ACTIVE MAGNETIC BEARINGS FOR FAULT DETECTION IN A CENTRIFUGAL PUMP

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ABSTRACT
Active magnetic bearings (AMB), a typical mechatronic product, have been successfully applied in industrial turbomachinery. They are well suited to operate contactless as actuator and sensor elements in rotating machinery. The aim of the presented projects of the Special Research Program (SFB 241) supported by the German Research Council (DFG) is to use active magnetic bearings as an identification and diagnosis tool for turbomachines. The identification and diagnosis procedures are based on frequency response functions. For this type of diagnosis using transfer functions, an accurate force measurement as well as a precise model are crucial. The paper compares force measurement results and achievable accuracies of radial and axial magnetic bearings using different measurement techniques over a large operating range.

INTRODUCTION
The advances of turbomachinery as well as their numbers and variety have increased extremely over the last few decades. The users of these machines expect, that their machines are running safe and reliable and that they have a high efficiency and availability. In order to satisfy these requirements an integrated fault detection and diagnosis becomes increasingly important for these machines.

Today, monitoring systems are normally not an integral component of turbomachines. With these failure detection systems, the relative and/or absolute motions of the rotor are measured as output signals. After signal processing, certain features (threshold values, orbits, frequency spectra etc.) are created from the measured data. With the deviations of these features from a faultless initial state, the diagnosis attempts to recognize possible faults. The difficulty with these procedures is that the causes of the modifications of the output signals can not be detected clearly. The reason might either be a change of the process or a modification of the system itself.

An improvement of the existing diagnostic techniques can be achieved by using AMBs. They are well suited to operate contactless as actuator and sensor elements in rotating machinery. Hence, an integrated diagnostic system can be developed based on input/output relations instead of only measuring the system responses (fig. 1).

In various applications the feasibility and profitability of using AMBs in turbomachines have been proven (e.g. [1], [8]). Furthermore, the application of AMBs as a force measurement tool to determine hydraulic forces acting on pump rotors was validated by Guinzburg, *et al.* [6] and Baun, *et al.* [2]. Both showed good agreements between experimental and theoretical data. Pottie, *et*

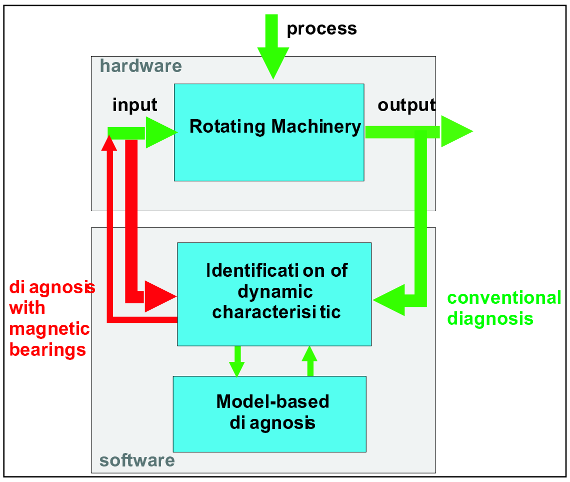

FIGURE 1: Failure diagnosis with AMBs.

al. [10] went further and evaluated measured frequency response functions (FRF) for the identification of fluid-structure-interaction forces. They suggested that a FRF measurement between on-line measured magnetic forces and displacements can potentially be used to perform fault diagnosis. Especially, the necessary improvement in the quality of the force measurement to yield accurate FRFs was pointed out by Pottie, *et al.* [10].

FORCE MEASUREMENT

Basically, the magnetic bearing force can be computed from

$$\vec{F}_{AMB} = \frac{A_{pole}}{2\mu_0}\sum_{k=1}^{8}\vec{B}^2_{pole_k}, \qquad (1)$$

with A_{pole} the cross-section area of the poles and μ_0 the permeability of vacuum.

The different force measurements in an AMB can now be divided into two main groups. The first is based on the measurement of coil currents and rotor displacements (*i*-*s*-method, reluctance network model) and the second method uses the direct measurement of the magnetic flux density B_{pole} with a Hall sensor at each pole. The drawback of the latter method is, that the air gap has to be enlarged to integrate the Hall sensors resulting in a decrease of load capacity of the bearing. To partially overcome this problem, a modified force measurement was presented by Gähler, *et al*. [5], where only the north poles were equipped with Hall sensors, while the fluxes at the south poles were computed using an on-line approximation.

