Active Magnetic Bearings as a Tool for Parameter Identification of Journal Bearings

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***Abstract*—In this overview, it is presented how Active Magnetic Bearings (AMBs) can successfully be used for the measurement of equilibrium positions and the identification of stiffness and damping coefficients of journal bearings. Therefore, the theory of force measurements in AMBs by using either a theoretical or an experimentally derived relation is provided. Different operation modes of AMBs are considered, including the operation as a magnetic actuator. The successful parameter identification of journal bearing properties using AMBs is demonstrated by measurements.**

I. Introduction

Active Magnetic Bearings (AMBs) are a powerful tool for investigating rotating and floating machine parts such as journal bearings. The challenge of testing journal bearing dynamics is the changing shaft position within the bearing depending on the rotational speed of the rotor. The changing shaft position affects possible excitation signals as well as measurement tools and methods. AMBs can handle these varying conditions quite well as they operate contact-free. They also allow for very high rotational speeds. Furthermore, AMBs do not only support the rotor, but they can simultaneously be used for the application of defined additional loads to the rotor as well as for measuring the system's reaction at the same position.

These capabilities of AMBs can advantageously be used for the identification of journal bearing properties. The most interesting parameters of journal bearings with respect to their dynamic behavior are

- the equilibrium position of the shaft in the bearing depending on the operation conditions and
- the linearized, speed dependent stiffness and damping coefficients.

Experimental identification of these parameters usually involves the measurement of the resulting journal bearing force for a defined position of the shaft. The journal bearing force can be measured either by conventional means such as piezoelectric force sensors or strain gauges at the journal bearing base or alternatively by using additional AMBs. In this work, two methods of AMB force measurements are described in Section II. An alternative approach for experimental identification of journal bearing properties is the measurement of the shaft position for a defined additional force on the rotor. Therefore different operation modes for AMB control are considered in Section III.

A. Equilibrium Position of the Shaft

To identify the equilibrium position of the shaft, Glienicke, [2], applied a static force on the bearing housing using air pressure bellows and measured the distance between bearing and shaft with capacitive position sensors. Knopf, [3], utilized AMBs for setting the rotor to a defined radial position and measured the journal bearing force by Hall sensors inside the AMBs. This method is adopted to force measurements with the AMBs by Baumann, [1], and applied to the test rig shown in Figure 1 (presented in Section IV).

B. Stiffness and Damping Coefficients of Journal Bearings

The identification of journal bearing stiffness and damping coefficients usually requires a small additional rotor excitation (either deflection or force) at the specific equilibrium position and the measurement of the resulting journal bearing forces respectively the resulting rotor deflections. Knopf, [3], and others applied a sinusoidal motion on the shaft and measured the AMB force using Hall sensors. Someya, [4], and Glienicke, [2], applied a small sinusoidal force on the bearing using a vibration exciter. Hagg and Sankey, [5], used an unbalance force for the excitation of the shaft. Nordmann, [6], applied an impact force to the shaft and performed parameter identification in the frequency domain. This method generally lacks repeatability of the applied force. Baumann, [1] and [7], introduces a method using AMBs simultaneously for excitation and measurement, which is presented in Section V.

Section VI comments on the possile sources of errors for the presented measurement techniques.

Compared to previous approaches using AMBs for journal bearing parameter identification, both excitation and measurement can simultaneously be executed in a wide frequency domain by using only one device as actuator and sensor at the same time. Once the test setup is arranged, identification can be performed very quick and with a high degree of repeatability. Advantage can be taken of the already existing position sensors and magnetic actuators in the AMB.

