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Magneto-structural modelling of micro machining spindles supported by active magnetic bearings

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Abstract
In high precision applications, such as micro machining, machine tool spindles supported by active magnetic bearings are a suitable alternative to air bearing spindles. The use of active magnetic bearings enables to implement an active vibration control. As such, both random vibrational excitations, e. g. caused by factory influences, as well as speed-dependent excitations, such as centrifugal forces, thermal expansion and cutting forces, can be compensated by means of suitable control systems.
To develop and analyse magnetic bearings suitable for high precision applications, electromagnetic, structural dynamics, and control engineering analyses have to be combined. As such, the development and analysis of a magnetic bearing system requires a coupled-fields analysis, in which the input of one field analysis depends on the results from another field analysis.
In this paper, a coupled magneto-structural simulation model of a hybrid spindle concept, featuring both magnetic and air bearings, is set up and analysed. The current control of the magnetic bearings is implemented by means of two PID controllers. An analytical model is used to calculate the magnetic actor's electromagnetic force acting on the rotor. The mechanical motion of the rotor is computed as a function of the electromagnetic force, the unbalance forces, and the air bearing characteristics using a numerical model. Both models are implemented as a coupled-fields model in a control loop. Using this coupled-fields model, the vibrational behaviour of a rotor startup is studied.
The method used for coupling the different physics fields is well suited for the implementation of complex mechanical models in control loops. Furthermore, it is possible to replace the analytical calculation of the electromagnetic force with a numerical electromagnetic model, which allows the analysis of magnetic bearings that cannot be described by analytical formulas.

Magnetic bearing, Mechatronic, Finite Element Method (FEM), Vibration

1. Introduction

Micro milling has several advantages when compared to other manufacturing processes [1]. One of the advantages of micro milling is the possibility to manufacture products and structures with complex geometries [2]. However, complex geometries require several changes of the feed rate. At constant rotational speeds of the spindle, changes of the feed rate per tooth result, which has a negative impact on the wear of the tool and the quality of the machined structure. To achieve a constant feed per tooth, the spindle's rotational speed can be coupled to the feed rate. The resulting adjustment of the spindle's rotational speed, however, leads to variations in the milled groove width due to changing unbalance forces during spindle acceleration and deceleration.

For micro milling applications, air bearing spindles are often used due to the high rotational speeds, low radial error motion, sufficient load capacity, and low friction losses. However, air bearing spindles are prone to self-excited vibrations [3] and by default do not possess the ability to implement an active control.

Active magnetic bearings enable to actively compensate disturbing forces . Drawbacks of magnetic bearings are the complexity and hence the costs [4].

As such, a hybrid spindle for micro machining, featuring both air bearings and a single magnetic bearing, could provide both the required load capacity as well as the possibility to actively perform spindle/tool path corrections during changes of the rotational speed, if needed. Additionally, a single magnetic bearing may be sufficient for active control while keeping implementation costs at a minimum.

A hybrid bearing spindle with a combination of magnetic bearings and air bearings was already presented in [5]. With the additional magnetic bearings, self-excited vibrations induced by the air bearings could be damped and thus an increase of the maximum rotational velocity by approximately 30% was possible.

The development of a hybrid bearing spindle, where the rotational speed is to be varied during the cutting process, requires that the influence of the drive torque, the rotational acceleration and transient effects are considered in the system model. These influences were not considered in [5].

This paper presents a simulation model for the development of a hybrid bearing spindle, where the rotational speed can be varied during the cutting process. Thus, this paper provides a solution for the development of a hybrid spindle for micro machining, where the rotational speed can be changed to maintain a constant feed per tooth without introducing an additional error motion due to spindle acceleration and decelaration.

For this purpose, a suitable modelling technique is provided in chapter 2, considering the rotational acceleration and the speed-dependent system dynamics. The model implements the analytical calculation of the electromagnetic force, the numerical simulation of the spindle motion and vibration, and the air bearing characteristics through lookup-tables in a single

closed-loop simulation model. The rotational velocity is not constant but a function of a predefined drive torque. The stabilizing effect of the magnetic bearing over the entire speed range is demonstrated through the simulation of a spindle' rotor startup in chapter 3. Chapter 4 concludes with a short summary and avenues for future/further research.

2. Coupled magneto-structural model setup

The setup of the coupled model consists of the calculation of the electromagnetic properties and the setup of the numerical mechanical model. Both aspects are included in the closed loop model used for the control system analysis.

