DYNAMIC PROPERTIES OF AEROSTATIC JOURNAL BEARINGS

Jiří Šimek, Jan Kozánek, Pavel Steinbauer

Abstract: Aerostatic bearings are in specific cases used for high speed rotors, where most importance have problems of stability. The knowledge of gas film stiffness and damping is necessary for calculation of the rotor stability limit. Algorithms for calculation of aerostatic journal bearings stiffness and damping already exist, but up to now they were not experimentally verified. The stand for demonstration of rotor dynamic phenomena at CTU in Prague was supplemented with super-structure enabling identification of stiffness and damping matrices coefficients. The stiffness and damping coefficients are determined from the response to harmonic excitation, invoked in two different directions by means of piezo-actuator.

Key words: aerostatic bearings, dynamic properties, stiffness, damping, identification

1. INTRODUCTION

Aerostatic bearings are characterized by high precision of operation and they are aimed first of all for applications with low rotational speeds. However, in specific cases, e.g. in polluted environment or with high requirements to bearing stiffness, they are used also for high-speed applications. As an example of successful high speed applications of aerostatic bearings one can mention dental drills with maximum speed of 750.000 rpm or drilling and grinding spindles with operating speeds above 100.000 rpm. The most important problem of high-speed rotors is stability. The stability limit must be determined with sufficient precision in advance, because the rotor instability in reality ends immediately by heavy failure. The knowledge of gas film stiffness and damping is therefore essential condition for calculation of the rotor stability limit.

Existing programs for calculation of aerostatic journal bearings stiffness and damping could not be up to now verified due to the lack of suitable experimental equipment. The stand for demonstration of rotor dynamic phenomena – Rotor Kit Bently Nevada (RKBN) - installed at CTU in Prague has modular design, which makes it suitable for different modifications. RBKN was therefore supplemented with super-structure enabling identification of stiffness and damping matrices coefficients.

2. IDENTIFICATION OF BEARING STIFFNES AND DAMPING

Plain journal bearings with full fluid film (see Fig.1) have some special features. Journal movement in one direction results in generation of fluid in arbitrary direction. The
bearing force can be therefore resolved into radial component, which returns the journal to its original position, and tangential component, which is the cause of bearing instability. For securing the stability of high-speed rotors it is necessary to use bearings with more complex geometry of sliding surface – lemon, offset halves (Fig 1) or tilting pad type.

![Fig. 1 Basic types of journal bearings: cylindrical, lemon, offset-halves](image1)

The bearing dynamic properties are usually expressed in the form of stiffness and damping matrices, which are matrices of the order 2. The coefficients of stiffness and damping matrices are calculated as a response of fluid film to harmonic excitation. There are many methods of experimental determination of journal bearing dynamic properties [1], all of them having some positive and some negative features. Most widely used was the method based on measuring the bearing response to harmonic excitation in two different directions relatively to static load [e.g. 2]. The design of the test stand for such identifications, built by prof. Glienicke at TH Karlsruhe, can be seen in Fig. 2.

![Fig. 2 Experimental stand of prof. Glienicke - TH Karlsruhe](image2)

The test bearing 120 mm in diameter is located between the two supporting journal bearings. Static load is inflicted by 3 bellows, so that it is possible to change magnitude as well as direction of load. Dynamic harmonic force is actuated by somewhat unusual type of vibrator, which enables continual change of exciting force amplitude without variation of its frequency. Drive of the test shaft and both vibrators is provided by special gearbox. The position of vibrators ensures, that it is possible to achieve simultaneous excitation of the test bearing in two directions with practically arbitrary amplitude of exciting force. The above-described test stand was designed for experiments with maximum speed of 10,000 rpm.
Similar test stand for bearings 90 mm in diameter was designed at the National research institute for machine design (SVÚSS) at Běchovice. Maximum speed, corresponding to sliding speed of about 80 m/s, was 20,000 rpm. The DC motor drive with high-speed gearbox enabled to reach speeds up to 40,000 rpm, which was utilized at tests of bearings with smaller diameter. Maximum static load was about 12 kN, which corresponds to specific load of 2 MPa. Regarding the experimental research of tilting-pad bearings with stiffness and damping dependent on excitation frequency, the vibrators were provided with independent drive by high-frequency motor with maximum speed of 17,500 rpm. Vibrators were equipped with optical sensors of phase marker, which enabled to determine phase shift between the dynamic force and individual components of vibrations. Phase shift was evaluated by special apparatus, similar to those used in balancing machines, because only components with frequency of excitation should be taken into consideration. Computer program based on dynamic model of the test stand was devised for calculation of test bearing stiffness and damping coefficient [4].

