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ACTIVE COMPENSATION OF THE EIGEN-DYNAMICS OF ELECTROMAGNETIC ACTUATORS BY ECU-BASED NON-LINEAR FEEDBACK CONTROL

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ABSTRACT

Increasing demands on the dynamic behaviour of industrial machines lead to a rising use of active elements for an active optimization of the dynamic properties of those systems. Especially in mechanical engineering applications electromagnetic actuators with a unidirectional motion are often the most suitable ones. For the use of those actuator types in industrial systems a linear dependence of the actuator-force on the control current and an independence of the force from the displacement of the pull-disk in the magnetic field are most desirable.
In this paper, investigations are presented of an electromagnetic actuator with high level permanent-magnets to realize the bias-flux. But besides many advantages, the use of high level permanent-magnets also leads to a high instability and a poor controllability of the system over the entire operating range.
By the use of an electronic-control-unit (ECU)-based non-linear feedback control the negative stiffness of the system could be eliminated and the non-linear dependence of the actuator-force on the control current could be linearized. Above that, the actuator dynamics have been compensated by an active compensation of the inertia forces. Thus, in order to further develop the actuator towards a compact machine tool for an easy integration into mechanical systems, an electromagnetic actuator has been realized that can be handled with classical linear control concepts.

1 INTRODUCTION

Actuators are used in many fields of mechanical engineering since long. They can be found in applications where a defined vibration excitation is needed, like in shakers and screening technology, and they are also often applied to generate exact positioning movements. Furthermore, there is an increasing number of applications where actuators are employed for the active optimization of the dynamic behaviour of technical systems. This development can mainly be attributed to a rising use of lightweight constructions combined with increasing operating speeds at the same time. This leads more and more to structural vibration problems. Commonly, these vibrations are eliminated by measures of passive damping. But these methods can only achieve a limited reduction of the system vibrations which is not satisfying in many cases. Significant improvements, especially in mechanical engineering and in auto motive applications, can only be realized by the use of active control [1, 2, 3].
Actuators are key-elements in the process of active optimization because they realize the physical energy transfer that is needed for the control actions. Thus, in recent years emphasis has been placed on the development of powerful actuators. This led to a wide range of different actuator types based on multiple physical effects [4]. Most of these systems (apart from the piezoelectric actuators and, with limits, also the magnetic bearings) are prototypes especially developed for one application.
For a lot of industrial applications in mechanical engineering actuators are needed which can generate forces in the kN-range, strokes in the mm-range and frequencies up to 500 Hz. These characteristics should be coupled with a robust and compact design. Moreover, a linear dependence of the force on the control current and a force behaviour independent from the stroke are desired. Here, electromagnetic actuators are often the most suitable ones due to their characteristic advantages with respect to their performance. Above that, an additionally improved behaviour can be realized by an ECU-based com-

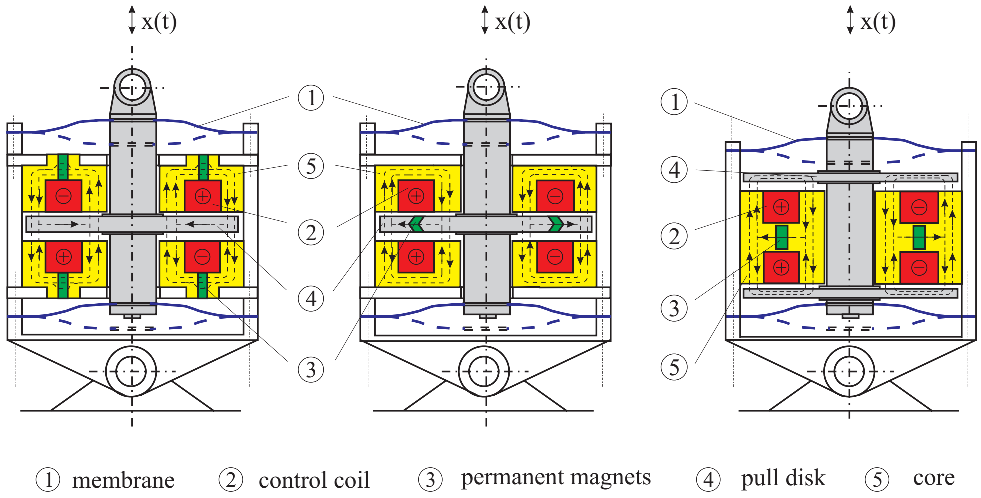

FIGURE 1: Different designs of the electromagnetic actuator

pensation of the nonlinear influences of the control force on the pull-disk position in the magnetic field as well as a linearization of the force-current dependence.

