MODELING OF RIGID AND ELASTIC STRUCTURES IN ACTIVE MAGNETIC BEARINGS

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Abstract

Active magnetic bearings (AMB) are very often presented as a progressive alternative of bearing for High Speed Cutting (HSC) applications. However, AMBs have also several significant disadvantages. One of the most crucial is very high complexity, which results in relatively high probability of failures. Properties of the AMBs are completely dependent on the quality of the control system. To improve properties of the overall system a multi level control system for HSC spindle with AMBs was proposed. The top level of such system (supervision level) performs several tasks, which allow us to eliminate negative properties of AMBs. This paper deals with modeling both rigid and elastic structures in active magnetic bearings as a part of the supervision system. Simple lumped mass, rigid beam, rigid rotor, elastic beam and elastic rotor models are presented. Simulation models in MATLAB were developed to be a part of above mentioned supervision system. Models were partly experimentally verified on two test rigs: (1) rigid rotor with one axial and two radial active magnetic bearings, and (2) elastic beam supported on one side by rolling element bearing and by the active magnetic bearing (opposed electromagnets).

1 Introduction

Active magnetic bearings are very often presented as a progressive alternative of bearing for High Speed Cutting applications. In comparison with so called conventional bearings, they have several significant advantages: absence of mechanical contact between shaft and bearing; very low wear; low power dissipation in the bearing; absence of lubricants; high speeds of rotation; possibility to adjust position between the shaft and the bearing; unbalance compensation; high accuracy; possibility to adjust bearing parameters (stiffness and damping) during the operation; ability to work in a broad spectrum of temperatures, in vacuum, in aggressive surroundings, etc.

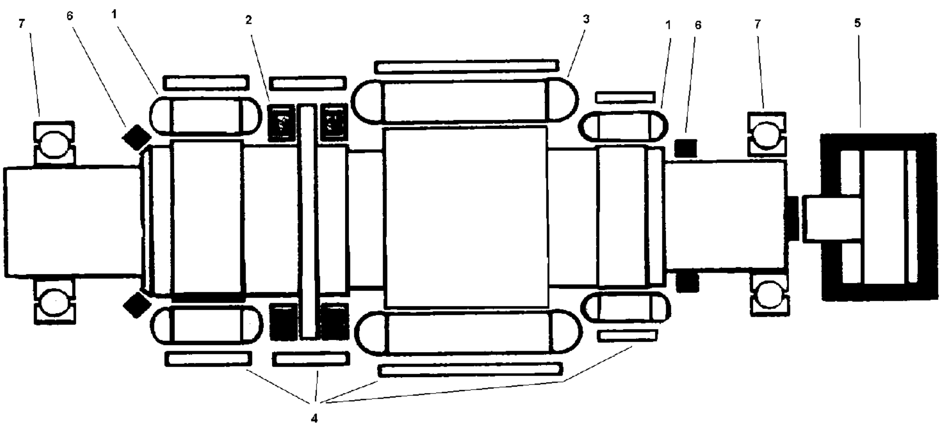

Fig.1 Scheme of cutting spindle with active magnetic bearings
(1-radial bearing, 2-axial bearing, 3-motor drive, 4-cooling,
5-tool-change mechanism, 6-sensors, 7-retainer bearings)

However, AMBs have also several significant disadvantages. One of the most crucial is very high complexity, which results in relatively high probability of faults or failures. Properties of the AMBs are completely dependent on the quality of the control system. Because AMBs are inherently unstable, they are not able to work correctly without permanent feedback (e.g. information about displacements measured by sensors).

