MAGNETIC ACTUATOR DESIGN FOR MECHANICAL ENGINEERING APPLICATIONS

Heinz Ulbrich
University of Essen, FB 12, Chair of Mechanics, D-45117 Essen, Germany
Yong-Xing Wang
University of Essen, FB 12, Chair of Mechanics, D-45117 Essen, Germany
Jens Bormann
University of Essen, FB 12, Chair of Mechanics, D-45117 Essen, Germany

ABSTRACT
Ever increasing demands on the speed of rotating machine elements cause dynamic effects that lead to vibrations which must be taken into account in order to meet the required specifications of the machines. Significant improvements can be expected by the use of elements that can actively control the dynamics of those machines, but this requires novel actuators. Introduced in this paper are some recently developed actuators. Their main features will be highlighted, although special attention is only given to magnetic actuators. A comparison between the different types of actuators is presented to provide guidelines for engineers in deciding, which of them matches best their special application.

INTRODUCTION

The intention to increase the productivity or efficiency of machines leads in almost every field of mechanical and automotive engineering to dynamical problems that cannot longer be solved by a mere adjustion of the design parameters. Improvements of already existing machines and vast innovations in new developments will be made possible by the use of special actuators and electronics in combination with control techniques.
The heart of such new applications are the actuators that control - depending on the objectives - the energy household of the system. In order to achieve this, some promising actuator concepts will be introduced and their strong and weak points will be discussed with regard to their industrial applications. Objective of the paper is to provide guidelines for practically oriented engineers to help them choose the appropriate actuators for their specific application.

POSSIBLE ACTUATORS

In the future, the dynamics of machines will be specifically and on purpose adapted to the requirements of the particular technical task by the use of actuators and electronics in combination with control techniques. A key position will thereby be held by the actuators which have to perform the energy transfer. They have the task to transform the information about the system - supplied by the sensors - into the desired physical actions. To realize these regulating actions, multiple and also very different physical effects can be used. In the following some actuator concept will be discussed and commented upon their advantages and disadvantages.

Piezo Actuators

Piezo actuators are especially well suitable when dealing with vibrations of small amplitudes or to adjust mirror systems or other components in high precision constructions within the µm-range [1]. They have also been investigated theoretically and experimentally for their ability to influence rotors via conventional bearings [2], [3], [4] and mechanisms in general [5]. Piezo actuators special merits are their high stiffness k_p and their performance in high frequency ranges. The realizable control forces can be calculated by

$$f_p(t) = L_p C_p k_p u(t) - k_p s(t) \quad . \qquad (1)$$

With L_p the piled height of the crystal wafers, C_p the electrostatic capacity (each crystal wafer is a capacitor) and k_p the stiffness of the actuator. The regulating distance is called $s(t)$, the control voltage $u(t)$.
The crucial disadvantages that limit the application range are the only small realizable regulating distances ($s(t) \approx L_p/1000$) and the very low material damping of the actuators (impacts are transfered almost undamped). Due to the dependency of the regulating distance on the

total height, the size of piezoelectric actuators can be quite large. Also, the permissable compressive load is very small, so usually the control forces can only be applied through additional transfer elements [2]. A typical transfer function of a piezo actuator is shown in Figure 1. It can be seen that a strong reduction of the amplitude begins at already 200 Hz and the phase lag progresses linearly with the frequency. At 1000 Hz the phase lag is already -180°.

