ON THE DYNAMIC STABILITY OF HIGH-SPEED GAS BEARINGS: STABILITY STUDY AND EXPERIMENTAL VALIDATION

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Abstract For high-speed applications, gas lubricated bearings offer very specific advantages over other, more conventional bearing technologies: a clean and oil-free solution, virtually wear-free operation, low frictional losses, wide operating temperature range, etc. However, the principal drawback involved in the application of high-speed gas bearings concerns the dynamic stability problem. Successful application therefore requires control of the rotor-bearing dynamics so as to avoid instabilities.

After a detailed study of the dynamic stability problem and the formulation of a convenient stability criterium, a brief overview is given of the currently existing bearing types and configurations for improving the stability. In addition, three strategies are introduced: (i) optimal design of plain aerostatic bearings; (ii) modification of the bearing geometry to counteract the destabilising effects in the gas film; and (iii) introduction of damping external to the gas film as to compensate for the destabilising effects.

These strategies are worked out into detail leading to the formulation of a series of design rules. Their effectiveness is validated experimentally at a miniature scale. In recent experiments a rotational speed of 1.2 million rpm has been achieved with a 6 mm rotor on aerodynamic journal bearings, leading to a record DN-number of 7.2 million.

Keywords air bearing, high speed, stability

1 INTRODUCTION

High-speed bearings constitute a key component of an increasing number of applications. In rotating equipment such as machine tool spindles, various kinds of turbomachines and dental drills, a clear trend has become evident towards an increase in rotational speed and towards a continual downscaling. Gas lubricated bearings offer, for these purposes, very specific advantages over other, more conventional bearing technologies. They provide a clean, oil-free bearing solution characterised by virtually wear-free operation, low frictional losses and a wide operating temperature range.

These advantages come forth from the fact that both bearing members are completely separated by a thin gas film, typically ranging from a few micrometers up to 50 µm in height. Through the generation of a positive pressure distribution in between both surfaces, the bearing is able to carry a load. This pressure generation can be the result of an external supply of pressurised air (aerostatic bearing), or of a combination of relative shearing motion and a converging gap height profile (self-acting or aerodynamic bearing).

Disadvantages, of course, also exist. The low specific load-carrying capacity and limited damping are the result of low viscosity of gasses and of their compressible nature. For high-speed applications, however, the principal drawback concerns the dynamic stability problem. Successful application therefore requires control of the rotor-bearing dynamics so as to avoid instabilities in the envisaged operational speed range.

This paper treats the dynamic stability problem of high-speed gas bearings. First, a study is performed to determine the underlying mechanism and to formulate a convenient stability criterium. After a brief overview of currently existing measures to improve the stability, the paper describes three additional strategies. Each of these strategies is experimentally validated at a miniature scale. Figure 1 illustrates how the stability performance of a conventional aerostatic bearing is improved by a modification of the film geometry (a-b). The introduction of external damping, finally, makes it possible to operate a self-acting bearing at a rotational speed of 1.2 million rpm (c). This translates to a DN-number[1] of 7.2 million DN, which represents, to our knowledge, a record for an air bearing of the self-acting type.

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[1] The DN-number is defined as the product of the diameter in mm and the rotational speed in rpm, and is a measure for the circumferential speed at the bearing surface. Although not dimensionless, it can be regarded as a size-independent performance indicator.

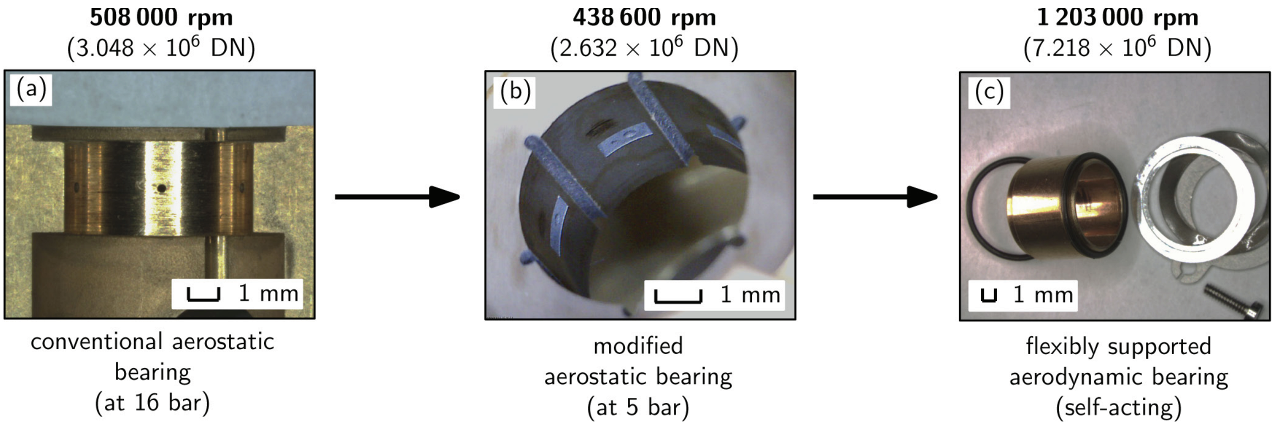

Figure 1: Illustration of the outcome of the three stabilising strategies: (a) conventional aerostatic bearings at high supply pressures; (b) aerostatic bearing with a modified film geometry; and (c) stabilisation by the introduction of 'external' damping.

