Influence of Viscoelastic Damping Elements on Magnetically Active Stabilized Degrees of Freedom

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Abstract: The application of viscoelastic damping elements is well suited for the minimization of radial and tilt vibrations of rotors featuring radial passive magnetic bearings. This paper discusses the influence of the damping element on the axial, magnetically active stabilized degree of freedom of such systems. The stabilizable axial excitations are calculated for the case of a rigid and for the case of an elastical stator-housing connection and are verified by means of measurements. Furthermore, the influence of the control parameters is discussed. In order to acquire an optimal stability against external vibrations, a genetic optimization of the controller parameters was carried out.

Keywords: Permanent Magnetic Bearing, Passive Magnetic Bearing, Viscoelastic Material, Damping, Stability

Introduction

The high mechanical and electrical complexity and particularly the high resulting costs are the main reasons for the reluctant spread of the magnetic bearing technology. The additional costs mainly arise from the actively controlled degrees of freedom, where sensors, signal processing, power electronics and actuators are needed. This paper deals with a magnetic bearing concept with relatively low constructive complexity. The radial and tilt stabilization are achieved by two passive magnetic bearings. The active element is used to control the axial rotor position, which is unstable due to the Earnshaw Theorem. Consequently, only one displacement sensor and one actuator are necessary for full magnetic levitation. Thus, the effort for sensors, power electronics, actuators and system control is quite low. However, passively stabilized degrees of freedom feature vanishingly low damping. Hence, the system is very sensitive to disturbances and operation close to a resonance frequency can be critical. Passing through such frequencies in order to reach supercritical operation points often leads to significant problems. In a parallely submitted paper Marth shows how viscoelastic damping elements can be utilized to damp the passively stabilized degrees of freedom [1]. These damping elements improve the sytem performance: the vibrations caused by unbalance, magnetic tolerances [2] or external excitations are suppressed effectively. As shown in Fig. 1, a configuration with one annular viscoelastic damping element at the bottom of the stator is investigated. A guidance, blocking the axial movement of the elastically mounted stator, is not used as it would lead to an enhanced construction effort and

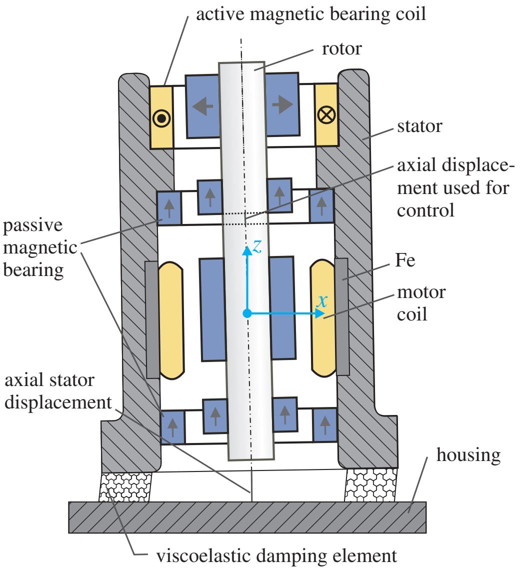

Fig. 1. Schematic representation of the entire assembly. The stator also performs axial deflections due to the elastic mounting.

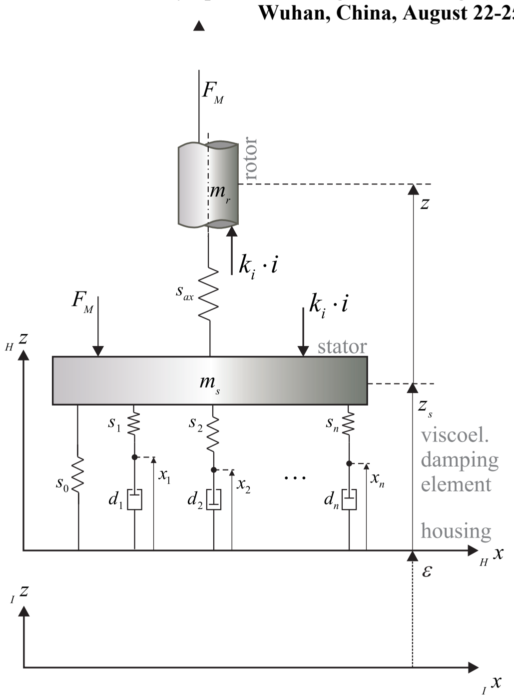

Fig. 2. Schematic representation of the vibrating bodies and the forces acting in axial direction. The viscoelastic damping element is modeled by a generalized Maxwell model.