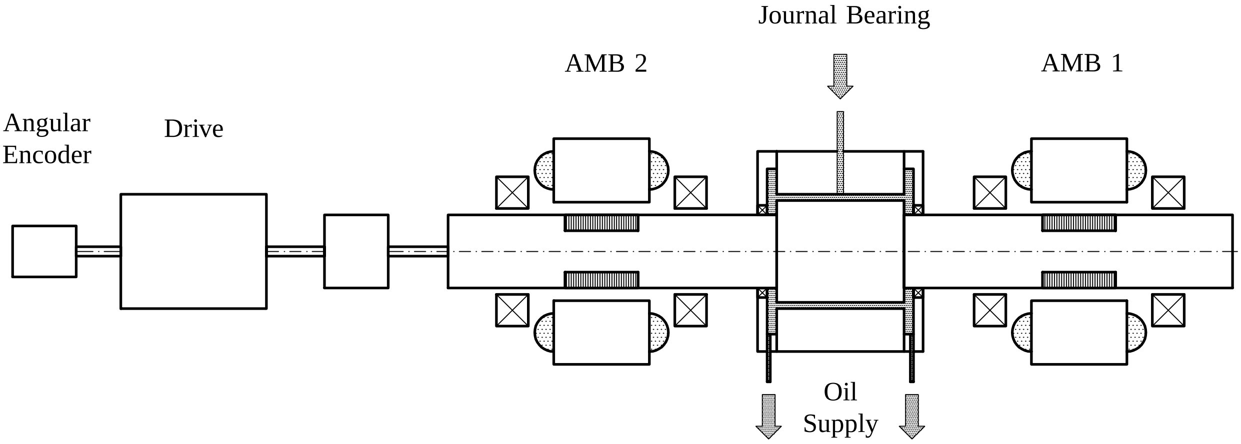

Figure 1. Test rig for parameter identification of a journal bearing using AMBs [1].

II. AMB Force Measurements

The AMB force measurement is generally based on the relation between the rotor position s, the AMB control current i and the resulting magnetic force F acting on the rotor. A first, straight-forward approach is to use the theoretical s-i-F-relation as shown for example by Hoffmann, [8]. But as AMBs are influenced by several nonlinearities, a second, experimental approach is used.

A. Theoretical s-i-F-Relation

Usually, there are at least four coils at the circumference of a radial AMB for applying magnetic forces on the rotor in four directions. A sketch of a typical AMB assembly is shown in Figure 2, [8]. The magnetic force in one direction composes from the two coils located on opposite sides,

$$F_{js} = F_{js+} - F_{js-} = k_{ML}\left[\left(\frac{i_V + i_{js}}{s_0 + s_j}\right)^2 - \left(\frac{i_V - i_{js}}{s_0 - s_j}\right)^2\right], \quad (1)$$

where s is either the z or y direction, $F_{js} = F_{jz}$ or F_{jy} respectively and $j = 1, 2$ (describing AMB 1 or 2 in the test rig, see Figure 1). Herein, s_0 is the length of the initial air gap in the magnetic bearing, i is the current in the coils, generally consisting of a control current i_{js} and a constant premagnetization current i_V. s_j is the distance of the shaft from the center position of the AMB and k_{ML} is a characteristic constant of the magnetic bearing resulting from material properties and geometry.

B. Interpolation of an Experimentally Derived s-i-F-Map

Unfortunately, AMBs are influenced by several nonlinearities which cannot be considered in equation (1). Therefore, an experimental approach for determining the AMB force was used by Hoffmann, [8], and Baumann, [1]. The idea is to map the magnetic force for numerous combinations of rotor position and control current within the reasonable operating range of the AMB. Based on this data base, the magnetic force can then be inter- or even extrapolated for arbitrary rotor positions and control currents, see Figure 3.

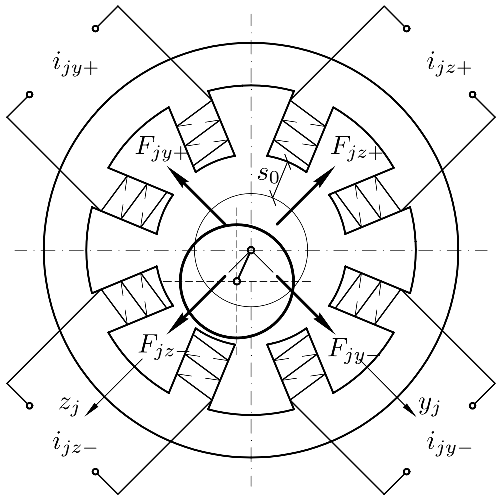

Figure 2. Assembly of an Active Magnetic Bearing [8].

