Active Magnetic Bearings as Actuators and Sensors in Journal Bearing Measurements

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Abstract: The present paper shows the successful usage of active magnetic bearings (AMBs) to identify the dynamic characteristics of journal bearings in the frequency domain. A test rig is operated using two AMBs supporting a rigid rotor with a centric journal bearing. An identification procedure for the stiffness and damping coefficients of journal bearings is presented. This procedure incorporates the successful application of AMBs for positioning, supporting, sensing and actuating the rotor simultaneously. For identification the real excitation force is needed. Two methods to derive a force-current-position relationship are outlined and compared: an idealized mathematical relation and a measured force-current-position map. Finally, results for applying the pre-measured map and an example of the identified stiffness and damping coefficients are given.

Keywords: Active Magnetic Bearings, Actuator, Sensor, Application, Identification of Dynamic Properties

Introduction

Some of the fundamental elements in accurate description of rotordynamic systems are the dynamic properties of the supporting bearings. The identification of bearing parameters is crucial for adequate interpretation of rotating machinery performance and is necessary to validate or calibrate predictions from computational models. Journal bearings strongly influence the dynamic behavior of rotor systems. The dynamic properties of journal bearings are usually taken into account by linearized stiffness and damping coefficients depending on the rotor speed. These coefficients are either calculated or measured and are available in tabulated form for various types of journal bearings, see for instance the classical references [1, 2].

The aimof many experimental studies on journal bearings is the determination of the linearized stiffness and damping coefficients in dependency of the actual rotor speed. However, these coefficients can be measured indirectly only. A conventional approach is to excite the journal of a rigid rotor, to measure the corresponding displacement and velocity orbits and to identify the underlying dynamic coefficients. This approach may lead to difficulties in the experimental realization of the excitation force. In addition, the shaft position lies within a fraction of the radial clearance which makes highly accurate measurements difficult. The merit of the present study is to determine these coefficients experimentally for a specific design of a journal bearing by utilizing the advantages of AMBs

In the proposed setup, active magnetic bearings (AMBs) are introduced. The main advantage of employing AMBs is that they are capable of supporting, sensing and exciting the rotor at the same time and at the same location – they can be used as collocated sensor and actuator pairs. AMBs are especially suited for experimental rotordynamic studies as they wor k contact-free and almost loss-free. Furthermore, AMB-supports enable the advantages of easily positioning the rotor in any desired position within the bearing clearance, a feature that is impossible to come by with conventional ball bearings. Hence, different static equilibrium positions of the journal bearing under consideration can be employed. Furthermore, AMBs enable to apply additional static and dynamic forces to the rotor by providing an additional disturbance current to the magnets on top of the control current.

