FAULT DIAGNOSIS IN ROTATING MACHINERY USING ACTIVE MAGNETIC BEARINGS

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ABSTRACT

The number of rotors in active magnetic bearings (AMBs) is increasing over the last few years. These systems offer a great variety of advantages compared to conventional systems. The aim of the presented project is to use the AMBs together with a developed built-in software for identification, fault detection and diagnosis in a centrifugal pump. A single stage pump representing the turbomachines is investigated. During full operation of the pump, the AMBs are used as actuators to generate defined motions respectively forces and as very precise sensor elements for the contactless measurement of the responding displacements and forces. In the linear case, which means small motions around an operating point, it is possible to derive compliance frequency response functions from the acquired data. Based on these functions a model-based fault detection and diagnosis is developed, which facilitates the detection of faults compared to state-of-the-art diagnostic tools, which are only based on the measurement of the systems outputs, i.e. displacements. In the paper, the different steps of the model-based diagnosis, which are modeling, generation of significant features respectively symptoms, fault detection, and the diagnosis procedure itself are presented and it is shown how two exemplary faults are detected and identified in particular.

INTRODUCTION

In various technical areas rotating machinery are in operation, like turbines, pumps, compressors, motors, generators, etc. Users 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 failure detection and diagnosis becomes increasingly important for these machines.

On the other hand, the demand on using magnetic bearings in turbomachines has strongly increased over the last five years. This is stated in several latest publications like [1], [2], and in the proceeding of the last magnetic bearing conference [3]. Most important hereby are the active magnetic bearings (AMBs), a typical mechatronic system. Rotors in AMBs already offer a variety of advantages compared to conventional systems. Some of them are the tuning possibilities for stiffness and damping, the absence of wear, the reduction of friction, the high running speeds, and possible unbalance compensation. In various applications the feasibility and profitability of using AMBs in turbomachines have been demonstrated, e.g. [4], [5].

However, there is much more potential in such systems than using them as a simple bearing. AMBs have to be used as sensor and actuator elements, too. They work in these new generation of turbomachines as an integrated identification and diagnosis tool. In this way, it will be possible to design new machines with higher performance, higher reliability and longer lifetime.

Equation (1) shows the linear description of the dynamic behavior of a rotor with stiffness-, damping- and inertia- characteristics, expressed by the matrices **M**, **D** and **K**. We assume, that the rotor matrices are time-invariant, but depend on the running speed and the actual operating condition.

$$\mathbf{M}\ddot{x}(t) + \mathbf{D}\dot{x}(t) + \mathbf{K}x(t) = \boldsymbol{F}(t) \tag{1}$$

The forces $\boldsymbol{F}(t)$ are considered as the system inputs and the displacements $x(t)$ as the system outputs. When input-output relations are considered in the frequency domain, equation (1) can be transformed to the following complex frequency response function

$$\hat{x}(\omega) = (\mathbf{K} - \omega^2\mathbf{M} + j\omega\mathbf{D})^{-1}\bar{F}(\omega) = \bar{\mathbf{H}}(\omega)\bar{F}(\omega) \quad (2)$$

where $H_{kl}(\omega)$ is the system response (amplitude and phase) of the displacement $x_k(\omega)$ due to a unit force excitation $\hat{F}_l(\omega)$. The frequency ω represents the excitation frequency, which is not necessary the running speed.

Today, monitoring and diagnosis systems are normally not an integral component of the turbomachines. These fault detection and diagnosis systems mainly measure the output signals $x(t)$, the relative and/or absolute motions of the rotor. 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 non-faulty initial state, faults are detected. Subsequently, 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 respectively the input or a modification of the system itself.

If AMBs are already integrated in the turbomachines, an improvement of the existing diagnostic techniques can be achieved without any additional hardware installation. AMBs are well suited to operate as sensor elements, being able to measure not only the outputs, but also the inputs. And furthermore, AMBs can be used as actuators to excite the system with defined signals, consequently receiving input-output relations. Hence, AMBs are able to measure all three components of equation (2), the input, the output, and the system characteristics itself. It should be mentioned, that the inputs and outputs of the system are restricted to the degrees of freedom at the bearings (figure 4 and equation (3)).