additional losses. Furthermore, an accurate model including the sliding friction would be hard to obtain. However, if the stator has no radial guidance, the damping elements do not only influence the radial and tilt vibrations but also the actively controlled axial direction.
In the following sections the influence of viscoelastic damping elements on the closed loop control of the axial rotor position is analyzed. An optimization with respect to the stability against external disturbances will be shown. The corresponding schematic representation of the controlled system is shown in Fig. 2.

Control structure

A mathematical model of the controlled mechanical and electrical system and of the controller algorithm is necessary for the calculation and optimization of the closed loop behavior. In the setup at hand, the relative axial rotor position z is regulated by a position controller with subordinate current control, see Fig. 3. In order to compensate stationary forces, a superimposed controller is used. This controller ensures an offset-free current in the actuator coil through an axial displacement between the rotor and stator magnets. With this method, the rotor weight force can be compensated with the axial magnetic force of the permanent magnetic radial bearings.
The axial rotor position is stabilized by a Lorentz force bearing. The axial force on the rotor is directly proportional to the coil current ($F_{AMB} = k_i \cdot i$). An optimized design of such an active magnetic bearing is shown in [3]. In a first step, the damping element is modeled by one spring and damper unit with the stiffness s_{ve} and the damping d_{ve}. The controlled system can be described by the linear differential equations

$$\begin{bmatrix} \dot{i} \\ \dot{z} \\ \dot{v} \\ \dot{z}_s \\ \dot{v}_s \end{bmatrix} = \begin{bmatrix} -\frac{R}{L} & 0 & -\frac{k_i}{L} & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 \\ \frac{k_i}{m_{red}} & -\frac{s_{ax}}{m_{red}} & 0 & \frac{s_{ve}}{m_{red}} & \frac{d_{ve}}{m_{red}} \\ 0 & 0 & 0 & 0 & 1 \\ -\frac{k_i}{m_s} & \frac{s_{ax}}{m_s} & 0 & -\frac{s_{ve}}{m_s} & -\frac{d_{ve}}{m_s} \end{bmatrix} \begin{bmatrix} i \\ z \\ v \\ z_s \\ v_s \end{bmatrix} + \begin{bmatrix} \frac{1}{L} & 0 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & \frac{1}{m_{red}} \\ 0 & 0 & 0 \\ 0 & -1 & -\frac{1}{m_{red}} \end{bmatrix} \begin{bmatrix} u \\ \ddot{\varepsilon} \\ F_M \end{bmatrix}, \quad (1)$$