Experimental Setup

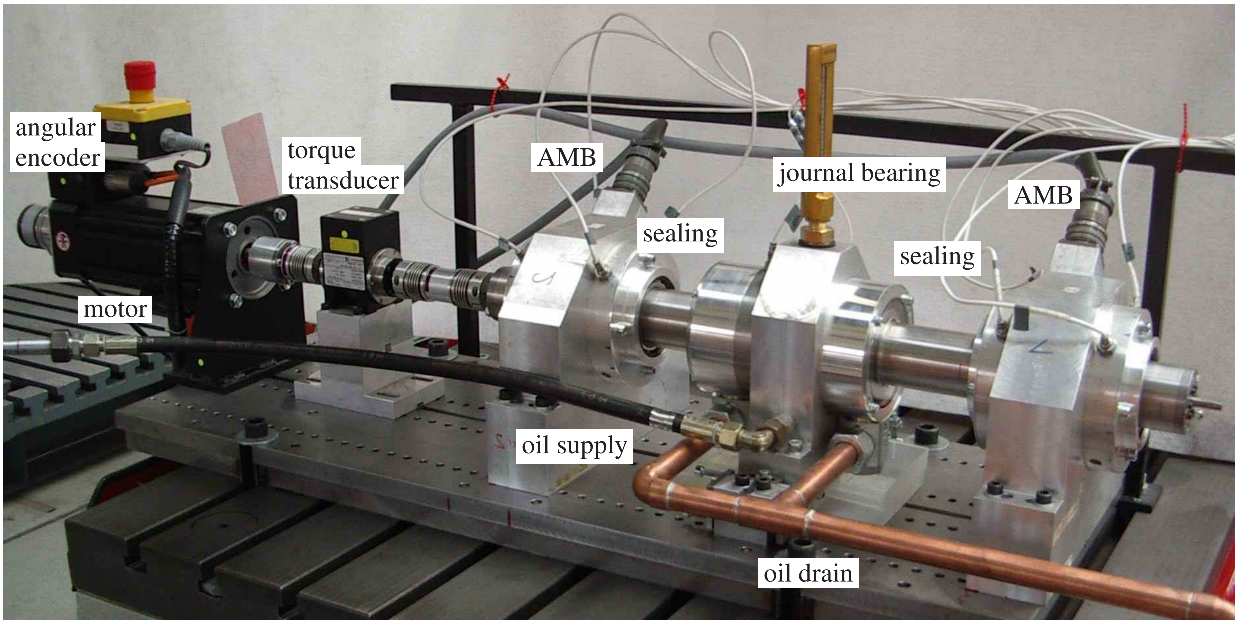

Figure 1: Journal bearing test rig at the Structural Dynamics Laboratory

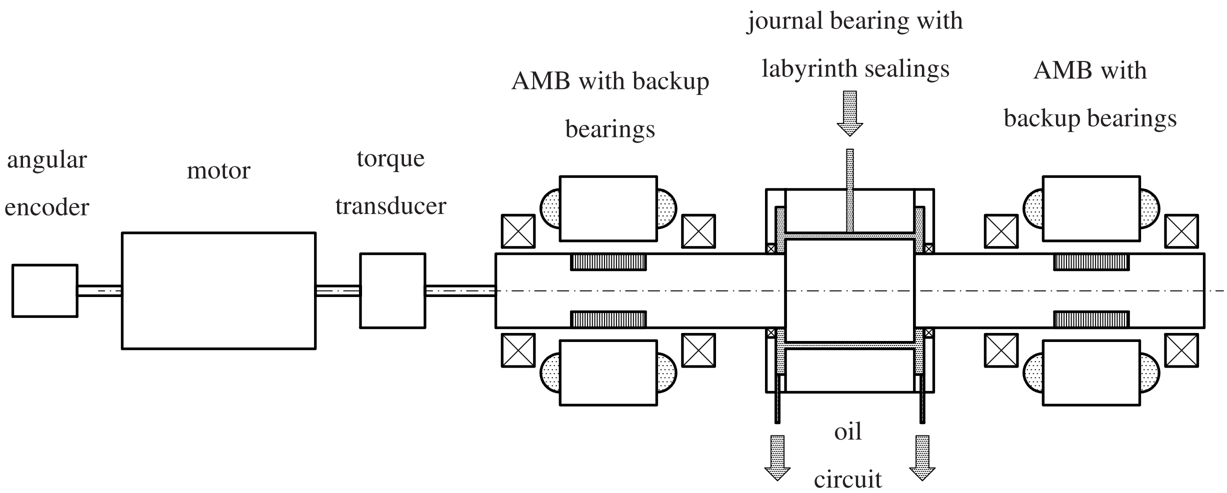

Figure 2: Sketch of the journal bearing test rig in Fig. 1

Figures 1 and 2 show the journal bearing test rig assembled at the Structural Dynamics Laboratory. A rigid rotor runs in the centered journal bearing and is supported by two AMBs at both rotor ends. The positions of the rotor at the journal bearing and at both AMBs are measured via contact-free eddy current displacement sensors. The rotor is driven by an electric motor which speed is controlled via an external signal. The torque provided by the motor and acting on the rotor is measured by a torque transducer between rotor and motor. The coupling between the rotor and the torque transducer is highly flexible. This allows for large radial misalignment within the bearing clearance which is needed for identification at different rotor positions in the journal. An angular encoder mounted on the motor side determines the rotation of the rotor. This encoder is placed on the motor side in order to keep the friction loss small. The losses in the AMBs are inevitable, however, they are small. A dSpace Control System is implemented for the active control of the magnetic bearings as well as for data acquisition. This setup enables the identification of dynamic stiffness and damping coefficients at constant rotor speeds as well as for run-up and run-down processes.