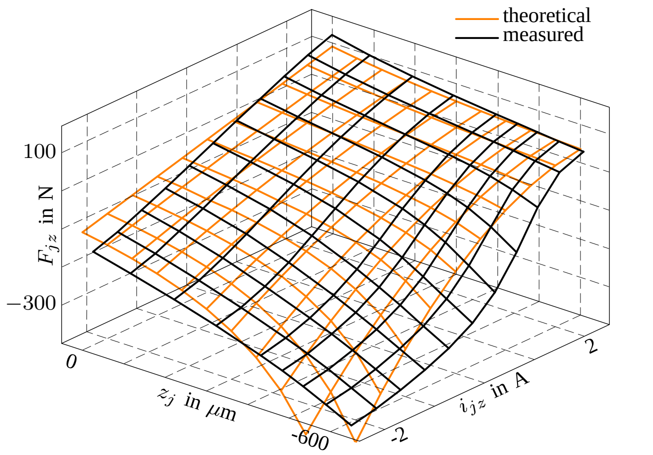

Figure 3. Comparison between the theoretical and the measured s-i-F-relation for the AMB used in [1] (with $i_{jy}=0$, $y_j=0$ and $i_V=2.5\,$A).

For the force mapping, Hoffmann and Heiland, [8] and [9], developed a test rig in which a non-rotating shaft was supported by thin legs carrying strain gauges. The shaft was then fixed in an AMB at the desired positions and a defined control current was applied on the AMB magnets. The magnetic force acting on the shaft was then measured by strain gauges and mapped. For mapping and for interpolation, it was taken advantage of the symmetry of the magnetic bearing.

III. Alternative Operation Modes of AMBs

Concerning the necessary control of AMBs and as well their application as actuators, different operation modes have to be considered:

A. Position Controlled AMBs

AMBs are generally used as contactless bearings which shall support a rotor at a specified position. In this case, the rotor shaft is forced to a defined position s_{contr} by adjusting the magnets' current i_{js} and consequently the magnetic force F_{js} by a PID control as shown in Figure 4. The possibility to change the rotor position in the AMBs allows the identification of journal bearing parameters at any position in the fluid film bearing independently from the rotor speed. Also, misalignment effects can be considered.

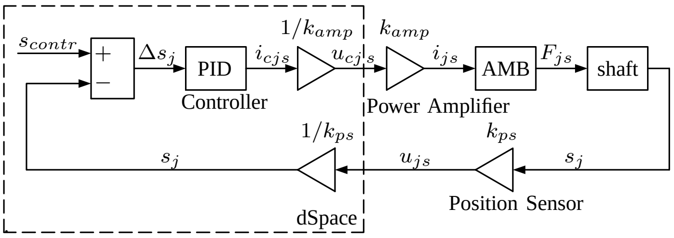

Figure 4. Position controlled Active Magnetic Bearing.

B. Force Controlled AMBs

If the AMB is primarily used as an actuator, it is more suitable to control the magnetic force of the AMB instead of the position, see Figure 5. In [10], the following approach is presented: First the magnetic force F_{js} of the magnetic actuator is estimated (F_{est}) using the AMB control current i_{cjs} and the position measurement of the shaft s_j. For this, different estimation functions F_{est} can be used: either the theoretical s-i-F-function or the experimentally derived s-i-F-map. In a second step, the estimated magnetic force of the actuator is controlled by a PID control, delivering the final control current i_{cjs} for the AMB. In [10], the control of the AMB is realized by a dSpace system and the magnetic force is estimated using the theoretical s-i-F-function (equation 1).

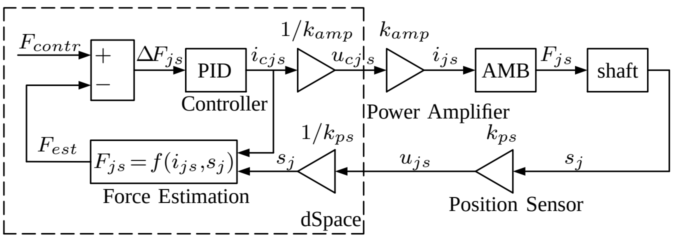

Figure 5. Force controlled Active Magnetic Bearing.