In this paper, an electromagnetic actuator with a unidirectional function will be investigated [5]. With this actuator the desired linearization of the force-current relation and the elimination of the system's axial stiffness will be discussed and realized by a non-linear ECU-based feedback control. Additionally, the actuator dynamics will also be compensated with the goal to realize an electromagnetic actuator with an exclusive proportional output control force-control current relation.

2 OPERATING PRINCIPLE OF THE ELECTROMAGNETIC ACTUATOR

The investigated actuator system is a unidirectional electromagnetic actuator that can generate forces up to ±600 N, a maximum stroke of 1 mm and has a cut-off frequency beyond 500 Hz. This performance is realized with a compact design at a volume of 1.4 dm^3 (diameter: 100 mm, height: 180 mm). The operation principle of the electromagnetic actuator can be seen from Figure 1, where three different designs of the presented actuator are sketched.

The axial forces of the actuator are generated by a system consisting of a pull-disk, fixed on a shaft and placed between two oppositely mounted pot-shaped magnets, each with an integrated control coil. Two annular membranes are used for a frictionless support of the shaft. This gives the system a high flexibility in axial and a high stiffness in radial direction. To generate the actuator force, a high energy pre-magnetization circuit (the bias-flux) is superimposed by a magnetic control field, excited by the control coils operated in a differential mode. To achieve a control force of several 100 N the pre-magnetization of the actuator is realized by strong permanent magnets. The solid core of the pot-shaped magnets is produced out of a soft magnetic material with the objective to minimize the arising eddy-current-effects at higher frequencies. This results into a more favorable dynamic behaviour.

The use of permanent magnets for a strong pre-magnetization leads to several advantages like high forces of several 100 N together with a simultaneous reduction of the actuator volume, less energy consumption and a drop of the system temperature compared with electrically pre-magnetized actuators. However, a main disadvantage of the use of strong permanent magnets is that a very high pre-magnetization leads to a strong negative stiffness, i.e. a strong mechanical instability and a bad controllability of the system over the entire operating range.

Investigations on the use of permanent magnets for the pre-magnetization of actuator systems have been done in [6] and they show that the manner of integration of the permanent magnets (e.g. integration in the core or in the pull-disk) has a significant influence on the actuator's performance. Figure 2 shows the test rig for the investigations.

3 NON-LINEAR FORCE BEHAVIOUR OF THE ACTUATOR

The static force behaviour of the electromagnetic actuator is substantially marked by two phenomena. First, there is the spring-force F_{mem} of the two annular membranes and second, the non-linear dependence of the magnetic force $F_{mag}(i, s)$ on the control current i and the distance s between the

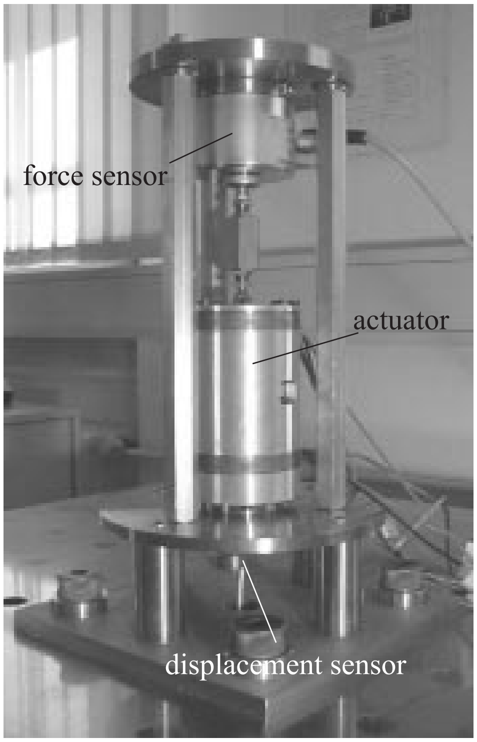

FIGURE 2: Test rig for experimental investigations

pull-disk and the pot-shaped magnets.