As was mentioned above, the quality of the control loop is limiting factor for the properties of the overall system with AMBs, and a failure occurred in the loop can have fatal consequents for the process (e.g. machined part) or for the equipment (e.g. spindle, tool, etc.).
Because AMBs are not robust itself, the desired robustness must be added by the control system. In the paper [4] a multi level control system for HSC spindle with AMBs was proposed. The top level of such system (supervision level) performs several tasks, which allow us to eliminate negative influence of uncertainties in the spindle subsystem, cutting process, etc.
One of the tasks, performed by proposed supervision level, is detection and correction of system component faults. Component faults or complete failures, when not treated properly, can result into incorrect behavior of the control loop, causing a decreasing of the cutting process accuracy, touch between the shaft and auxiliary bearings, or in the extreme case an unstable behavior and damage of the system.
System component faults can be classified (according to [3]) as external or internal to the magnetic bearing control system. A fault is considered to be external when either it manifests itself as or its effect can be replicated by an external disturbance acting on the system.
External faults include

- rotor impact,
- rotor mass loss,
- base motion,
- rotor deformations
- sudden changes in loading
- rotor rub
- cracked rotor

External faults usually cause abnormal rotor vibrations, and can be treated by sufficient control force and suitable controller design ([3]). Changes in control system can include simple adjustment of controller parameters, adjustment or adaptation of the control algorithm. Therefore the development of models of the hovering body (rotor, beam or lumped mass) is necessary.

2 Structure models

Levitating structure can be modeled by using different approaches.
The simplest one model is the lumped-mass model. The lumped-mass model can be used, for example, for the axial bearing model.
When the frequencies of the disturbing signals are lower then the first resonant frequency of the structure, it is possible to use rigid rotor or rigid beam model (Fig 2). Both models are formally very similar. The rigid rotor model includes the influence of the gyroscopic effect, which causes coupling between movements in the XZ and YZ planes. In the case of rotor, we can further neglect the gyroscopic effects, and the model can be divided into two separate rigid beam models, each of them describing the movements in the XZ, resp. YZ planes.

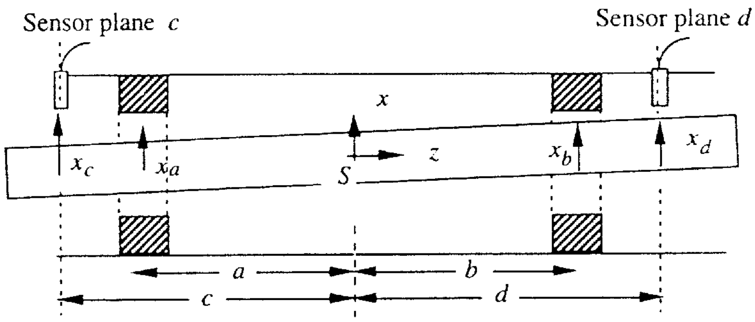

Fig.2 Rigid rotor/beam in the AMBs

The next approach used, is assuming rotor or beam as an elastic structure. The continuous system must be dicretized and the matrixes, describing the system, can be obtained by means of several discretization methods, e.g. the finite element method. Because the resulting model is very high order, it is reduced by means of modal truncation method.

The levitating structure model is described by the equations (1). The meaning of the symbols is clarified in the table 1. C_R matrix defines the relation between sensor output (measured values) and rotor displacement.

$$\begin{aligned} \dot{x}_R &= A_R(\Omega)x_R + B_R u \\ y_R &= C_R x_R \end{aligned} \tag{1}$$

TABLE 1: MODELS

	A_R	B_R
Lumped-mass model	$\begin{bmatrix} 0 & 1 \\ M^{-1}K & 0 \end{bmatrix}$	$\begin{bmatrix} 0 \\ M^{-1} \end{bmatrix}$
Rigid beam	$\begin{bmatrix} 0 & 1 \\ M^{-1}K & 0 \end{bmatrix}$	$\begin{bmatrix} 0 \\ M^{-1} \end{bmatrix}$
Rigid rotor	$\begin{bmatrix} 0 & 1 \\ M^{-1}K & -M^{-1}\Omega G \end{bmatrix}$	$\begin{bmatrix} 0 \\ M^{-1} \end{bmatrix}$
Elastic beam	$\begin{bmatrix} 0 & I \\ -M^{-1}K & -M^{-1}B \end{bmatrix}$	$\begin{bmatrix} 0 \\ M^{-1} \end{bmatrix}$
Elastic rotor	$\begin{bmatrix} 0 & I \\ -M^{-1}K & -M^{-1}(B+\Omega G) \end{bmatrix}$	$\begin{bmatrix} 0 \\ M^{-1} \end{bmatrix}$

Models of the hovering body are combined with models of other components of the control loop, i.e. sensors, actuators and power amplifiers, controllers, filters, etc. Resulting combined model can be used analysis of the system and for controller adjustment.

