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INVESTIGATIONS IN A SOFTWARE-BASED DESIGN OF LINEAR ELECTROMAGNETIC ACTUATORS

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ABSTRACT

In a lot of industrial applications actuators are required to realize an active optimization of the static and dynamic behavior of technical systems. Here, electromagnetic actuators are often the most suitable ones due to their characteristic performance. They are integrated in mechatronic systems, where they have to realize the physical energy transfer required e. g. in an active vibration excitation or an active vibration isolation. This paper deals with an electromagnetic actuator that realizes unidirectional control forces of about 1 kN and strokes of 2 mm. The actuator forces are simulated by means of a parametric nonlinear model, that is also used for an improvement of the actuator design. Furthermore, an enhancement of the system performance is realized with an electronic-control-unit (ECU) based feedforward controller. It is designed based on the developed parametric model and is used to linearize the static and dynamic force behavior of the electromagnetic actuators. Finally, the optimized system is tested by means of an active vibration excitation, used in the automotive industry to analyze vehicle components.

INTRODUCTION

In the last years, the number of mechatronic solutions to improve the dynamic behavior of technical systems in mechanical or in automotive engineering is increasing [1]. Hereby, actuators are the connectors between the technical system and the technology that processes the information about the system – e.g. a controller strategy. Caused by the fact that the actuator realizes the physical energy transfer, needed for the control actions, it will be a significant key element in the process of active optimization. Here, active vibration excitations like in shakers, exact positioning movements or active vibration isolations are required for a significant improvement of the static and dynamic behavior of technical systems.

The choice of a suitable actuator type depends on the application. Especially a lot of implementations in the already mentioned branches of industry need actuators that can generate unidirectional control forces in the kN-range as well as medium strokes of about 2-3 mm. The required frequency range depends on the application, but mostly, a frequency band of $0\,\mathrm{Hz} \leq f \leq 250\,\mathrm{Hz}$ is demanded. These characteristics should be coupled with a very compact and robust design and a simple transfer characteristic. With respect to these demands, electromagnetic actuators are often the most suitable ones for a lot of operations [2]. Nevertheless, there are just a few applications where electromagnetic systems are in use, normally as prototypes. One reason for its rare use is the very strong nonlinear behavior between the control forces, the control currents and the strokes of the system and the extensive optimization process for a model-based linearization of the actuator.

Thus, the aim of this paper is the realization of unidirectional electromagnetic actuators with the above required performance and a simple transfer characteristic. Therefor, some design rules are discussed and a realized electromagnetic actuator is presented. The developed system has a very strong premagnetization, caused by high energized permanent magnets. This leads to an enormously strong instability and a bad controllability of the actuator. Hence, the use of permanent magnets to generate the bias-flux seems to be a significant disadvantage. But relief could be achieved using a digital nonlinear feedforward control to compensate the negative nonlinear stiffness of the actuator and to linearize the dependence of the actuator forces on the control current. The optimized electromagnetic actuator is finally tested by means of an active vibration excita-

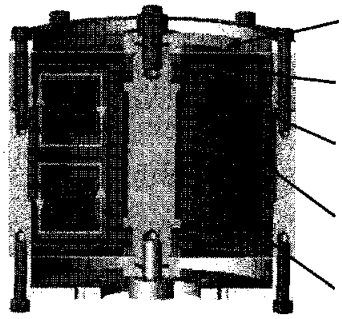

FIGURE 1: Design of the electromagnetic actuator

tion, an application used in automotive engineering to analyze vehicle components.

ACTUATOR DESIGN

The cross-sectional view in Figure 1 shows the design of the investigated electromagnetic actuator. It mainly consists of two pull-disks, fixed on a shaft and positioned at the outer ends of a solid pot-shaped magnetic core. The shaft is connected to the outer shell by means of two annular membranes. The membranes realize a frictionless support of the movable shaft with a very high radial stiffness and axial flexibility. The magnetic core is designed out of a NiFe-based soft-magnetic material with the objective to minimize eddy-current-effects at higher frequencies. The bias-flux is generated by high energized permanent magnets, integrated in the magnetic core as shown in Figure 1. To realize tensile as well as compressive forces, two control coils, working in a differential mode, are integrated in the pot-shaped magnet. Thus, a first linearization of the dependence of the magnetic forces on the control currents is achieved and, as a further advantage, only one power-amplifier is required to control the actuator. In this work, a power-amplifier of 300 W is used that is working as an ideal current-amplifier to control the actuator in a no-load operation at higher frequencies.