C. Force and Position Controlled AMBs

If the AMB is only used as an actuator as presented in Section B, additional bearings are needed to carry the rotor weight and to ensure the position of the shaft. At the test rig shown in Figure 1, the AMBs are primarily controlling the correct rotor position in the fluid film bearing. In order to run the journal bearing with different additional external loads applied by the AMBs, a combination of force and position control is necessary. In addition to the force control, Roy [11] controls the inclination angle of the rotor system to zero with forces acting in opposite directions in the two AMBs. This allows the shaft to move in lateral direction in the journal bearing depending on rotational speed and external load. Applying additional loads to the journal bearing using the AMBs gives the opportunity to change the operating conditions (Sommerfeld number, eccentricity, stiffness and damping coefficients) of the journal bearing at constant speed. Also, time dependent loads can be applied by the AMBs based on the s-i-F-map, simulating real load characteristics. In a test

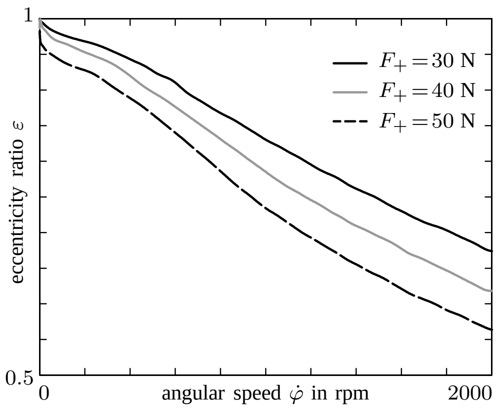

Figure 6. Eccentricity ratios ε depending on angular speed $\dot{\varphi}$ and three load conditions F_+, [11]

three different additional static forces F_+ acting in opposite direction of the gravity are applied. The resulting eccentricity ratios ε of the equilibrium positions in the journal bearing depending on the angular speed $\dot{\varphi}$ and the load conditions F_+ are shown in Figure 6.

IV. Measurement of the Journal Bearing Equilibrium Position with AMBs

The position control proposed in section III is used within the method for measuring the equilibrium rotor position in a journal bearing introduced by Knopf, [3], and revised by Baumann, [1]: For defined radial rotor positions, the resulting journal bearing force is measured with the help of the AMBs. As the resulting force in a journal bearing acts in the same direction as the applied load, the attitude angle γ_0 is known from the direction of the resulting force.

In detail, the rotor is placed at several defined radial positions along one of the magnetic axes of the AMBs by controling the AMB position. Then the rotor position, the control currents of the AMBs, and the rotational speed are measured in a first run at constant speed (with journal bearing operation, subscript n) and in a second run at zero rotational

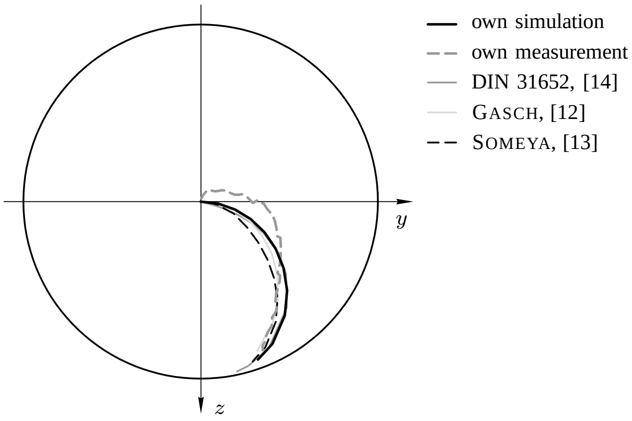

Figure 7. Equilibrium rotor position of the tested journal bearing, [1]

speed (without journal bearing operation, subscript 0). In the next step, the AMB force components F_{1z}, F_{2z}, F_{1y} and F_{2y} for both magnetic bearings 1 and 2 are either calculated using the theoretical approach (1) or interpolated using the experimentally derived s-i-F-map. The journal bearing force F_G then results from the difference between the magnetic forces in both runs (with and without journal bearing operation),

$$\begin{aligned} F_{Gz} &= F_{1z_0} + F_{2z_0} - F_{1z_n} - F_{2z_n} \\ F_{Gy} &= F_{1y_0} + F_{2y_0} - F_{1y_n} - F_{2y_n}\,. \end{aligned} \tag{2}$$