Operation of AMBs

Both AMBs are nominally identical and possess two perpendicular radial axis z_j and y_j, with $j = 1, 2$ for the two AMBs, see Fig. 3. Each AMB consists of two opposite electromagnets in each radial direction. The current fed through the electromagnets is controlled actively in dependency of the actual rotor position. During active control of the system, the rotor position is measured by the position sensors, the corresponding control current for a desired rotor position is calculated within the control system and passed to a power amplifier that supplies the electromagnets. The electromagnetic compatibility of these components is crucial for a successful assembly. The characteristic values of the AMBs are listed in Table 1.

Table 1: Test rig properties

Rotor	speed	$n = 0...6000\,\mathrm{rpm}$
	mass	$m = 13.6\,\mathrm{kg}$
Journal bearing	type	circular, one axial groove
	diameter	$D = 60\,\mathrm{mm}$
	width	$B = 60\,\mathrm{mm}$
	clearance ratio	$\psi = 8.3^0/_{00}$
	kinematic viscosity at 25 °C	$\nu = 16.5\,\mathrm{mm^2/s}$
Magnetic bearings	number of electromagnets	4 in each bearing
	magnetic bearing constant	$k_m = 3.36 \cdot 10^-6\,\mathrm{Vsm/A}$
	controller sampling frequency	8192 Hz
	bias current	$i_V = 2.5\,\mathrm{A}$
	maximum current	5 A
	clearance	0.78 mm
	rotor diameter	56 mm

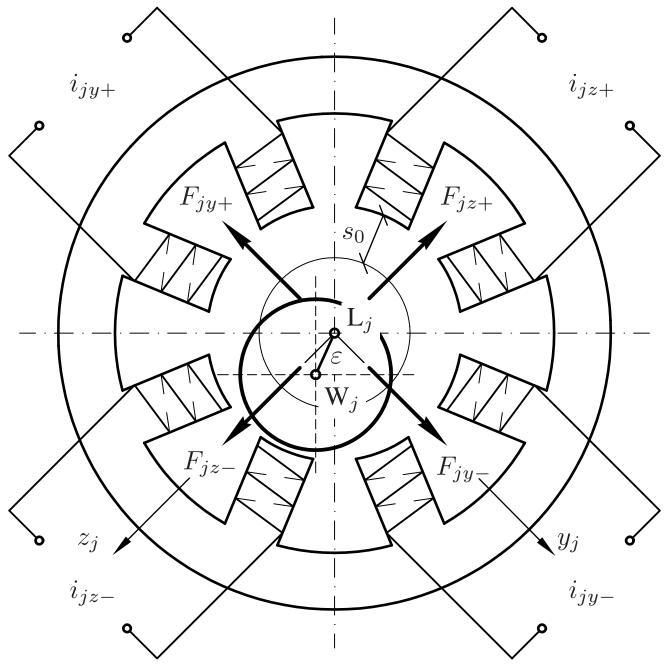

Figure 3: Design of the magnetic bearing

An AMB applies an electromagnetic force $F_j = F_{js+} - F_{js-}$ to the rotor along the coordinate axis $s = z, y$. This force depends strongly nonlinear on the rotor position $s_{Wj} = z_{Wj}, y_{Wj}$ and the control currents i_{js+} and i_{js-}. In order to reduce this strongly nonlinear dependency of the electromagnetic force, a difference control scheme is applied for which the control current supplied to the electromagnets is split into two parts: a constant magnetic bias current i_V and a control current i_S. The total current supplied to a pair of opposite electromagnets follows then to be $i_{js+} = i_V + i_S$ and $i_{js-} = i_V - i_S$.

The AMBs are operated decentralized and both radial movements z_j and y_j of the rotor are controlled independently. The operation axis of the electromagnets enclose 45 degrees with the vertical axis. This ensures that both directions are contributing equally to the static bearing load and, consequently, possess identical force characteristics, such that the same controller scheme can be applied for both directions. The condition for collocation of the position measurement is fulfilled by mounting two sensors at each axis. Then, the radial position of the rotor is determined by measuring the position of the bearing stud on the left and right hand side of the electromagnets and taking the mean value. This is justified for rigid rotors. In total four sensors are used for a single AMB. Because of the collocation, a stable rotor support can be realized by a simple linear PD-controller. An additional integral controller component is used for compensation of a static deflection error due to the rotor weight. Therefore, PID-controllers with the control law