To generate the actuator forces, the high energized pre-magnetization circuit (signed as a pale grey line in Figure 1) is superimposed by the magnetic control field (black line), produced by the control coils. Due to the high energy bias-flux, the investigated electromagnetic actuator realizes control forces in the kN-range with strokes of $\Delta x = 2\,\mathrm{mm}$. This performance is realized with a compact design at a volume of $V = 1.35 \cdot 10^{-3}\,\mathrm{m}^{-3}$ that corresponds to a height of $h = 0.11\,\mathrm{m}$.

DESIGN ASPECTS

The maximum achievable actuator forces of systems presented in the last section, mainly depend on different design parameters.

- Choice of the soft-magnetic material with a suitable magnetic flux density B and the required pole faces A_δ for the design of the magnetic core. Caused by the relationship (with the permeability μ)
$$F_M = \frac{B^2 A_\delta}{2\mu}$$
especially the magnetic flux density influences the magnetic force F_M. The magnetic materials with the highest flux densities, currently available, are CoFe-alloys with $B \approx 2,3\,\mathrm{T}$. Compared to a NiFe-alloy with $B \approx 1,55\,\mathrm{T}$, CoFe-materials can realize more than double high magnetic forces.
- Design of the bias-flux. Here, the selections of the magnetic material (NdFeB-alloys allow highest remanent inductions and major coercive forces), the outer dimensions of the high energized permanent magnets and its positioning in the magnetic core are significant for the performance of the electromagnetic actuator.
- Dimensioning of the control coils.
- Design of the membranes integrated in the electromagnetic actuator. The main task of the membranes, working like springs, is the guidance of the shaft. Therefore, a high radial stiffness is necessary. Though, the actuator has to realize strokes in the mm-range. On the one hand side, this demands axial flexibilities, high enough to minimize the mechanical load of the membranes. On the other hand side, the membrane stiffness reduces the strong negative stiffness of the magnetic system. Hence, a compromise has to be made between the needed axial flexibility and the useful high membrane stiffness.

These design-parameters have to be co-ordinated in a numerical computation procedure with the aim to gain maximum control forces.

MODEL OF THE ACTUATOR

Quasi-static force behavior

The control forces F of the investigated electromagnetic actuator

$$F(i,x) = F_M(i,x) - F_S(x) \tag{1}$$

results from the superposition of the magnetic forces F_M and the restoring forces F_S of the used membranes. Both are nonlinear functions depending on the control current i and the displacement x of the pull-disk.

For the quasi-static case, the reluctance forces of the investigated electromagnetic actuator result from the difference between the magnetic forces generated by the two control coils and are calculated by

$$F_M(i,x) = k_i(i,x)\,i + k_x(i,x)\,x \qquad (2)$$

with the parameter $k_i(i,x)$ standing for the force-current-factor and $k_x(i,x)$ representing the negative stiffness of the magnetic system. Both factors are nonlinear functions depending on different electrical, magnetical and geometrical parameters (k_1, k_2) of the actuator design. The force-current-factor is calculated, using

$$\Lambda = \frac{k_\mu^2(i,x)\,k_1}{k_2\left((2\,R_{pm} + k_2\,x_0)\,x_0 - k_2\,x^2\right)^2} \quad ,$$

by means of

$$k_i(i,x) = \Lambda\,w\,\Theta_{pm}\left((2\,R_{pm} + k_2\,x_0)\,x_0 + k_2\,x^2\right)$$

respectively the force-displacement-factor with

$$k_x(i,x) = \Lambda\left(\Theta_{pm}^2\,k_2\,x_0 + w^2\,i^2\,(2\,R_{pm} + k_2\,x_0)\right).$$

The factor Λ takes into account the saturation effects of the used soft-magnetic material by the parameter

$$k_\mu(i,x) = \frac{\mu_r(H)}{\mu_{r,\max}} \quad ,$$

described in more detail in [3]. The coefficient k_μ considers the dependence of the relative permeability $\mu_r(H)$ on the magnetic field strength H. In the equations to calculate the describing force-factors k_i and k_x, the bias-flux is represented by Θ_{pm} and the reluctance of the permanent magnets is R_{pm}. The initial air-gap between one pull-disk and the magnetic core is denoted by x_0 and the number of windings of the control coil is represented by the parameter w.