3 Experiments

Models described in the paper were used for modeling of two test rigs.
First of them is a rigid rotor (in the further text called rotor MECOS) in one axial and two radial eight-pole magnetic bearings. Maximum speed of the rotor is 30000 rpm; the diameter of the rotor is cca.48mm (measured across the lamination sheets). A picture of the rotor can be found in the Figure 3. When assuming the rotor an elastic structure, it was discretized by means of finite element method. The rotor was divided into several "shaft elements". The axial bearing, motor and radial bearing lamination were modeled as discrete discs added into particular nodes. The model is shown in the Figure 3

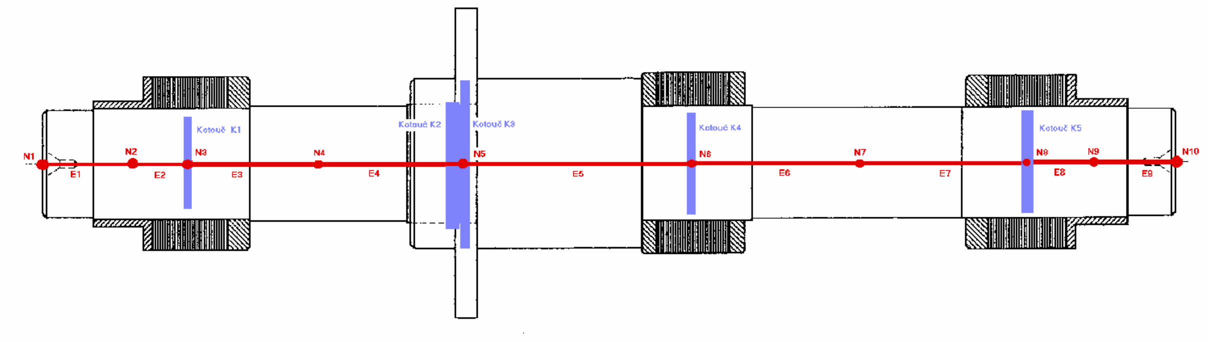

Fig.3 Rotor MECOS and it's dicretization

The second test rig is a beam supported in rolling element bearings on one side and the magnetic actuator on the other. The test rig consists of the prismatic beam, revolute joint (ball bearing), magnetic actuator and sensors. Displacements are measured by BALUFF inductive sensors. Signals

from sensors are processed in PC using MATLAB. The PC is equipped with two AD/DA cards (AD512 and MF614) and Extended Real Time Toolbox as an interface to MATLAB.
Detailed dimensions are given in the Figure 4.
The beam was divided into several "beam elements". The roller joint was defined in the rolling element bearing position.

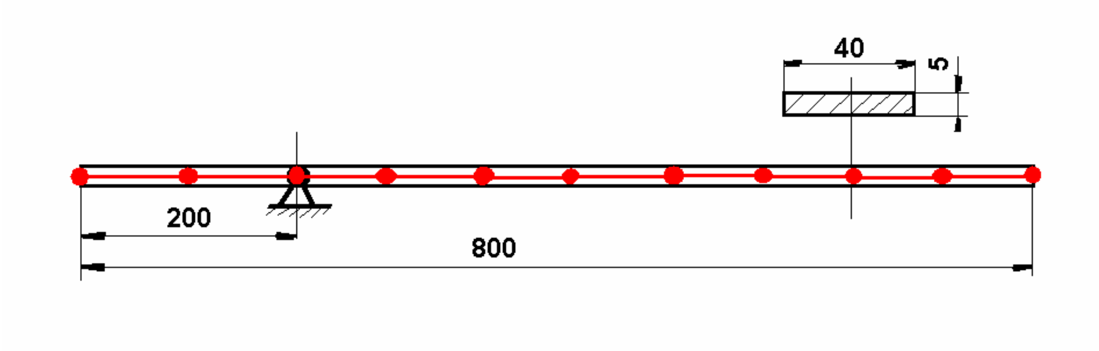

Fig.4 Beam and it's discretization

As an example the modeshapes of the rotor and beam are shown in the Figure 5 and Figure 6. Models obtained by means of finite element method were reduced by modal truncation method. Only first three, respectively five modes were taken into account in the case of rotor, respectively beam.
The others components of the AMB system were programmed in the MATLAB too.