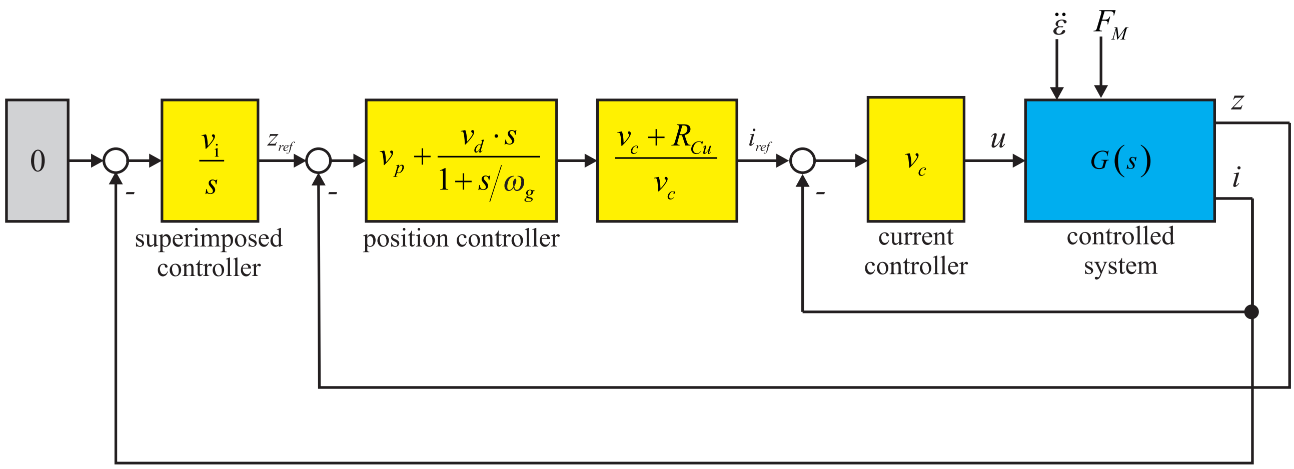

Fig. 3. Control structure used for the stabilization of the axial degree of freedom

where z_s and v_s is the position and the velocity of the axial stator deflection, while v is the velocity of the relative rotor deflection. R and L denote the resistance and the inductance of the actuator coil. The negative axial stiffness s_{ax} of the permanent magnetic journal bearings can be calculated analytically if no ferromagnetic material is near the magnetic circuit [4]. The variable m_{red} is equal to $m_{red} = m_s \cdot m_r/(m_s + m_r)$, m_s and m_r denote the mass of the stator and the mass of the rotor. The acceleration $\ddot{\varepsilon}$ of the housing and the axial force F_M act as external disturbances. F_M can be caused by the motor unit illustrated in Fig. 1. The impressed voltage u of the actuator coil is the only actuating variable of the system.

Enhancement of the model for a viscoelastic damping element: The stiffness and damping of a viscoelastic material is highly dependent on the frequency, on the temperature and on the pre-pressing. The memory of the material is described by means of a generalized Maxwell model, which uses $n+1$ parallel spring and damper units. These units and the corresponding states x_1 till x_n are shown in Fig. 2.

In order to obtain the dynamic behavior of the viscoelastic damper, it is necessary to calculate the quasi-static stiffness s_0 in a first step. In a second step, the thermo-viscoelastic properties (the Prony parameters) have to be derived from the master curve of the material [5]. With use of this information, the parameters $s_0 \ldots s_n$ and $d_1 \ldots d_n$ can be calculated. In this paper the frequency dependency is modeled by means of a generalized Maxwell model of 15th order. Thus, the controlled system is described by 20 linear differential equations

$$\begin{bmatrix} \dot{i} \\ \dot{z} \\ \dot{v} \\ \dot{z}_s \\ \dot{v}_s \\ \dot{x}_1 \\ \vdots \\ \dot{x}_n \end{bmatrix} = \begin{bmatrix} -\frac{R}{L} & 0 & -\frac{k_i}{L} & 0 & 0 & 0 & \ldots & 0 \\ 0 & 0 & 1 & 0 & 0 & 0 & \ldots & 0 \\ \frac{k_i}{m_p} & -\frac{s_{ax}}{m_p} & 0 & \frac{s_g}{m_p} & 0 & -\frac{s_1}{m_s} & \ldots & -\frac{s_n}{m_s} \\ 0 & 0 & 0 & 0 & 1 & 0 & \ldots & 0 \\ -\frac{k_i}{m_s} & \frac{s_{ax}}{m_s} & 0 & -\frac{s_g}{m_s} & 0 & \frac{s_1}{m_s} & \ldots & \frac{s_n}{m_s} \\ 0 & 0 & 0 & \frac{s_1}{d_1} & 0 & -\frac{s_1}{d_1} & \ldots & 0 \\ \vdots & \vdots & \vdots & \vdots & \vdots & \vdots & \ddots & \vdots \\ 0 & 0 & 0 & \frac{s_n}{d_n} & 0 & 0 & \ldots & -\frac{s_n}{d_n} \end{bmatrix} \begin{bmatrix} i \\ z \\ v \\ z_s \\ v_s \\ x_1 \\ \vdots \\ x_n \end{bmatrix} + \begin{bmatrix} \frac{1}{L} & 0 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & \frac{1}{m_p} \\ 0 & 0 & 0 \\ 0 & -1 & -\frac{1}{m_p} \\ 0 & 0 & 0 \\ \vdots & \vdots & \vdots \\ 0 & 0 & 0 \end{bmatrix} \begin{bmatrix} u \\ \ddot{\varepsilon} \\ F_M \end{bmatrix}. \quad (2)$$