3.1 Influence of the membranes

The membranes of the actuator have a significant influence on the force behaviour of the system. Their main task is the guidance of the shaft and the mounted pull-disk with a high radial stiffness. Furthermore, the positive stiffness of the membranes reduces the negative stiffness of the pre-magnetization – in the ideal case, the negative stiffness can be completely eliminated. One of the main demands on the actuators's performance is a stroke in the mm-range. To reduce the mechanical load of the membranes, only flexible membranes with small thicknesses t_{mem} shall be used in the actuator. On the other hand, membranes with small thicknesses lead to a strong non-linear relation between the elastic-force F_{mem} and the deflection x of the center point of the membranes (see Figure 3). The spring-force and the deflection, respectively, can be approximately calculated by applying the energy method which leads to

$$\begin{aligned} F_{mem} &= -\frac{E \cdot t_{mem}^3}{(1-\nu^2)} \left\{ a_1 + a_2 \cdot \left(\frac{x}{t_{mem}} \right)^2 \right\} \cdot x \\ &= -k_{mem}(x) \cdot x \qquad , \end{aligned} \tag{1}$$

where E represents the YOUNG modulus of the spring steel, ν denotes the POISSON ratio and a_1 and a_2 are geometrical parameters of the membranes. Enhanced investigations on the design and the load distribution in these elements have been executed in [7].

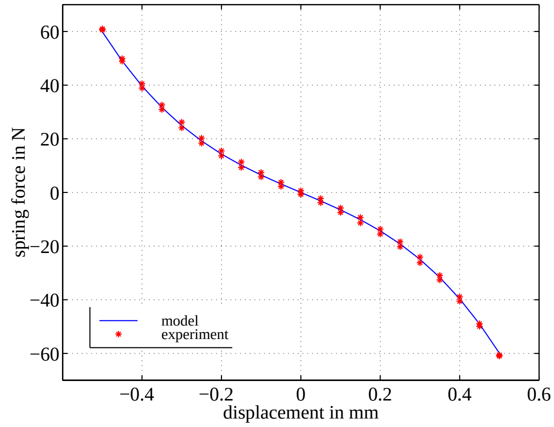

FIGURE 3: Non-linear spring-force of the membranes

3.2 Non-linear magnetic force

As already described above, two oppositely mounted pot-shaped magnets with differential control coil arrangement are used to generate the magnetic force F_{mag}. Together with the spring-force of the membranes the actuator will finally generate the system force

$$F = F_{mag} + F_{mem} \qquad . \tag{2}$$

The magnetic force field, covered in [6] in great depth, results from the two coil-magnets. The attainable magnetic force of the actuator for the quasi-static case is a function of the control distance $s = x_0 \pm x$ (where x_0 represents the initial air-gap) and the magnetic flux $w \cdot i$ due to the control current i

$$\begin{aligned} F_{mag} &= F_{mag,1} - F_{mag,2} \\ &= k_1 \cdot \left\{ \frac{(\Theta_{pm} + w \cdot i)^2}{\left(R_{pm} + k_2 \cdot (x_0 - x)\right)^2} - \ldots \right. \\ & \qquad \left. \ldots \frac{(\Theta_{pm} - w \cdot i)^2}{\left(R_{pm} + k_2 \cdot (x_0 + x)\right)^2} \right\} \end{aligned} \tag{3}$$

where w is the number of loops in the coil, Θ_{pm} is the bias-flux generated by the permanent magnets, R_{pm} stands for the magnetical resistance of the permanent magnets and k_1 and k_2 represent the geometric and electric parameters of the system. Finally, the control force of the actuator system results out of the non-linear functions for the magnetic force and the spring-force, equations (1) and (3), to the following description

$$F = k_i(i,x) \cdot i + [k_x(i,x) + k_{mem}(x)] \cdot x. \tag{4}$$

non-linearities in equation (4) cannot be neglected for large displacements of the pull-disk and high control currents. Therefore, a linearization of this equation around a working point of the operating range is not appropriate.