The configuration of a hybrid spindle is shown in figure 1a), the used coordinate system can be seen in figure 1b). The spindle rotor is supported by two aerostatic journal bearings and a single magnetic bearing, located near the rotor's front end. The configuration of the heteropolar, 4-pole-pair active magnetic bearing is shown figure 1c). A differential winding [4] is employed. Herein, one electromagnet is operated with the sum of the bias current i_0 and the control current i, while the opposite electromagnet uses the difference. This ensures that the current and thus the force increase in one electromagnet to the same extent as they decrease in the other electromagnet.

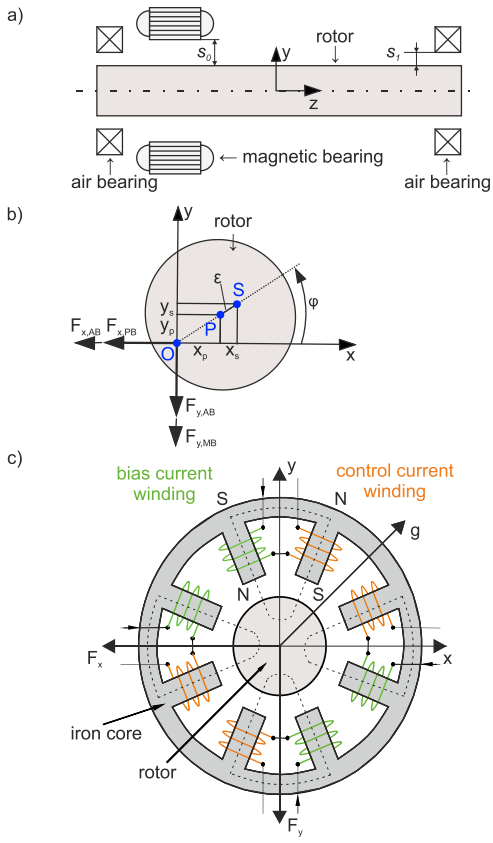

Figure 1. a) hybrid spindle featuring both air bearings and a single magnetic bearing, b) free-body diagram of the rotor, c) schematic view of the heteropolar 4-pole-pair active magnetic bearing

2.1. Electromagnetic force calculation

The calculation of the pulling forces acting on the spindle is derived from the magnetic field energy W. The electromagnetic pulling force f can be calculated using the principle of virtual displacement from the field energy and becomes [4]:

$$f = \frac{1}{4}\mu_0 n^2 A_l \frac{i^2}{s^2} cos(\alpha) \qquad (1)$$

where μ_0 is the vaccum permeability, n is the number of turns of a single coil, A_l is the pole face area, i is the current and α is the angle between the rotor surface and the pole area (see figure 1).

For this magnetic bearing with differential winding, the total force in x- and y-direction is calculated by replacing the current in equation (1) with the sum $(i_0 + i)$ or difference $(i_0 - i)$ of the bias current i_0 and the control current i. The distance s is replaced by the sum $(s_0 + x)$ or difference $(s_0 - x)$ of the nominal bearing gap s_0 and the instantenous rotor position x [4]. With these substitutions, the electromagnetic pulling force f_x in x-direction is obtained as follows:

$$f_x = f_+ - f_- = \frac{1}{4}\mu_0 n^2 A_l \left(\frac{(i_0+i)^2}{(s_0+x)^2} - \frac{(i_0-i)^2}{(s_0-x)^2}\right) cos(\alpha) \qquad (2)$$

The electromagnetic force f_y in y-direction can be calculated similarly.

2.2. Spindle motion simulation

Applying Newton's second law of motion to the free rotor (figure 1b)) yields the equations of motion for this spindle configuration:

$$m\ddot{x} = -F_{x,MB} - F_{x,AB} + mgsin\left(\frac{\pi}{4}\right) + m\varepsilon(\ddot{\varphi}sin(\varphi + \dot{\varphi}^2 cos(\varphi)) \qquad (3)$$

$$m\ddot{y} = \left(F_{y,MB} + F_{y,AB}\right) + mgcos\left(\frac{\pi}{4}\right) + m\varepsilon(-\ddot{\varphi}cos(\varphi + \dot{\varphi}^2 sin(\varphi)) \qquad (4)$$

$$J\ddot{\varphi} = -\left(F_{x,MB} + F_{x,AB}\right)\cdot(y + \varepsilon\, sin(\varphi)) + \left(F_{y,MB} + F_{y,AB}\right)\cdot(x + \varepsilon\, cos(\varphi)) + M \qquad (5)$$

where m is the rotor mass, ε is the rotor eccentricity, $F_{x,MB}$ and $F_{y,MB}$ are the magnetic bearing forces, $F_{x,AB}$ and $F_{y,AB}$ are the air bearing forces, g is gravity, φ is the rotational angle, $\dot{\varphi}$ is the rotational velocity, $\ddot{\varphi}$ is the rotational acceleration, J is the rotor's moment of inertia about the z-axis and M is the drive torque of the rotor.