$$i_S(t) = c_P\,\Delta s(t) + c_D\,\Delta\dot{s}(t) + c_I \int_0^t \Delta s(\tilde{t})\,d\tilde{t} \tag{1}$$

are implemented for AMB operation. The control current $i_S(t)$ depends linear on the offset along the corresponding coordinate axis

$$\Delta s = s_{Wj} - s_S \qquad \text{with} \qquad s = z_j, y_j. \tag{2}$$

Herein $s_{Wj} = z_{Wj}, y_{Wj}$ depicts the measured rotor position and $s_S = z_S, y_S$ the set value for the rotor position, see Fig. 3. The control law in Eq. 1 comprises of the proportional parameter c_P, the derivative parameter c_D and the integral parameter c_I. These control parameters define the dynamic behavior of the AMB with respect to its stiffness and damping. The nominal control parameters assure a stable AMB operation (without the journal bearing) and minimize the rotor vibration.

Force measurement with AMBs

AMBs are suited for force measurements if the force-current-position relationship is known. Then, the acting electromagnetic force is determined by sensing the rotor position and by the calculated control current. Reversely, a desired (additional) force can be applied to the rotor system by the AMB by an additive current fed through the electromagnets. Both, force determination and force application, originate contact-free and in arbitrary radial directions. The actual electromagnetic force acting on the rotor can be identified by two methods. The first method is based on an idealized equation for the electromagnetic force. The second method determines the force by interpolation within an experimentally pre-determined force-current-position map. Both procedures allow the identification of the dynamic coefficients of the centric journal bearing.

The first method incorporates the idealized electromagnetic force which acts along a radial direction. For the difference control scheme, this force can be written as ([6])

$$F_s = F_{s+} - F_{s-} = k_M \left[\left(\frac{i_V + i_S}{s_0 + s} \right)^2 - \left(\frac{i_V - i_S}{s_0 - s} \right)^2 \right] \text{ with } F_s = F_{jz}, F_{jy} \text{ and } j = 1, 2\,. \quad (3)$$

This force is by definition positive for a positive rotor deflection, see Fig. 3. The relation in Eq. 3 represents an idealized approximation. Examples of system immanent nonlinearities not introduced in Eq. 3 are the saturation of the core material, the magnetic hysteresis and the electromagnetic coupling of the radial directions z and y. Especially in the border area of the force-current-position map, the approximated force in Eq. 3 will strongly deviate from the real electromagnetic force acting. In particular, for a vanishing air gap an infinitely large force is predicted which is not achieved in reality due to the saturation and the magnetic resistance.

The second method bypasses this inadequacy by pre-measuring a force-current-position map. During identification, the actual force is determined by interpolation from the pre-measured force-current-position map, see [3, 4]. Doing so, the identified force matches the real force sufficiently well. The identification of the force-current-position map is carried out in a separate instrumentation described in detail in [5, 6]. One rotor stud is clamped between two flexible components and placed within the AMB at a desired position. Strain gauges are placed on this fixture. Each electromagnet is powered by a separate current such that all combinations of current and rotor position are reached. The comparison between the measured map with the idealized relation in Eq. 3 is shown in Fig. 4 and emphasizes the inaccuracy of the idealized relation in the region of large rotor deflection, i.e. small air gap and large attraction force by neighboring electromagnets.

Identification of journal bearing stiffness and damping coefficients

In general, the journal bearing stiffness and damping coefficients can be determined experimentally by applying small additional forces to the rotor and measuring the resulting vibrations or vice versa. The linearized coefficients can afterwards be calculated from the relation between force and position and between force and velocity, respectively. A benefit of using