If a direct measurement of the flux density is not possible or desired and if the AMB is driven in differential mode, eq. (1) can be linearized for small rotor displacements around the center

$$F_x = k_i i_x + k_s s_x \qquad (2)$$

where i_x is the control current and s_x the rotor displacement in the *x*-direction. The constants k_i, k_s depend on the chosen design point of the magnetic bearings (bias current i_0 and air gap s_0). A more advanced force measurement based on coil currents and rotor displacements is a reluctance network model. The linear network model used here accounts much better for eccentric rotor positions and cross-coupling than the *i*-*s*-method. Depending on the requirements, the model can further be extended to consider leakage and fringing effects, eddy currents, nonlinear material behavior, and hysteresis etc. as it is described, e.g. in Meeker, *et al.* [9] and Springer, *et al.* [11].

Force Measurement Results for the Radial Bearing

In the following, results of the four different force measurement techniques described above were applied to the AMBs. For each method the calibration was performed for an operating range of up to 25% of nominal bearing clearance and of a force level up to maximum bearing force. The use of the measurements over the entire force range without giving away any capacity is the most attractive case with respect to a possible industrial application. The key data of the radial AMBs is summarized in table 1. The stator component of the bearing is composed of separated horseshoe-shaped SiFe-laminations with 8 magnetic poles.

TABLE 1: Key data of the radial and thrust bearings

	Radial Bear.	Thrust Bear.
Bearing clearance	1.3 mm	1.2 mm
Bias current	4 A	3.6 A
Coil turns	306	286
Pole area	864 mm^2	2720 mm^2
Maximum force in pole direction	750 N	2200 N

Measured data was gained, while the rotor was floating in the AMBs and different external static forces F_{Ref} were applied. An external load cell was used to obtain the reference force F_{Ref}. More details about the calibration can be found in the work done by Förch, *et al.* [4] and Knopf, *et al.* [7].

TABLE 2: Static force error of the radial bearing.

	Centric Rotor range: ± max. bearing force	Eccentric Rotor range: ± max. bearing force
i-*s*-method	9%	9%
reluctance network	8%	7%
4 Hall sensors + approx.	2%	3%
8 Hall sensors	<1%	3%

Table 2 and Fig. 2 show the force error for the centric load case. The percentage values of the force errors are related to the maximum bearing force. Data presented on the 8 Hall sensor method is obtained from Knopf, *et al.* [7]. It can be seen, that the force measurements of the flux based methods (using 4 respectively 8 Hall sensors) are clearly more accurate than the current-displacement based ones (*i*-*s*-method, reluctance network). Analogues can be found for the eccentric case, where the rotor was moved out of the center of about 25% of the air gap. The error of the current-displacement based methods stays about the same. But it should be mentioned, that there is already a substantial force error of the *i*-*s*-method at