The fluid forces of the air bearings can be represented through stiffness and damping coefficients c_{ij} and k_{ij} $(i, j = x, y)$, respectively, through the following equations [6]:

$$F_{x,AB} = -c_{xx}\dot{x} - c_{xy}\dot{y} - k_{xx}x - k_{xy}y \qquad (6)$$

$$F_{y,AB} = +c_{xy}\dot{x} - c_{xx}\dot{y} + k_{xy}x - k_{xx}y \qquad (7)$$

The coefficients c_{ij} and k_{ij} are dependent on the rotational velocity. One method for the calculation of the air bearing coefficients is given in [3].

The right-hand sides of equations (3), (4), and (5) represent gravitational force, unbalance force considering rotational acceleration, the pulling force of the magnetic bearing, and drive torque.

2.3. Control system model setup

Distributed proportional-integral-derivative (PID) control [4] is used for the current control of the magnetic bearing. Herein, each controlled variable (currents i_x, i_y) is assigned an input (positions x, y). A single-variable PID controller is designed and implemented for each pair of inputs and outputs. Implementing independent PID controllers for the current control in both x- and y-direction, the electromagnetic force calculation (section 2.1), the equations of motion and the air bearing characteristics (section 2.2), one obtains a model of the closed loop, which can be used for the system simulation and the controller design. The closed loop can be seen in figure 2a), with a detailed schematic of the plant given in figure 2b). The model dynamics can be described as follows:

During a time step Δt and for a given target position r_x and r_y of the spindle rotor, the PID controllers determine the deviation e_x and e_y from the instantaneous positions x and y and accordingly alter the control currents i_x and i_y. The updated control currents get passed to the plant subsystem (figure 2b)) along with the instantaneous positions x and y of the spindle and the torque of the spindle's driving motor.

Within the plant subsystem, the electromagnetic force is calculated and updated based on equation (2). The updated electromagnetic force is then transferred to the mechanical model, consisting of the equations (3), (4) and (5). In order to solve these equations, additional data is required. This additional data consists of the instantaneous values of rotational acceleration $\ddot{\varphi}$, rotational velocity $\dot{\varphi}$, rotational angle φ, position x and y and velocitiy $\dot{x}$ and $\dot{y}$.

Further, the corresponding air bearing stiffness and damping coefficients are required. Since these coefficients are dependent on the rotational velocity $\dot{\varphi}$, they are implemented as lookup-table data.

Within these lookup-tables, the stiffness and damping coefficients are evaluated by looking up or interpolating said values as a function of $\dot{\varphi}$.

When all data is evaluated, the equations of motion are solved numerically to obtain the accelerations $\ddot{x}$ and $\ddot{y}$. Accelerations are then integrated twice to obtain both updated instantaneous velocities and positions. With the updated positions x and y, the next time step can be computed.