Since an analytical description of the magnetic forces is scarcely possible with respect to all non-linearities (like the magnetical saturation or the influence of the heat generation by the current coils) a formulation of the magnetic forces with respect to these restrictions will not be discussed in this paper. Instead, for the description of the compensation of the influence of the pull-disk displacement on the actuator force, a non-linear characteristic field $F(i,x)$ is numerically drawn out of experimental results, shown in Figure 4. The measuring points are used as knot points for

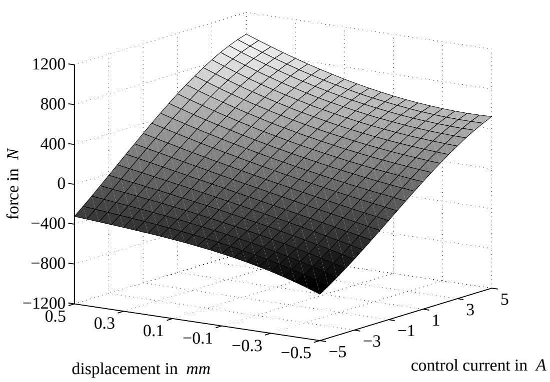

FIGURE 4: Performance graph of the actuator force

the interpolation of the measurement by a regression field of order three. Thus, equation (4) can be written in the form

$$\begin{aligned} F(i_m, x_n) &= k_i(i_m, x_n) \cdot i_m + \ldots \\ & \quad [k_x(i_m, x_n) + k_{mem}(x_n)] \cdot x_n, \end{aligned} \tag{5}$$

where i_m and x_n represent a working point somewhere within the operating range.

4 ACTIVE COMPENSATION BY ECU-BASED NON-LINEAR FEEDBACK CONTROL

4.1 Compensation of the system stiffness and linearization of the force-current-relation

To linearize a non-linear system two different procedures can be chosen. Either the system will be 'linearized' by means of an optimized robust control concept or the non-linear effects are actively compensated using a feed back mechanism. Using the second possibility, a linearized system with an easy handling can be realized and the dynamic behaviour of the system can subsequently be treated using classic linear control concepts.

With respect to the second way, [8] has to be mentioned. There, the non-linear force field of a magnetic bearing was successfully linearized by the use of a non-linear compensation.

In this paper, the compensation method shall be applied to the presented actuator. A non-linear compensation field is used to take into account the displacement influence on the actuator force and to eliminate by this way the strong negative stiffness and the poor controllability of the system. In addition to the compensation of the negative stiffness of the actuator, the remaining non-linearity of the magnetic forces in relation to the control current will also be compensated. To determine the required compensation-field, for every working point (i_m, x_n) of the non-linear force behaviour presented in Figure 4, a compensation current $i_{c,x}(i_m, x_n)$ has been calculated to eliminate the force resulting from a displacement of the pull-disk. Above this, an additional current $i_{c,lin}(i_m)$ has been determined to eliminate the non-linear dependence of the magnetic force on the control current. By summing these two compensation components to

$$\begin{aligned} i_c(i_m, x_n) &= i_{c,s}(i_m, x_n) + i_{c,lin}(i_m, x_n) \\ &= \frac{(k_i^* - k_i\,(i_m, x_n)) \cdot i_m}{k_i(i_m, x_n)} - \ldots \\ & \quad \frac{(k_x\,(i_m, x_n) + k_{mem}\,(x_n)) \cdot x}{k_i(i_m, x_n)} \end{aligned} \tag{6}$$

the final field of compensation currents $i_c(i_m, x_n)$ is generated, shown in Figure 5. This field is then

