AERODYNAMIC JOURNAL BEARING WITH A FLEXIBLE, DAMPED SUPPORT OPERATING AT 7.2 MILLION DN

Tobias Waumans1*, J. Peirs1, F. Al-Bender1, D. Reynaerts1
1Departement of Mechanical Engineering, Katholieke Universiteit Leuven, Leuven, Belgium
*Presenting Author: tobias.waumans@mech.kuleuven.be

Abstract: As the bottleneck for the successful application of ultra-miniature gasturbines is predominantly imposed by limitations in currently available high-speed bearing technology, new bearing concepts have to be developed. This paper reports on the achievement of a record speed of 1 203 000 rpm (= 7.2 million DN) on aerodynamic journal bearings. The concept and implementation of the flexible, damped bearing support are outlined and its effectiveness for the stabilisation of high-speed gas bearings is demonstrated experimentally. The following related topics are briefly studied: (i) characterisation of different elastomeric support materials; (ii) discussion of observed measurement artefacts at high speed; and (iii) an analysis of the frictional loss sources.

Keywords: air bearings, high speed, stability

INTRODUCTION

The bottleneck for the successful application of ultra-micro gasturbines is predominantly imposed by limitations in currently available high-speed bearing technology. Gas bearings are able to meet the stringent requirements imposed by this application if the stability issue is tackled.

In this paper, a new bearing concept is described which uses a flexible, damped support to enhance the bearing stability. With this aerodynamic bearing, a record speed of 7.2 million DN1 has been achieved. The following sections will first address the implementation details of this stabilisation technique, after which the experiment leading to the speed record is described. Then, the following topics are addressed: (i) characterisation of different elastomeric support materials; (ii) discussion of observed measurement artefacts at high speed; and (iii) an analysis of the frictional loss sources.

STABILISATION WITH A FLEXIBLE, DAMPED SUPPORT

Introducing external damping

As is generally known, any gas bearing is prone to a self-excited whirl instability which is caused by cross-coupling effects in the gas film. The attainment of stable, high-speed operation is only possible by taking measures either within the gas film itself, or by the introduction of damping from outside of the gas film, i.e. external damping. An optimal design of the film height profile or the incorporation of pumping grooves are examples of the former stabilisation technique. However, it seems that the introduction of external damping is the only fundamental solution to the stability problem since the sufficient damping is only obtained within the gas film by reverting to very small values of the nominal radial clearance and, more importantly, since the damping capacity of any gaseous film is practically zero at high frequencies.

Description of the proposed implementation

The proposed bearing concept to introduce external damping consists of a non-rotating bearing bush which is flexibly supported with respect to the bearing housing. The bearing bush has a self-acting wave-shaped height profile (diameter 6 mm, slenderness ratio 1) on its inner side which combines load-generating capability with favourable intrinsic stability properties [1]. On the outside, 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 or elastomeric material.

1 The DN-number is used here as a measure for the achieved rotational speed. This number is defined as the product of the rotational speed in rpm with the bearing diameter in mm.
SPEED RECORD: 7.2 MILLION DN ON AERODYNAMIC BEARINGS

Initial tests were performed to determine the optimal support parameters [1]. Thereafter, successful operation was demonstrated up to a rotational speed of 683 280 rpm (= 4.1 million DN). Not the manifestation of self-excited instabilities, but the limited driving power of the impulse turbine prevented reaching even higher speeds.

In more recent experiments on the same bearing configuration, helium has been used to drive the turbine. Due to the higher sonic outlet velocity (ca. 1000 m/s), the maximal attainable rotational speed could be increased to 1 203 000 rpm (= 7.2 million DN). This achievement represents, to our knowledge, the speed record in terms of DN-number for an air bearing of the self-acting type (Table 1). The waterfall diagram recorded during this experiment is shown in Fig. 2.

CHARACTERISATION OF DIFFERENT SUPPORT MATERIALS

Since the support parameters should be chosen in accordance to the gas film properties [2], it might be useful to determine the characteristics of different elastomeric O-ring materials. The quasi-static characterisation is performed by applying a controlled load to the test bearing while measuring the resulting deflection (Fig. 3).