Finally, the attitude angle γ_0 is calculated from

$$\gamma_0 = \arctan\frac{F_{Gz}}{F_{Gy}}\,. \tag{3}$$

Figure 7 presents the experimental and some numerical results for the tested journal bearing in comparison to several reference data from GASCH, NORDMANN and PFÜTZNER, [12, Tab. 12.1], from SOMEYA, [13, Calc. No. 2], and from DIN 31652, [14, Tab. 18]. The experimental results agree with the reference data in an accaptable manner. Only for small rotor deflections ($\varepsilon < 0.5$), some moderate differences have to be noticed.

V. Identification of Journal Bearing Stiffness and Damping Coefficients with AMBs

A. General Measurement Method

In general, the stiffness and damping coefficients of journal bearings are linearized at the equilibrium position for small deflections, [12]. Therefore, they can be identified by applying a force on the rotor and measuring the rotor's deflection or vice versa, [2] or [4]. As the equilibrium position of the shaft is speed dependent, the stiffness and damping coefficients are speed dependent as well.

AMBs are quite useful for the identification as they allow to place the rotor at a desired (equilibrium) position and as they can apply small additional magnetic forces on the rotor simultaneously. The resulting small rotor deflections around the equilibrium position can also be measured by the AMB displacement sensors.

Baumann, [1], proposes the following method for the identification of stiffness and damping coefficients of journal bearings using AMBs:

For the measurement, the rotor is placed either at the equilibrium position depending on the rotational speed or at the corresponding radial position at one of the AMB axes. In the latter case, the final stiffness and damping coefficients have to be converted by a geometric transform connecting the equilibrium position with the chosen AMB axis. Now, small additional excitation currents i_{ejs} are defined and added to the control currents i_{cjs} for applying additional excitation forces on the rotor by the AMB. Then, the AMB currents with and without the additional excitation currents, the rotor position, and the rotational speed are measured. For each operating point, several samples are recorded.

Afterwards, the additional rotor deflections $\Delta\mathbf{r} = [\Delta y, \Delta z]^T$ around the equilibrium position are calculated by subtracting the equilibrium position from the measured rotor position. The additional excitation force $\mathbf{F} = [F_y, F_z]^T$ is obtained from the difference between the total magnetic force (resulting from the total AMB current) and the magnetic control force (only due to the control current). Because of the AMB's nonlinearities, it is not advisable to estimate the excitation forces directly from the excitation current. For describing the forces, the theoretical approach or preferably the experimental s-i-F-map can be used.

Now, the speed dependent journal bearing stiffness and damping coefficients c_{kl} und b_{kl} with $k, l = y, z$ can be determined from the complex coefficient matrix

$$\mathbf{K}(\overline{\Omega}) = \begin{bmatrix} c_{yy} + i\overline{\Omega} b_{yy} + \overline{\Omega}^2 m_{yy} & c_{yz} + i\overline{\Omega} b_{yz} + \overline{\Omega}^2 m_{yz} \\ c_{zy} + i\overline{\Omega} b_{zy} + \overline{\Omega}^2 m_{zy} & c_{zz} + i\overline{\Omega} b_{zz} + \overline{\Omega}^2 m_{zz} \end{bmatrix} \tag{4}$$

derived from the linearized equation of motion

$$\widehat{\mathbf{F}}(\overline{\Omega}) = \mathbf{K}(\overline{\Omega})\ \Delta\widehat{\mathbf{r}}(\overline{\Omega})\,. \tag{5}$$

The complex coefficient matrix can be calculated from the cross power spectral density matrix $\widetilde{\mathbf{S}}_{Fr}$ between the excitation $\mathbf{F}$ and the response $\Delta\mathbf{r}$, and from the auto spectral density matrix $\widetilde{\mathbf{S}}_{rr}$ of the responses $\Delta\mathbf{r}$ by (for details see [1])

$$\mathbf{K}^*(\overline{\Omega}) = \widetilde{\mathbf{S}}_{Fr}(\overline{\Omega})\,\widetilde{\mathbf{S}}_{rr}^{-1}(\overline{\Omega})\,, \tag{6}$$