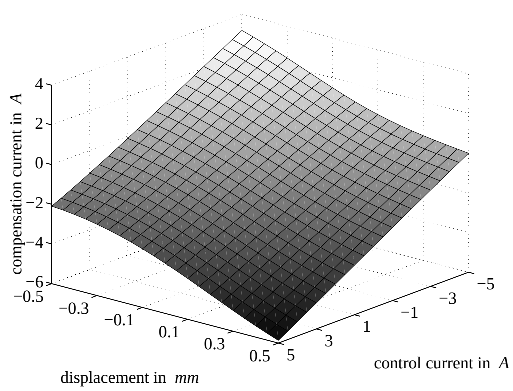

FIGURE 5: Characteristic field of the compensation currents $i_c(i_m, x_n)$

loaded onto an ECU and by adding this currents $i_c(i_m, x_n)$ to the control current i_m an actuator with a linear force-current-behaviour and no system-stiffness is generated:

$$F(i_m) = k_i^* \cdot i_m. \tag{7}$$

The parameter $k_i^*(i_m, x_n)$ in equation (7) represents the new constant force-current-factor resulting from the realized compensation and linearization.

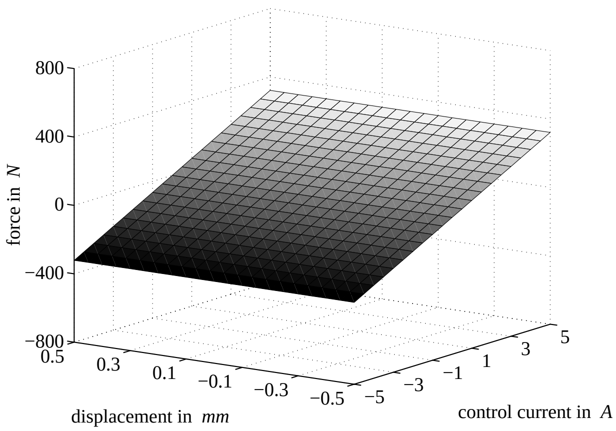

FIGURE 6: Linearized and compensated force-field

Figure 6 shows the calculated force field described by equation (7). Figure 7 contains experimental results of force-step responses carried out by the initial and the compensated actuator. For both cases, the step-response is measured at three different positions x_n of the movable pull-disk.

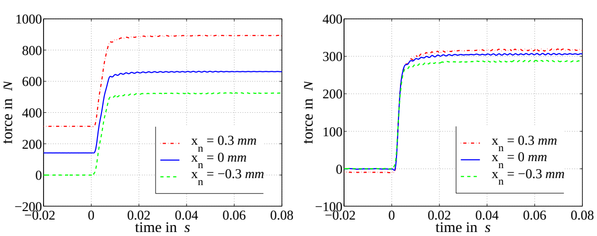

FIGURE 7: Force-step responses; left: initial actuator, right: compensated system

4.2 Compensation of the inertia forces

The transfer function $F(t)/u(t)$ of the magnetic actuator's force as a function of the input voltage of the power amplifier of the investigated system is characterized by the dynamics of the used power amplifier and the electromagnets. The amplifier's transfer characteristic $g_{amp}(t)$ can be described as a PT_1-element with a delay time T_t. Thus, it is taken into account that the amplifier does not work in a no-load operation and does not generate a proportional current to an input voltage at high frequencies. The transfer characteristic $g_{mag}(t)$ of the electromagnets is caused by the magnetic reversal losses and can be modeled as a PT_1-element, too. Thus, the transfer function of the complete magnetic circuit extended by another PT_2-element representing the dynamic properties of the used test rig ($g_{tr}(t)$) will be described in the LAPLACE-domain as

$$\begin{aligned} G(s) &= \frac{F(s)}{U(s)} = G_{tr}(s) \cdot G_{mag}(s) \cdot G_{amp}(s) \\ &= \frac{1}{T_{2,tr}^2\, s^2 + T_{1,tr}\, s + 1} \cdot \frac{k_i}{T_{mag}\, s + 1} \cdot \ldots \\ &\quad \ldots \frac{k_{amp}\, \mathrm{e}^{-T_t\, s}}{T_{amp}\, s + 1}, \end{aligned} \tag{8}$$

where k_i is the force-current-factor, k_{amp} represents the proportional factor of the power amplifier and $T_{1,tr}$, $T_{2,tr}$, T_{mag} and T_{amp} are the time constants of the test rig, of the magnets and of the amplifier. The model and the experimental results are presented in Figure 8.