3. Modification of Rotor Kit for Identification of Bearing Dynamic Coefficients

Rotor Kit Bently Nevada (RKBN) is designed as modular and therefore it is almost ideal basis for possible super-structures. That is why this equipment available at CTU in Prague was used for identification of dynamic coefficients of aerostatic bearings. Aerostatic bearings were selected because available maximum speed of the RKBN - 10,000 rpm - is not sufficient for experimental research of aerodynamic bearings.

Principle changes of the RKBN are evident from the cross section shown in Fig. 3.

![Fig. 3 Super-structure for identification of aerostatic bearings dynamic properties](image)

The test shaft 1 is supported in two rolling bearings 18, which are inserted into bearing bodies 2 and 3 fastened to original RK frame. The test head 4 with aerostatic bearing 5 is located between rolling bearings. The rolling bearing outer diameter is smaller than test bearing diameter, so that the change of the bearing would be as simple as possible. Piezo-electric actuator 12 for excitation of the bearing by harmonic force is connected to the test head by means of joint 7. The actuator is at its upper end connected to rigid frame 15, once more by joint 8. Usage of joints is necessary, to eliminate forces in undesirable directions, which would otherwise act on the bearing. Type P-845.40 piezo-actuator of the firm PI is used, with push/pull force capacity of 3000/700 N. Load cell 13 is situated between actuator and test head, enabling recording...
of exciting force together with pertinent responses. Load cell Bruel&Kjaer type 8200, with measuring range of 1000 N for tension and 5000 N for compression, was considered. The actuator frame is fastened directly to support bearing bodies.

Necessary conditions for getting correct results are to limit all clearances and to achieve maximum possible stiffness of all parts of the stand. The frame eigenvalues were determined by experimental modal analysis [3]. Although lower modes of vibration of detected eigenfrequencies are not in direction of dynamic excitation force of piezo-actuator, it will be advisable to avoid them during tests. The body of RKBN was mounted on rigid, heavy plate and whole stand table was placed on thin rubber pads with resulting eigenfrequency of about 5 Hz. Thus the experiment is isolated from vibration excitation of the environment and the foundation eigenfrequency is far from excitation frequencies used during experiment.

Stiffness and damping coefficients will be determined from the bearing response to harmonic excitation, invoked in two different directions by means of above-mentioned piezo-actuator. The test methodology is based on experience with aerostatic bearings and with experimental research of dynamic properties of journal bearings. Static load in direction of dynamic force is adjusted by shifting the test bearing through piezo-actuator by means of differential screw 16. The force generated by the gas film at certain journal eccentricity is measured by the load cell and the corresponding eccentricity is determined by relative sensors 14. The relation force – eccentricity is then used for setting the static load in direction perpendicular to dynamic harmonic force; the eccentricity is adjusted by means of screw 26, which can be designed also as differential one. Relative sensors 14 will be used also for measurement of test bearing response to harmonic excitation by piezo-actuator. Signal from the load cell will be recorded together with the response in both directions. The bearing is equipped with relative sensors at both ends, so that it will be possible to determine the mean values at the bearing centre plane. All four coefficients of stiffness and damping matrices can be determined from magnitude and relative phase shifts of the response.

Contrary to oil lubricated bearings, where viscosity is strongly dependent on temperature, dependence of dynamic viscosity of gases on temperature is minimal. That is why it is possible to carry out the whole series of tests with excitation in direction of static load and then perform for the same parameters (journal eccentricity, air inlet pressure, speed) the series of measurements with excitation perpendicular to static load. Because it is not necessary to wait for stabilization of temperature, the proceeding could be very fast. The design of the stand enables fast exchange of test bearings, so that it would be possible to get great amount of experimental data in relatively short time.

4. Dynamic model of test stand

The universal dynamic model of the test stand used in [4] is schematically outlined in Fig. 4.
The model considers flexibility of supporting sliding bearings and test stand foundation. Basic equations of motion are as follows:

\[ M_2 \ddot{\mathbf{x}}_2 + 2Z \mathbf{x}_i = f_k, \quad k=1,2, \text{ for test bearing} \tag{1} \]

where \( M_2 = \begin{bmatrix} M_2 & 0 \\ 0 & M_2 \end{bmatrix} \), \( M_2 \) ... mass of test bearing,

\[ \mathbf{x}_2 = \begin{bmatrix} x_2 \\ y_2 \end{bmatrix} \] ... excursions of the test bearing,

\[ \mathbf{x}_i = \begin{bmatrix} x_i \\ y_i \end{bmatrix} \] ... excursions of the shaft,

\[ \mathbf{x}_r = \begin{bmatrix} x_r \\ y_r \end{bmatrix} = \begin{bmatrix} x_2 - x_1 \\ y_2 - y_1 \end{bmatrix} \] ... excursions of the test bearing relative to the shaft,

