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Letter to the Editor

On the active stabilization of tilting-pad journal bearings

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1. Introduction

Conventional mechanical bearings, such as rolling-element and fluid-film bearings, are passive devices in the sense that they cannot adjust their dynamic behavior in response to changes in the operating conditions. To overcome this limitation, *active* bearings have been suggested as an instrument for improving the stability and performance. In particular, active bearings can potentially reduce rotor vibration, adjust the load stiffness, provide an automatic rotor balancing capability, and compensate for rotor misalignment, changes in rotor speed, and dynamic loads.

Magnetic bearings are the most popular type of active bearing [1], and are well suited for high-speed applications because of their unique ability to suspend the imposed load with virtually no friction. However, due to its low load-carrying capacity and high cost relative to mechanical bearings, the active magnetic bearing (AMB) is not the most cost-effective solution for many low-/medium-speed, high-load applications. In addition, since the AMB is open-loop unstable (i.e., when operated with no feedback control), it requires backup ball bearings in case an electromagnet failure occurs. As a result of these limitations, one may wonder: can conventional *mechanical* bearings be adapted for active use? Although this question has been answered in the affirmative in several papers published since the 1980s (e.g., see Refs. [2–10] and the references therein), it remains a relatively unexplored area within the *traditional control* research community. The purpose of this note is to present a contribution to the active mechanical bearing literature from a new perspective. Specifically, we first discuss the general concept of an active mechanical bearing by drawing an analogy with the AMB. We then outline the preliminary step in developing an advanced feedback control framework for a type of active tilting-pad journal bearing (TPJB), which is founded on adaptive non-linear control theory. A series of simulations are also presented with the intent of illustrating the stabilizing effect of the proposed adaptive controller on otherwise unstable operating conditions of the rotor–TPJB system, which can damage the bearing and possibly cause a catastrophic failure.

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2. Active mechanical bearings

To answer the question posed in the previous section, it is necessary to first understand what enables an active bearing, such as the AMB, to change the dynamic behavior of the rotor–bearing system. Fundamentally, there needs to be a mechanism for on-line adjustment of the force transmitted from the journal to the bearing. In the AMB, this occurs by adjusting the flux field produced by the electromagnet (i.e., $F(t) = k\phi^2(t)$ where $F(t)$ is the magnetic force, k is a constant, and $\phi(t)$ is the flux [1]). The flux, in turn, is actively controlled through the voltage applied to the electromagnet. In extending this idea to a mechanical bearing, we need to identify an analogous mechanism. To that end, we can view the pressure field in a fluid-film journal bearing as the equivalent of the flux field in the AMB. The issue then becomes: How does one actively adjust the pressure field? A direct approach is to control the supply pressure of the fluid injected in a hydrostatic bearing [8–10] in response to the rotor motion.[1] In the analogy with the AMB, one can view the active hydrostatic bearing as equivalent to the current-controlled AMB [1], where the electromagnet current (or flux) is assumed to be the control input.

An indirect approach is to adjust the geometry and thickness of the fluid film in a hydrodynamic bearing. The reasoning behind this approach can be understood by examining the Reynolds equation for laminar, incompressible, Newtonian, inertialess, thin-film flows [11],

$$\frac{1}{r^2}\frac{\partial}{\partial\theta}\left(h^3\frac{\partial P}{\partial\theta}\right) + \frac{\partial}{\partial z}\left(h^3\frac{\partial P}{\partial z}\right) = 6\mu\omega\frac{\partial h}{\partial\theta} + 12\mu\frac{\partial h}{\partial t}, \tag{1}$$

where $P(\theta, z, t)$ is the pressure field between the journal and bearing, θ and z are co-ordinates of the bearing system, $h(\theta, z, t)$ is the film thickness, r is the journal radius, $\omega(t)$ denotes the journal speed, and μ is the lubricant viscosity. The right-hand side of Eq. (1) can be viewed as the "control input", whereby adjusting $h(\theta, z, t)$ provides a means of manipulating the pressure field. Hence, an active hydrodynamic bearing is the analog of the voltage-controlled AMB [1], where the electromagnet voltage is the control input.

