DESIGN, OPTIMISATION AND TESTING OF A HIGH-SPEED AERODYNAMIC JOURNAL BEARING WITH A FLEXIBLE, DAMPED SUPPORT

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Abstract: This paper reports on the design and experimental validation of a self-acting journal bearing that is stabilised by means of a flexible and damped support structure. Design guidelines are formulated to determine the optimal support parameters. Test results show successful operation up to a rotational speed of nearly 700 000 rpm without observing excessive rotor whirl.

Keywords: air bearing, high speed, stability

INTRODUCTION

The bottleneck for the successful application of high-speed turbomachinery is predominantly imposed by limitations in currently available high-speed bearing technology. The combined requirement of high rotational speed and elevated temperatures, stress the need for innovative bearing solutions. Gas bearings are able to meet these stringent requirements if the stability issue is tackled.

STABILITY PROBLEM OF HIGH-SPEED GAS BEARINGS

As is generally known, gas bearings are prone to a self-excited whirl instability when operated at high speed. Their successful application requires therefore a sound understanding of this phenomenon to identify the relevant parameters and to propose remedies that postpone the onset of self-excited whirling.

A linear stability analysis reveals the cross-coupled stiffness $k_{ij}$ as the destabilising factor. A necessary stability condition, for a Jeffcott rotor-bearing system, may be formulated as [1]:

$$k_{ij} \leq \frac{k_{ii}}{m} c_{ii},$$

where $m$ is the rotor mass, while $k_{ii}$ and $c_{ii}$ represent respectively the direct gas film stiffness and damping coefficients.

This simple stability criterium allows us to derive three basic strategies for improving the stability: (i) increase of direct stiffness $k_{ii}$, e.g. by an optimal choice of the restrictor geometry in case of aerostatic bearings; (ii) reduction or elimination of the aerodynamically induced cross-coupling $k_{ij}$ through a stability-optimised film geometry such as found in e.g. tilting-pad bearings; and (iii) increase in direct damping $c_{ii}$. The latter strategy can only be effected by reverting to very small values of the radial clearance (in the order of a few micrometers for miniature gas bearings). This leads to various practical problems: increased viscous frictional losses, alignment issues and the inability to cope with centrifugal or thermal rotor growth.

A variant of this last strategy consists in the introduction of damping to the rotor-bearing system outside of the gas film. This approach of adding external damping to the system, compensates then for the destabilising effects within the gas film itself.

INTRODUCING EXTERNAL DAMPING

Literature overview

Implementations for introducing external damping to a rotor-bearing system exist in various forms. The most widespread implementation is found in oil-based squeeze-film dampers as a support for rolling element bearings in aircraft gas turbine engines [2]. In the field of gas bearings, a common implementation consist of an aerostatic bearing bush supported by an elastomeric material such as rubber O-rings [3, 4].

Within the context of this paper, the work performed by [5] is worth mentioning. He was able to attain, with a spiral grooved journal bearing on rubber O-rings, a rotational speed of 509 000 rpm for a diameter 6 mm rotor of 2.35 g (= 3 054 000 DN).

Although fairly easy to realise, this implementation has some disadvantages. In order to improve the stability, a correct combination of support stiffness and damping is required, as will be shown later on. The achievement of this optimal combination seems not always possible with elastomeric materials. Another difficulty lies in the characterisation of the complex dynamic behaviour of the support material and its dependence on the temperature.
Proposed implementation

In the here proposed concept, the bearing bush is supported by rubber O-rings which provide in the support stiffness and partly account for the external damping (Fig. 1). They however also serve as seal for an oil filled squeeze-film cavity. The amount of damping that is hereby introduced to the system, can be tuned by varying the oil viscosity. The support stiffness may be adjusted by changing the O-ring preload. This (partly) separation of stiffness and damping contribution of the support is a first novelty of the concept.

Fig. 1: Schematic view of the proposed implementation with O-rings.

Rotordynamic model

Fig. 2 outlines the dynamic model used to investigate the effect of the support parameters on the stability. The gas film is represented by eight dynamic coefficients which depend on the steady-state working conditions (eccentricity and rotor speed $\omega$) and on the perturbation frequency $\nu$. The support model is limited to a constant support stiffness $k_e$ and damping coefficient $c_e$, while the bush mass is denoted by $m_b$.

In order to formulate a set of dimensionless design guidelines, the support parameters are related to the gas film properties in the following way:

$$M_b = \frac{m_b}{m}, \quad K_e = \frac{k_e}{k_{ii}}, \quad C_e = \frac{c_e}{c_{ii}}.$$

Since the optimal value of the support parameters will depend on the ‘quality’ of the bearing itself, its properties in terms of stability are represented by:

$$\omega_n = \sqrt{\frac{k_{ii}}{m}}, \quad \zeta_n = \frac{c_{ii}}{2m\omega_n}, \quad \kappa = \frac{k_g}{k_{ii}}.$$

Optimal support parameters

As stated by Lund [6], the improvement of the stability may only be effected by a proper choice of the support parameters. This is also evidenced by the set of graphs shown in Fig. 3. These graphs visualise the damping ratio of the least stable mode for different combinations of $K_e$ and $C_e$. The influence of the bush mass $M_b$, cross-coupling ratio $\kappa$ and gas film damping $\zeta_n$ on the optimal value of the support parameters is illustrated by respectively the top, middle and bottom row of graphs.