PRINCIPLE OF MODEL-BASED FAULT DIAGNOSIS

In general, the methods used for fault[1] diagnosis are classified into two main groups: the ones based on mathematical or physical models and the ones only based on measured signals. Classical diagnostic methods are signal-based. But as Natke [6] already stated: 'Such a verified and validated mathematical model with sufficiently small errors is the best available knowledge base' (p. 14). Hence, the work presented here concentrates on model-based methods.

1. Fault is any deviation from the normal behavior of the plant.

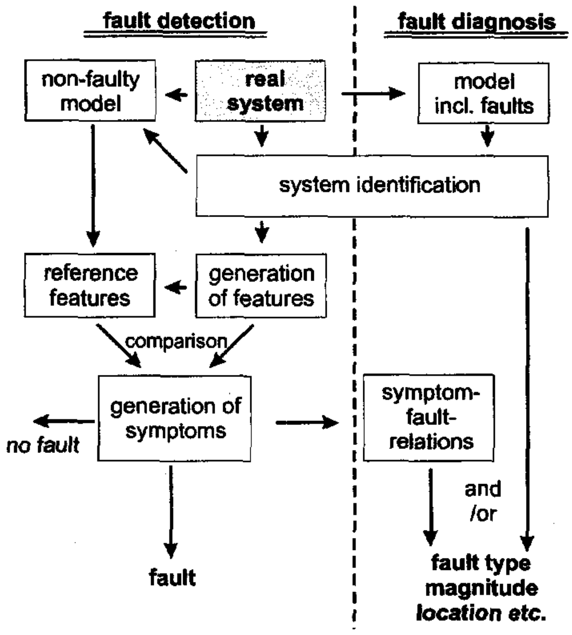

FIGURE 1. General concept of model-based diagnosis.

The entire model-based diagnosis procedure is split up into two tasks (figure 1), i.e. the fault detection and the actual diagnosis. The faults acting on the system can further be divided into two groups. One group is altering the system $\mathbf{H}(\omega)$ and the second one is altering the input F, respectively the output x. These groups are often referred to as multiplicative respectively additive faults [7]. In this presentation, the focus lies on faults belonging to the first group.

The main steps of the *fault detection* are:

- Data acquisition and signal processing
- Modeling of the non-faulty plant
- Generation of features
- Generation of symptoms respectively residuals
- Quality criterion - no fault / fault detected

The most important part of the entire diagnosis procedure is the generation of features. The generation of the residuals of the features respectively the symptoms is then a comparison between the features of the non-faulty state and the actual state of the plant [8]. The comparison results in a statement that something is wrong and a fault occurred respectively that everything is still alright.

The main steps of the *fault diagnosis* procedure are:

- Isolation: Determination of the faulty component
- Deriving symptom-fault relations

and / or

- Modeling of possible faults with their effects
- Identification of fault magnitude, location etc.

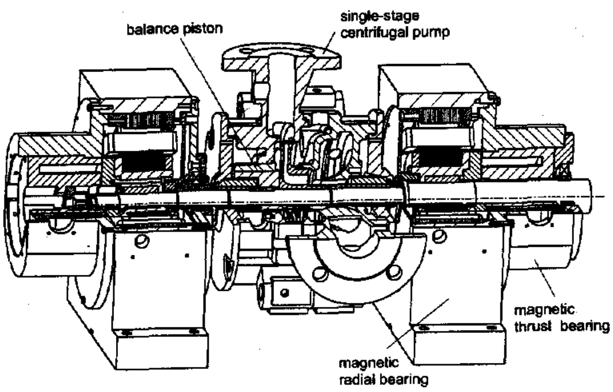

FIGURE 2. Scheme of the single-stage pump in AMBs.

