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Tilting pad gas bearings for high speed turbomachinery: Design, Fabrication and Validation

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Abstract
A prototype tilting pad gas bearing is proposed for small-scale turbomachinery. It is shown that the bearing is capable to support a 130 gram shaft at 220.000 rpm without dynamic instability and with minimal frictional losses. In addition, experimental imbalance response data is provided which show limited rotor whirl motion for a relative large rotor imbalance. The proposed bearing is therefore useful for turbo applications where rotor tip-clearance must be guaranteed over a large speed range.

Keywords: ultra-high speed, self-acting bearing, gas bearing

1. Introduction

Small-scale turbomachinery can easily exceed a rotational speed of a few hundred thousand rpm. Such a high rotational speed, in combination with high temperature, cannot be obtained with conventional rolling element bearings. In contrast, hydrostatic and hydrodynamic oil-film bearings can resist these extreme conditions, but at the cost of high frictional losses. These frictional losses reduce the device efficiency in many applications to an unacceptable level, such that gas bearings or active magnetic bearings are the only remaining solutions. The latter is unfortunately more complex and expensive while the former is often fraught with aerodynamic instability problems [1].

However, aerodynamic instability can be greatly reduced by implementing gas bearings of the tilting pad type with geometrical pivot offset [2]. Yet, this superior bearing technology is not general used in small-scale turbomachinery; mainly because they are not available as off-the-shelf components and due to the scarce knowledge on how to design and fabricate them for small journal sizes.

This paper will therefore review the design, fabrication, and experimental validation of a 16 mm ID prototype tilting pad bearing. The purpose of this bearing is to achieve a speed of 200.000 rpm in an experimental setup. A second objective is to limit the rotor whirl motions within 10 µm, even for an extreme imbalanced shaft.

2. Dynamic stable tilting pad bearing

A typical tilting pad bearing is made up of 3 or more pads. Each of these pads can tilt around an individual pivot point which is either rigidly or flexibly mounted to the outer bearing casing. In addition, the pivot points are typical located at a certain radial or tangential offset from its geometrical concentric position, as designated with ε and η in Figure 1 respectively. This pivot offset will result in a wedge-shaped gap between pad and shaft, even before any aerodynamic forces are acting on the pad. Such geometrical offset, often referred to as pad pre-load, will significantly improve the bearing load capacity and its dynamic stability. It is therefore the preferred method to stabilize large-scale conventional hydrodynamic oil-film bearings. However, implementing pivot offset for small-scale gas-bearings is more difficult due their small size and tighter tolerances.

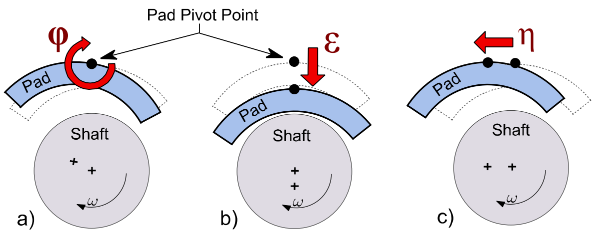

Figure 1. Pad pivot point offset directions. {ϕ,ε and η}

2.1 Monolithic design

In this paper, a monolithic design is therefore proposed to realize precisely the required geometrical pivot offset within a limited volume. This design, as shown in Figure 2, has flexible structures to provide both a tilt and radial degree of freedom to the pads. In addition, tree separate set-screws are incorporated into the design to displace each pivot point towards the bearing center by 30 µm, thereby creating the required geometrical pivot offset.

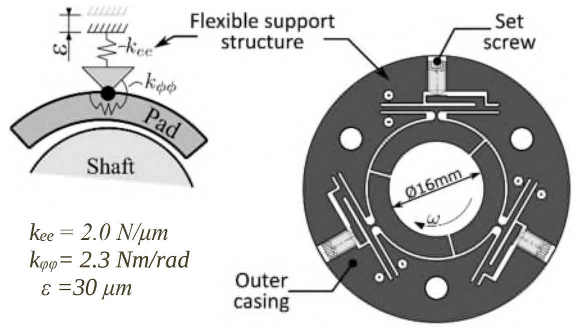

Figure 2. Theoretical and actual 2D design

Fabrication of the monolithic bearing is achieved by a combination of micro-milling, diamond turning and wire-EDM machining [3]. That is, an accurate circular center bore is made by precision diamond turning; giving the bearing inner surface both a superior form and roughness quality. Wire-EDM, in a sequential step, forms the flexible support structure and separates the individual pads. In the assembly stage, a fully operational bearing cartridge is formed by joining two of these bearings with the use of a spacer ring, as shown in Figure 3.

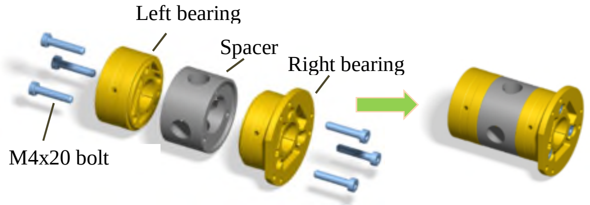

Figure 3. Exploded view; bearing cartridge consisting of two bearings

3. Experimental validation

Performance of the assembled bearing cartridge is evaluated with the setup shown in Figure 4. As depicted in this figure, a 130 gram dummy rotor with two air impulse turbines on each side is placed inside the cartridge. Compressed air on these turbines drives the rotor to high rotational speeds, while an optical tacho sensor and multiple proximity probes measure the shaft rotational speed and radial position respectively.