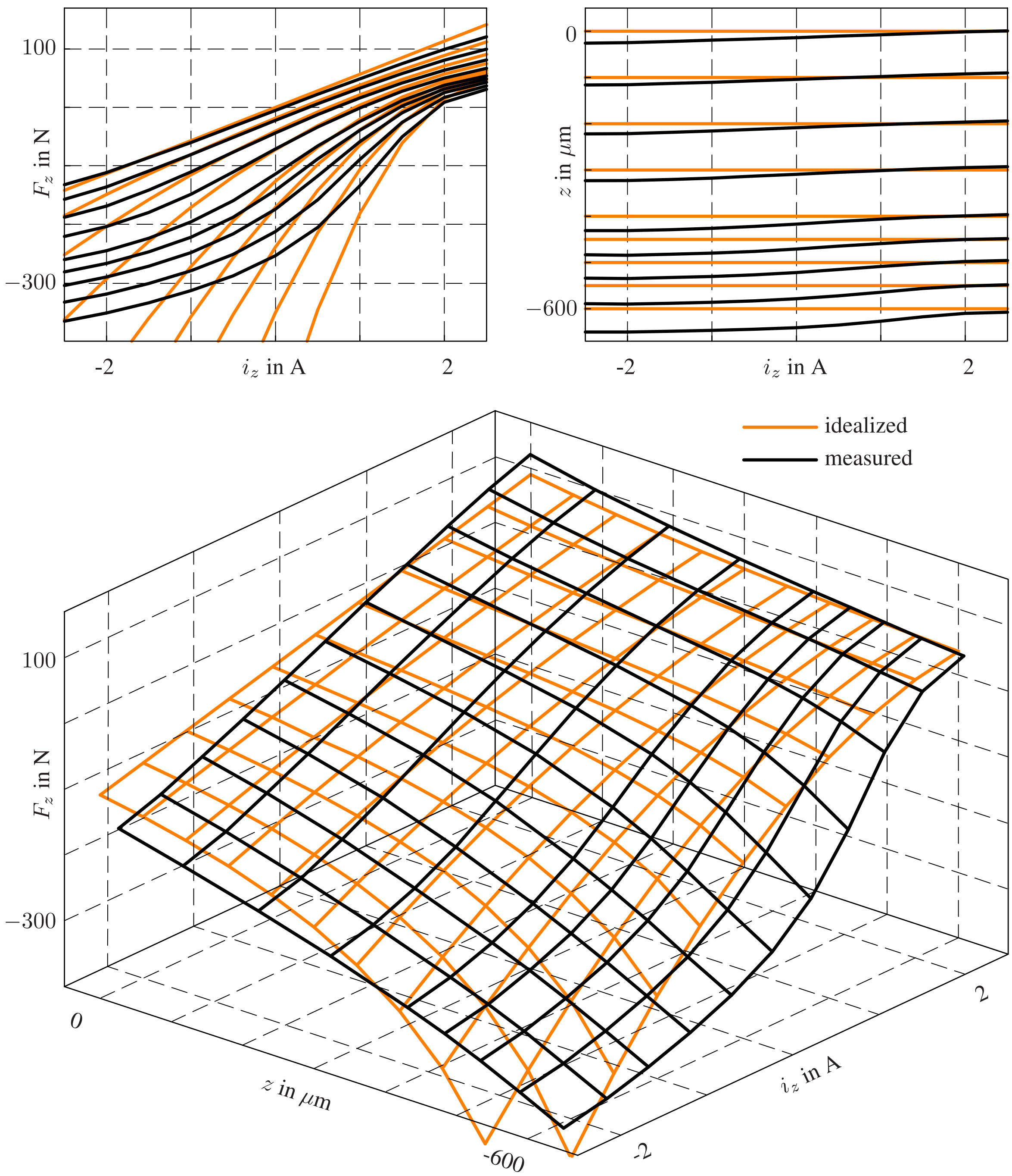

Figure 4: Comparison between measured and ideal force-current-position map of one AMB at control current $i_y = 0$, bias current $i_V = 2.5\,\mathrm{A}$ and centric position $y = 0$.

AMBs is that they support the rotor in every position and simultaneously act as sensors and actuators, respectively, at the same location. The AMBs' sensor operation involves measuring the rotor position and applying the control current to the electromagnets.

Within the successfully realized measurement procedure, the additional excitation currents for the bearing magnets are applied as uncorrelated noise in the frequency range of rigid body vibrations in two orthogonal directions. The magnitude of the additional forces is interpolated from the force-current-position map, which was experimentally determined earlier by [3], using the measured magnetic bearing rotor positions and the applied excitation currents. The AMBs are also used for positioning the rotor. Since the journal bearing stiffness and damping coefficients depend on the rotor speed and the static equilibrium eccentricity, respectively, different measurement positions are chosen throughout the bearing clearance. At each measurement position 16 measurements were taken for averaging. The rotor speed is kept either constant (at different fixed values) or fulfills a short run-up around a constant reference speed at different constant rotational accelerations.