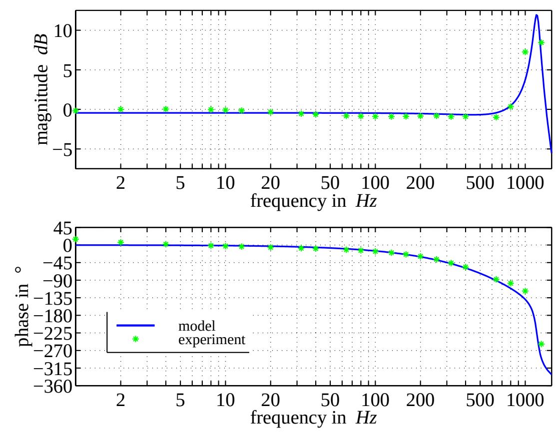

FIGURE 8: Transfer characteristic of the electromagnets (effects above 500 Hz are due to test rig)

After the negative stiffness of the actuator has been compensated, the dynamic behaviour of the electromagnetic actuator is still affected by the eigen-dynamics of the actuator caused by the inertia of the moving masses (shaft and the pull disk). To further develop the actuator towards a machine tool with a proportional dynamic force-current-behaviour at the interface between actuator and controlled system, the inertia force shall be compensated too by applying appropriate counteracting forces onto the system. Therefore, the axial acceleration $\ddot{x}(t)$ of the pull-disk is measured and the inertia forces are calculated on-line and then fed back. The effect of the compensation of the eigen-dynamics is presented using the actuator with compensated displacement-influence which has been transformed into a simple

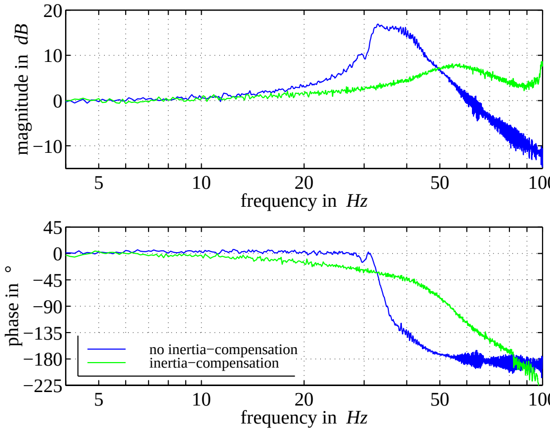

FIGURE 9: Influence of the compensation of the eigen-dynamic on the transfer characteristic

spring-mass-damper-system with an eigen-frequency of 35 Hz using a classical PD-controller. In Figure 9 the non-compensated and the eigen-dynamics-compensated systems are presented. The required counteracting-forces $F_c(t)$ are determined by the movable mass m, the applied stroke x and the actuation frequency f and can be calculated by

$$F_c(t) = m\,x(t)\,(2\pi f(t))^2 \leq \max\left[|F(i_m,t)|\right].$$

5 CONCLUSIONS

Actuators are key elements in the active optimization of the dynamics of machines.

In this paper, investigations of an electromagnetic actuator with a pre-magnetization realized by strong permanent magnets are presented. The use of the permanent pre-magnetization leads to a non-energy-consuming but strong negative stiffness and a poor controllability of the actuator. These effects are mainly due to the non-linear stiffness of the membranes, the strong negative stiffness of the magnetic system and the non-linear relation between the magnetic force and the control current. By eliminating the non-linearities of the system, using an ECU-based compensation mechanism, the disadvantages could be overcome and a proportional force-steering voltage behaviour of the actuator could be realized that is independent from the displacement of the pull-disk. In addition to the compensation of the negative stiffness and the linearization of the force behaviour, the inertia forces of the moving parts were compensated, too. Thus, an electromagnetic actuator with a proportional characteristic in the quasi-static *and* dynamic range up to a limiting cut-off frequency was realized.

The future research work will focus on the extension of the realizable stroke together with increasing actuator forces, a frequency range up to 300 Hz and towards a more compact and robust design.

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