The restoring forces of the two annular membranes are calculated with

$$F_S = \frac{E\,t_S^3}{(1-\nu^2)}\left\{a_1 + a_2\left(\frac{x}{t_S}\right)^2\right\} = k_S(x)\,x, \quad (3)$$

where E is the Young's modulus, ν the Poisson ratio, t_S stands for the thickness of the membranes and a_1 and a_2 are geometrical parameters.

The control forces of the electromagnetic actuator according to equation (1) result from the equations (2) and (3). They are described with

$$F(i,x) = k_i(i,x)\,i + \left(k_x(i,x) - k_S(x)\right)x. \qquad (4)$$

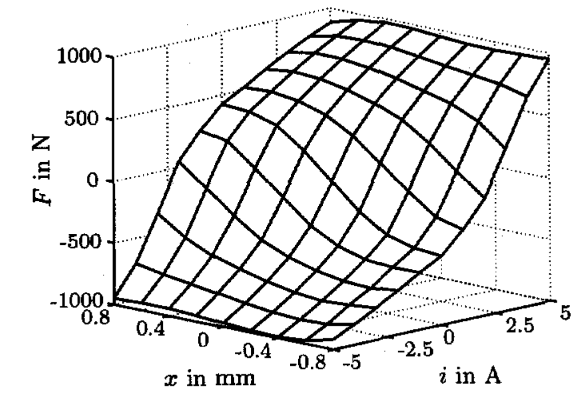

FIGURE 2: Force-field of the actuator

Figure 2 shows the quasi-static force-field $F(i,x)$, obtained by variation of the control current i and the displacements x.

Transfer characteristic

The dynamics of the investigated system are specified in the Laplace-domain by the transfer function $G_S(s) := \mathscr{L}\{F(t)/i(t)\}$. It describes the actuator forces depending on the control current and is characterized by the dynamics of the magnetizations $G_M(s)$ and the power amplifier $G_A(s)$. The transfer behavior of the electromagnets is caused by the reversal losses in the solid core evoked by hysteresis and eddy-current effects. It is described my means of a PT_1-element extended by a rational function that lowers the amplitude and the phase. The power amplifier corresponds to a PT_1-element and takes into account that the amplifier does not control the actuator in a no-load mode at high frequencies as a result of the limited output-voltage of $|u_{o,\max}| = 40\,\text{V}$. This results to the transfer function

$$G_S(s) = \frac{k_i\,k_A\,(1 + s\,T_{M,n})}{(1 + s\,T_M)\,(1 + s\,T_{M,d})\,(1 + s\,T_A)}\,, \quad (5)$$

where $T_{M,n,d}$ represents time parameters ($T_{M,n} > T_{M,d}$) of the electromagnet and k_A and T_A are coefficients determining the dynamic behavior of the power amplifier. The frequency response function $G_S(j\omega)$ is shown in Figure 5 as the dotted line and is compared with experimental results.

LINEARIZATION OF THE FORCE BEHAVIOR

The strong nonlinear relationship between the control forces, the control currents and the displacements results from the saturation effects of the magnetic material and the nonlinear dependence of the control forces on the displacements. To realize an