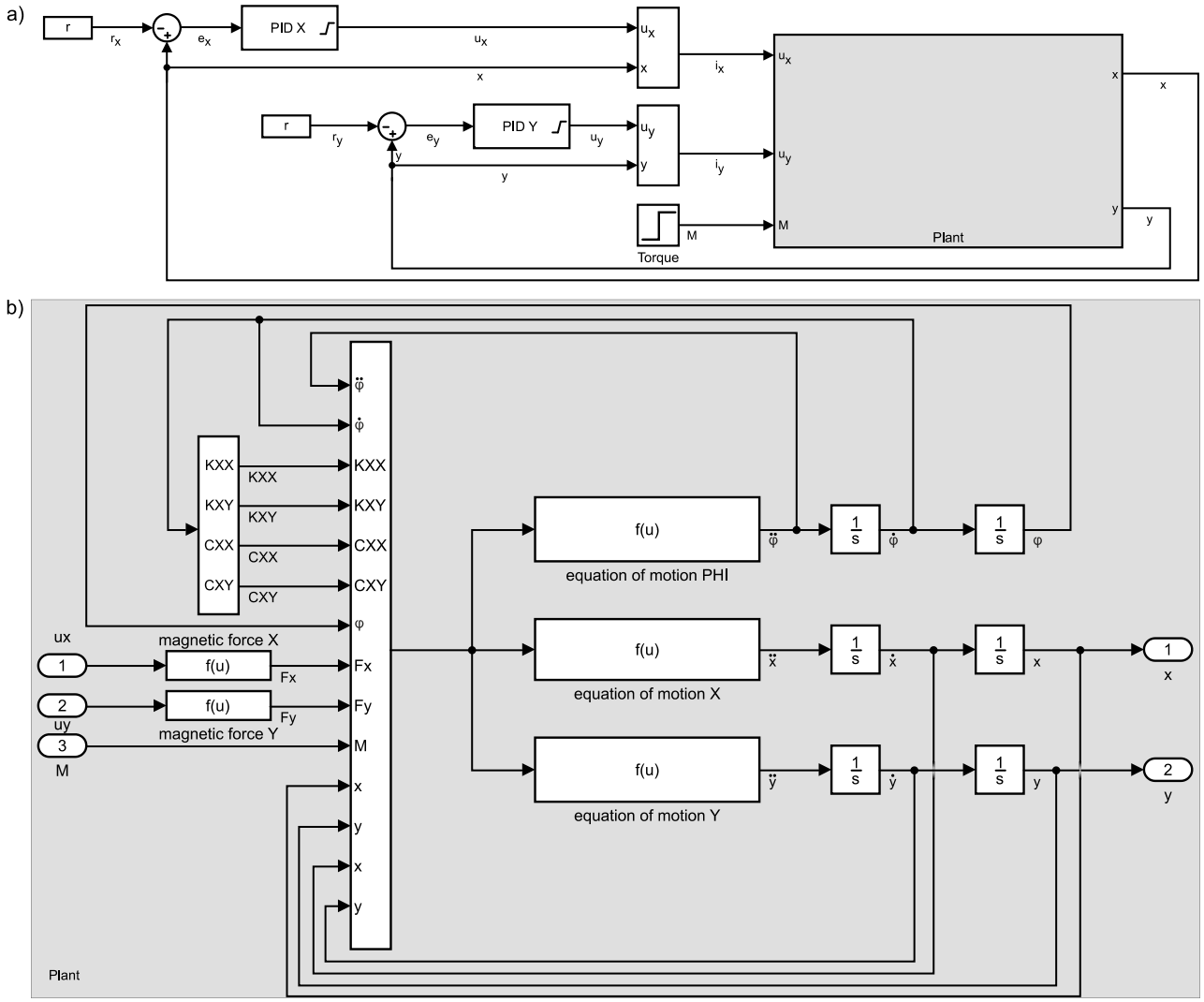

Figure 2. Control system model for hybrid spindle concept featuring both air bearings and a single magnetic bearing

3. Results

The speed-dependent stiffness and damping coefficients of the air bearings, the required simulation settings, as well as the data for the rotor, the magnetic actuator and the controller are given in table 1 and table 2. Simulink R2019b is used as system simulation software.

The control system model (figure 2) is used to simulate a rotor startup. In the first phase of the simulation ($t = 0 \dots 1s$) the rotor is brought into levitation, in the second phase ($t = 1 \dots 2s$) a constant torque of $1\ Nm$ is applied. For the listed simulation data, the constant torque of 1 Nm generates a constant slope of the rotational velocity, ranging from $0\,rpm$ at $t = 1s$ to $135{,}000\,rpm$. The results for the vibration amplitude of the rotor in x- and y-direciton for this speed range can be seen in figure 3.

Figure 3a) shows the simulated vibration amplitude without the use of an additional magnetic bearing. The first critical speed is passed at approximately 1.4 seconds. During this time range, the peak-to-peak vibration amplitude of the spindle increases from $4\ \mu m$ up to $16\ \mu m$.

The stabilizing effect of the additional magnetic bearing is shown in figure 3b). The rotor's vibration amplitude during the pass of the critical speed is effectively controlled through the active magnetic bearing. The peak-to-peak vibration amplitude is kept in the range of $4\ \mu m$.

Although the vibration amplitude could be reduced in the range of the critical speed, the remaining vibration amplitude of $4\ \mu m$ could not be reduced any more. Instead, the period length of this vibration decreases, leading to a strongly oscillating behaviour (figure 3b, right side). This oscillation could not be further reduced with the control method used.

