New Test Techniques Using Magnetic Bearings

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Summary

The appearance of a multitude of excitation mechanisms in rotor systems under working conditions is a well known fact. A magnetic bearing is used to experimentally generate these effects. It not only allows a perfect simulation of the real excitation forces, but in addition enables the generation of arbitrary test forces in order to evaluate the effectiveness of active controlled supports of rotors. The magnetic bearing along with its electronic components will be presented as a force generator. By the use of a computer, a fully automated test run with arbitrary excitation mechanisms can be realized. Subsequently, possible applications of magnetic bearings for research purposes will be demonstrated experimentally. Two simple examples will be outlined more in detail.

Introduction

The advantages of magnetic bearings in comparison to conventional bearings are becoming increasingly interesting for industrial applications in such fields as space or vacuum technology, and as bearings in tooling machines, in turbomachinery, in centrifuges, etc., i.e. /1, 2, 3, 4/.

The use of magnetic bearings as "active" machine elements is not only of interest as an optimal bearing for rotating machinery. They are also well suited for producing realistic simulations of specific excitation forces. Aside from thus enabling significant reductions in the amount of experimental efforts for fundamental studies, a separation of the various effects is possible. On the actual machinery, in contrast, the sum of all occuring effects is present at all times. In connection with controlled rotor systems, a practical means of testing control concepts is given. These points are at the center of the following discourse. First, the magnetic bearing along with its electronic components will be presented as a force generator.

Later, the "active" machine element is integrated into an existing test facility at a location chosen with regard to control conditions. Theoretical correlations will be briefly outlined. Subsequently, possible applications of magnetic bearings for research purposes will be discussed in conjunction with experimental results attained on the test facility.

Magnetic bearing as force generator

The complete configuration of the active element including its necessary electronic components is schematically shown in Fig. 1. It consists mainly of the magnetic bearing with four elec-

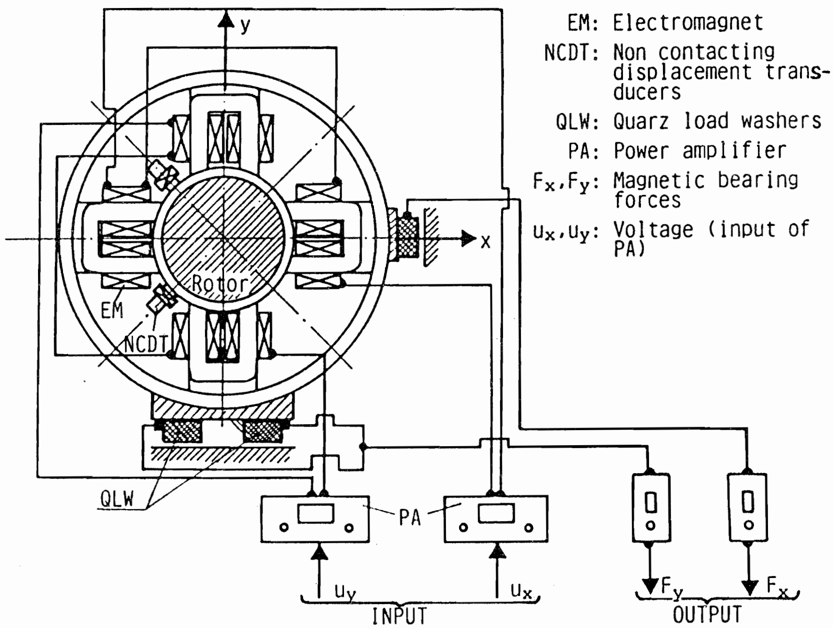

Fig. 1: Schema of the magnetic bearing including the sensor and electronic device

tromagnets (EM) switched in a differential arrangement, which produces a linear I/O-characteristic. The operation of the power amplifiers (PA) as current source yields the following equation between the input voltages u_x, u_y and the resulting forces F_x, F_y:

$$\begin{bmatrix} F_x \\ F_y \end{bmatrix} = \begin{bmatrix} k_u & 0 \\ 0 & k_u \end{bmatrix} \begin{bmatrix} u_x \\ u_y \end{bmatrix} + \begin{bmatrix} k & 0 \\ 0 & k \end{bmatrix} \begin{bmatrix} x \\ y \end{bmatrix} , \qquad (1)$$

where k_u represents the force-voltage-factor, k the force-displacement-factor, and x, y the radial displacements of the rotor in the respective directions.