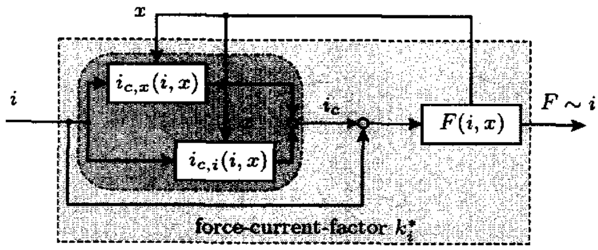

FIGURE 3: Schematic of the ECU-based feedforward control

electromagnetic actuator with a simple transfer characteristic all significant nonlinearities and the strong negativ system-stiffness has to be linearized and compensated, respectively. To linearize the magnetic force-field, a nonlinear feedforward controller is inserted (see also [4]), already practiced in [5] at electromagnetic actuators with a pre-magnetization by permanent magnets. Figure 3 shows a scheme of the used ECU-based feedforward strategy. The essential compensation currents $i_c(i,x)$ are computed with the equation

$$\begin{aligned} i_c(i,x) &= i_{c,x}(i,x) + i_{c,i}(i,x) \\ &= \frac{k_S(x) - k_x(i,x)}{k_i(i,x)}\,x + \frac{k_i^* - k_i(i,x)}{k_i(i,x)}\,i \end{aligned} \tag{6}$$

based on the already modelled coefficients k_i and k_x. The factor k_i^* in equation (6) represents the new, constant force-current-factor of the optimized elec-

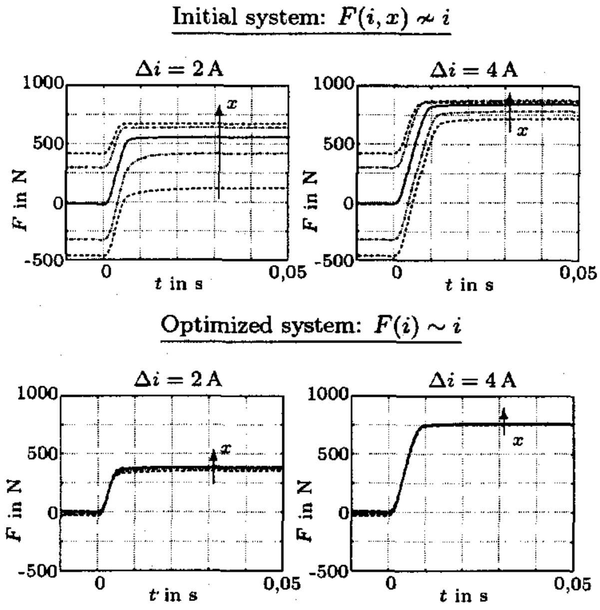

FIGURE 4: Force-step responses at different displacements: $--$ $x = \pm 0,6\,\mathrm{mm}$, $-\cdot-$ $x = \pm 0,3\,\mathrm{mm}$, — $x = 0\,\mathrm{mm}$

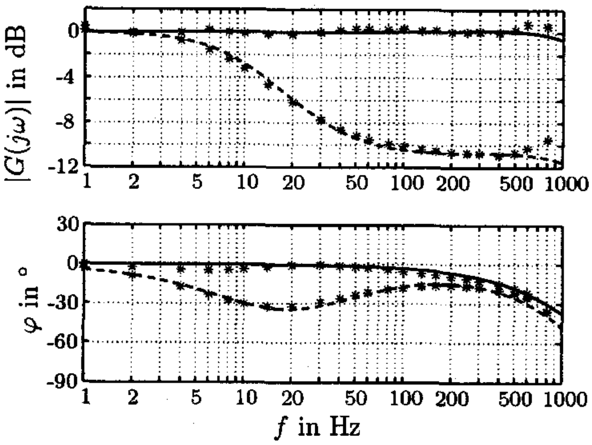

FIGURE 5: Frequency responses of the electromagnetic actuator: — optimized system $G_S^*(s)$, $--$ initial system $G_S(s)$, $*$ experiment

tromagnetic system. Figure 4 shows force-step responses of the initial and optimized electromagnetic actuator. The active optimization by means of the nonlinear feedforward control has realized an actuator with no system-stiffness and a linearized static force behavior with a constant force-current-factor $k_i^* \approx 190\,\mathrm{N/A}$.