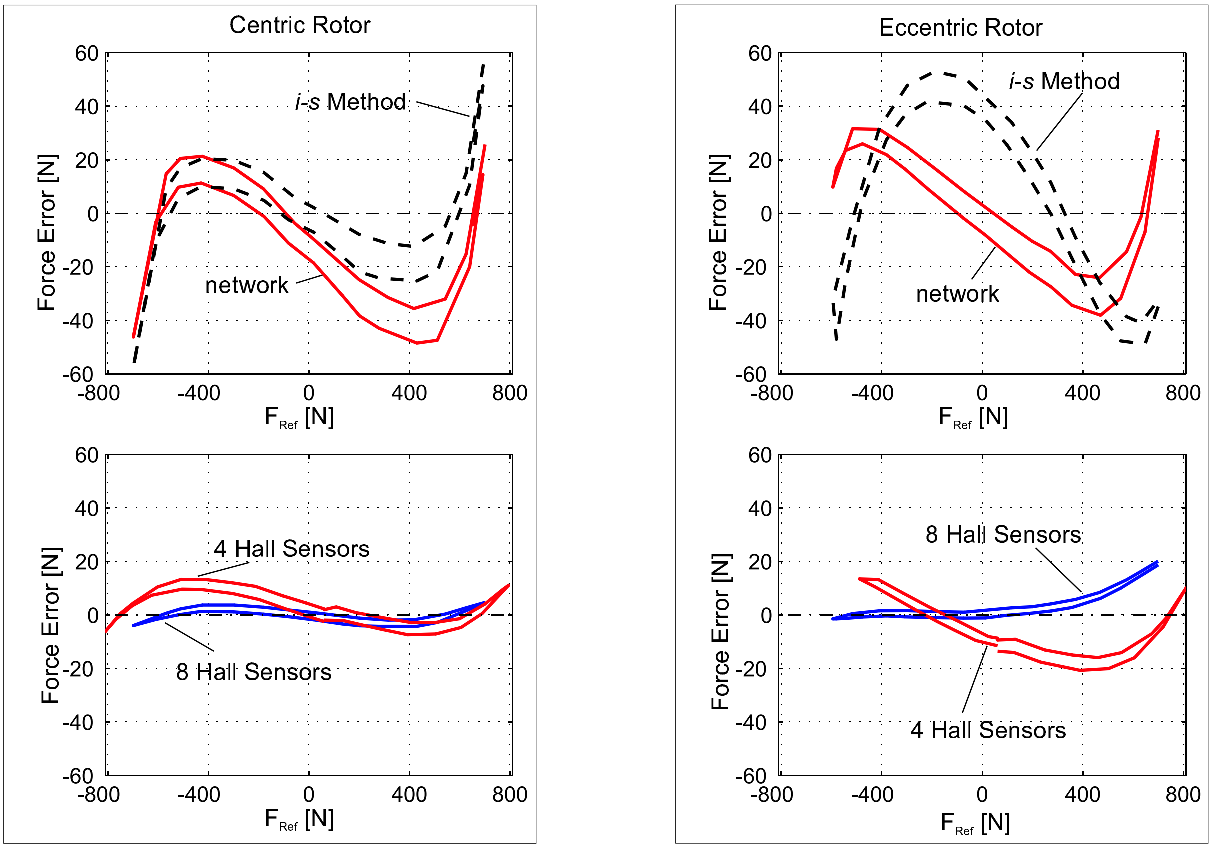

FIGURE 2: Comparison of four different force measurements for the radial bearing.

small loads. That is due to keeping the error relatively small at maximum loads. The 8 Hall sensor method compared to the 4 Hall sensor method is a little more accurate and cross-coupling effects are less.

In summary, the advantages of the flux based methods are:

- they are clearly more accurate,
- influences of hysteresis and non-linearities of the magnetic material, power amplifier, etc. are diminished or completely eliminated,
- on-line implementation is possible (with some restrictions for the 4 Hall sensors method + approx.),

while the advantages for the current-displacement method are:

- no additional hardware is needed,
- *i-s*-method is the easiest one to implement in an on-line procedure (restricted applicable for the network).

Force Measurement Results for the Thrust Bearing

The two force measurement methods used for the thrust bearing are again based on current-displacement signals (*i-s*-method) respectively on flux signals. The thrust bearing only consists of a single axis, i.e. no cross-coupling occurs. Hence, it is sufficient to integrate the Hall sensors only at the north pole. To account for the tilting of the disc, four Hall sensors are installed at the north pole. The key data of the thrust bearing is also listed in table 1.

TABLE 3: Static force error of the thrust bearing.

	Centric Rotor range: max. bearing force	Eccentric Rotor range: max. bearing force
i-s-method	32% (1.5%)*	34% (8%)*
Hall sensors	<1%	1.4%

* when force range restricted to 50% of max. bearing force

A similar calibration procedure as for the radial bearings was performed. The results are shown in table 3 and fig. 3. The force error of the Hall sensor method is less than 1% for forces up to the maximum bearing force and eccentricities of about 15% of nominal bearing clearance. It is noticeable, that the non-linearity of the iron does not influence the Hall sensor method at all, as it is expected. For getting acceptable accuracies for the *i-s*-method the operating range had to be restricted to 50% of maximum bearing force, because with increasing loads or eccentricities the force error of the *i-s*-method escalates dramatically. Further should be mentioned, that with a thrust bearing configuration like it is used here