Rigid Stator–Housing Connection

In a first step, the system behavior of the closed loop is investigated for a rigid stator–housing connection (very high stiffness s_{ve}, $z_s = 0$). A shaker is used to impress sinusoidal excitations on the housing, modelled by the time-dependent coordinate ε in Fig. 2. In steady-state operation the active magnetic bearing stabilizes the rotor by applying a sinusoidal current. If the necessary

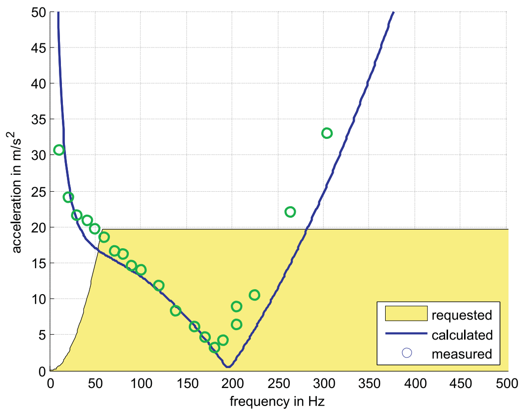

Fig. 4. Stabilizable accelerations of the housing for a freely chosen set of controller parameters (rigid stator–housing connection)

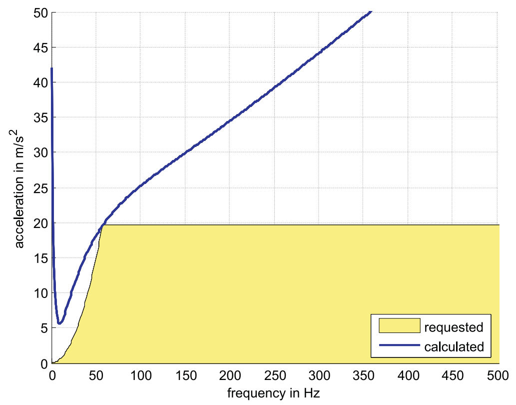

Fig. 5. Stabilizable accelerations of the housing for an optimized set of controller parameters (rigid stator–housing connection)

voltage u does not saturate (due to the voltage limitation), the amplitude and phase of the current can be calculated with the linear transfer function

$$T_{i,a}(j\omega) = \frac{\underline{i}(j\omega)}{\underline{\ddot{\varepsilon}}(j\omega)}. \tag{3}$$

It should be noted that $T_{i,a}$ has been calculated for discrete states and includes the delay of the controller algorithm. Fig. 4 shows the external accelerations $|\underline{\ddot{\varepsilon}}(j\omega)|$ that can be stabilized with the maximum permanently possible current $|\underline{i}(j\omega)| = \hat{i}$ for a freely chosen set of controller parameters. The stabilizable accelerations calculated with equation (3) are plotted as a solid line. The circles show the stabilizable accelerations gained by measurements. The filled area was taken from the environmental testing norm for sinusoidal vibrations [6]. The active magnetic bearing has to stabilize this minimum of acceleration so that the device under test is not damaged by these external vibrations. In case of the freely chosen controller parameters used in Fig. 4 only very low axial accelerations can be stabilized around 200 Hz. Therefore, the requested acceleration profile is not satisfied.

Thus, in the next step, a multi-objective optimization of the controller parameters was carried out. All parameters of the controller (v_i, v_p, v_d, ω_g, v_c, see Fig. 3) have been varied in a genetic optimization run. The boundary conditions of this optimization were the stability of the closed loop and the capability of the actuator to lift the rotor from the axial stop to the force-free axial equilibrium position. The first objective was the compliance of the vibration norm [6]: the calculated curve should not cross the filled area shown in Fig. 5. A second optimization criteria was a low amplification of the position- and the current-sensor noise. Fig. 5 shows the calculated transfer function for an optimized set of controller parameters. Thus, it is possible to meet the requested transfer function in the case of a rigid stator–housing connection.

Elastically Mounted Stator

It has been demonstrated that the desired axial stability can be achieved in case of a rigid stator-housing connection. However, a viscoelastic mounted stator is necessary for the minimization of the radial and tilt vibrations of the rotor and the considered system is now enhanced by a damping element (see Fig. 1). This section investigates the influence of this viscoelastic element on the stability of the closed loop. Fig. 6 shows the stabilizable accelerations of the housing. It reveals a good agreement between calculation and measurement. Although an optimization of the controller parameters was carried out in the same way as for the rigid stator–housing connection, no set of parameters could be found that satisfies the desired standard [6] for the selected damping element.