where $\mathbf{K}^*(\overline{\Omega})$ is the complex conjugate stiffness matrix $\mathbf{K}(\overline{\Omega})$. At last, the stiffness and damping coefficients c_{kl} and b_{kl} as well as the mass parameters m_{kl} are determined by splitting the complex coefficient matrix $\mathbf{K}(\overline{\Omega})$ in its real and imaginary part,

$$\mathrm{Re}\{k_{kl}\} = c_{kl} + \overline{\Omega}^2\, m_{kl} \quad \text{and} \quad \mathrm{Im}\{k_{kl}\} = \overline{\Omega}\, b_{kl}\,, \tag{7}$$

and a final least square fitting.

B. Types of Excitation

The use of AMBs for exciting rotating systems allows a fast and reproducible identification of journal bearing parameters due to the short measurement period for each sample also at different operation conditions. Depending on the measurement focus, several types of excitation can be realized:

- Noise excitation: Providing a broadband white noise signal, all relevant frequencies are excited.
- Sine sweep excitation: Providing a sine sweep as excitation current, a specific frequency range is excited at an increased energy level compared to the noise excitation.
- Impulse excitation: A short force impulse excites all relevant frequencies of the system. The excitation time is minimal.

The proposed identification method was already tested for a simoultaneous, but uncorrelated noise excitation in two independent directions at the test rig in Figure 1. The identification method was not only applied at constant rotational speeds but also at rotor run-up processes in a short speed interval around the nominal rotational speed, [1]. Some results of these tests are shown in Figures 8 and 9. The tests with sine sweep and impulse execution at the same test rig are still going on.

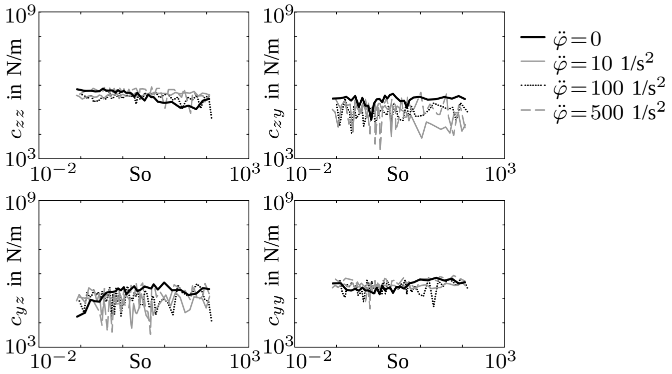

Figure 8. Measured journal bearing stiffness coefficients c_{kl} at stationary operation and during run-up processes depending on the Sommerfeld number So, [1]

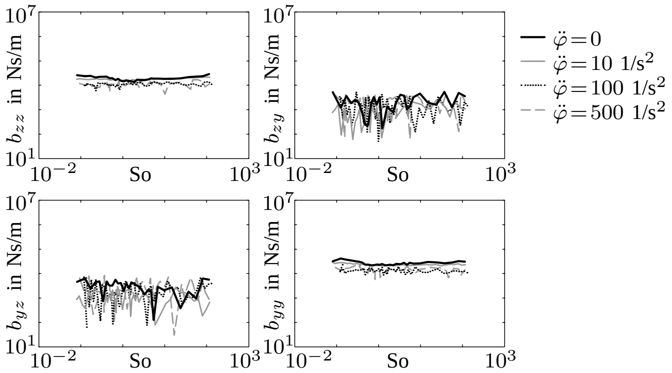

Figure 9. Measured journal bearing damping coefficients b_{kl} at stationary operation and during run-up processes depending on the Sommerfeld number So, [1]

VI. Possible Sources of Errors

The correct identification of journal bearing characteristics strongly depends on the exact measurement of the journal bearing force. As the journal bearing force is determined from the difference of the AMB forces in two separate runs respectively with/without the small excitation current, situations with almost identical or very small AMB forces are critical to the exactness. These critical situations are at the one hand especially measurements with small journal bearing forces as they occur for small Sommerfeld numbers or small rotor eccentricities in the journal bearing, and at the other hand measurements with small AMB forces which occur for large Sommerfeld numbers respectively for large rotor eccentricities in the journal bearing.

