more precise computational model of TPJB, the friction in the pivots, mainly in the ball-and-socket coupling, should be considered because the stick and slip phases of the pad can occur during the operation. The friction has a negative role on the pad tilting behaviour, and the global bearing behaviour can be close to the bearing with fixed geometry if the pad is in the sticking phase. The conformal contact theory [18] should be considered for the description of the contact pressure in the case of a ball-and-socket pivot. The most used friction models are the Coulomb, Bengisu-Akay and LuGre models [19]. The latter two models are susceptible to the control parameters, and it is necessary to focus more on their establishment during the modelling.

## 5. Study case: Pad fluterring in a four-segment TPJB

Above stated computational model [10] and typical phenomena related to the tilting pad journal bearing are demonstrated on the nonlinear steady-state response to harmonic excitation of four-segment TPJB. This configuration of TPJB has two upper pads which are almost unloaded during operation, and thus they might be susceptible to pad fluttering. The TPJB was adopted from [20] and is presented in Fig. 3 and Table 1.

The response of the system to the out-of-balance force was analysed in the range of rotor speeds from 1,500 to 13,000 rpm. The responses depicted in Figs. 4 and 5 are obtained for the case of bearing without preload. The results are shown in the form of spectrograms and bifurcation diagrams and include information about the Lyapunov exponents:

- The motion of the journal and all four pads were analysed using the fast Fourier transform (FFT), and the results are depicted as spectrograms – see Fig. 4. These diagrams allow identifying qualitative changes in the response with changing rotor speed.
- For more precise insight into the dynamic behavior of the system, bifurcation diagrams were constructed – see Fig. 5. These diagrams depict local extremes at the particular rotor speed employing different colors for local maxima (red) and minima (black). The bifurcation diagrams help to analyse the steady-state response and distinguish between regions with periodic, quasiperiodic and chaotic motions. This diagram is supported by an evaluation of the Lyapunov exponents to quantify the nature of the excited motion of the journal and individual pads.

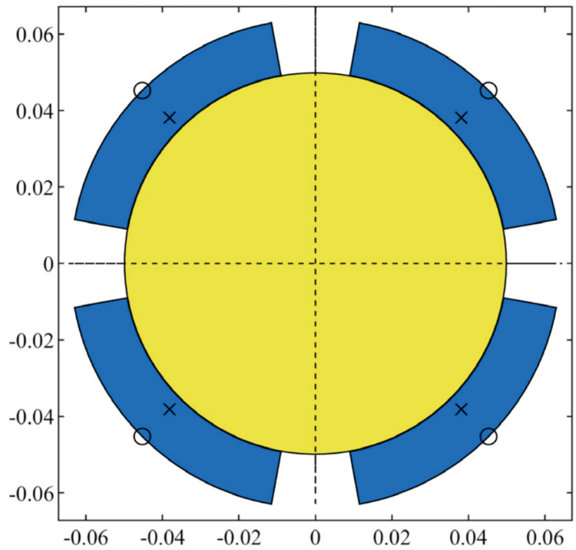


*Figure 3. A scheme of the studied TPJB (taken from [20]). Pivots are situated at 45, 135, 225 and 315 deg (measured from the horizontal axis). Pads centres of mass are denoted by crosses (x) and pivots by circles (o)*

*Table 1. Parameters of the studied TPJB (taken from [20]). The pivot ratio is the fraction of the distance between the leading edge and the pad pivot point to the complete pad arc length.*

| Parameter | Value | Unit |
|---|---|---|
| Journal radius | 49.9 | mm |
| Journal weight (static load) | 19.6 | kN |
| Journal static unbalance | $5 \cdot 10^{-3}$ | kg m |
| Maximum speed | 13000 | rpm |
| Pad inner radius | 50 | mm |
| Pad axial length | 100 | mm |
| Pad thickness | 14 | mm |
| Pad material density | 8400 | kg $m^{-3}$ |
| Pivot radius | 64 | mm |
| Pivot ratio | 0.5 | |
| Oil dynamic viscosity | $19 \cdot 10^{-3}$ | Pa s |
| Oil density | 860 | kg $m^{-3}$ |
| Ambient and supply pressure | 0 | Pa |
| Reynolds number | 307.3 | |
| Reduced Reynolds number | 0.62 | |