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

Table 1: Overview of high-speed bearing achievements (in order of increasing DN-number).

2 THE DYNAMIC STABILITY PROBLEM

In general, two types of dynamic instability can be encountered when dealing with gas lubricated rotor-bearing systems: pneumatic hammering and subsynchronous rotor whirl. Both are self-excited instabilities, but with a different underlying mechanism.

For an externally pressurised gas bearing system, the possible occurrence of pneumatic hammering exists due to a loss of film damping. This loss of damping is caused by a time-lag effect due to the compressible nature of gasses. In literature, this first type of instability has already been studied extensively ([2],[3],[4],[5]) and it is therefore not treated in this paper.

A second type of instability, which is of greater importance here, is generally referred to as half-speed (or more generally sub- or non-synchronous) whirling. The underlying source of this type of instability involves rotation which causes a cross-coupling effect in the gas film. This can lead to sudden sub-synchronous shaft whirling being very destructive in nature.

Before discussing the nature and underlying mechanism of self-excited whirl in high-speed gas bearings, a clear distinction has to be made between the various types of whirling encountered in rotating machinery.

2.1 Synchronous versus self-excited whirl

When dealing with rotating machinery on gas lubricated bearings, the gas film supporting the rotating shaft is rarely in a steady-state condition. Shaft whirling can always be observed. This whirling, however, does not automatically preclude the stability of the non steady-state working condition. Therefore, it is important to distinguish between different types of whirling and their implications on stability.

Whirling that occurs *synchronous* with the rotational speeds is, in fact, the passive response of the rotor-bearing system to excitation induced by residual imbalance. Depending on the presence of respectively static or dynamic rotor imbalance, cylindrical or conical whirling will be observed (although a combination of both is more likely in practice). As shown in Figure 2, the synchronous response features a maximum at the resonance frequency of the rotor-bearing system. This operating point is referred to as a critical speed in rotordynamic terms. Once above a critical speed, the synchronous response amplitude reduces gradually before reaching a constant value which reflects the mismatch between geometrical axis of rotation and the mass line of the rotor. This shift in axis of rotation is referred to as inversion [9]. Depending on the balancing conditions and the amount of damping present in the system, safe passage can be assured through the encountered critical speeds. When compared to rolling element bearings or hydrodynamic/static bearings of similar size, gas bearings usually possess lower stiffness values. This often makes that for a high-speed application, the design speed lies in the supercritical operating range.

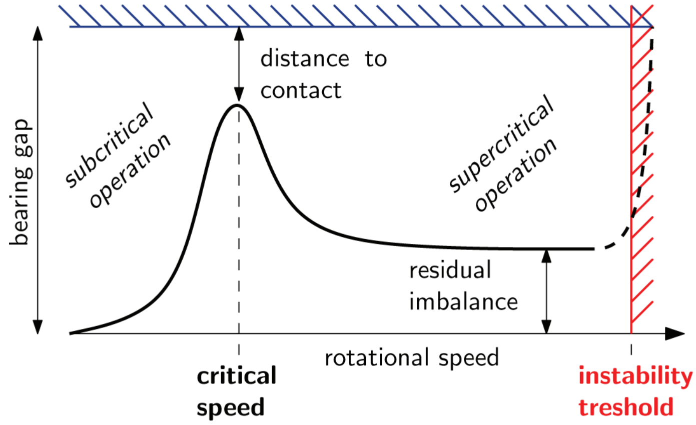

Figure 2: Typical (and qualitative) behaviour of the synchronous bearing response for a system with a supercritical operational speed range.

Once the system is operating at a supercritical speed, a further increase of the rotational speed will inevitably lead to the occurrence of sudden and destructive whirling. In contract to synchronous whirl, this dynamic instability is of a self-excited nature and manifests itself at a *nonsynchronous* (and mostly *subsynchronous*) frequency. The operating point at which self-excited whirl sets in, is referred to as the threshold or onset speed of the rotor-bearing system (see Figure 2). Due to the sudden increase in amplitude with respect to speed, this most often corresponds to the maximal attainable rotational speed of the system (see Figure 3). Postponing the onset speed of this whirling therefore poses the greatest challenge in a high-speed gas bearing design. In literature, this instability is also referred to as half-speed or half-frequency whirl [6],[3].