This study enables us to formulate qualitative design rules which should be followed to make the external damping effective: (i) try to keep the bush mass a factor ten smaller than the rotor mass, i.e. $M_b < 0.1$; (ii) the cross-coupled stiffness of the gas film may not exceed the direct stiffness, i.e. $\kappa < 1$; and (iii) provide for an amount of external stiffness that is certainly smaller than the film stiffness, i.e. $K_e < 0.5$.

Of course, each of these design rules must be regarded as a rule-of-thumb rather than as a guarantee for stability. The design process should therefore be backed by a more extensive stability evaluation for various operation conditions.

EXPERIMENTAL VALIDATION

Overview

The test setup for validating the stability of various miniature high-speed bearings is shown in Fig. 4. It consists of a cylindrical rotor (dia. 6 x 30 mm and mass $m = 6.67$ g) supported by two identical journal bearings and two aerostatic thrust bearings. The rotor is driven to the required speed by an impulse turbine.
Fig. 3: Damping ratio of least stable mode for different values of the dimensionless support parameters $K_e$ and $C_e$ (shaded gray areas reflect unstable behaviour).

Instrumentation consists of two fiber optical displacement probes located at either end of the rotor, and an optical fiber embedded into one of the thrust bearings for recording the speed.

Aerodynamic bearing geometry

Fig. 5 depicts the aerodynamic film geometry of the test bushes. It combines favourable intrinsic stability properties with ease of manufacturing at a miniature scale. The wave-shaped geometry is obtained by a controlled elastic deformation when machining the bearing surface on a precision lathe (Fig. 6), after which the diverging sections are removed by wire-EDM.

The geometrical design parameters of the test geometry are as follows: radius $r = 3$ mm, length $L = 6$ mm, radial clearance $c = 10 \mu$m, wave amplitude $a_{\text{wave}} = 5 \mu$m, section angle $\theta_s = 120^\circ$ and groove angle $\theta_g = 36^\circ$. The bushes are manufactured out of bronze and have a mass $m_b = 0.74$ g.

Fig. 5: Wave-shaped film geometry of the test bush.

Test results

The optimal support parameters were determined from a series of experiments using different combinations of O-ring preload and oil viscosity. The
threshold speed observed in these experiments is given by Table 1. It is clear from these results that for a given value of the support stiffness, there exists an optimal amount of external damping, and that the support stiffness may not exceed the gas film stiffness ($k_0 = 0.30$ N/µm at $\omega = 5000$ Hz (= 300 000 rpm)).

**Table 1:** Threshold speed of self-excited whirling for different combinations of support stiffness and damping.

<table>
<thead>
<tr>
<th>support stiffness $k_e$ [N/µm]</th>
<th>oil viscosity [cSt]</th>
<th>threshold speed [Hz]</th>
</tr>
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<tbody>
<tr>
<td>0.04</td>
<td>no oil</td>
<td>5250</td>
</tr>
<tr>
<td></td>
<td>10</td>
<td>5450</td>
</tr>
<tr>
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<td>22</td>
<td>5160</td>
</tr>
<tr>
<td>0.14</td>
<td>no oil</td>
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<tr>
<td></td>
<td>1</td>
<td>2272</td>
</tr>
<tr>
<td></td>
<td>5.3</td>
<td>2420</td>
</tr>
<tr>
<td></td>
<td>22</td>
<td><strong>4418</strong></td>
</tr>
<tr>
<td></td>
<td>32</td>
<td>2826</td>
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<tr>
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<td>no oil</td>
<td>2300</td>
</tr>
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<tr>
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<tr>
<td></td>
<td>68</td>
<td>2500</td>
</tr>
<tr>
<td></td>
<td>150</td>
<td>2230</td>
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</table>

For the combination $k_e = 0.04$ N/µm and an oil viscosity of 22 cSt, a runup experiment has been performed up to 683 280 rpm. The waterfall diagram of Fig. 7 reveals the occurrence of random whirl for speeds above 360 000 rpm. This type of self-excited whirl does not appear at a fixed subsynchronous frequency. Its amplitude remains however low (< 2 µm) and fairly constant up to the maximum attainable speed. The rotor speed could not be increased further due to the limited driving power of the turbine.

**CONCLUSION**

This paper has discussed the design aspect and experimental validation of a flexibly supported aerodynamic journal bearing. Design rules are formulated for the optimal choice of the external support parameters. Test results show successful operation up to 683 280 rpm (= 4 100 000 DN), which represents to our best knowledge and with the exception of foil bearing technology, the highest achieved DN-number for a self-acting bearing operated in air.

Further work should concentrate on the nonlinear study of the random whirl phenomenon and on implementations that are compatible with a high-temperature environment (capillary seal, integral wire mesh damper etc.).

**REFERENCES**