The rotor motion can be measured by built-in non contacting displacement transducers. Forces on the rotor are simultaneously determined in x-and y-directions by quarz load washers. In order to decouple the forces into the two measurement directions, special axial ball guides are used. This statically and kinematically defined assembly permits the application of defined forces into the rotor at the location of the magnetic bearing. Naturally, all conceiveable force functions, including nonlinear types, are possible.

Experience has shown that a targeted experimental investigation can only be conducted with a reasonable amount of effort, if a preceding theoretical study of the system has been completed. The modelling of a rotor system as a hybrid multibody system (HMBS) has proven to be useful, especially in respect to including active components. The HMBS-modell contains rigid bodies(e.g. bearing units) and elastic subsystems (e.g. rotors, blades), as determined by specific needs /5/.

System formulation

The setting-up of the equations of motion for such HMBS is best carried out by a computer. A very effective method is the direct evaluation of the principle of d'Alembert (/6/). In order to separate the distributed coordinates, one makes use of the product method by Ritz:

$$v(z,t) = \mathbf{v}(z)^T \mathbf{c}_V(t), \qquad (2)$$

where $\mathbf{v}(z)$ is the vector of permitted shape functions (natural modes of vibration of the nonrotating rotor) and $\mathbf{c}_V(t)$ is the vector of the matching time functions. This leads to the equations of motion in their usual form:

$$\mathbf{M}\ddot{\mathbf{q}}(t) + \mathbf{P}\dot{\mathbf{q}}(t) + \mathbf{Q}\mathbf{q}(t) = \sum_{i=1}^{k} \mathbf{h}_i(t,\mathbf{q},\dot{\mathbf{q}}) , \qquad (3)$$

where $\mathbf{q} \in \mathbb{R}^f$ represents the displacement vector, with $(\dot{})$ the time derivative, f the number of degrees of freedom which is the sum of f_r and f_e, f_r are the degrees of freedom of rigid bodies, and f_e are the degrees of freedom of elastic bodies, $\mathbf{M}$ is the fxf mass matrix, $\mathbf{P}$ is the fxf matrix of forces proportional to velocities, $\mathbf{Q}$ is the fxf matrix of forces proportional to displacements, $\mathbf{h}_i$ is the vector of i-th external force which is dependent on the acting excitation forces (i.e.: unbalance, internal dissipative effects, fluid forces etc.).

In addition to excitation forces that are due to the nature of the system, almost arbitrary further excitations can be achieved with an appropriate location of the magnetic bearing. This shall be demonstrated by two examples.

First example: non-conservative forces

In the presence of non-conservative forces, such as originate from unsymmetrical flow in turbomachinery, unstable vibrations can be induced (/7/). This typ of force appears in (3) on the right hand side. It can be written in the form

$$\mathbf{h}_i = \int_0^L \mathbf{J}^T \mathbf{f}_i \, \delta(z-z_i)\,dz, \quad 0 \leq z_i \leq L \,, \tag{4}$$

where $\mathbf{J}$ is the Jacobian matrix of translation (see ref. /5/), L is the length of the elastic structure $\delta(z-z_i)$ is the Dirac function, with which the location where the force acts is taken into account. The non-conservative force $\mathbf{f}_i$ is given by

$$\mathbf{f}_i = \begin{bmatrix} 0 & -q \\ q & 0 \end{bmatrix} \begin{bmatrix} u(z_i,t) \\ v(z_i,t) \end{bmatrix} \,, \tag{5}$$

where $u(z_i,t)$ and $v(z_i,t)$ represent the radial displacements of the rotor at the point z_i in the respective directions. The degree of influence this force has on the natural vibrations can be judged by the elements of the Jacobian matrix $\mathbf{J}$ (controllability). These elements assume the value of the amplitudes of the natural modes of vibration at the point z_i (shape function, eq. (2)). Those natural modes of vibration possessing a mode at this point are not influenced by the force.