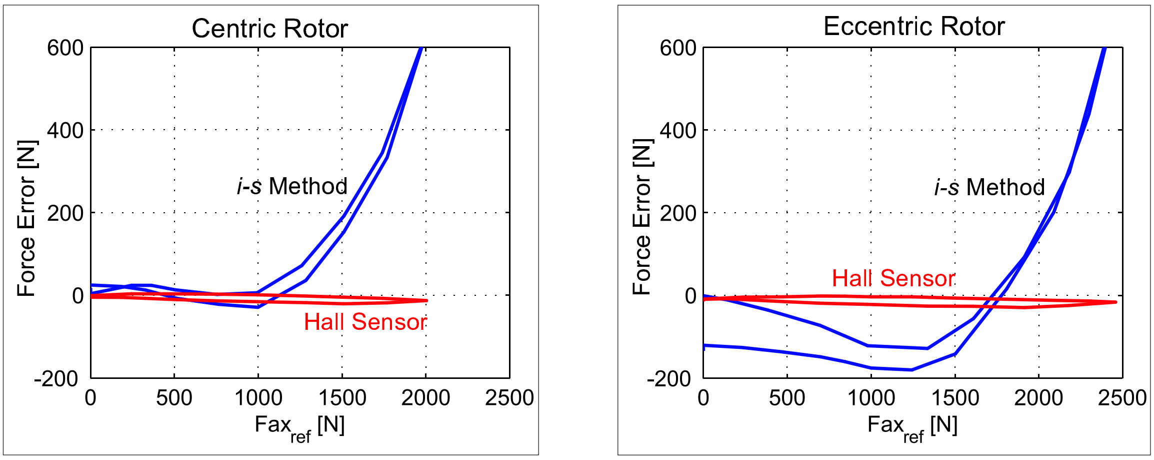

FIGURE 3: Comparison of two different force measurements for the thrust bearing.

(see fig. 5), the accuracy of the *i*-*s*-method is strongly depending on temperature elongation of the shaft, due to the dependence of the k_i, k_s-constants on the nominal air gap.

MODELLING OF THE MECHATRONIC SYSTEM

At first, a detailed model of the mechatronic system is required for the development of a robust controller. The robustness has to account for changes in the dynamic characteristic of the plant due to altering operating points (part load, over load), running speeds, and fault levels. Secondly, the model based diagnosis requires a detailed rotordynamic model including fluid-structure-interaction elements to detect possible faults.

The mechatronic system investigated consists of the pump rotor, the (fluid) forces acting on the rotor, and the active magnetic bearing system, i.e. magnetic actuators, amplifiers, controllers, and position sensors (fig. 4).

The rotor of the pump is modelled through a finite element (FE) model. The model update of the plain mechanical structure is performed by an experimental

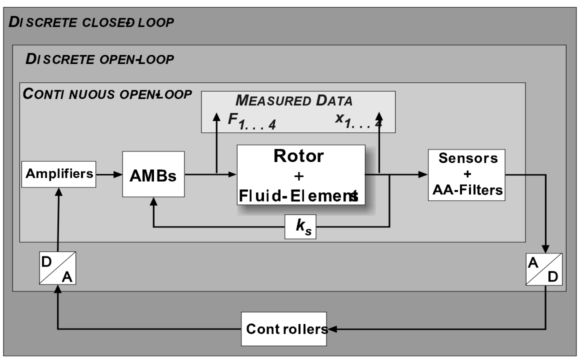

FIGURE 4: Scheme of the model of the mechatronic system.

modal analysis leading to an accurate rotor model. The fluid forces acting on the rotor in the seals (impeller, balance piston) are expressed by rotordynamic coefficients. These coefficients are computed using finite-difference methods.

For a further usage in the modeling procedure of the mechatronic system, the FE-model including the fluid forces in the seals is transformed into a state-space formulation. To reduce the order of the FE-model, a simple modal truncation is performed. Hence, the rigid body modes and the first four bending modes are used to construct the state-space model. Modal damping is used to model the structural damping and a value of 0.5% was arbitrarily chosen. The linear state space model obtained, couples the radial degrees of freedom due to gyroscopic and fluid-structure-interaction effects. The axial degree of freedom is decoupled and consequently modelled separately.

The AMBs are described using eq. (2), with the constants k_i, k_s determined from calibration. The switching power amplifier used to drive the magnetic coils is simply modelled by a constant factor. The position sensors are eddy current probes and modelled using low pass filters (PT_2-characteristic).

The open loop plant is assembled to the cascade of the different subblocks shown in fig 4. The continuous description of the system is then discretisized with a sampling frequency of 3.5 kHz. The necessary controller design to stabilize the unstable rigid body modes is based on the developed model and follows a PID-strategy with some additional filters accounting for the flexible modes.