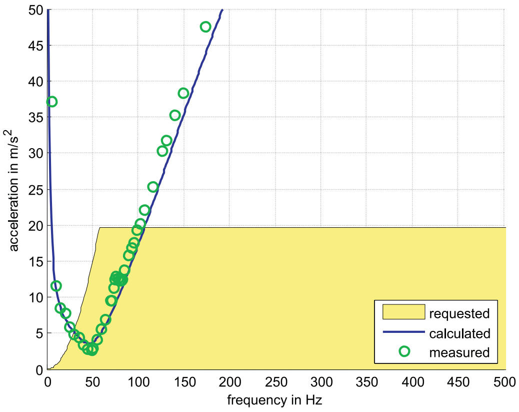

Fig. 6. Stabilizable accelerations of the housing for an optimized set of controller parameters. The requested transfer function cannot be achieved with the used damping element.

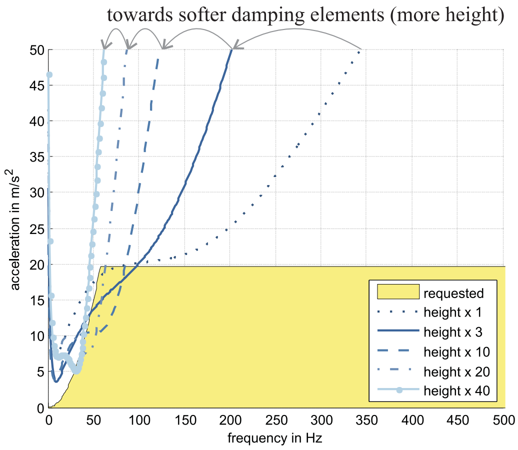

Fig. 7. Stabilizable accelerations of the housing for varying axial stiffness of the damping element. The controller parameters are optimized for each configuration.

Fig. 7 reveals how the stiffness of the damping element influences the stability curve. It shows the stabilizable accelerations for different heights of a damping element made from the same viscoelastic material. The higher the damping element, the lower the axial stiffness. The damping value is directly proportional to the axial stiffness [5]. For each configuration a genetic optimization of the controller parameters was carried out as described in the previous section. For damping elements with either relatively low or relatively high axial stiffness controller parameters could be found so that the requested curve can be fulfilled. Fig. 7 reveals that systems built of damping elements with sufficient height (low axial stiffness) have a very stable behavior against external disturbances. This is particularly true for high excitation frequencies. Furthermore, one has to consider the axial deflections of the relative rotor–stator position z as well as the axial deflections z_s of the stator. They may violate the specified values if the stiffness of the damping element is getting too low.

Conclusion

This paper shows a systematic way to analyze the stability of an active magnetic bearing against external disturbances. The investigations reveal that the choice of the appropriate controller parameters is crucial whether a requested acceleration curve is achievable or not. The influence of the five controller parameters of the used control scheme is not trivial, since the controlled system is of order five (rigid stator–housing connection) or twenty (elastically mounted stator). Therefore, a genetic optimization was carried out to find the appropriate set of controller parameters.

In the case of a rigid stator–housing connection the closed loop system can meet the acceleration curve (taken from the vibration norm) in axial direction. However, the radial and tilt vibrations of the rotor are nearly undamped for a stiff mounted stator. That is why an elastically mounted stator is necessary for the investigated type of magnetic bearing.

In the case of a viscoelastic stator–housing connection the stability against sinusoidal vibrations depends on the stiffness of the damping element. For very high stiffnesses similar transfer functions (between an external axial vibration and the current of the active magnetic bearing) as in the case of the rigid stator–housing connection are achieved. For relatively low stiffnesses (e.g. soft material, damping element cross-section with big height/width ratio) it is also possible to meet the acceleration standard. In the analyzed construction exists an area between these two extremes where it is not possible to find controller parameters so that the desired stability of the active magnetic can be achieved. The calculated transfer functions have been successfully verified by measurements.

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