The dynamics of the electromagnetic actuator has a significant drop of the amplitude and phase response. This has to be compensated, too. Therefor, the additional transfer function

$$G_c(s) = \frac{1 + s\,T_{M,d}}{1 + s\,T_{M,n}} \tag{7}$$

is connected in incoming circuit to the feedforward controller, that is shown in Figure 3. This results in the dynamic behavior $G_S^*(s) = G_S(s)\,G_c(s)$ of the optimized electromagnetic actuator presented in Figure 5. Here, the amplitude response of the transfer function $G_S(s)$ is compensated up to frequencies of 500 Hz and the dynamics correspond to a proportional element aside from the remained phase lag.

CONTROL UNIT DESIGN

In this section the optimized electromagnetic actuator is tested by means of an industry like application. Therefor, an active vibration excitation is selected. A mass Δm, representing e. g. a vehicle component, that has to be analyzed, is fixed on top of the developed electromagnetic actuator. Figure 6 shows a photograph of the experimental system.

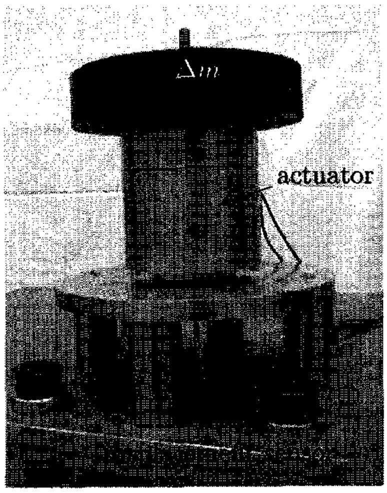

FIGURE 6: Experimental setup for active vibration excitation

The active optimization of the force behavior leads to an electromagnetic actuator with no system stiffness and a simple transfer characteristic. On that score, often in contrast to a lot of applications of active magnetic bearings, the improved system can be stabilized over the whole operating range of the actuator by means of a classical linear control concept. Figure 7 shows the block diagram of the closed control loop.

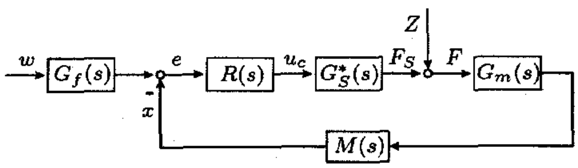

FIGURE 7: Control loop of the optimized system

The reference reaction of the control loop

$$G_{cl}(s) := \frac{X(s)}{W(s)} = \frac{G_0(s)\, G_f(s)}{1 + G_0(s)} \tag{8}$$

with

$$G_0(s) = M(s)\, G_m(s)\, G_S^*(s)\, R(s)$$

is mainly characterized by the transfer function of a classical $PIDT_1$-controller $R(s)$ and the dynamics of the prefilter $G_f(s)$. In addition, $M(s)$ represents the used displacement sensor and corresponds to a proportional element. The transfer function $G_m(s)$ takes into account the dynamic behavior of the inertia mass Δm, supported in axial direction by the two annular membranes.

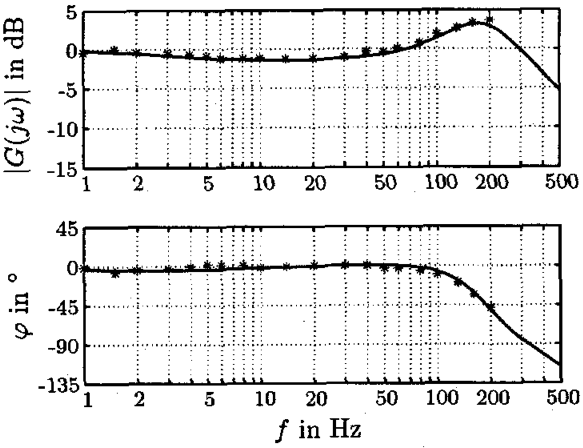

FIGURE 8: Stabilized electromagnetic actuator with an additional mass Δm: — model, $*$ experiment