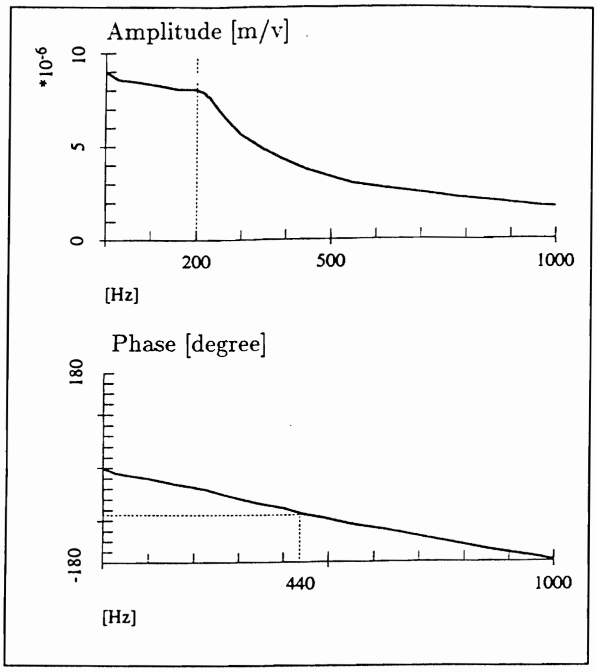

FIGURE 1: Measured frequency characteristics of a piezo actuator with regard to the regulating distance *s(t)* over a sine-shaped control voltage *u(t)*

Hydraulic Actuators

For a very different kind of applications a hydraulic actuator has been designed (Figure 2). The actually operating part consists of two cylindrical chambers which are sealed in working direction (vertical direction in Figure 2) by elastic membranes. The pressure difference Δp between the two chambers 1 and 2, necessary to create the regulating operation, is supplied by a servo valve.

The special design of the actuator yields a high radial stiffness and ensures a corresponding safety against tilting effects (depending on the distance between the two membranes as well as upon their thickness).

Through this arrangement, the regulating movements can be applied completly without friction. The length l_H of the hydraulic actuator is changed by the length *s(t)* due to the regulating action. The control force can be expresssed by an equation similar to that of the piezo system:

$$f_H(t) = A^* \Delta p(t) - k_M s(t) \quad (2)$$

With A^* the characteristic membrane area, $k_M = 2c_L$ the effective membrane stiffness, c_L the stiffness of one membrane and *s(t)* the regulating distance in the direction of the force. The regulating force is generated by the pressure difference Δp and is not (different to the piezo actuator system) proportional to the control voltage. Unfortunately, the regulating pressure strongly depends on all fluiddynamic effects of the hydraulic and the servo valve as well. Thus, the realizable regulating frequencies are mainly determined by the cut-off frequency of the used servo valve and the influence of the pipe elasticity and the fluid compressibility. The actuator has basically an integrating frequency characteristic with respect to the regulating distance [6].

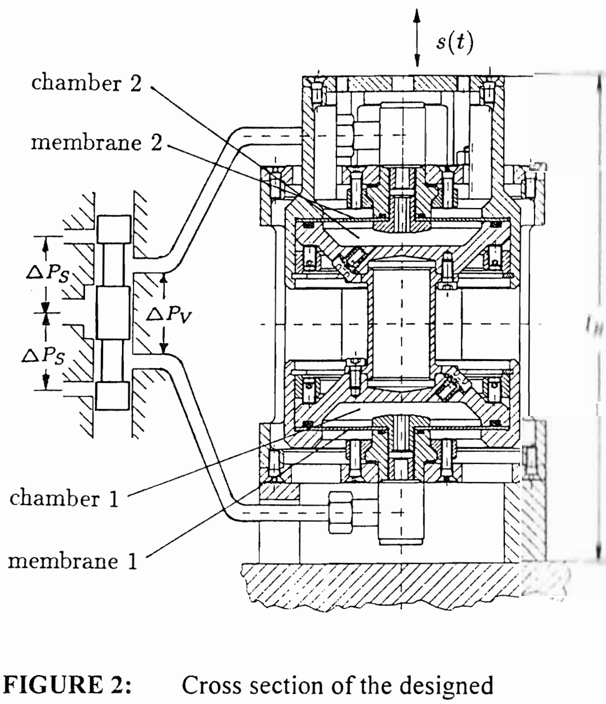

FIGURE 2: Cross section of the designed hydraulic actuator

Due to the choice of membranes (diameters, thickness) a multitude of statical characteristic curves can be realized. Also, the maximum control forces and regulating distances can be adapted in wide ranges with an adequate choice of the membranes. The given design limit is always the maximum material stress in the membranes. The thicker the membranes at a given diameter, the larger the stiffness of the actuator and the smaller the maximum regulating distance. Regarding this it should also be noted that the physics comply with the regulating actions since in the deflected state the controller generally has to apply forces in opposite direction to the deflection $s(t)$. This means the term $k_M s(t)$ from equation 2 will be superimposed by the control force $A \cdot \Delta p(t)$. The functional dependencies are shown in Figure 3.