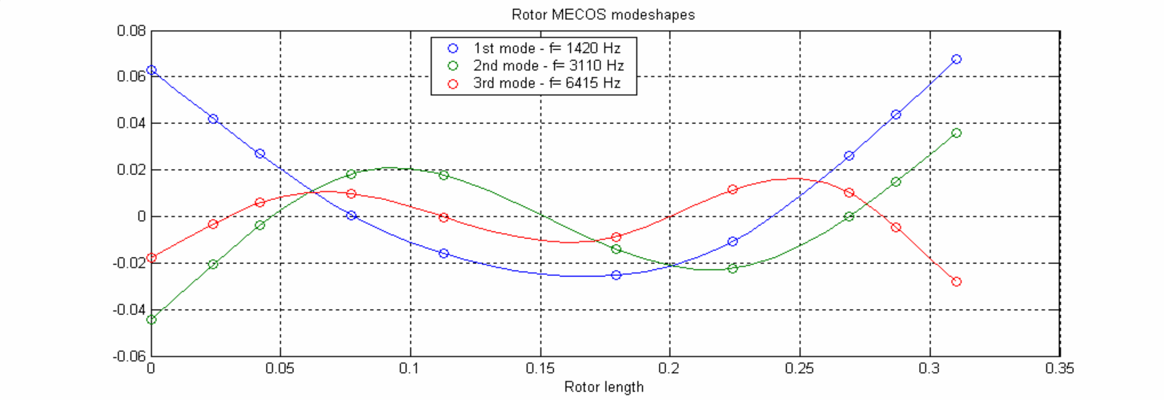

Fig.5 Modeshapes of the rotor MECOS

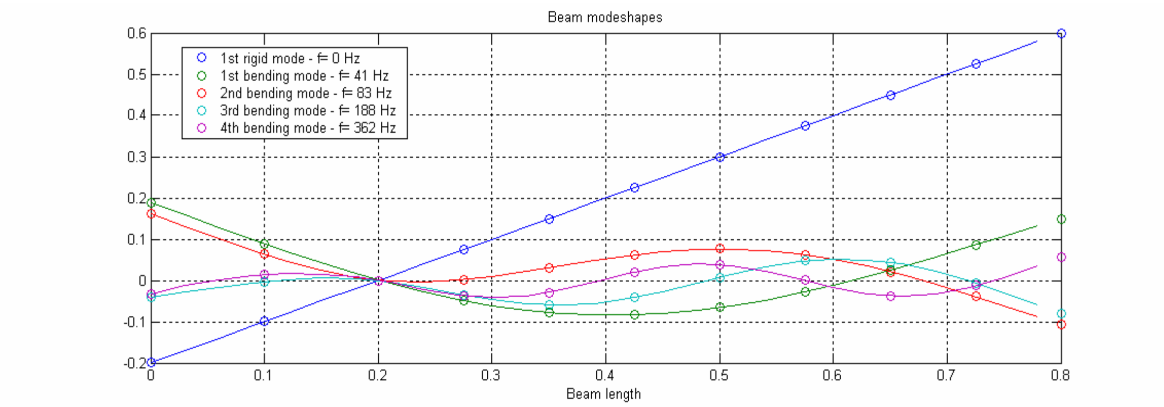

Fig.6 Modeshapes of the beam

4 Conclusion

Mathematical models of the components of an active magnetic bearing system are described in the paper. Models for rigid and elastic structures (either beam or rotor) were developed. They were programmed in the MATLAB software. Models are intended to be a part of a supervision system for the process monitoring, fault detection and on-line controller adaptation being developed by the author's team.

Acknowledgement

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