The O-rings under investigation had the following properties: 6.8 mm x 0.50 mm, nitrile rubber - 70 shore and silicone rubber - 25 shore (from Apple Rubber). Figure 4 compares the characteristics of both elastomeric materials under different preload conditions. It is clear that the nitrile rubber O-rings under high preload (green dash-dotted curve) yield a support stiffness that is too large in comparison to the gas film stiffness (ca. 0.2 N/µm versus respectively 0.3 N/µm at 300 000 rpm). This was confirmed by the fact that no stable behaviour could be observed with this combination. The nitrile rubber O-rings under low preload (red dashed curve) and the silicon rubber O-rings (blue solid curve) provide a support with suitable stiffness properties (ca. 0.01 N/µm). Furthermore, a relatively large amount of hysteresis is found to exist in both latter support combinations.

Table 1: Overview of high-speed bearing achievements.

<table>
<thead>
<tr>
<th>author</th>
<th>affiliation</th>
<th>year</th>
<th>dia. [mm]</th>
<th>rotational speed [rpm]</th>
<th>DN-number</th>
<th>type</th>
</tr>
</thead>
<tbody>
<tr>
<td>H. Signer</td>
<td>NASA Lewis Research Center</td>
<td>1973</td>
<td>120</td>
<td>25 000</td>
<td>3 000 000</td>
<td>ball bearings</td>
</tr>
<tr>
<td>C. Zwyssig</td>
<td>ETH Zürich</td>
<td>2008</td>
<td>3.17</td>
<td>1 000 000</td>
<td>3 175 000</td>
<td>ball bearings</td>
</tr>
<tr>
<td>S. Tanaka</td>
<td>Tohoku University</td>
<td>2003</td>
<td>4</td>
<td>1 250 000</td>
<td>5 000 000</td>
<td>hydroinertia</td>
</tr>
<tr>
<td>S. Tanaka</td>
<td>Tohoku University</td>
<td>2009</td>
<td>8</td>
<td>642 000</td>
<td>5 136 000</td>
<td>foil bearing</td>
</tr>
<tr>
<td>A. Epstein</td>
<td>MIT</td>
<td>2006</td>
<td>4.2</td>
<td>1 700 000</td>
<td>7 140 000</td>
<td>aerostatic</td>
</tr>
<tr>
<td>T. Waumans</td>
<td>K.U.Leuven</td>
<td>2010</td>
<td>6</td>
<td>1 203 000</td>
<td>7 218 000</td>
<td>aerodynamic</td>
</tr>
<tr>
<td>J. W. Beams</td>
<td>University of Virginia</td>
<td>1937</td>
<td>9</td>
<td>1 300 000</td>
<td>11 700 000</td>
<td>aerostatic</td>
</tr>
<tr>
<td>J. W. Beams</td>
<td>University of Virginia</td>
<td>1946</td>
<td>0.521</td>
<td>37 980 000</td>
<td>20 130 000</td>
<td>magnetic</td>
</tr>
</tbody>
</table>

Fig. 2: Waterfall diagram recorded during the speed record experiment up to 1 203 000 rpm. The synchronous whirl response is indicated by ‘1x’.

Fig. 3: Quasi-static characterisation of the elastomeric support material.

Fig. 4: Comparison of elastomeric O-ring material characteristics.
**INSTRUMENTATION ARTEFACTS**

The waterfall diagram of Fig. 2 shows nonsynchronous whirling above rotational speeds of ca. 300 000 rpm. Initially, this particular whirling phenomenon was attributed to the nonlinear behaviour of the rotor-bearing configuration as observed in for instance limit-cycle orbits. A somewhat similar behaviour was also reported in measurements performed by [3].

To conclude on whether this so-called ‘random whirl’ is not a measurement artefact, a runup experiment is performed during which the whirling behaviour is recorded simultaneously by the in-house developed fiber optical measurement system, and by a commercial laser vibrometer (Polytec OFV 2200).