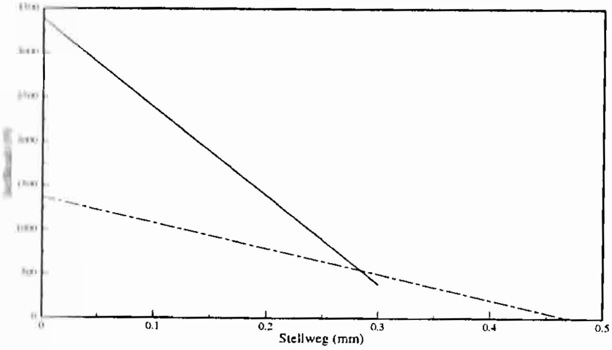

FIGURE 3: Force-distance plots for the hydraulic actuator
— · — · — membrane thickness t = 1.2 mm
———— membrane thickness t = 0.8 mm

When using these actuators to control rotor vibrations, two actuators with radial working directions can be positioned perpendicular to each other if necessary [7].

Magnetic Actuators

Magnetic Actuators can be classified into two types. One type are the magnetic bearings that permit a non-contact force transfer. The other type are the electromagnetic actuator systems where the regulating actions - as with the other presented actuator concepts - are applied indirectly, i.e. usually via conventional bearings. In both types the magnetic forces can be controlled by either current or voltage. Here we use control by current (the power amplifiers are working as a current source). The relationship between the realizable force $f(t)$, the control current $i(t)$ and the regulating distance $s(t)$ is strongly non-linear. Under investigation are two types of magnetic actuators. The difference between the two types is basically the realization of the pre-magnetizing: electrically (type I) or by permanent magnets (type II), respectively. For type I, where the pre-magnetization is done electrically, the resulting force can be expressed by

$$f(t) = \frac{s_0^4}{(s_0^2 - s^2)^2} \cdot \left[k_i\, i + (k_s - k_M)\, s + \frac{k_s}{k_w i_0^2 s_0} (k_w i_0 i s^2 + s_0 s i^2) \right] \qquad (3)$$

where s_0 is the air gap in neutral position, i_0 is the constant pre-magnetizing current and k_i, k_s, k_w and k_M are constants that mainly depend on the geometry of the actuator design, the used magnetic material and the employed electrical components. For type II, where the pre-magnetization is done by permanent magnets, the relation between force, control current and regulating distance can be expressed by

$$f(t) = \frac{(R_{pm} + k_f s_0)^4}{\left[(R_{pm} + k_f s_0)^2 - (k_f s)^2\right]^2} \cdot \left\{ k_i\, i + (k_s - k_M)\, s + \frac{k_s}{(R_{pm} + k_f s_0)\Theta_{pm}^2} \cdot \left[\Theta_{pm} k_f w\, i\, s^2 + (R_{pm} + k_f s_0) w^2 s\, i^2\right] \right\} \qquad (4)$$

where R_{pm} is the magnetic resistance of the integrated permanent magnets, Θ_{pm} is the magnetic potential of the permanent magnets and k_f is a constant that depends on the geometry of the actuator design. In comparison to the actuator of type I, where the pre-magnetizing is realized electrically, the magnetic potential $\Theta_{pm} \triangleq i_0 w$, with w the number of turns in the pre-magnetizing coil. With the assumption of small control currents $i(t) << i_0$ and small regulating distances $s(t) << s_0$ the following linear correlation between the control current and the resulting force can be obtained:

$$f(t) = k_i i(t) + (k_s - k_M)\, s(t)\,. \qquad (5)$$

The constants k_i (force-current coefficient), k_s (force-displacement coefficient) and $k_M = 2c_L$ (stiffness factor of the assembly in working direction, c_L stiffness of one membrane) permit large varieties in layout and design of the actuators. Equation 5 also shows the destabilizing effect on such a magnetic actuator system by a negative stiffness indicated by the $+k_s$-factor. The most severe case of this destabilizing effect occurs in non-contact magnetic bearings ($k_M = 0!$).