The isolation of a fault is relatively simple, due to the possibility of the AMBs to derive separated features based on the output x, input F, and system $\bar{\mathbf{H}}(\omega)$ information. The main part of the diagnosis is to develop the relationships between the changes of the symptoms and the acting fault(s). One way in finding the most probable fault is to use inference mechanisms like fuzzy-logic or classification methods like neural network algorithms, or a combination of both. These procedures are not covered within this paper. In addition or instead of using the change of symptom pattern to identify a possible fault, a model-aided diagnosis procedure is used. This method will be presented for two exemplary faults. The basic idea is to integrate different fault models into the updated non-faulty reference model, study their effects and determine the most possible fault.

EXPERIMENTAL SETUP

Within the presented research project, a magnetically suspended centrifugal pump serves as a representative for the turbomachines. The designed test rig is used to validate and to demonstrate the performance of the developed model-based diagnosis.

The modular concept of the design enables an easy extension of a single-stage to a four-stage pump system, both of which are subject for investigations. The presented paper is restricted to the single-stage pump with respect to the experimental data. However, the developed method of the diagnosis procedure can be transferred to the four-stage pump as well as to any other turbomachine running in AMBs.

Figure 2 shows the scheme of the single-stage pump in AMBs. The pump is located between two active magnetic bearings levitating the rotor in five degrees of freedom. Besides the replacement of the conventional roller bearings through active magnetic bearings, the original pump system, especially the hydraulic part, remains unchanged.

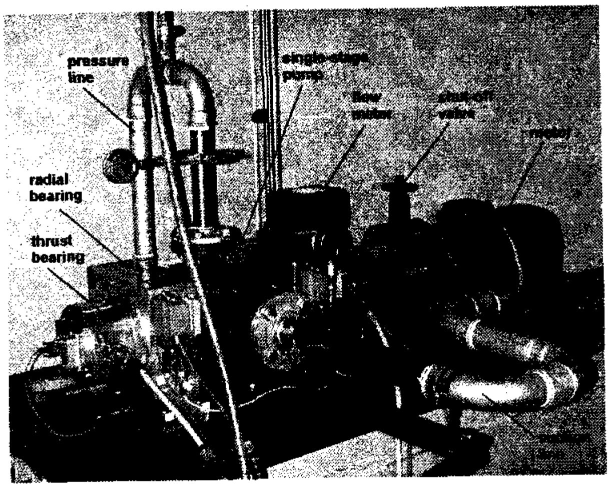

FIGURE 3. The test rig of the single-stage pump.

TABLE 1: .Technical data

Pump Data		
rotational speed [rpm]		2950
flow rate [m3/h]		18
head [m]		21
radial clearance piston seal [mm]		0.09
radial clearance impeller seal [mm]		0.12
Magnetic Bearing Data		
	radial	axial
air gap [mm]	1.3	1.2
AMB-Force (per axis) [N]	750	2200
premag. current [A]	4.0	3.6
windings per pole pair [-]	306	286
cross section area of pole [mm^2]	864	2720

In addition to the two mechanical seals sealing up the hydraulic part, the pump contains two contactless annular seals. One is placed at the suction side and one at the pressure side of the impeller, latter is the balance piston. The rotor is coupled to the motor by a flexible membrane coupling. It should be mentioned that the dimensions of the magnetic bearings are largely oversized and not optimally designed for the pump.

Some technical data of the pump as well as of the AMB systems is presented in table 1. The entire test rig configuration is shown in figure 3. The hydraulic

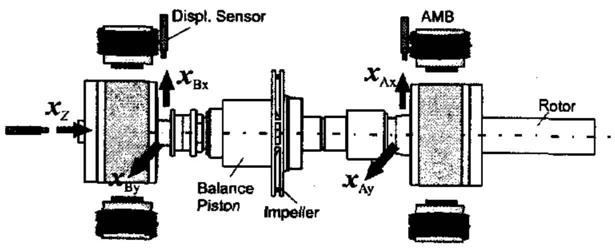

FIGURE 4. Actuator and sensor locations in the x-z-plane.

periphery in which the pump is embedded is demonstrated. It is an open circuit with the reservoir not shown. Water serves as process medium with a temperature kept constant at 20°C.