One method for implementing the indirect approach is to actively deform a flexible membrane segment on the surface of a plain journal bearing [2,5,6] and thereby, reshape the fluid film. On the other hand, to a certain extent, the mechanism behind the indirect approach intrinsically occurs in a TPJB, where the pads are free to partially rotate about a fixed pivot. With this in mind, a more versatile method is to individually actuate each pad of the TPJB in the radial direction [2,7], while still allowing the pads to freely tilt (see Fig. 1). This method can be taken a step further by using *feedback control* to actuate the pads [4] based on measurements of the pad and journal motions (i.e., an automatic preloading). In essence, the control mechanism of the active TPJB is designed to enhance the pressure-generating capacity of the normal squeeze action [11,12] (the second term on the right-hand side of Eq. (1)). In what follows, we discuss the design of a model-based, feedback control strategy for the above-described active TPJB.

[1] The active bearing proposed in Refs. [8,9] is more accurately categorized as a TPJB augmented with an active hydrostatic mechanism.

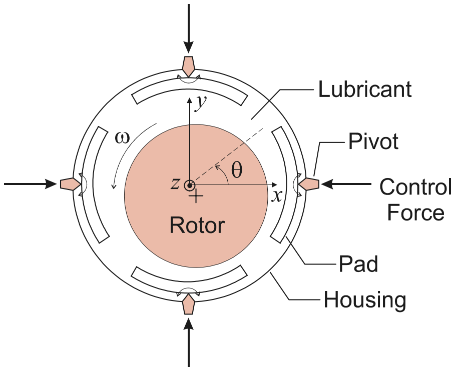

Fig. 1. Conceptual illustration of the active TPJB.

3. Active TPJB system model

Consider the four-pad, active TPJB system shown in Fig. 1. Let (x, y, z) be an inertial coordinate frame with the origin at the housing center. We assume that: (1) the rotor shaft is vertical, (2) the rotor, pivots, and pads are rigid, and (3) the x- and y-direction motions of the rotor are decoupled. Let $x_r(t)$, $x_1(t)$, and $x_2(t)$ denote the position of the rotor center, right pad/pivot pair, and left pad/pivot pair along the x-axis, respectively. Similarly, $y_r(t)$, $y_3(t)$, and $y_4(t)$ denote the position of the rotor center, top pad/pivot, and bottom pad/pivot along the y-axis, respectively. Let m_r and m_i be the mass of the rotor and ith pad/pivot, respectively. Finally, $F_i(t)$ denotes the control force applied to the ith pad/pivot. The equations of motion for the rotor–bearing system are given by

$$m_r \ddot{x}_r = \sum_{i=1}^{2} f_{h_i}, \qquad m_r \ddot{y}_r = \sum_{i=3}^{4} f_{h_i}, \tag{2}$$

$$m_i \ddot{x}_i = F_i - f_{h_i}, \quad i = 1, 2, \qquad m_i \ddot{y}_i = F_i - f_{h_i}, \quad i = 3, 4, \tag{3}$$

where $f_{h_i}(t)$ is the hydrodynamic force the fluid applies on the rotor and ith pad. Unfortunately, one cannot obtain a general, closed-form expression for f_{h_i} from first principles. As a result, a "control-friendly" model that approximates the behavior of f_{h_i} is required for the design of a control law. Typically, a linear spring–damper-like force as shown below is used

$$f_{h_i} = b_i(\dot{x}_i - \dot{x}_r) + k_i(x_i - x_r), \quad i = 1, 2, \tag{4}$$

$$f_{h_i} = b_i(\dot{y}_i - \dot{y}_r) + k_i(y_i - y_r), \quad i = 3, 4, \tag{5}$$

where b_i and k_i represent the bearing damping and stiffness coefficients, respectively. Typically, a perturbation analysis is performed about an equilibrium point to estimate constant values for the dynamic coefficients [13]. However, these values are limited to only a small region about the equilibrium point (e.g., small eccentricity ratios). Here, to account for variations in the rotor speed, lubricant viscosity, and/or lubricant temperature, we allow the dynamic coefficients to vary slowly with time.