Within the presented research project, a test rig of a magnetically suspended centrifugal pump (fig. 5) was designed in parallel. The modular concept of the design

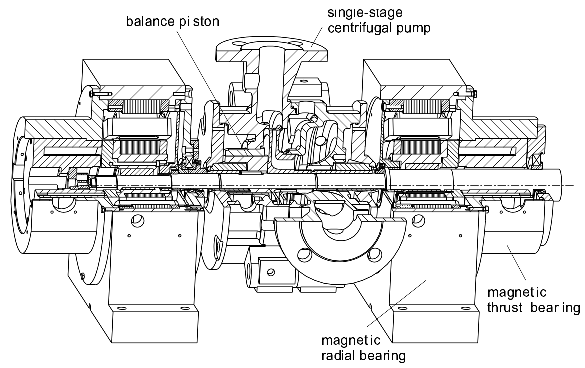

FIGURE 5: Scheme of the test rig of the single-stage pump in AMBs.

enables an easy extension of the single-stage to a multi-stage pump system, both of which are subject for investigations. This test rig is used to validate and to demonstrate the performance of the developed model based diagnosis methods. For the validation of the entire mechatronic model, the measurement of the compliance function of the closed loop was performed. The good agreement between the measured and simulated compliance function can be determined from fig. 6. The remaining deviations of the simulated curve to the measured one results from the dynamics of the substructure (e.g. housing), which is not modelled yet.

IDENTIFICATION AND FAULT DETECTION

As described earlier in this paper, the diagnosis process mainly depends on two procedures, namely on the accurate numerical modeling of the rotordynamic system (with and without failures) and on the experimental identification of the dynamic behavior of the system. Eq. (3) shows the linear dynamic description of a rotor with fluid structure coupling elements, where the matrices $\boldsymbol{M}$, $\boldsymbol{D}$ and $\boldsymbol{K}$ are time-invariant, but depending on running speed and operating conditions.

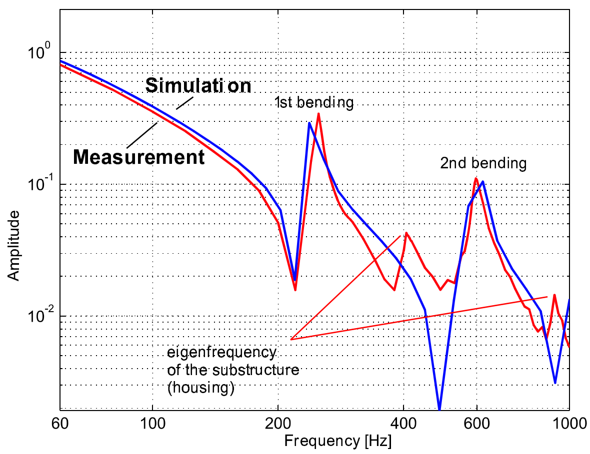

FIGURE 6: Compliance function of the closed loop of the mechatronic system.

$$\boldsymbol{M}\ddot{\boldsymbol{x}} + \boldsymbol{D}\dot{\boldsymbol{x}} + \boldsymbol{K}\boldsymbol{x} = \boldsymbol{F} \tag{3}$$

The identification of the system parameters is usually done by measuring input-output relations. In most cases and especially for flexible structures, it is much easier to use defined forces as an input and measure the output displacements, rather than vice versa. In the frequency domain this leads to compliance functions:

$$\hat{\boldsymbol{x}}(\Omega) = \underbrace{[\boldsymbol{K} - \Omega^2\boldsymbol{M} + i\Omega\boldsymbol{D}]^{-1}}_{= \overline{\boldsymbol{H}}(\Omega)} \hat{\boldsymbol{F}}(\Omega) \tag{4}$$