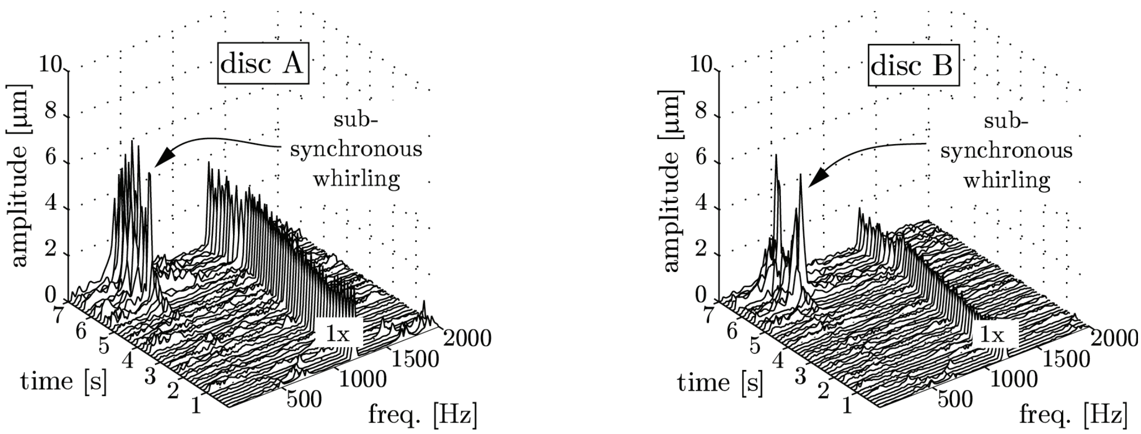

Figure 3: Waterfall diagram of a runup experiment showing sudden self-excited whirling at a subsynchronous frequency. The synchronous response is marked with '1x'.

2.2 Formulation of a stability criterium

Before considering solutions to this stability problem, it might be interesting to determine its underlying mechanism and to investigate how and to what extent the different rotor-bearing parameters affect the situation. To this end, a relatively simple Jeffcott configuration is adopted [9] consisting of a rotordynamic system with only two degrees of freedom (x and y) as shown in Figure 4. The linearisation of the dynamic gas film behaviour around the steady-state operating point leads to the formulation of a set of stiffness and damping coefficients. Two coefficients respectively represent the direct stiffness and damping behaviour (k_{xx}, k_{yy} and c_{xx}, c_{yy}) of the supporting film, while the two other coefficients describe the cross-coupled behaviour (k_{xy}, k_{yx} and c_{xy}, c_{yx}). This cross-coupling is best understood as a reaction of the gas film in a direction perpendicular to the applied perturbation.

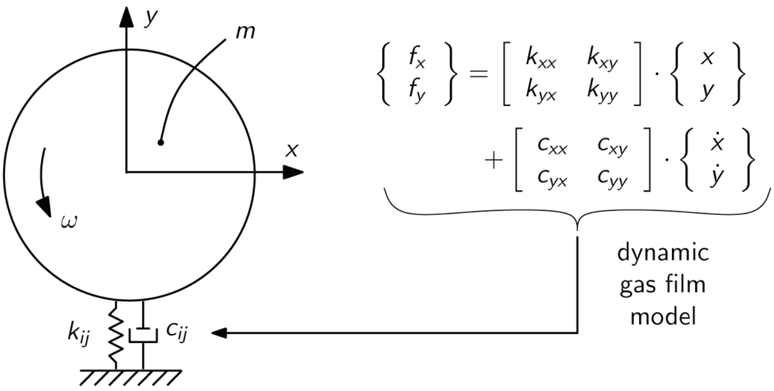

Figure 4: Dynamic model of a Jeffcott rotor-bearing system.

The formulation of the equations of motion of this system and the subsequent analysis of its eigenvalues, reveal that the cross-coupled stiffness acts as negative damping to the rotor-bearing system [1]. Furthermore, it is possible to derive a simple stability criterium which allows us to reason on the effect of the various parameters of the problem. To this end, a few simplifications have to be made first: the assumption of a symmetric rotor-bearing system, i.e. $k_{ii} = k_{xx} = k_{yy}$ en $k_{ij} = k_{xy} = -k_{yx}$; and neglecting the effect of the cross-coupled damping, i.e. $c_{xy} = c_{xy} = 0$. This stability criterium expresses the maximal allowable amount of cross-coupling as function of the other parameters:

$$\left|k_{ij}\right| \leq \sqrt{\frac{k_{ii}}{m}} c_{ii}$$

or, in terms of the ratio between the cross-coupled stiffness $k = k_{ij}/k_{ii}$ and the damping ratio at zero speed $\zeta_n = c_{ii}/(2m\omega_n)$

$$\left|\kappa\right| \leq 2\zeta_n .$$

When observing the typical behaviour of the dynamic gas film coefficients as a function of the speed (see top graph of Figure 5), the occurrence of self-excited whirl seems inevitable when no measures are taken. The left side of the above equations will only increase with speed since the destabilising cross-coupling effect (k_{ij}) originates from aerodynamic film action. The film damping (c_{ii}), on the other hand, decreases due to compressibility effects (at infinitely high perturbation frequencies a gaseous film offers no damping). At a certain operating point, the damping is unable to oppose the destabilising forces in the film which will result in the onset of self-excited whirl. This loss of damping is represented graphically in the bottom part of Figure 5. The onset speed is marked by the intersection point with zero line. At this point, the rotor-bearing system is marginally stable.