4. Adaptive control law

We now outline the design of a feedback controller for the active TPJB based on adaptive non-linear control theory [14,15]. The detailed mathematical development of the control law, inclusive of the closed-loop stability analysis, is described in Ref. [16]. The problem statement of interest in this note is as follows. Given the active TPJB dynamic model of Eqs. (2)–(5), we seek a control law to drive the rotor to the housing center, i.e.,

$$\lim_{t\to\infty} x_r(t) = 0 \quad \text{and} \quad \lim_{t\to\infty} y_r(t) = 0, \tag{6}$$

despite the coefficients $b_i(t)$ and $k_i(t)$ being *unknown* and slow time-varying. It follows, therefore, that the controller should automatically compensate for parametric uncertainties and variations in the bearing damping and stiffness caused by changes in the operating conditions. To solve this problem, we use an adaptive full-state feedback control law

$$\begin{aligned} F_i &= F_i(p_x, \dot{p}_x, \hat{\theta}_x), \quad i = 1, 2, \\ F_i &= F_i(p_y, \dot{p}_y, \hat{\theta}_y), \quad i = 3, 4, \end{aligned} \tag{7}$$

where

$$\begin{aligned} p_x &\triangleq [x_r \;\; x_1 \;\; x_2]^{\mathrm{T}}, \quad \hat{\theta}_x \triangleq [\hat{b}_1(t) \;\; \hat{k}_1(t) \;\; \hat{b}_2(t) \;\; \hat{k}_2(t)]^{\mathrm{T}}, \\ p_y &\triangleq [y_r \;\; y_3 \;\; y_4]^{\mathrm{T}}, \quad \hat{\theta}_y \triangleq [\hat{b}_3(t) \;\; \hat{k}_3(t) \;\; \hat{b}_4(t) \;\; \hat{k}_4(t)]^{\mathrm{T}} \end{aligned} \tag{8}$$

and the coefficient estimates are updated on-line by adaptation laws,

$$\dot{\hat{\theta}}_x = \dot{\hat{\theta}}_x(p_x, \dot{p}_x), \quad \dot{\hat{\theta}}_y = \dot{\hat{\theta}}_y(p_y, \dot{p}_y). \tag{9}$$

The design of Eqs. (7) and (9), as detailed in Ref. [16], is based on the integrator backstepping technique [14]. The use of this technique is motivated by the structure of the active TPJB dynamics (2)–(5), where the uncertainty in the model is not "matched" [14] with the control input (i.e., the uncertainty and control input do not appear in the same equation). More importantly, the backstepping technique provides the framework for tackling the control problems associated with future extensions of this work, e.g., the use of a more refined (non-linear) model for f_{h_i} in place of Eqs. (4) and (5).

5. Simulation results

A simulation model of the active TPJB system was developed to evaluate the proposed adaptive controller. Specifically, the Reynolds equation was solved for the pressure field, which was then integrated along each pad surface area to give the *actual* hydrodynamic force.[2] The resulting force was used in Eqs. (2) and (3) together with the control inputs given in Ref. [16] to simulate the dynamic behavior of the closed-loop system. The parameters of the TPJB used to develop the simulation model are shown in Table 1. Two simulations were conducted to illustrate the stabilizing effect of the active control strategy when the system is subjected to operating

[2] Note that the force model in Eqs. (4) and (5) was only used for control design purposes.

Table 1
TPJB parameters

Rotor mass (m_r)	3 kg
ith pad mass (m_i)	0.821 kg
ith pad inertia	4.475×10^{-5} kg m^2
Pad arc	60°
Pad radius of curvature (R)	50.27 mm
Rotor radius (r)	49.37 mm
Pad length ($L = 2r$)	98.60 mm
Nominal clearance ($c = R - r$)	0.9 mm
Lubricant viscosity (μ) at 37°C	0.05 Pa · s
Rotor speed (ω)	3000 r.p.m.

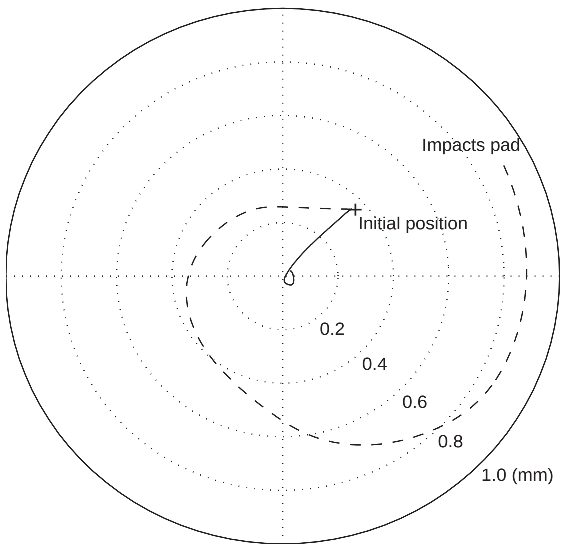

Fig. 2. Simulation 1: orbit of rotor center within clearance circle. (Dashed line: passive case; solid line: active case.)

conditions that yield unstable rotor motions for the conventional (passive) TPJB. The control inputs of the active strategy were limited to ± 150 N in all simulations.