The modelled frequency response $G_{cl}(j\omega)/G_f(j\omega)$ of the stabilized system is shown in the Bode-plot of Figure 8. The experimental results certify the well conformity of the developed model with the real system. Finally, with respect to the model-based design of the output control to operate the electromagnetic actuator, the prefilter has to be laid out. The aim of it is to realize a reference reaction of the control loop that corresponds to a quasi-proportional element. Therefor, the transfer characteristic of the prefilter $G_f(s)$ is adapted with the function

$$G_f(s) = \frac{\sum_m s^m\, b_m}{\sum_n s^n\, a_n} \quad , \quad n \geq m \tag{9}$$

to the ideal inverse transfer function $G_{cl,inv}(s) = G_{cl}^{-1}(s)\, P(s)$ where $P(s)$ considers the pole surplus. The frequency responses of $G_f(j\omega)$ and $G_{cl,inv}(j\omega)$

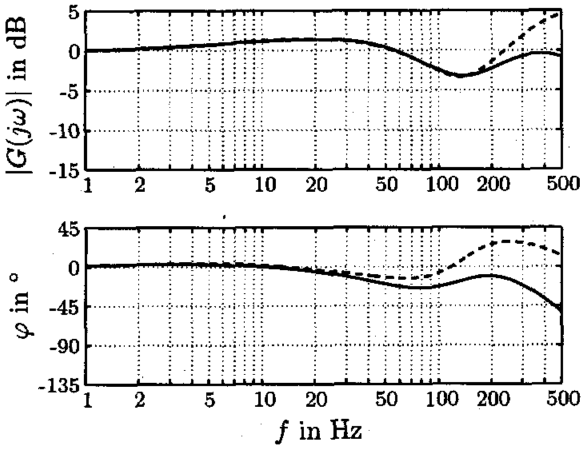

FIGURE 9: Design of the pre-filter: — $G_f(j\omega)$, -- $G_{cl,inv}(s)$

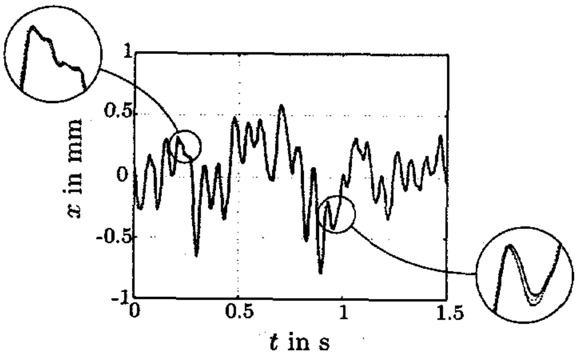

FIGURE 10: Experimental results of the positioning of a mass for an active vibration excitation: — output signal, – – input signal, ··· simulation

are shown in the diagram in Figure 9.

After designing the classical control unit, the operated optimized electromagnetic actuator is tested by an active vibration control. The aim of the selected application is, that the fixed mass Δm follow a generated displacement signal as exact as possible. The used signal corresponds to a twice integrated acceleration signal that is measured during a test drive of a car on a test track. Figure 10 shows a comparison of the simulated and measured output signal with the generated input signal. Obviously, good accordance between the experimental and simulation results is achieved.

CONCLUSIONS

Actuators are key elements in mechatronic systems. They realize the physical energy transfer necessary to achieve an active optimization of the dynamics of technical systems. Especially in a lot of industrial applications, in mechanical and automotive engineering, electromagnetic actuators are often the most suitable ones. They realize the demanded performance of actuation forces in the kN-range, strokes up to several mm and frequencies beyond 250 Hz.

In this paper, an electromagnetic actuator with a bias-flux realized by strong permanent magnets was presented. The use of the permanent premagnetization leads on the one hand side to a non-energy-consuming and higher magnetic forces but on the other hand to a strong nonlinear negative stiffness and a poor controllability of the actuator. Beyond this, the dependence of the control forces on the control currents are nonlinear, too. The nonlinear force-field was modelled and the mathematical description was used for an active optimization of the system transfer characteristic. By eliminating the non-linearities of the system, using an ECU-based feedforward control, the disadvantages could be overcome and a proportional force-current behavior of the actuator could be realized that is independent from the displacement of the pull-disk. Due to the simple transfer characteristic of the improved electromagnetic actuator, a classical linear control concept was applied to operate the system in an industry like application, where an active vibration excitation was realized.

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