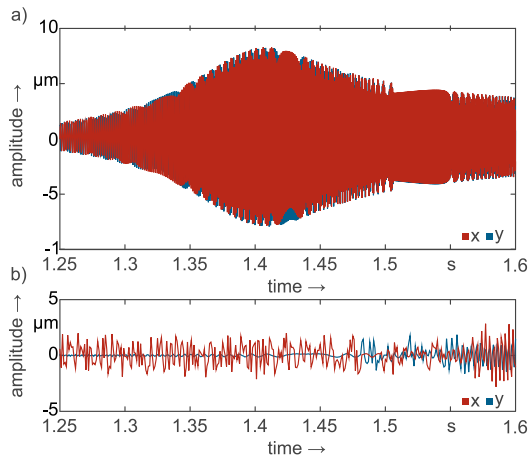

Figure 3. Rotor vibration amplitudes in x- and y-direction for an a) air bearing spindle and a b) hybrid spindle concept featuring both air and a single magnetic bearing

Table 1. Air bearing stiffness and damping coefficients used for the analysis

Rotational velocity	Direct stiffness	Cross-coupled stiffness	Direct damping	Cross-coupled damping
$\Omega/(min^{-1})$	$k_{xx}/(\frac{MN}{m})$	$k_{xy}/(\frac{kN}{m})$	$c_{xx}/(\frac{Ns}{m})$	$c_{xy}/(\frac{Ns}{m})$
600	9.25	27.12	885.27	-3.95
30,000	9.99	1124.77	808.10	-176.30
60,000	11.53	1406.57	651.37	-253.75
90,000	13.03	1044.90	507.97	-251.80
120,000	14.26	478.71	401.12	-218.59

Table 2. Spindle and magnetic bearing data used for the analysis

Parameter	Value
Simulation time t	$2\ s$
Step size Δt	variable
Maximum step size Δt_{max}	$1 \cdot 10^{-4}$
Solver	ode45
Rotor mass	$0.61\ kg$
Rotor moment of inertia about z-axis	$7.01 \cdot 10^{-4}\ kg \cdot m^2$
Rotor eccentricity	$2 \cdot 10^{-6}\ m$
Coaxial magnetic bearing gap s_0	$1 \cdot 10^{-3}\ m$
Torque	$1\ Nm$
Magnetic bearing constant	$2.56 \cdot 10^{-6} Nmm^2/A^2$
Bias current	$2.5\ A$
Control current	$\pm 2.5\ A$
Proportional gain	$7200\ A/m$
Integral gain	$55800\ A/ms$
Differential gain	$194.7\ As/m$
Target position in x- and y-direction	$0\ m$

4. Conclusion and future work

This paper investigated the stabilizing effect of an active heteropolar 4-pole-pair magnetic bearing in an air bearing spindle, intended for applications with varying rotational speeds. In particular, a method for the development of a hybrid spindle, intended for micro milling with a constant feed rate per tooth without introducing an additional error motion, was presented. For this purpose, a simulation model of the control system was setup, incorporating the calculation of the magnetic bearing's electromagnetic force, the speed-dependent dynamic characteristics of the air bearings and the simulation of the spindle motion under the consideration of the drive torque, unbalance forces and the current control of the magnetic bearing through two PID controllers. A comparison of a rotor startup as a function of the drive torque both with and without the magnetic bearing reveal the stabilizing effect of the magnetic bearing. With the chosen controller settings, it was possible to effectively reduce the rotor's vibration motion amplitude during the pass of critical speeds.

The following conclusions can be drawn from this analysis:

- The presented modelling technique is suitable for the analysis of hybrid spindles intended for applications with varying rotational speeds;
- Magnetic bearings have the ability to stabilize air bearing systems and to reduce spindle vibrations when passing critical speeds;
- Lookup tables can be used to implement the dynamic characteristics of air bearings, making it possible to include important dynamic properties of the air bearings while keeping simulation time (and thus controller design and optimization time) as small as possible since no finite element analysis must be employed during each time step;
- Lookup tables can be used to implement relevant system characteristics which cannot be modelled through analytical formula. For example, it is possible to replace the analytical calculation of the electromagnetic force with a lookup table filled with values obtained from a numerical electromagnetic model, making it possible to analyse control system models of complex magnetic bearing configurations without having to integrate a time-consuming full finite element analysis in the system simulation;
- Distributed PID control can be used for the initial design of hybrid spindle systems;
- More sophisticated control methods are required to further reduce the error motion of hybrid spindles.

Since the distributed PID control acts as if the individual control loops for the control currents i_x and i_y would not influence the motion in the perpendicular direction ($i_x \leftrightarrow y$, $i_y \leftrightarrow x$), the chosen control method may not best suited for the control of a hybrid bearing spindles for micro machining, since the motion in x- and y-direction is tightly coupled and thus can cause too much undesired interference between the control loops in x- and y-direction. In future work, additional control methods, such as centralised control and distributed control with decoupling networks will be investigated and implemented.

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