A simulation of the effects of non-conservative forces by the magnetic bearing can be formulated in the following manner:

$$\mathbf{f}_i = \begin{bmatrix} F_x \\ F_y \end{bmatrix} = \begin{bmatrix} k_u & 0 \\ 0 & k_u \end{bmatrix} \begin{bmatrix} u_x \\ u_y \end{bmatrix} + \begin{bmatrix} k & 0 \\ 0 & k \end{bmatrix} \begin{bmatrix} x \\ y \end{bmatrix} = \begin{bmatrix} 0 & -q \\ q & 0 \end{bmatrix} \begin{bmatrix} x \\ y \end{bmatrix}, \qquad (6)$$

where $x = u(z_i,t)$ and $y = v(z_i,t)$. To satisfy equation (6) the control vector is determined as

$$\begin{bmatrix} u_x \\ u_y \end{bmatrix} = \begin{bmatrix} k_{11} & k_{12} \\ k_{21} & k_{22} \end{bmatrix} \begin{bmatrix} x \\ y \end{bmatrix}, \qquad (7)$$

and the amplification factors assume following values:

$$k_{11} = k_{22} = -k/k_u \; ; \quad k_{21} = -k_{12} = -q/k_u \; . \qquad (8)$$

<u>Second Example: test forces (here as step-function)</u>

The step-function excitation is often used to examine the dynamic behavior of systems. It also can be easily generated and accurately reproduced by the magnetic bearing. The equation for a step-function force in x-direction is given as follows:

$$\mathbf{f}_j = \begin{bmatrix} F_x \\ F_y \end{bmatrix} = \begin{bmatrix} k_u & 0 \\ 0 & k_u \end{bmatrix} \begin{bmatrix} k_{11} & k_{12} \\ k_{21} & k_{22} \end{bmatrix} \begin{bmatrix} x-x_o \\ y \end{bmatrix} + \begin{bmatrix} k & 0 \\ 0 & k \end{bmatrix} \begin{bmatrix} x \\ y \end{bmatrix} = \begin{bmatrix} Q_x \\ 0 \end{bmatrix}, \qquad (9)$$

where the amplification factors assume the values

$$k_{12} = k_{21} = 0, \quad k_{11} = k_{22} = -k/k_u \wedge x_o = Q_x/k \; . \qquad (10)$$

These two examples indicate the numerous of applications that a magnetic bearing permits in simulating excitation forces. Furthermore, the excitation forces can be generated, individually or as the sum of various effects.

Test facility

Fig. 2 shows the facility used in examining such possible applications.

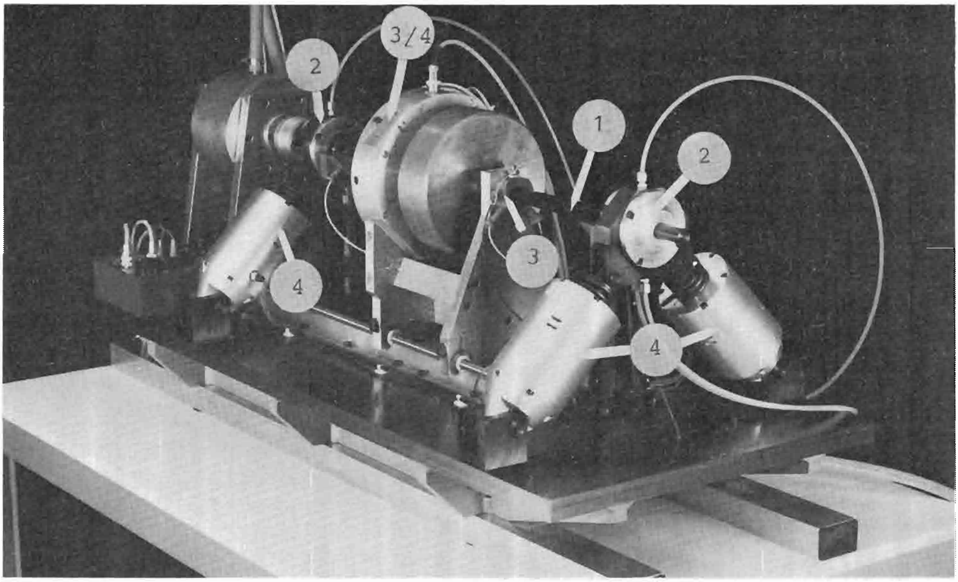

Fig. 2: Test rig (without electronics and oil supply system)

The facility mainly consists of the, elastic rotor structure ① , the bearing units (ball- or journal bearings) ② , the sensors ③, the actuators ④, the oil supply system (not shown), and the electronics (not shown).