In the first simulation, the initial conditions of the rotor and pads were prescribed as follows:

$$\begin{aligned}
&x_r(0) = y_r(0) = 0.25 \text{ mm}, \quad \dot{x}_r(0) = \dot{y}_r(0) = 0,\\
&x_1(0) = y_3(0) = 50.27 \text{ mm}, \quad x_2(0) = y_4(0) = -50.27 \text{ mm}, \quad \dot{x}_i(0) = 0 \quad (i = 1, 2, 3, 4),\\
&\varphi_1(0) = -0.08^\circ, \quad \varphi_2(0) = \varphi_3(0) = 0^\circ, \quad \varphi_4(0) = -0.31^\circ, \quad \dot{\varphi}_i(0) = 0 \quad (i = 1, 2, 3, 4), \quad (10)
\end{aligned}$$

where $\varphi_i(t)$ denotes the tilt angle of the ith pad.

Fig. 2 compares the orbit of the rotor center during the passive and active TPJB operations. The comparison of the rotor center position versus time is shown in Fig. 3. Notice the diverging rotor orbit during the passive operation, indicating that the initial conditions (10) are possibly outside

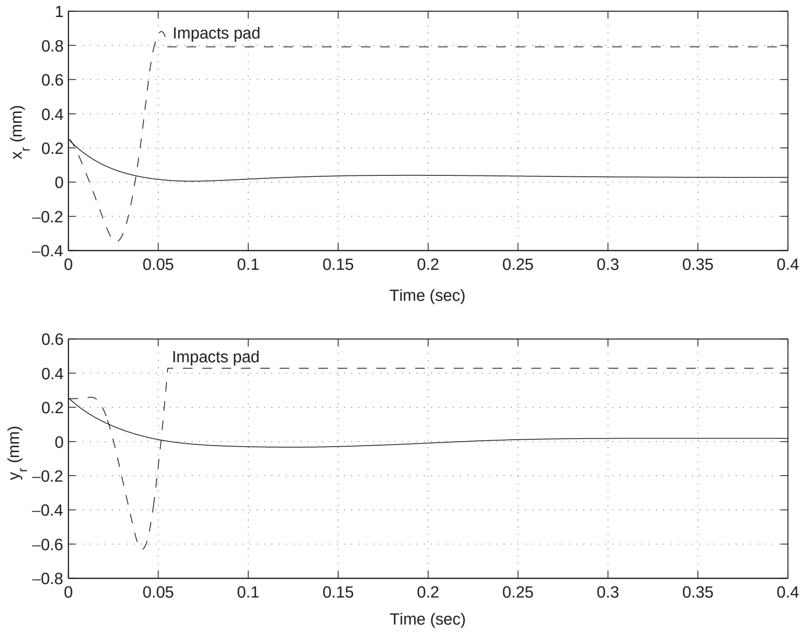

Fig. 3. Simulation 1: position of rotor center $(x_r(t), y_r(t))$. (Dashed line: passive case; solid line: active case.)

the stability region of the system. In fact, the rotor impacts the right pad at $t \approx 0.055$ s, at which point the simulation was stopped. On the other hand, the adaptive controller, through the translation of the pads, was able to stabilize the rotor close to the housing center in approximately 0.25 s.

To further illustrate the utility of the adaptive controller, we considered the TPJB with the following initial conditions:

$$\begin{aligned} &x_r(0) = y_r(0) = 0.6\ \text{mm}, \quad \dot{x}_r(0) = \dot{y}_r(0) = 0, \\ &x_1(0) = y_3(0) = 50.27\ \text{mm}, \quad x_2(0) = y_4(0) = -50.27\ \text{mm}, \quad \dot{x}_i(0) = 0 \quad (i = 1, 2, 3, 4), \\ &\varphi_i(0) = 0.005^\circ, \quad \dot{\varphi}_i(0) = 0 \quad (i = 1, 2, 3, 4). \end{aligned} \tag{11}$$

The passive bearing with the initial conditions (11) runs in a stable mode. Now, consider an impulse-like external disturbance applied to the rotor along the x- and y-directions having the form

$$d_x(t) = d_y(t) = \begin{cases} 950\ \text{N}, & 0.1 \leqslant t \leqslant 0.12\ \text{s}, \\ 0\ \text{N} & \text{otherwise}. \end{cases} \tag{12}$$

The comparison of the rotor orbit and position in time, under the above-described conditions, are shown in Figs. 4 and 5, respectively. Observe how the disturbance causes the rotor orbit to diverge in the passive case until the rotor impacts the top pad at $t \approx 0.105$ s. The adaptive controller, however, is able to overcome the disturbance force and stabilize the rotor close to the housing center in approximately 0.3 s.