DATA ACQUISITION AND SIGNAL PROCESSING

Sensor Capability of the AMB system

An important role in the development of the fault diagnosis comes up to the precise measurement of the input/output signals (displacements and forces). The contactless measurement of displacements is state-of-the-art using inductive or eddy current sensors, whereas the contactless measurement of the forces acting on the rotor is more difficult. Latest results showed that with the integration of Hall probes into the air gap of radial AMBs [9], [10] and of axial AMBs [11] sufficient accuracy of the force measurement can be gained, too.

A detailed comparison of the reachable accuracies of the force measurement methods for different operating ranges can be found in [11], [12].

Actuator Capability of the AMB system

The AMB-system is digitally controlled and hence offering a great flexibility. With an on board sine wave generator, the AMB system can be excited during regular operation with defined frequencies from 0 - 1 kHz. The performance of a step sine procedure and sequential excitation of all five inputs of the system with five linear independent force excitation patterns, leads to measurable 25 transfer functions:

$$\begin{bmatrix} \hat{x}_{Ax} \\ \hat{x}_{Ay} \\ \hat{x}_{Bx} \\ \hat{x}_{By} \\ \hat{x}_{Z} \end{bmatrix} = \begin{bmatrix} \overline{H}_{11} & \overline{H}_{12} & \overline{H}_{13} & \overline{H}_{14} & \overline{H}_{15} \\ \overline{H}_{21} & \overline{H}_{22} & \overline{H}_{23} & \overline{H}_{24} & \overline{H}_{25} \\ \overline{H}_{31} & \overline{H}_{32} & \overline{H}_{33} & \overline{H}_{34} & \overline{H}_{35} \\ \overline{H}_{41} & \overline{H}_{42} & \overline{H}_{43} & \overline{H}_{44} & \overline{H}_{45} \\ \overline{H}_{51} & \overline{H}_{52} & \overline{H}_{52} & \overline{H}_{54} & \overline{H}_{55} \end{bmatrix} \begin{bmatrix} \hat{F}_{Ax} \\ \hat{F}_{Ay} \\ \hat{F}_{Bx} \\ \hat{F}_{By} \\ \hat{F}_{Z} \end{bmatrix} \tag{3}$$

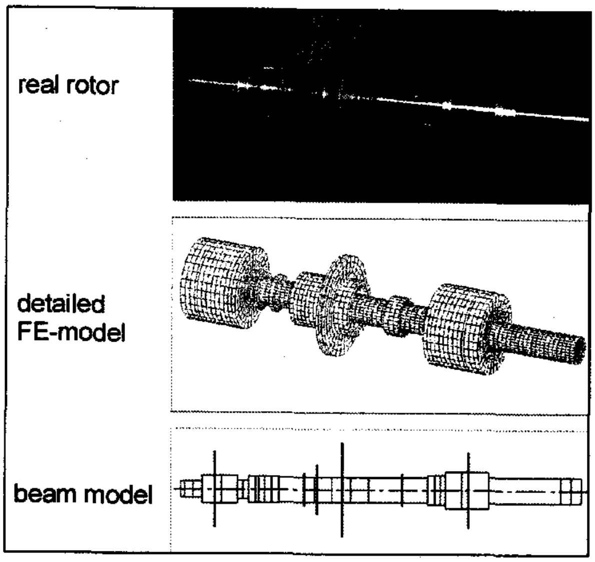

FIGURE 5. Real rotor and its FE-models.

with A, B standing for the two different radial bearings, and x, y specifying the radial axes respectively z the axial axis.