Two types of actuators are used. Forces can be applied directly onto the rotor through the magnetic bearing or indirectly via the bearing housings through electromagnetic actuators. They can act as control forces or as any conceiveable kind of excitation force (e.g. an impulse, periodic force, circulatory force, fluid induced force, force due to bearing damage, foundation vibration (earthquake simulation), etc.). Furthermore, special electronics were developed for the adaptation of analog control voltages with the aid of a computer. All measurement and computer systems and components are interconnected by corresponding data buses (IEC). A fully automated test operation is thereby possible.

Test results

The results shown in Figs. 3 and 4 were gained on the facility. Fig. 3a depicts the locus diagram (radial displacement) of the rotor and in Fig. 3b the supplementary locus diagram of the effective non-conservative (circulatory) force produced by the magnetic bearing. One can recognize the equal sense of direction for both paths of locus diagrams resulting in an increasing amplitude and transverse force and thus evidencing the correlation given by eq. (6). In Fig. 4 the magnetic bearing served as test force generator. The test force was used for controller testing purposes in the regarded form of an impulse force. The control forces act upon the rotor and are generated by electromagnetic actuators via the journal bearing housing. A destinct improvement of system damping by the use of the controllers can be ascertained. In comparing Figs. 4a and 4b one

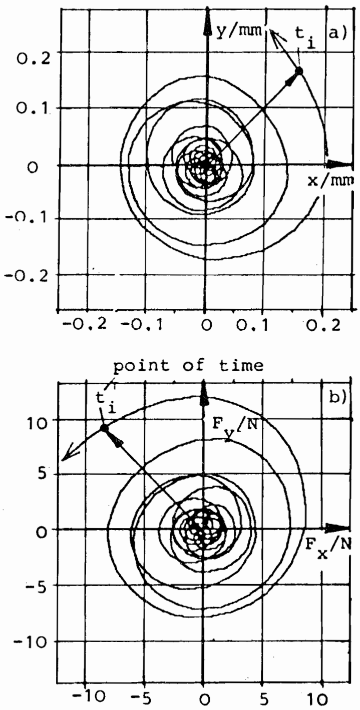

Fig.3: Locus diagrams with regard to the magnetic bearing location
a) displacement of the rotor
b) acting magnetic bearing force

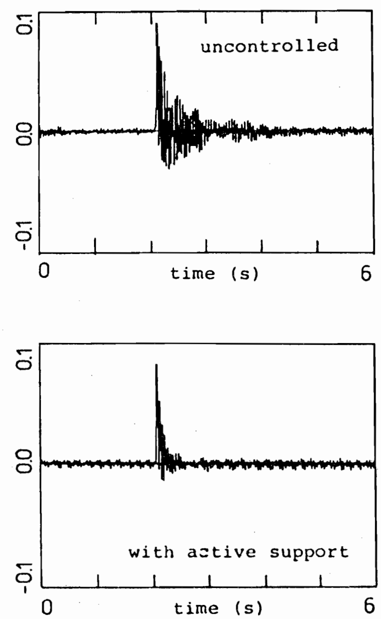

Fig.4: Transient behavior of system (impulse force produced by the magnetic bearing)

also recognizes the reproduceable amplitude of the transverse impulse generated by the magnetic bearing.

Conclusions

The use of magnetic bearings in rotor systems is not only advantageous as a substitute for conventional bearings and/or a vibration controller. They also open new possibilities in the field of test techniques. Realistic excitation processes can be simulated. This often means a considerable reduction in the scope of necessary experimental efforts. In addition, magnetic bearings are very well suited for testing the dynamic behavior of rotor systems by creating reproduceable disturbance forces. In connection with controlled rotor systems a means of testing various controllers is given. And through the use of a computer, a fully automatic operation of test cycles with various excitations is possible.

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