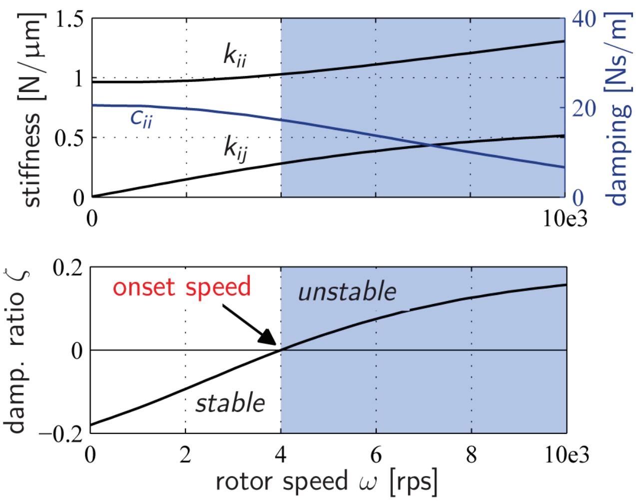

Figure 5: Typical behaviour of the dynamic coefficients k_{ii}, k_{ij} and c_{ii} with respect to rotor speed ω (top); and resulting damping ratio ζ (bottom).

3 STRATEGIES FOR IMPROVING THE STABILITY

Literature reports on a wide variety of bearing types and film geometries with improved stability performance [3],[6],[7],[8]. A detailed overview and description hereof would be outside the scope of this paper. In most cases, the enhancement of the bearing stability is achieved by one or a combination of the below measures:

- A *decrease of the nominal radial clearance*. This approach results in a significant increase in film damping (damping scales inversely proportional with the third power of the air gap). However, it also comes at the cost of increased viscous frictional losses, more stringent manufacturing/assembly tolerances and a reduced ability to cope with thermal or centrifugal distortions.
- A *modification of the film geometry* through the introduction of stabilising surface features either on the rotating or non-rotating bearing member (e.g. spiral grooves). The principal consequence of this approach is an increase of the direct film stiffness due to the viscous pump effect.
- The usage of bearing types with a *conformable film geometry* such as found in for instance tilting-pad bearings or foil bearings. The superior stability behaviour of these bearing types comes forth from a combination of effects: (i) the ability to safely operate at small values of the radial clearance; (ii) optimal gap height profile featuring a low cross-coupling ratio; and (iii) the introduction of damping by means of the support structure.

3.1 Modular test setup

The test setup for validating the stability of various miniature high-speed bearings is shown in Figure 6. 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. 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.

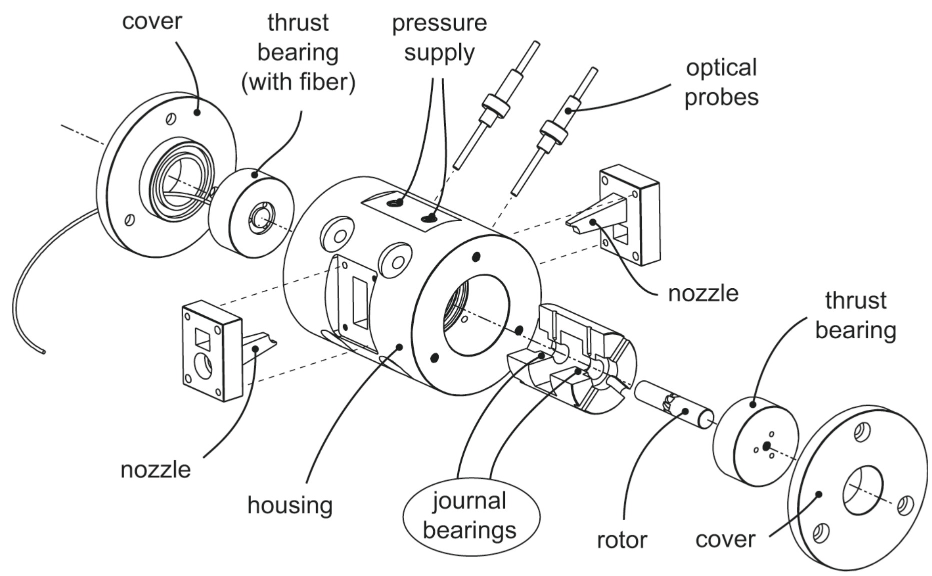

Figure 6: Exploded view of the test setup for validating the stability of various bearing types.

3.2 Strategy 1: Optimal design of conventional aerostatic bearings

Aerostatic bearings have always been an obvious choice for high-speed gas bearing applications since they suffer less from self-excited instabilities than their aerodynamic counterparts. Nevertheless, most aerostatic gas bearing applications still require a sound and well-considered bearing design to prevent self-excited whirling from occurring within the envisaged operation range. A first strategy therefore investigates the potential of conventional, plain aerostatic bearings for high-speed applications. One of the goals of this study concerns the formulation of a series of dimensionless design guidelines which assist the bearing designer during the determination of the bearing parameters such as radial clearance value, bearing length and feedhole diameter (only inherently restricted bearings are considered).

The conclusions from this study may be summarised as follows: (i) the stability is unconditionally enhanced by a reduction of the radial clearance and an increase in bearing length; (ii) for each combination of bearing clearance and length, there exists an optimal value of the feedhole diameter; and (iii) the bearing can be stabilised to some extent by increasing the supply pressure. A more elaborate discussion of the results may be found in [1].