\[ 2Z = \begin{bmatrix} 2Z_{xx} & 2Z_{xy} \\ 2Z_{yx} & 2Z_{yy} \end{bmatrix} \] ... matrix of test bearing complex stiffness,

\[ 2Z_{jk} = K_{jk} + i\Omega H_{jk}, \quad i = \sqrt{-1}, \]

\[ f_1 = \frac{\sqrt{2}}{2} \begin{bmatrix} F_{d1} \\ F_{d1} \end{bmatrix} e^{i\Omega t}, \quad f_2 = \frac{\sqrt{2}}{2} \begin{bmatrix} -F_{d2} \\ F_{d2} \end{bmatrix} e^{i\Omega t} \] ... exciting dynamic forces,

\[ M_1 \ddot{\mathbf{x}}_i + 2^1Z(\mathbf{x}_i - \mathbf{x}_3) - 2^2Z \mathbf{x}_i = 0, \quad \text{for the shaft} \tag{2} \]

where \( M_1 = \begin{bmatrix} M_1 & 0 \\ 0 & M_1 \end{bmatrix} \), \( M_1 \) ... shaft mass,

\[ \mathbf{x}_i = \mathbf{x}_2 - \mathbf{x}_r, \quad \mathbf{x}_i = \begin{bmatrix} x_i - x_3 \\ y_i - y_3 \end{bmatrix} = \mathbf{x}_2 - \mathbf{x}_r - \mathbf{x}_3, \]

\[ \mathbf{x}_3 = \begin{bmatrix} x_3 \\ y_3 \end{bmatrix} \] ... excursions of the frame,

\[ ^1Z = \begin{bmatrix} ^1Z_{xx} & ^1Z_{xy} \\ ^1Z_{yx} & ^1Z_{yy} \end{bmatrix} \] matrix of supporting bearing complex stiffness,

\[ M_1 \ddot{\mathbf{x}}_3 - 2^1Z(\mathbf{x}_1 - \mathbf{x}_3) - 2^2Z \mathbf{x}_3 = -f_k, \quad k=1,2, \text{ for the frame} \tag{3} \]

where \( M_3 = \begin{bmatrix} M_3 & 0 \\ 0 & M_3 \end{bmatrix} \) .... \( M_3 \) ... mass of the frame,

\[ ^3Z = \begin{bmatrix} ^3Z_x & 0 \\ 0 & ^3Z_y \end{bmatrix} \] ... matrix of the frame support complex stiffness.
Assuming the solution in the form
\[ \begin{align*}
\chi_j &= \hat{\chi}_j e^{i\Omega t}, \quad j=1, 2, 3, \\
\ddot{\chi}_j &= -\Omega^2 \chi_j e^{i\Omega t},
\end{align*} \tag{4} \]

after substitution into (1), (2) and (3) we get
\[ \hat{Z} \hat{\chi} = \hat{f}_k, \quad k=1,2, \tag{5} \]

where
\[
\hat{Z} = \begin{bmatrix}
\begin{array}{ccc}
\Omega^2 M_1 - 2 Z^2 + Z & -\Omega^2 M_2 & 0 \\
\Omega^2 M_1 + 2 Z - 2 Z^2 & -\Omega^2 M_1 + 2 Z & -2 Z \\
2 Z & -2 Z & -\Omega^2 M_3 + 2 Z + Z^2
\end{array}
\end{bmatrix},
\]

\[
\hat{\chi} = \begin{bmatrix}
\chi_1 \\
\chi_2 \\
\chi_3
\end{bmatrix}, \quad \hat{f}_1 = \frac{\sqrt{2}}{2} F_{d1}, \quad \hat{f}_2 = \frac{\sqrt{2}}{2} F_{d2}.
\]

To determine the test bearing stiffness and damping coefficients it is sufficient to solve equation (1) for vectors of complex amplitudes \( \chi_2, \chi_r \) measured at two different directions of excitation force \( F_{d1}, F_{d2} \).

The dynamic model of RKBN could be probably simplified, because stiffness of the frame and supporting rolling bearings will be much higher than expected stiffness of aerostatic journal bearings. The movement of the foundation and of the shaft in support bearings will be very small in comparison to with the excursions of test bearing relative to shaft and could be therefore probably neglected. This assumption must be however confirmed by measurement.

5. **Test bearings**

Four variants of aerostatic journal bearings were proposed for experimental verification of dynamic characteristics. Bearing diameter is 30 mm, their length to diameter ratio \( l/D \) is equal either to 1,5 (Fig. 5) or to 1,0 (Fig. 6).