The outcome of this experiment is shown in Fig. 5. In contrast to the waterfall diagram as recorded by the fiber optical system, the one obtained by the laser vibrometer is free from any ‘random’ whirling. This confirms the particular nonsynchronous whirling as being a measurement artefact. Apart from this conclusion, the synchronous amplitude recorded by the laser vibrometer seems to be somewhat larger (0.5 to 1 µm difference) at high values of the rotational speed.

The exact explanation of this artefact is not entirely clear. But, it has been found that the problem becomes more prominent when the optical measurement surface is of poor quality. Since the fiber optical measurement system is based on the amount of light that is reflected back into the fiber, any surface irregularity in the form of fingerprints, scratches or indentations will induce problems.

**ANALYSIS OF BEARING AND WINDAGE LOSSES**

A deceleration experiment has been carried out to give an estimation of the frictional losses. To this end, the rotor is accelerated to nearly 700 000 rpm after which the supply to the driving turbine is closed instantly. The speed-versus-time data is recorded and analysed accordingly to obtain the frictional power loss and friction torque (Fig. 6).

Apart from the losses in the journal bearings, the following dissipative sources can be identified in the system: (i) windage losses originating from a smooth cylindrical shaft which is rotating in its housing; (ii) viscous frictional losses in the thrust bearings; and (iii) losses caused by the turbine driving system.

Below, these different loss sources are discussed and quantified.
Journal bearing losses

The frictional losses in the bearings can be split up into a viscous and a hydrodynamic loss contribution. Both sources are included in the gas film model which assumes laminar viscous flow in the bearing gap. The frictional losses versus speed of a single journal bearing are plotted in Fig. 7 (green dash-dotted curve). The centrifugal rotor growth is hereby taken into account (0.85 µm radial growth at 700 000 rpm).

Windage losses

The flow is however no longer laminar when considering the losses induced at the non-bearing areas of the rotor. To estimate these so-called windage losses, the following relationship based on turbulent flow between to parallel plates, is used [4]:

\[ P_{hub} = \pi C_d \rho R^4 \omega^3 L, \]  
(1)

with \( \rho \) the density of the gas, \( R \) the radius of the rotor, \( L \) its length and \( \omega \) the rotational speed. \( C_d \) is the so-called skin friction coefficient which is obtained from the following empirical equation:

\[ \frac{1}{\sqrt{C_d}} = 2.04 + 1.768 \ln(\text{Re} \sqrt{C_d}), \]  
(2)

with the Reynolds number \( \text{Re} \) defined as:

\[ \text{Re} = \frac{R \omega}{\nu}, \]  
(3)

in which \( c \) equals the radial gap and \( \nu \) the kinematic viscosity of the gas.

Equation (1) has been evaluated for increasing values of the rotational speed \( \omega \). The length \( L \) is hereby set at 9 mm (half rotor) while the average radial clearance value \( c \) over this length is chosen as 3 mm. The result is shown in Fig. 7 (blue solid curve).

Other loss sources: thrust bearings and turbine

The frictional losses attributed to the thrust bearings is rather limited (0.4 W per bearing for a 30 µm gap at 700 000 rpm). The losses originating from the Pelton turbine, on the other hand, will be more significant. A reasonable estimation of this loss source proves to be far from straightforward.

Figure 7 summarises the behaviour of the different frictional loss sources with respect to the rotational speed and their relative contribution to the total frictional loss (black solid curve). It is seen that the radial bearing loss comprises only one third of the overall loss. The unmodelled losses in the turbine (red dashed curve) prove to be as significant as the actual bearing losses.

CONCLUSION

This paper has reported on the achievement of a record speed of 7.2 million DN on aerodynamic journal bearings. The concept and implementation of the flexible, damped bearing support are outlined and its effectiveness for the stabilisation of high-speed gas bearings is demonstrated experimentally. The following related topics have been investigated: (i) characterisation of different elastomeric support materials; (ii) discussion of observed measurement artefacts at high speed; and (iii) an analysis of the frictional loss sources.

REFERENCES