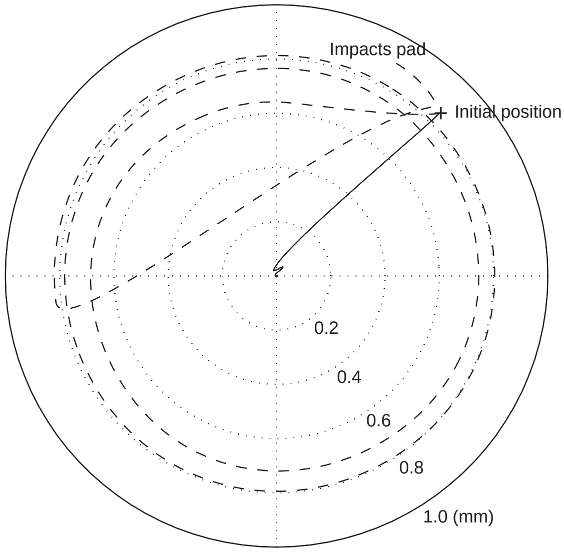

Fig. 4. Simulation 2: orbit of rotor center within clearance circle. (Dashed line: passive case; solid line: active case.)

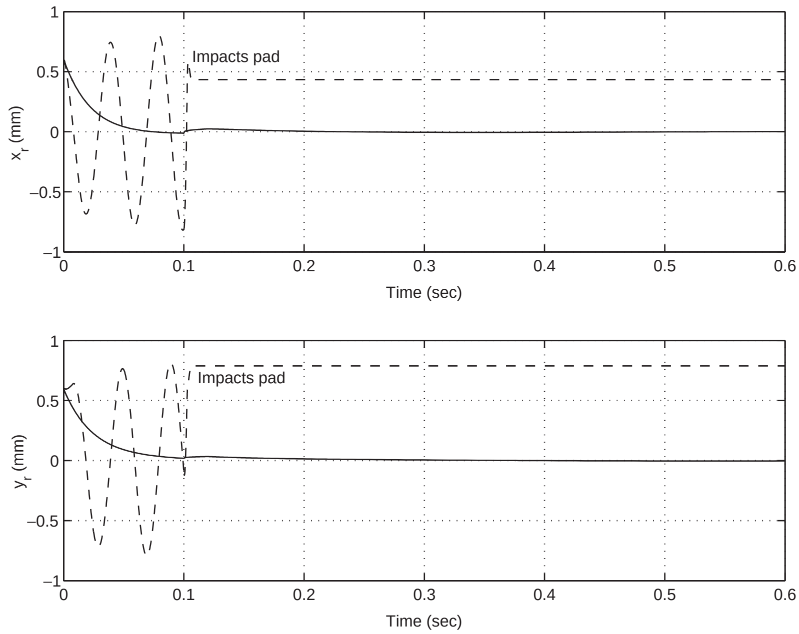

Fig. 5. Simulation 2: position of rotor center $(x_r(t), y_r(t))$. (Dashed line, passive case; solid line, active case.)

6. Conclusion

In this note, we discussed the general concept of active mechanical bearings by drawing an analogy with the principles of operation of active magnetic bearings. This discussion indicates that fluid-film (hydrostatic and hydrodynamic) bearings seem to be the most suitable for active operation by automatically adjusting the pressure field to stabilize the journal. We also outlined the initial effort in developing an advanced feedback control framework for a type of active tilting-pad journal bearing whose dynamic behavior is adjusted on-line by translating the pads. The control framework exploits adaptive non-linear control theory, and is motivated by the desire to expand the stable operating range of the rotor–bearing system beyond what is possible with standard linear control theory. Simulation results illustrated the stabilizing effect of the proposed adaptive controller on unstable operating conditions of the passive rotor–bearing system.

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