![Fig. 5 Test aerostatic journal bearing with I/D=1,5](image-url)
The advantage of greater $l/D$ ratio is higher value of angular stiffness, which may eliminate problems with tilting of the test bearing when it is excited by dynamic force. The shorter bearing, on the other hand, is closer to bearing geometry used in reality. Both bearing types have either two rows of feeding orifices distant one quarter of bearing length from the boundary, or one row of orifices in centre plane of the bearing. There are 8 orifices around the bearing periphery of 0.2 mm ($l/D=1.5$) or 0.3 mm ($l/D=1.0$) in diameter. The bearing diametral clearance varies from 0.03 mm to 0.045 mm, which corresponds to relative clearance of from $1.10^{-3}$ to $1.5.10^{-3}$. By increasing the feeding hole diameter or bearing clearance one can get another bearing geometry to broaden the measured dynamic characteristic data range.

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Fig. 6 Test aerostatic journal bearing with $l/D=1.0$

Calculated static and dynamic characteristics of test bearings with $l/D=1.5$ - radial clearance 0.0225 mm - and $l/D=1.0$ - radial clearance 0.015 mm- are summarized in Table 1.

Table 1  Expected bearing characteristics at relative eccentricity of 0.5

<table>
<thead>
<tr>
<th>type</th>
<th>$l/D$</th>
<th>inlet pressure (MPa)</th>
<th>load capacity (N)</th>
<th>stiffness (N.m$^{-1}$)</th>
<th>damping (N.s.m$^{-1}$)</th>
<th>flow rate (l/min)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 row of orifices</td>
<td>1.5 0.0225</td>
<td>0.3</td>
<td>45</td>
<td>6.5.10$^6$</td>
<td>4.2.10$^3$</td>
<td>2.8</td>
</tr>
<tr>
<td></td>
<td>1.0 0.015</td>
<td>0.5</td>
<td>74</td>
<td>9.9.10$^6$</td>
<td>3.9.10$^3$</td>
<td>4.9</td>
</tr>
<tr>
<td>2 rows of orifices</td>
<td>1.5 0.0225</td>
<td>0.3</td>
<td>21</td>
<td>1.1.10$^7$</td>
<td>4.1.10$^3$</td>
<td>2.4</td>
</tr>
<tr>
<td></td>
<td>1.0 0.015</td>
<td>0.5</td>
<td>54</td>
<td>1.8.10$^7$</td>
<td>3.2.10$^3$</td>
<td>4.7</td>
</tr>
<tr>
<td></td>
<td>1.5 0.0225</td>
<td>0.3</td>
<td>64</td>
<td>1.0.10$^7$</td>
<td>3.2.10$^3$</td>
<td>5.6</td>
</tr>
<tr>
<td></td>
<td>1.0 0.015</td>
<td>0.5</td>
<td>105</td>
<td>1.5.10$^7$</td>
<td>3.0.10$^3$</td>
<td>9.7</td>
</tr>
<tr>
<td></td>
<td>1.5 0.0225</td>
<td>0.3</td>
<td>28</td>
<td>1.5.10$^7$</td>
<td>3.0.10$^3$</td>
<td>4.7</td>
</tr>
<tr>
<td></td>
<td>1.0 0.015</td>
<td>0.5</td>
<td>72</td>
<td>2.5.10$^7$</td>
<td>2.3.10$^3$</td>
<td>9.5</td>
</tr>
</tbody>
</table>

It can be seen, that bearing with 2 rows of orifices has about 50% higher load capacity and stiffness, but also at the same rate increased compressed air consumption. Lower clearance results naturally in greater stiffness, even if the bearing active area, i.e. $l/D$ ratio, is reduced by 30%.
6. Conclusions

Rotor Kit Bently Nevada was extended by super-structure enabling identification of dynamic characteristics of aerostatic journal bearings. Individual parts of super-structure were manufactured and their dynamic properties were determined either by calculation or by modal analysis. Stiffness and damping coefficients will be determined from the bearing response to excitation by harmonic force. The exciting force will be generated by piezo-actuator, so arbitrary force series can be easily generated. General dynamic model of the test stand was proposed, which could be probably simplified due to low stiffness of test bearings relative to support bearings and the frame. The most important condition for correctness of identified stiffness and damping coefficients is to achieve the highest possible accuracy of determining the amplitudes of test bearing response and namely of the phase shift between exciting force and the test bearing response.

Rotor Kit Bently Nevada with its maximum speed of 10.000 rpm is suitable for identifications of aerostatic bearings. For identification of dynamic properties of more widely used aerodynamic bearings could be after some modifications used one of the stands, designed for demonstration of rotor-dynamic phenomena at TU Liberec [5].

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7. References