In normal configuration with two radial bearings and a thrust bearing, five linearly independent excitation patterns can be produced by the AMBs:

$$\begin{bmatrix} \hat{x}_1 \\ \hat{x}_2 \\ \hat{x}_3 \\ \hat{x}_4 \\ \hat{x}_5 \end{bmatrix} = \begin{bmatrix} \overline{H}_{11} & \overline{H}_{12} & \overline{H}_{13} & \overline{H}_{14} & \overline{H}_{15} \\ \overline{H}_{21} & \overline{H}_{22} & \overline{H}_{23} & \overline{H}_{24} & \overline{H}_{25} \\ \overline{H}_{31} & \overline{H}_{32} & \overline{H}_{33} & \overline{H}_{34} & \overline{H}_{35} \\ \overline{H}_{41} & \overline{H}_{42} & \overline{H}_{43} & \overline{H}_{44} & \overline{H}_{45} \\ \overline{H}_{51} & \overline{H}_{52} & \overline{H}_{52} & \overline{H}_{54} & \overline{H}_{55} \end{bmatrix} \begin{bmatrix} \hat{F}_1 \\ \hat{F}_2 \\ \hat{F}_3 \\ \hat{F}_4 \\ \hat{F}_5 \end{bmatrix} \tag{5}$$

For example, if the AMB produces a harmonic force in the direction 1 (fig. 7) at discrete frequencies Ω and if the displacements of the rotor are measured in the direction $1-5$, one column of the transfer matrix can be identified. For linear systems with symmetric matrices, this measurement is sufficient to get a complete modal description of the system, but some fluid-structure-elements can have matrices that show no symmetrical characteristics. In this case, the complete transfer matrix $\overline{H}(\Omega)$ has to be measured. Instead of fitting the physical parameters assembled in M, D, K it is much easier to identify an equivalent set of system parameters, the so called modal parameters. These parameters consist of a set of natural frequencies ω_i, damping values α_i and corresponding eigenvectors ϕ_i (i = 1, 2,...,2N) and offer also a complete description of the dynamic system. The description of the transfer function $\overline{H}(\Omega)$ can also be formulated in the modal space, as described in [3].

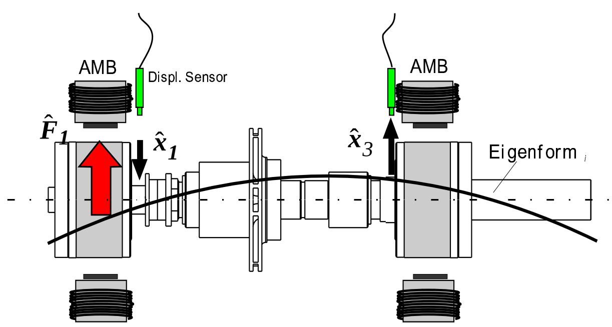

FIGURE 7: Identification scheme using AMBs (Dof x_2, x_4 and x_5 not shown)

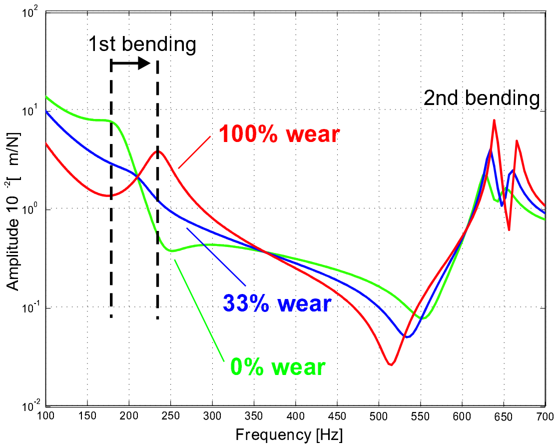

FIGURE 8: Change of the compliance functions of the plant at 3000rpm due to different states of wear of the balance piston.

The investigations are starting with the detection of the wear of the balance piston, where some simulation results are presented in fig. 8. The simulated compliance functions represent three different worn out states of the piston (new, 33% worn, and worn out). The influence upon the rotordynamic behaviour of the system can clearly be seen. The detection of a deviation from some initial state, is the starting point for the following diagnosis procedure. During the diagnosis process, several transfer functions are calculated by inserting different faults into the complete rotordynamic model to find the best transfer function meshing the measured ones. Afterwards, different faults like wear of seals, a crack in the rotor, loosening parts, cavitation, increase of the imbalance, etc. are planned to be determine regarding their type, location, and extend.

SUMMARY

In this paper the model based diagnosis using active magnetic bearings in a centrifugal pump is introduced. For this type of diagnosis using frequency response functions an accurate force measurement is crucial. The paper compares force measurement results and achievable accuracies of radial and axial magnetic bearings using different measurement techniques over a large operating range. The most accurate method is the Hall sensor method. The achieved accuracy of the force measurement seems to be sufficient to work with frequency response functions. Furthermore, the linear state-space model of the mechatronic system representing the faultless initial state is presented.

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