Equation 5 does not yet reveal the difference between type I and type II. The difference becomes only obvious when we look at the expressions of the factors k_s and k_i:

Type I (electrical)

$$k_s = \frac{2(1-\sigma)^2 W_s^2 k_w^2 i_0^2}{k_f s_0^3} \qquad (6)$$

$$k_i = k_s \cdot \frac{s_0}{k_w i_0}\,; \quad k_w = \frac{W_0}{W_s} \qquad (7)$$

Type II (permanent magnets)

$$k_s = \frac{2(1-\sigma)^2 k_f^2 \Theta_{pm}^2}{(R_{pm} + k_f s_0)^3} \qquad (8)$$

$$k_i = k_s \frac{(R_{pm} + k_f s_0)\, W}{k_f \Theta_{pm}}; \quad W = W_0 + W_s \qquad (9)$$

The newly introduced constants are: σ loss factor taking into account stray flux, W_0 turns of bias coil, W_s turns of control coil.
There are several advantages in using permanent magnets for realizing the bias flux:

- there is no temperature problem
- the turns for the control coil are doubled because the winding space has doubled
- the over-all magnetic resistance is increased which coincides with a possible boost of the stored energy by the right choice of permanent magnets represented by the magnetic potential Θ_{pm}.

Figure 4 shows cross sections of the two actuators, that are currently being developed.

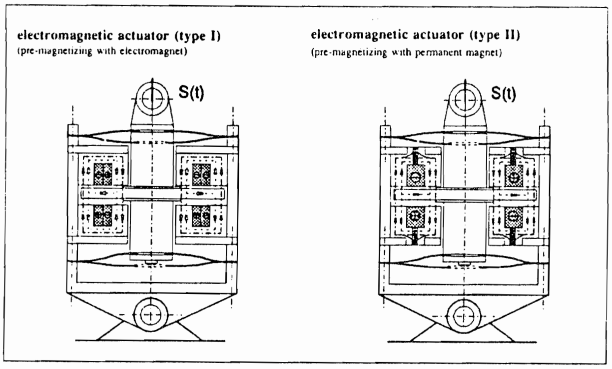

FIGURE 4: Cross sections of the two types of magnetic actuators

Tuning both systems with the objective of maximizing the control forces at a constant actuator size reveals, that using permanent magnets for the pre-magnetizing results into a control force about 2.5 times as high as in the case of indirect pre-magnetizing. The bias current i_0 or Θ_{pm}, respectively, was choosen in such a way, that if the control current $i = i_{max} = 5\,A$, the magnetic saturation flux density is 50 % below the magnetic saturation limit. Figure 5 shows the calculated maximum control forces as a function of the control current.
The results shown in Figure 5 have been verified by experiment. The first prototype of the actuator of type II is displayed in Figures 6, 7 and 8. To give an idea of the components, the actuator is first shown dismantled (Figure 6). In Figure 7 the mounted actuator system is displayed. To illustrate the arrangement of the permanent magnets, Figure 8 shows a top view of the actuator when the upper membrane is taken off.

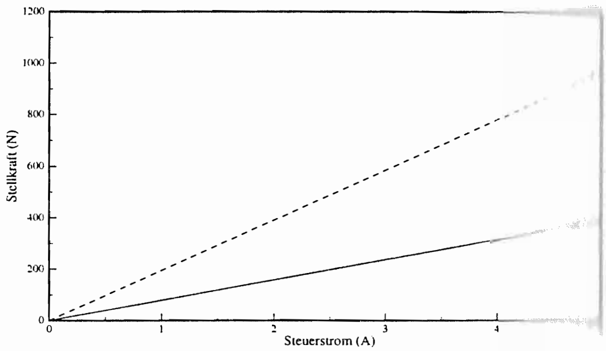

FIGURE 5: Possible control forces in neutral position (air gap $s_0 = 1.0\ mm$)
——— Type I
($\Theta_0 = i_0 W_0 = 500\ A$)
- - - - Type II
($\Theta_0 = \Theta_{pm} = 4000\ A$)

The measured frequency characteristic of this actuator is plotted in Figure 9 up to a frequency range of 300 Hz. The results were gained using power amplifiers switched to the current mode and driven by a sine-shaped input voltage. It turns out that the phase lag at 300 Hz is already about -60°. Substantial improvements can be expected by the use of new soft magnetic materials. The experimental investigations are still in progress.
Table 1 gives an overview about the presented actuator systems with regard to their realizable regulating distances and frequency ranges. It gives hints which actuator concept might fit best when facing a special application.