These conclusions have some important implications for practical bearing design. Acceptable performance in terms of high-speed stability can only be attained by reverting to small values of the radial clearance. This, in turn, demands closer control of the manufacturing tolerances, leads to increased viscous frictional losses and creates problems due to thermal and centrifugal distortions. Stabilisation by means of a supply pressure increase, on the other hand, is also not interesting from a practical point of view.

The limitations of conventional aerostatic bearings are illustrated by a first series of experiments performed by the author. By means of a modular setup, shown in Figure 6, plain aerostatic journal bearings with various design parameters have been evaluated at high speed. Rotational speeds up to 506 880 rpm and 3.1 million DN could only be attained through the combination of a relatively small clearance value (ca. 7.5 µm) and elevated supply pressures up to 16 bar.

3.3 Strategy 2: Attempts to reduce the cross-coupling

In a second strategy, the root cause of self-excited whirling is tackled by a modification of the film and entrance geometry. For this purpose, an aerostatic bearing with an innovative geometry has been designed, as shown in Figure 7. The sectioned and strongly asymmetric geometry enables the counteraction of the aerodynamically induced cross-coupling in the film. In contrast to previous implementations of this strategy by means of angled injection [10], the current implementation allows for a complete elimination of the driving force of self-excited whirling up to high values of the rotational speed. It proves even to be possible to over-compensate for the cross-coupling present in a film, which results in backward self-excited whirl rather than the normally observed forward whirl.

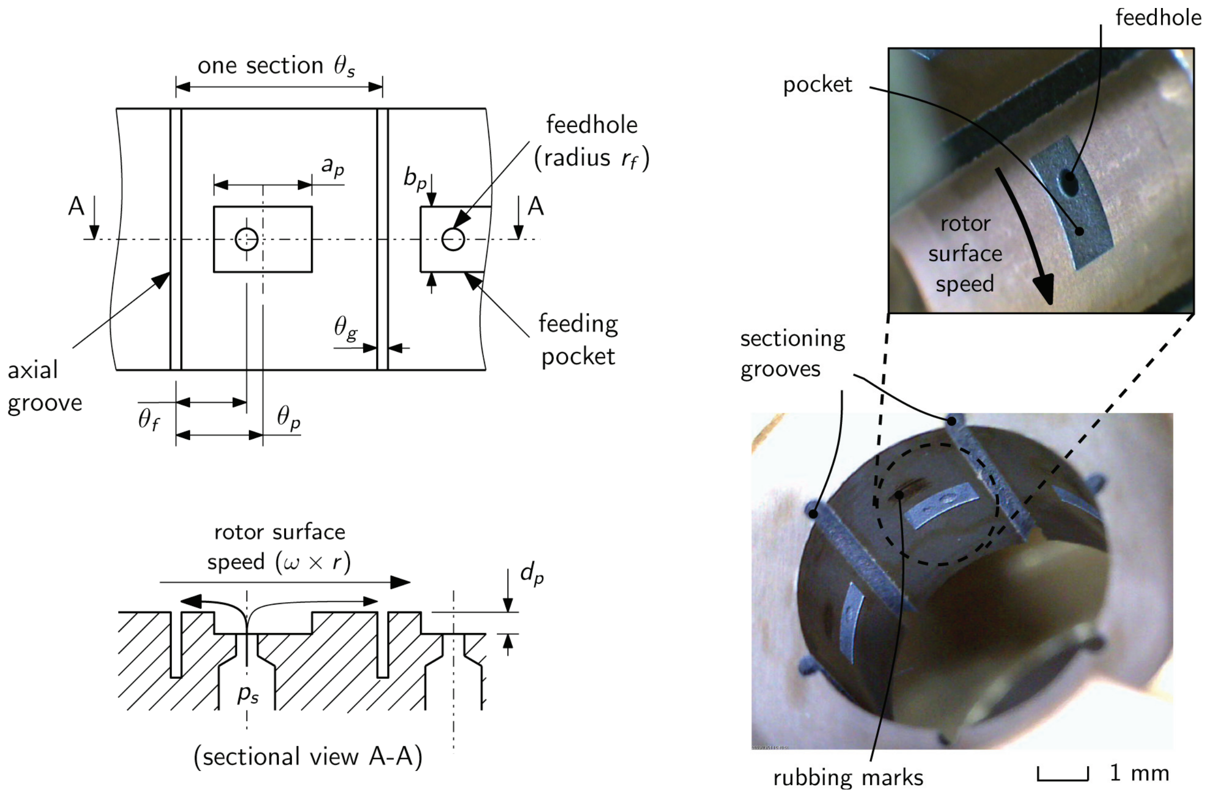

Figure 7: Schematic drawing of proposed geometry to counteract cross-coupling in the film (left); and detail view of the realised test bearing (right) [1].

Due to the pronounced pressure-dependency of the counteraction effect, the supply pressure must be kept between a lower and upper threshold value to prevent respectively forward or backward whirling from setting in. These threshold values shift upwards with speed, resulting in a lower and upper stability boundary. This is illustrated by the stability map of Figure 8. In practice this pressure-dependency implies that the supply pressure has to be increased in a controlled way when speeding up the rotor in order to remain within the stable operating range.