The AMBs' actuator operation involves the addition of a small excitation current added to the control current. This exciting current introduces additional electromagnetic forces acting on the rotor leading to additional small displacements within the bearings. The bearing stiffness and damping coefficients are calculated by relating the additive forces to the resulting additional displacements. The calculation of the linearized stiffness and damping coefficients from the measured data is realized in the frequency domain using spectral densities. By assuming small forced vibrations within the excitation frequency range of 3 to 100 Hz, the stiffness and damping coefficients are evaluated from the complex coefficient matrix and averaged over the

Settings:

rotor position within the bearing clearance

possible rotor speeds:
a) constant (e.g. 300 or 600 rpm)
b) short run-up around nominal speed (e.g. 550..650 rpm)

Rotor excitation:

force excitation in two orthogonal directions by additive current

broadband, uncorrelated noise signals (filtered) (3-100 Hz)

Data acquisition:

identification of applied excitation force by force-current-position map

resulting displacements and velocity within the journal bearing

averaging over 16 measurements

Postprocessing:

linearity check

calculation of the complex coefficient matrix in the frequency domain by spectral densities

stiffness and damping coefficients determined from real and imaginary part

least-squares approximation over the excitation band

Figure 5: Flowchart for the identification of journal bearing coefficients.

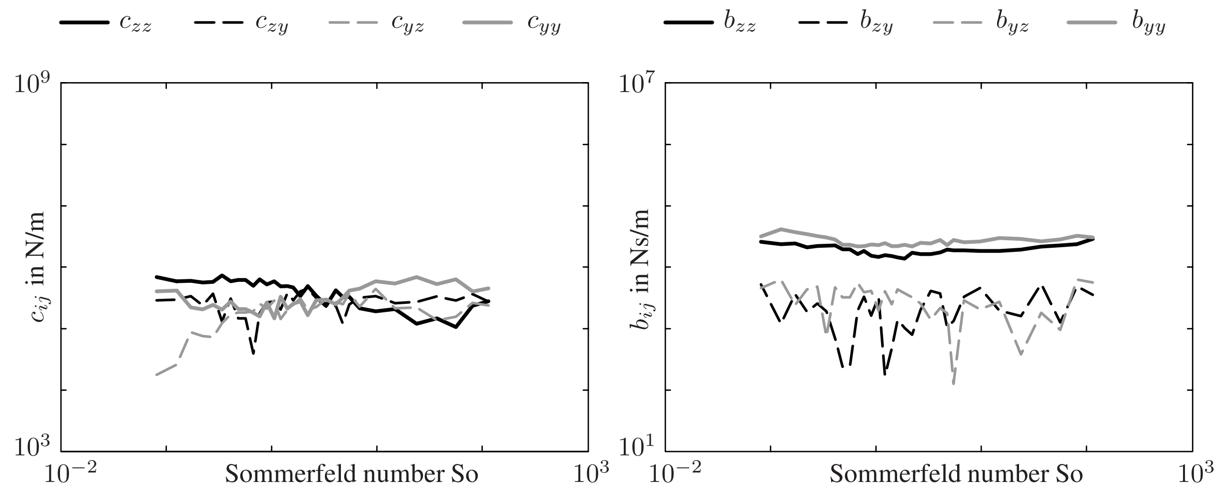

Figure 6: Identified journal bearing stiffness and damping coefficients at constant rotor speed.

excitation frequency range by a least-square method, see [7] for more details. Figure 5 shows the identification procedure in detail and Figure 6 displays an example of the experimentally determined stiffness and damping coefficients for the test journal bearing at different constant rotor speeds.

Summary

In the present study an identification procedure for the stiffness and damping coefficients of journal bearings is described. This procedure incorporates the successful application of AMBs for positioning, supporting, sensing and actuating the rotor simultaneously. For identification the real excitation force is needed. Two methods to derive a force-current-position relationship are outlined and compared: an idealized mathematical relation and a measured force-current-position map. Finally, results for applying the pre-measured map and an example of the identified stiffness and damping coefficients are given.

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