MODELING OF THE NON-FAULTY PLANT

Obviously, one step of the model-based diagnosis is the deriving of a parametric model of the plant. This is usually not a trivial process and may not for all diagnosis procedures justify the extra effort it requires. But as mentioned above, the pump is part of the active magnetic bearing system and unlike conventional bearing systems a controller is necessary to stabilize the unstable suspense state of the rotor, caused by the magnetic field. In the last years, especially for digital controlled bearings, the controller is mainly designed using model-based methods. That is exclusively the case the more the rotor has an elastic rotordynamic behavior, which is usually the case for multi-stage pump systems, where the impeller is placed between the bearings. Hence, a result of the controller design is the non-faulty model of the plant as it is a part of the entire model of the mechatronic system. In summary, the modeling of the plant on one hand enables the design of a controller with high performance and on the other hand a more detailed diagnosis.

The rotor of the pump is modeled through a finite element model (figure 5). The model update of the plain mechanical structure is performed by an experimental modal analysis to assure an accurate description of the reality. The detailed FE-modeling was necessary to account for the dynamics of the entirely assembled rotor including several contact faces, strengthening of rotor through the shaft nut etc. In a second step, the detailed

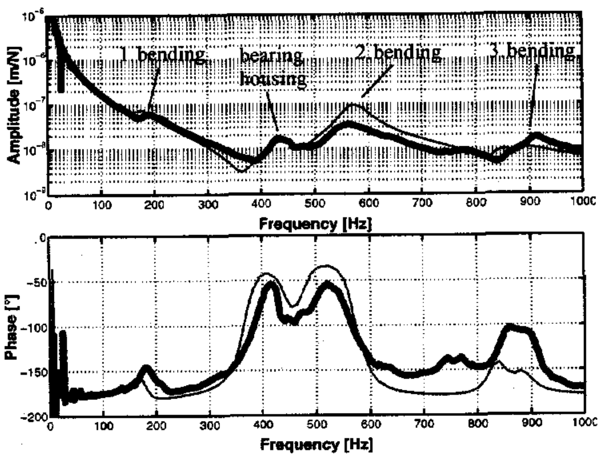

FIGURE 6. Measured (dark) and modeled (thin) frequency response function $\boldsymbol{H}_{11}$.

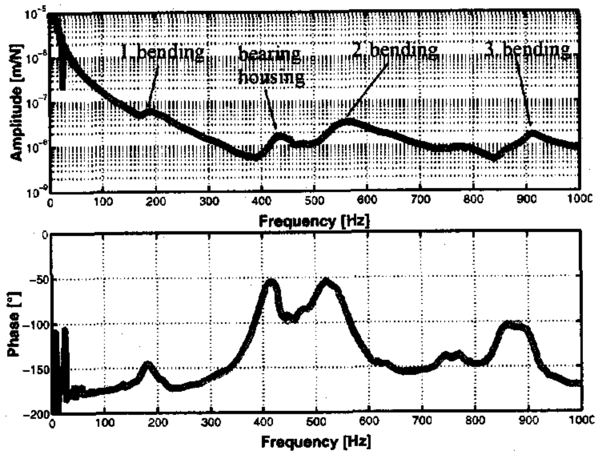

FIGURE 7. Measured (dark) and fitted (light) frequency response function $\boldsymbol{H}_{11}$.

FE-model is transferred to a beam model (figure 5) to add the gyroscopic effects. The model reduction is simply performed by modal truncation. The rigid body modes as well as the first four bending modes are considered. Modal damping is used to model the structural damping and a value of 0.75% was arbitrarily chosen.

Special routines are available as finite difference codes to consider the fluid forces acting on the rotor in the seals (impeller, balance piston), expressed as rotordynamic coefficients. Additionally, the influence of the mechanical seals as well as the bearing housing is added to the model in terms of single mass-spring-dampers. The radial degrees of freedom are coupled due to gyroscopic and fluid-structure-interaction effects. The axial degree of freedom is decoupled and not further investigated troughout this paper.