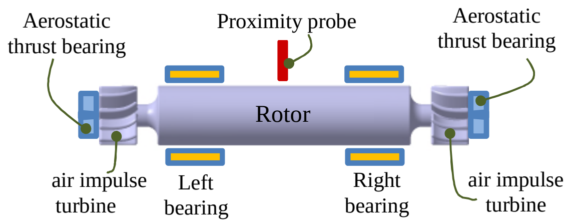

Figure 4. Schematic representation of the bearing setup

During initial tests, rotational speeds of up to 220.000 rpm could easily be reached without observing any dynamic rotor instability, as shown in Figure 5. The bearing could theoretically sustain rotational speeds above 220.000 rpm (equal to 3.520.000 DN), but this is not experimentally validated due to the sincere risk of rotor disintegration by centrifugal forces [3].

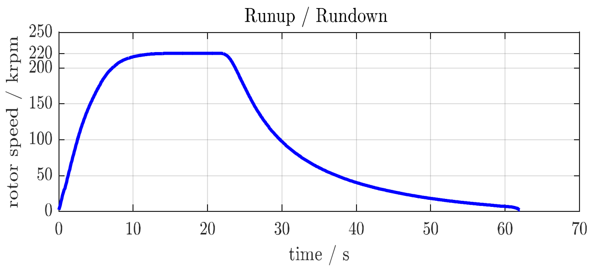

Figure 5. Measured rpm. Compressed air is applied to the setup up-till t=22 seconds. After that, air is cutoff and the rundown phase begins.

Since no instability occurred, and to validate the dynamic stiffness, load on the bearings is increased by attaching 310 mg·mm imbalance masses to each end of the rotor.

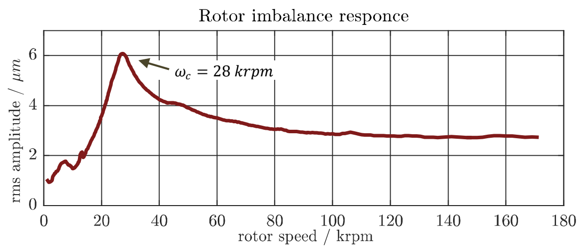

Figure 6. Measured imbalance response for 310 mg·mm per bearing

Under this fairly extreme imbalance condition, the shaft whirl radius stayed within a reasonable 6.5 µm, as shown in Figure 6.

3.1 Bearing power loss

An attempt has been made to validate the theoretically computed bearing losses with experimental measurements. Unfortunately, these losses could not be measured directly with the available setup, since it has only sensors for the speed and position of the shaft. The shaft deceleration, during the rundown phase, can however be used to estimate the bearing power loss. That is, when compressed air to the impulse turbines is cutoff, the shaft starts to deaccelerate immediately as shown in Figure 5. The deceleration is caused by the power loss of the two bearings and the drag torque of the two free spinning turbines at each end of the shaft. The combined drag torque of the two turbines depends on the rotational speed ω and is experimental determined as $T_t = 1.84e^{-11}\omega^2 + 4.40e^{-8}\omega$. Substitute T_t and the experimental obtained relation $\dot{\omega}(\omega)$ into equation (1) gives an estimate of the power loss W_b of a single bearing, i.e.:

$$W_b(\omega) = ½\omega\{I_s\dot{\omega}(\omega) - T_t(\omega)\} \quad (1)$$

where I_s is the inertia of the rotor, which is $3.7e^{-6}$ *kgm²* for the used shaft in the setup. Based on the speed-curve of Figure 5, the experimental determined $W_b(\omega)$ for the prototype bearing is showed beside the theoretical computed curve in Figure 7.

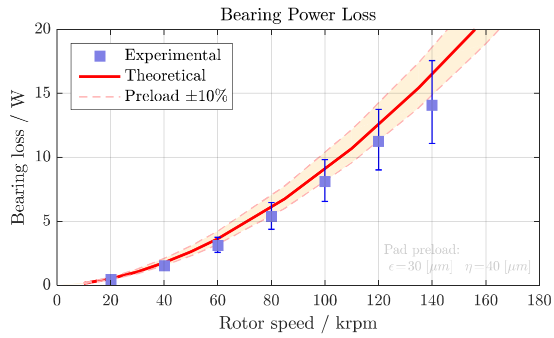

Figure 7. Bearing power loss as function of rotational speed

As can be seen, the experimental and theoretical computed bearing losses are in reasonable agreement. The small deviation can be explained by a possible over estimation of the theoretical values due to an overestimation of the bearing actual pivot point offset ε. Changing this value by only ±10% (or ±3 µm) will results in bearing loss according to the two dashed lines around the solid line. Also the uncertainty on the experimental obtained values can be a reason for the small nonconformity.

4. Conclusion

A prototype tilting pad gas bearings is proposed for high speed turbo machinery. The prototype bearing utilize set-screws to precisely set the radial pad pivot point offset to 30 um after circular turning of the center bore. Practical experiments confirm the stable operation of the bearing up to 220.000 rpm, even with an extreme rotor unbalance of 310 mg·mm. Whirl radius stayed in all cases below 6.5 um, which is more than adequate to prevent mechanical contact between shaft and bearing surface. Bearing power loss depends on the rotational speed ω and is in the order of 10 W at 100 krpm, which is significant lower than comparable oil-film bearings.

References

[1] Peirs J, Waumans T, Vleugels P, Al-Bender F, ..., Reynaerts D 2007 Micropower generation with microgasturbines: a challenge *J. of Me-chanical Engineering Science* **221** 489-500

[2] Czolczynski 1999 *Rotordynamics of gas-lubricated journal bearing systems*

[3] Nabuurs M J H W, Al-Bender F, Reynaerts D, Mayer M and Vogt A 2018 Fabrication and validation of a flexible tilting pad gas bearing for high speed turbo machinery *21st International Colloquium Tribology, Germany*