Figure 8 also provides an experimental proof-of-principle of this counteraction strategy at a moderate speed of 120 000 rpm. The frequency spectra at the lower and upper stability boundary indicate respectively forward and backward subsynchronous whirling, while stable operation is seen at a supply pressure value between both boundaries. By gradually increasing both the rotational speed and supply pressure, a maximal speed of 438 600 rpm has been achieved (= 2.6 million DN).

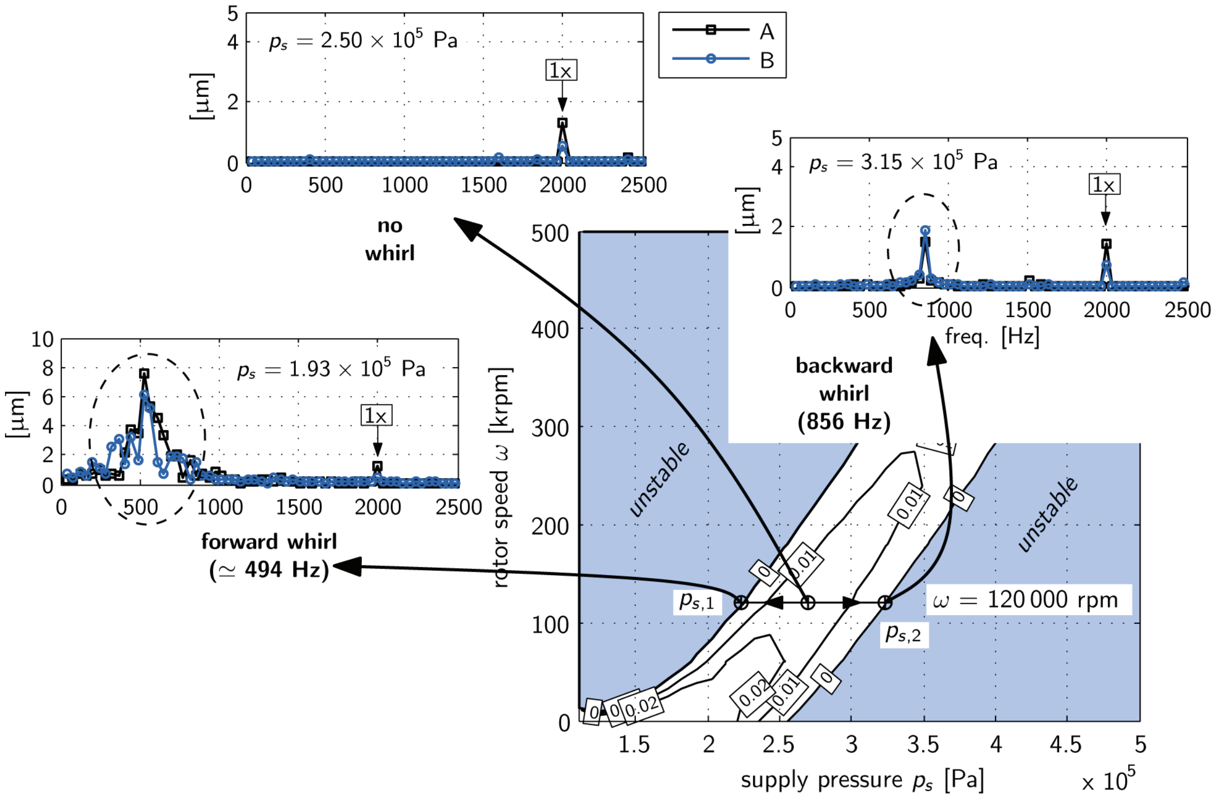

Figure 8: Experimental proof-of-principle at 120 000 rpm showing the frequency spectra respectively at the lower stability boundary, in between both stability boundaries, and at the upper stability boundary.

When compared to conventional aerostatic bearings, this bearing geometry offers the capability of stable high-speed operation at lower values of the supply pressure, making it more attractive for practical use. The main drawback, however, proves to be the lack of film damping caused by the measures taken to counteract the cross-coupling.

3.4 Strategy 3: Introducing external damping

In a last strategy, the destabilising forces are compensated for outside of the gas film, rather than eliminated within the film itself. This introduction of so-called external damping proves to be the only fundamental solution to the stability problem of high-speed gas bearings, since any gaseous film provides only little or negligible damping at high frequencies. This concept of adding damping is however not confined to the field of gas bearings. In aircraft gas turbine engines, rolling element bearings are supported by oil-based squeeze-film dampers to prevent excessive rotor vibrations when passing resonance frequencies [11].

3.4.1 Proposed implementation of the concept of external damping

In our case, damping is introduced to the rotor-bearing system by a flexibly supported bearing bush (see Figure 9 and [12]). The bearing bush does not rotate, but is able to give way when the shaft starts whirling. This flexible support is realised by means of elastomeric O-rings. The oil film surrounding the bush acts in this way as a squeeze-film damper to the system. This configuration combines the low friction properties of a gas film with the controllable, high damping capability of a liquid film. On the inside, the bearing bush has a wave-shaped aerodynamic film geometry which combines a relatively high direct stiffness with favourable stability characteristics [12].