Figure 6 shows a comparison between the measured frequency response function $\overline{\boldsymbol{H}}_{11}$ (see eq. (3)) of the non-faulty pump rotor at nominal running speed (thick curve) which serves as the reference, and the calculated frequency response function of the non-faulty model (thin curve). The high quality of the model is clearly demonstrated.

Once such a good model state is reached, the modal parameters, i.e. the undamped eigenfrequencies ω_0 respectively the damped eigenfrequencies ω_d, the damping ratio D as well as the eigenmodes can be computed from the finite element model for the non-faulty system. The parameters of the relevant modes concerning the diagnosis procedure are listed in the table 2 in the last two columns. The resonances of the second and third bending modes as well as the resonance of the bearing housing are clearly detectable in figure 6. Whereas the first bending mode can hardly be observed, due to the position of a vibration node in the vicinity of the sensor. Further can be observed that the water surrounding the rotor adds remarkable damping to the system.

GENERATION OF (REFERENCE) FEATURES

There are four different methods of generating features, respectively symptoms in model-based fault detection: Kalman filter, diagnostic observer, parity relations, parameter estimation [7]. The paper focuses on the generation of features from the measured frequency response functions, characterizing the system behavior. Therefore the parameter estimation is very suitable. The parameters, which have to be identified, are coefficients of a matrix fraction description, which reduces to a simple numerator/denominator representation for SISO[2]-models:

$$\overline{H}(\omega) = \frac{b_0 + b_1\omega + b_2\omega^2 + \dots}{a_0 + a_1\omega + a_2\omega^2 + \dots}. \tag{4}$$

The software used for identification is the FREQID-toolbox written by *de Callafon and Van den Hof* from the Delft University of Technology. The estimation of the model is a least square curve fitting routine based on frequency domain data. For a more detailed discussion on the procedure, one is referred to [13].

Figure 7 shows the result of the curve fitting of the frequency response function of the non-faulty pump. The thicker dark curve again represents the measured transfer function $\overline{\boldsymbol{H}}_{11}$. The lighter curve represents the identified SISO-system model, with a model order of 14. The high quality of the fitted model is demonstrated. With the identified model a second set of modal parameters (i.e. poles of equation (4)) can be computed,

2. Single Input Single Output

which are listed in table 2 again. It should be mentioned, that here only the eigenfrequencies and the damping ratio can be extracted, because of the insufficient information receiving from two sensor planes with respect to the reconstruction of the mode shapes. Due to the good agreement of the identified and calculated modal parameters, it is straight forward to establish a model-based diagnosis procedure.

The first advantage of the model-based procedure is demonstrated by the first column of that table. Without the knowledge added from the physical model, it would be impossible to distinguish and classify the different eigenfrequencies.

TABLE 2: Reference modal parameters of the non-faulty system.

	Identified model		Physical model	
eigen-modes	eigen-freq. f_d [Hz]	damping ratio D [%]	eigenfreq. f_d [Hz]	damping ratio D [%]
1. bending	**187**	**9.4**	BW:174 FW:182	BW:6.3 FW:8.9
bearing housing	**436**	**4.0**	444	5.0
2. bending	**561**	**6.3**	BW:566 FW:582	BW:1.9 FW:2.4

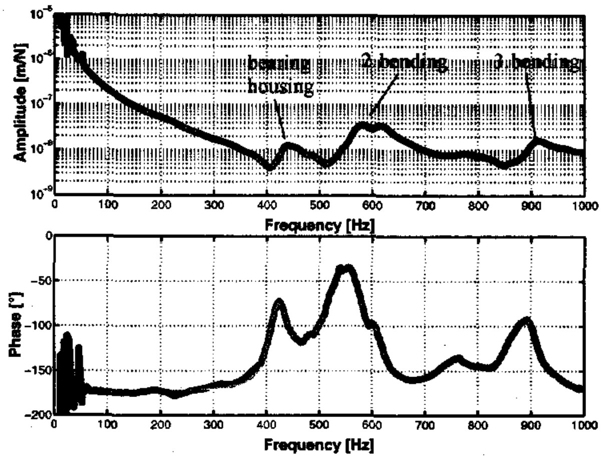

FIGURE 8. Measured (dark) and fitted (light) frequency response function H_{11} of the pump system with worn out balance piston.