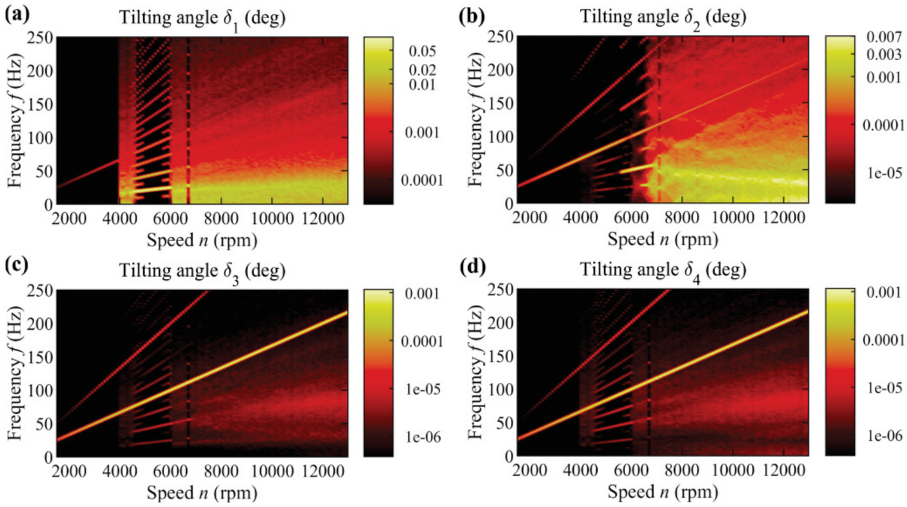


*Figure 4. FFT analysis of the nonlinear steady-state response to harmonic excitation of TPJB without preload (adopted from [10])*

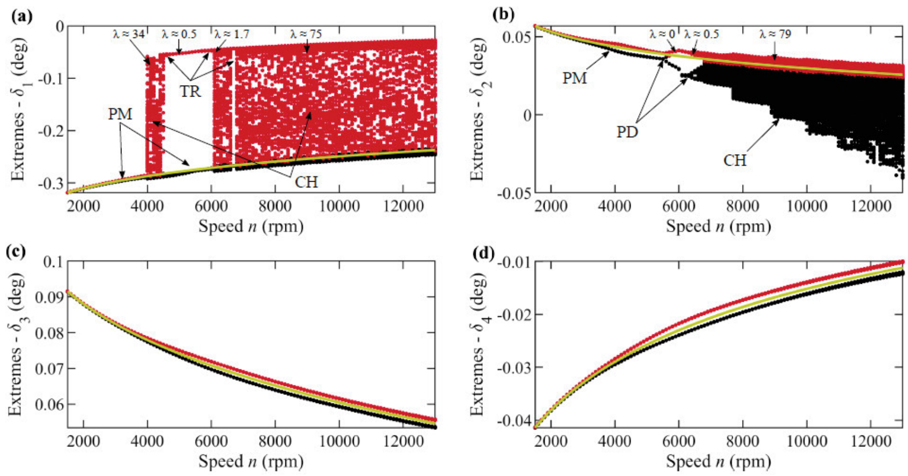


*Figure 5. Bifurcation diagrams of TPJB without preload (adopted from [10])*

Based on the FFT analysis of the nonlinear steady-state response to harmonic excitation (see Fig. 4) and from bifurcation diagrams (see Fig. 5) can be summarised following conclusions:

- The dominant response of the bottom pads 3 and 4 is synchronous (1X). The 1X component is caused due to the out-of-balance force. The second harmonic (2X) is also present, but approximately two orders of magnitude smaller than 1X. The 2X component stems from the nonlinearity of the system, and it is also caused due to the out-of-balance force. See Fig. 4c and Fig. 4d.
- The motion of pads 3 and 4 is periodic – see Fig. 5c and Fig. 5d.
- The response of upper pads 1 and 2 differs significantly from the response of the bottom pads – see Fig 4a, Fig. 4b, Fig. 5a and Fig. 5b.

- The motion of pad 1 can be classified as chaotic almost in the whole investigated speed range. The chaotic nature of this motion arises from repeated single-sided impacts of the pad's edge to the journal. At lower speeds (4,000 rpm), a single-sided contact occurs, while increasing rotor speed. Transient zone (TR) between periodic (PM) and chaotic (CH) motion. See Fig. 4a and Fig. 5a.
- The response of pad 2 can be divided into three main regions with qualitatively different behaviour. At low rotor speeds, the response is synchronous with negligible harmonics and sub-harmonics due to the nonlinearity. The period-doubling (PD) occurs after approximately 5,500 rpm and 6,400 rpm. The period-doubling (PD) events are apparent from the formation of sub-synchronous components 0.5X and 0.25X from the bifurcation diagram (see Fig. 5b). After surpassing 6,700 rpm, the pad motion becomes chaotic, with cascading increments in oscillations minima. See Fig. 4b and Fig. 5b.

Complete results, discussion and more study cases are presented in [10].

## 6. Conclusions

The phenomena causing nonlinear vibrations of journal bearings were briefly introduced in this paper, with the main focus devoted to the tilting pad journal bearings (TPJBs). The structure of the TPJB and its possible instabilities spragging and flattering were presented. An approach to the TPJB computational modelling was briefly introduced. For example, the nonlinear steady-state response of a four-segment TPJB to the harmonic excitation calculated using the introduced computational model was presented. It has been demonstrated that the TPJB can exhibit quasi-periodic and even chaotic vibrations due to self-excited motion of the upper unloaded pads. Although such behaviour is not as dangerous as oil whirl and oil whip, it should be considered during the design of the TPJB and avoided, if possible.

## Acknowledgements

This work was supported by project 22-29874S of the Czech Science Foundation and by the institutional support for the long-time conception development of the research institution provided by the Ministry of Industry and Trade of the Czech Republic to Research and Testing Institute Plzen.

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