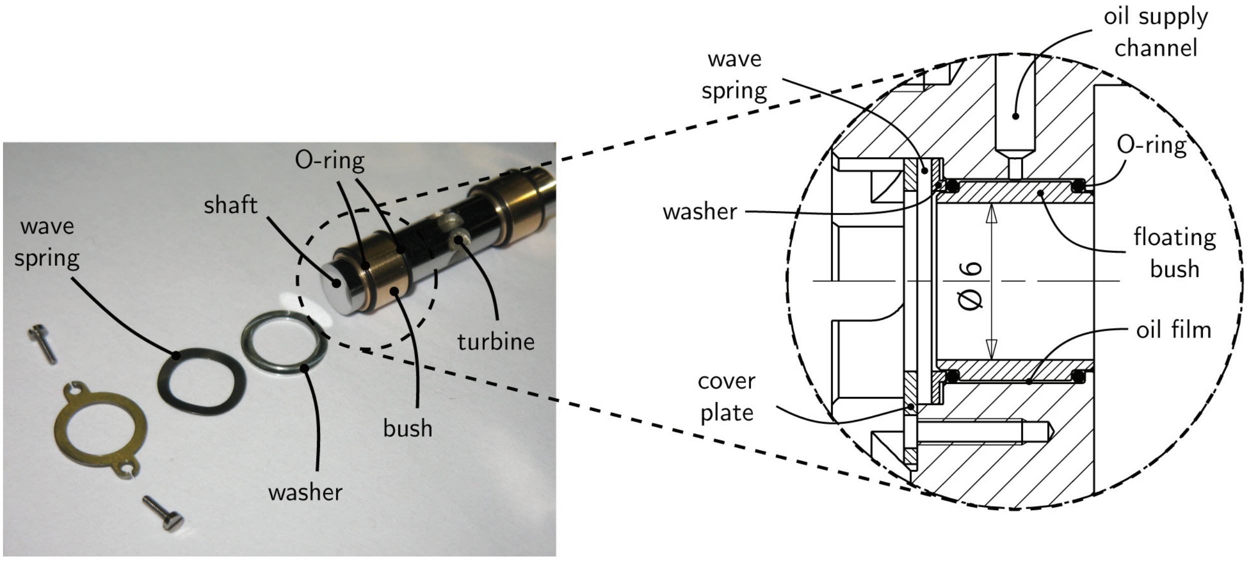

Figure 9: Implementation of the concept of external damping by means of a flexibly supported bearing bush.

3.4.2 *Rotordynamic model to determine the optimal support parameters*

External damping is however only effective through a proper selection of the support parameters, as already observed by [13]. In order to derive design guidelines for this purpose, a rotordynamic model has been set up which takes into account the effect of a flexible bearing support. This model starts from the gas film representation of Figure 4, but includes a bush mass m_b supported by a spring with a constant stiffness k_e and a damper with a constant damping coefficient c_e (see Figure 10).

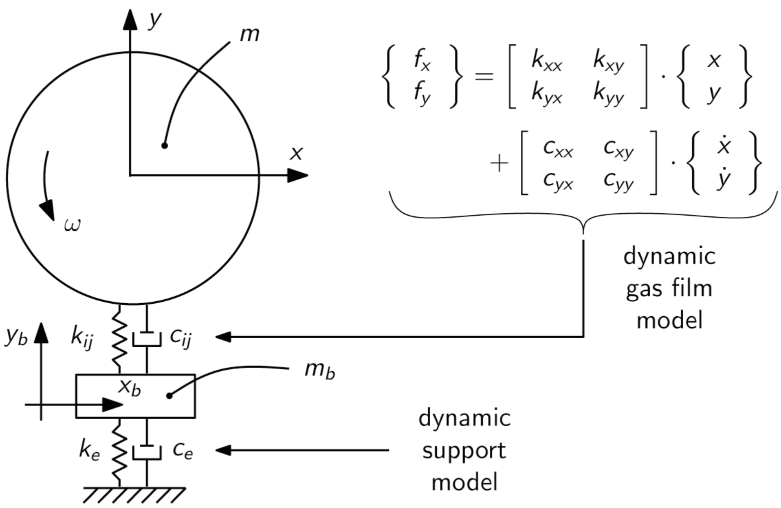

Figure 10: Rotordynamic model of a flexibly supported gas bearing.

The outcome of the stability study may be summarised by the following set of dimensionless design guidelines: (i) try to keep the bush mass a factor ten smaller than the rotor mass, i.e. $m_b/m < 0.1$; (ii) the cross-coupled stiffness of the gas film may not exceed the direct stiffness, i.e. $|k_{ij}/k_{ii}| < 1$; and (iii) provide for an amount of external stiffness that is certainly smaller than the film stiffness, i.e. $k_e/k_{ii} < 0.5$. Of course, each of these guidelines 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.

3.4.3 *Experimental validation*

In a first series of experiments, the optimal value of external support parameters of the system in Figure 9 has been determined. This is done by performing a series of runup experiments with different values of the support stiffness and support damping. These support parameters are tuned by respectively varying the axial preload of the rubber O-rings and by filling the squeeze film cavity with oils of different viscosity.