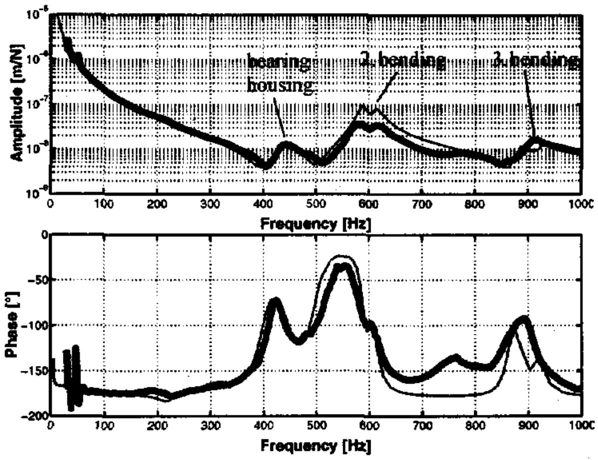

FIGURE 9. Measured (dark) and modeled (light) frequency response function H_{11} of the pump system with worn out balance piston.

TWO EXEMPLARY FAULTS

The diagnosis procedure developed using frequency domain data received from the AMBs, is demonstrated for two exemplary fault. As mentioned earlier, one assume a fault, which alters the system $\mathbf{H}(\omega)$. Analogous to the deriving of the reference features of the non-faulty system, features (i.e. modal parameters) of the faulty system are identified. Figure 8 shows such a measured transfer function (dark curve) of a faulty system together with the identified model (light curve). The faulty condition here is the worn out balance piston. Identifying the modal parameters and computing the deviations to the reference ones from table 2 leads to the values listed in table 3. This comparison of the features (modal parameters) generates symptoms for the faulty system, which significantly confirm that something is going wrong. A few facts are already recognizable from the table. The resonance of housing A is unchanged, hence something is wrong in the rotordynamic system. Furthermore, all rotor eigenfrequencies are shifted to higher frequencies. It should be mentioned, that the first bending mode is not observable in the frequency response function $\boldsymbol{H}_{11}$. Therefore, the function $\boldsymbol{H}_{33}$ is used additionally to identify the corresponding modal parameters.

A second fault investigated is the dry-run condition. Analogous to the first fault, the modal parameters are identified from the measured transfer function (figure 10)[3] and the deviations from the reference parameters are again listed in table 3. It can be seen, that the increase in the rotor eigenfrequencies is even larger than for the worn out condition of the balance piston.

3. The observable rigid body modes are due to stick-slip effects of the mechanical seals, because the dry-run condition was measured at standstill for preserving the seals.

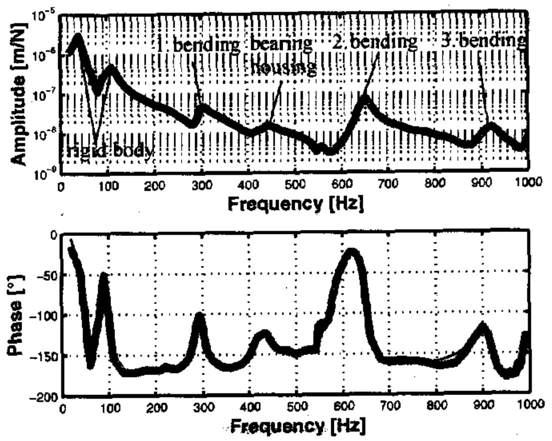

FIGURE 10. Measured (dark) and fitted (light) frequency response function H_{11} of the pump system under dry-run configuration.