Additionally, the measurement accuracy of the AMB rotor position and the accuracy of the AMB control chain have to be kept in mind. Furthermore, the s-i-F-map has been derived experimentally by using the same technical devices and additional strain gauges.

For the used test rig, the s-i-F-map itself has a measurement inaccuracy of 6,4 %, and the force measurement by the s-i-F-map of the AMBs has an inaccuracy of 12,2 %. Summed up, the determination of the attitude angle and the journal bearing rotor position have a measurement inaccuracy of 24.4 % respectively 5 %. This means that the above experimental results are quite acceptable.

Besides the measurement inaccuracies, the influence of oil grooves and other form deviations of the journal bearing have to be considered, and the exact positioning of particularily the AMB sensors has to be ensured.

VII. Summary

Active Magnetic Bearings are a powerful tool for the parameter identification of journal bearings. The position of the shaft in the bearing and the loading of the shaft can be set independently in a certain range. Additionally, it is possible to apply a reproducible excitation on the bearing at the same time easily.

The equilibrium position and the stiffness and damping coefficients of a journal bearing have been measured successfully by using AMBs as support, sensors and actuatos simultaneously. For this, it was necessary to determine the magnetic forces in the AMB. It has been shown that the presented idealized nonlinear equation is not sufficient, but the use of an experimentally derived s-i-F-map delivers good results. This s-i-F-map can be measured with a certain effort.

The described methods using AMBs for parameter identification in rotating systems can be adapted to other applications, for example squeeze film dampers, tilting pad journal bearings or components with elastomers.

References

[1] K. Baumann, *Dynamische Eigenschaften von Gleitlagern in An- und Auslaufvorgängen.* Dissertation TU Darmstadt, 2010.

[2] J. Glienicke, *Feder- und Dämpfungskonstanten von Gleitlagern für Turbomaschinen und deren Einfluss auf das Schwingungsverhalten eines einfachen Rotors.* Dissertation TH Karlsruhe, 1966.

[3] E. Knopf, *Identifikation der Dynamik turbulenter Gleitlager mit aktiven Magnetlagern*. Dissertation TU Darmstadt, 2001.

[4] T. Someya, *Stabilität einer in zylindrischen Gleitlagern laufenden, unwuchtfreien Welle – Beitrag zur Theorie des instationär belasteten Gleitlagers.* Dissertation TH Karlsruhe, 1962.

[5] A. C. Hagg and G. O. Sankey, "Some dynamic properties of oil-film journal bearings with reference to the unbalance vibration of rotors," *Journal of Applied Mechanics*, pp. 302–306, 1956.
[6] R. Nordmann and K. Schöllhorn, "Ermittlung der Feder- und Dämpfungskonstanten von Gleitlagern durch parametrische Identifikation," *VDI-Berichte*, vol. 381, 1980.
[7] K. Baumann, F. Dohnal, and R. Markert, "Active magnetic bearings as actuators and sensors in journal bearing measurements," in *Proceedings on The Twelfth International Smposium on Magnetic Bearings (ISMB12)*, 22.-25. August 2010, pp. 686–693.
[8] K.-J. Hoffmann, *Integrierte aktive Magnetlager*. Dissertation TU Darmstadt, 1999.
[9] R. Heiland, *Konstruktion einer Vorrichtung zur Messung von Magnetlagerkräften.* Studienarbeit TU Darmstadt, 1995.
[10] G. B. Daniel, K. L. Cavalca, and O. Alber, "Application of active magnetic actuator for excitation of rotating systems supported by tilting pad journal bearing," in *22nd International Congress of Mechanical Engineering (COBEM 2013)*, 2013.
[11] H. Roy, *Force Control of Magnetic Bearing*. Master-Thesis TU Darmstadt/IIT Delhi, 2013.
[12] R. Gasch, R. Nordmann, and H. Pfützner, *Rotordynamik*, 2nd ed. Berlin: Springer-Verlag, 2006.
[13] T. Someya, *Journal-Bearing Databook*. Berlin: Springer-Verlag, 1989.
[14] "Gleitlager – hydrodynamische radial-gleitlager im stationären betrieb: Berechnung von kreiszylinderlagern." Berlin, 1996.