Hereafter, the system has been successfully tested up to 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. More recently, a runup experiment (Figure 11) has been performed by driving the turbine with helium instead of air. Thanks to the higher sonic exit speed at the driving nozzles (ca. 1000 m/s), the speed could be further increased to 1 203 000 rpm (= 7.2 million DN). This achievement represents to our knowledge the highest DN-number ever reported for a gas bearings of the self-acting type (see Table 1).

The above achievement shows the effectiveness of this last stabilising strategy. In contrast to other measures, the introducing external damping proves itself as a fundamental solution to the stability problem of high-speed gas bearings. The onset of self-excited whirl is not postponed, as is done in other methods, but its driving force is compensated for outside of the gas film.

Current research focuses on the study of alternative support structures which introduce damping in a controllable and reliable way but with an improved compatibility to high-temperature working conditions.

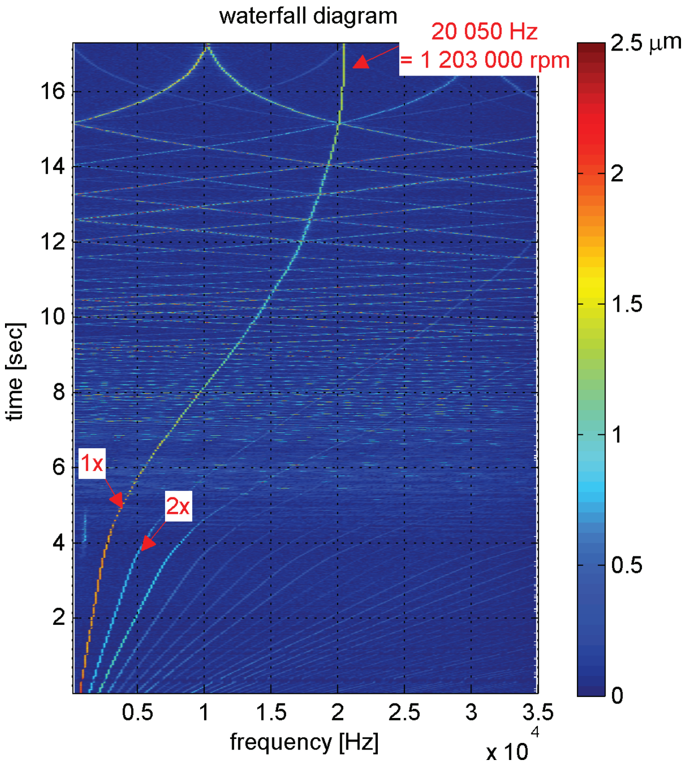

Figure 11: Waterfall diagram of a runup experiment with a helium-driven turbine up to 1 203 000 rpm.

3.4.4 Instrumentation artefacts

The waterfall diagram of Figure 11 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 [14]. 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 Figure 12. 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.

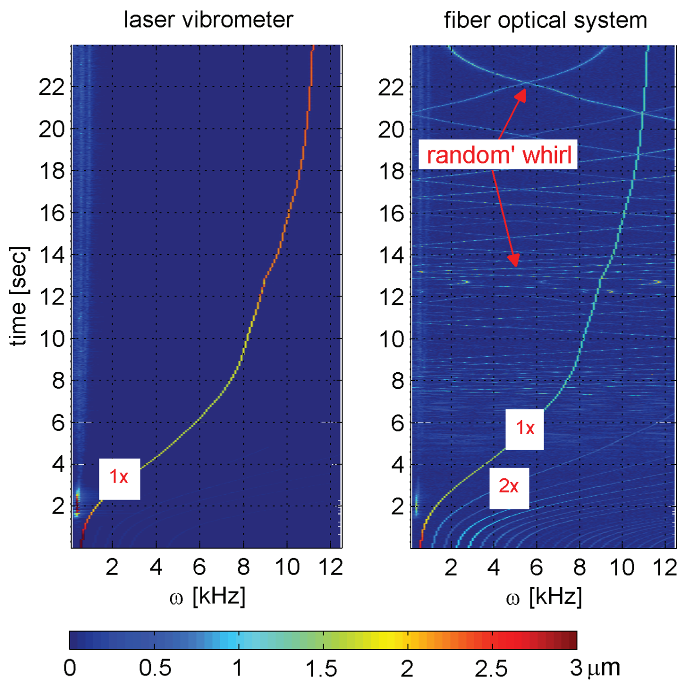

Figure 12: Comparison of waterfall diagram as recorded by a laser vibrometer (left); and an in-house developed fiber optical measurement system (right). The synchronous whirl response is indicated by ‘1x’.

4 CONCLUSIONS

This paper has treated the dynamic stability problem of high-speed gas bearings. The underlying mechanism of self-excited whirling has been identified and a convenient stability criterium has been formulated. The currently existing stabilising techniques are briefly discussed, as well as three additional strategies for further improving the stability. The last strategy poses a fundamental solution to the stability problem by the introduction of damping external to the gas film. The effectiveness of this approach is demonstrated by the successful operation of a 6 mm aerodynamic bearing at 1.2 million rpm, which translates to a record DN-number of 7.2 million.

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