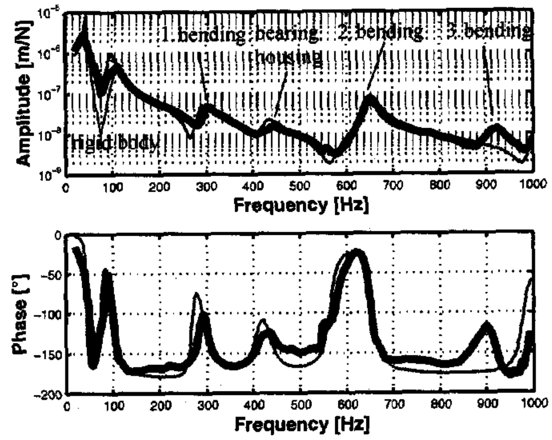

FIGURE 11. Measured (dark) and modeled (light) frequency response function H_{11} of the pump system under dry-run configuration.

In summary, the deviations of the modal parameters for these two faults are significant and hence fault detection is easily performed. The second step of the diagnosis part is to identify the acting fault. This can be done by evaluating the specific symptom pattern by using symptom-fault relations or by a model-based procedure. The non-faulty physical model is extended through different fault models. One of which is the change of the rotordynamic coefficients to represent the worn out respectively the dry-run condition. Evaluating the corresponding frequency response functions and comparing the simulated faulty ones with the measured ones leads to figure 9 respectively figure 11. A good agreement can be recognized again. Computing the modal parameters from the two models simulating the faulty conditions leads to the values listed in table 3. The comparison of the identified and modeled deviations shows the same good agreement and hence enables an entire model-based diagnosis for these faults.

TABLE 3: Deviations of the identified and modeled modal parameters with respect to the reference ones due to the faulty conditions worn out piston and dry-run.

eigen-modes		worn out balance piston: eigen-freq. f_d [Hz]	worn out balance piston: damping ratio D [%]	dry-run: eigen-freq. f_d [Hz]	dry-run: damping ratio D [%]
1.bending	ident.	**+37**	**+2.3**	**+115**	**-7.9**
1.bending	model	BW:+43 FW:+53	BW:+0.3 FW:-2.0	BW:+109 FW:+102	BW:+0.6 FW:-2.1
bearing housing	ident.	**+3**	**+0.1**	**+0**	**+1.9**
bearing housing	model	+0	+0	+0	+0
2.bending	ident.	**BW:+20** **FW:+55**	**BW:-3.9** **FW:-3.5**	**+90**	**-5.0**
2.bending	model	BW:+24 FW:+31	BW:-1.2 FW:-1.4	BW:+71 FW:+56	BW:-0.4 FW:-0.9

CONCLUSION

The procedure of model-based diagnosis for turbomachines running in active magnetic bearings has been described. A magnetically suspended centrifugal pump has been chosen as the application to demonstrate the developed routines. In particular, the special capabilities of an AMB system to measure the frequency response functions have been introduced. Further, a high quality model of the rotordynamic system including the fluid-structure-interaction in the seals for the model-based diagnosis has been described. The focus laid on the identification of the modal parameters to generate features respectively symptoms to detect faulty conditions, which are altering the system. Using this procedure and the updated model, two different faults have been detected and diagnosed. The future development concentrates on the integration of other faults like crack in the rotor, or loosening of a shaft nut. Furthermore, for a practical use of the developed model-based diagnosis not only the modal parameters should be used as features, but also information gained from the bearing forces. Then faults like misalignment [14], imbalance, or even part load and over load conditions are detectable. It should be mentioned, that this model-based diagnosis using AMBs not only works for centrifugal pumps but for all the other turbomachines running in AMBs, too. Hence, turbomachines equipped with such integrated model-based diagnosis lead to machines with higher performance, higher reliability and longer lifetime.

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