SUMMARY

Considerable improvements of the dynamics of machines can be expected through the application of active control techniques. It is just a matter of time that active bearings will be regularly employed in machine tools, turbine engines, engine suspensions, valve control and various other mechanisms. To account for these future requirements a practically oriented development of „custom designed“ actuators with simultaneous integration of the necessary sensors is of high priority. The aim of the paper is to contribute to this fast growing area of active vibration control and its transition into practical applications.

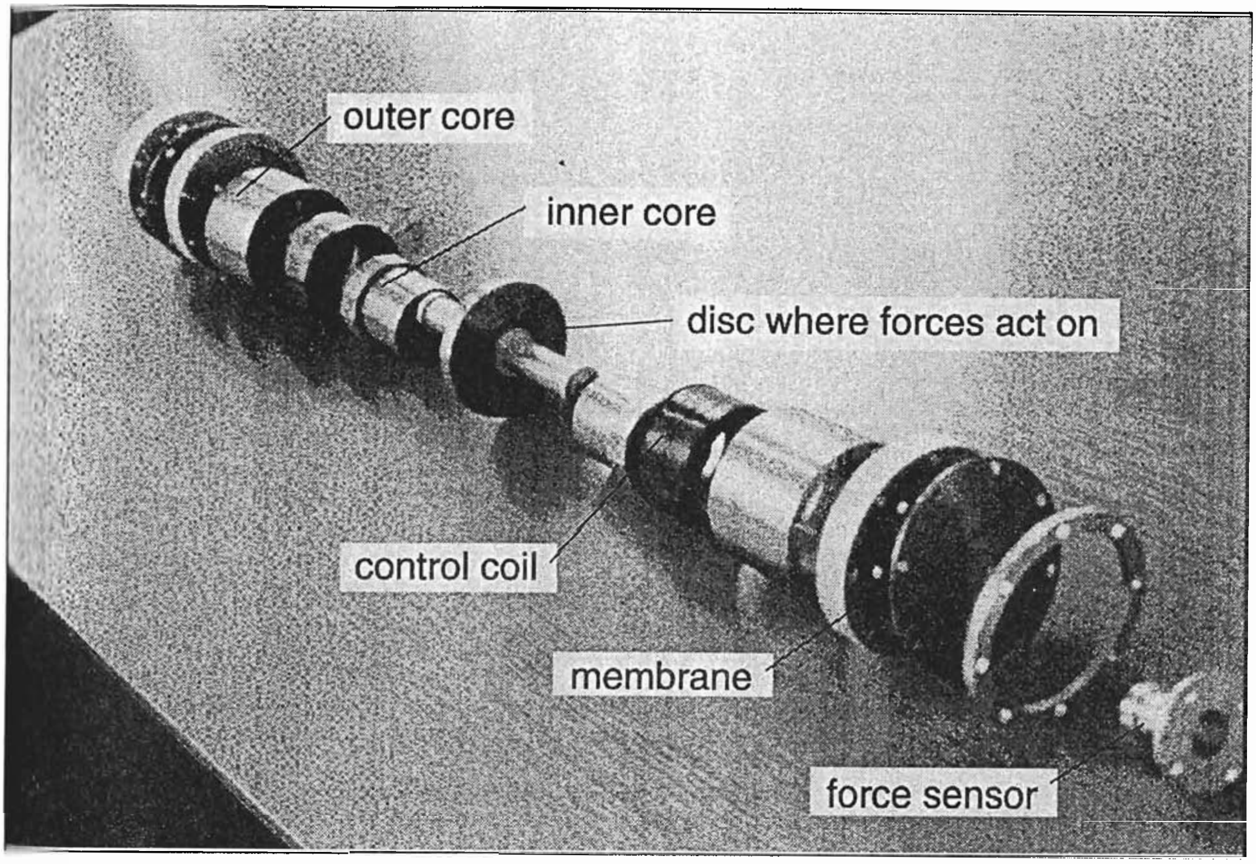

FIGURE 6: Magnetic actuator shown dismantled

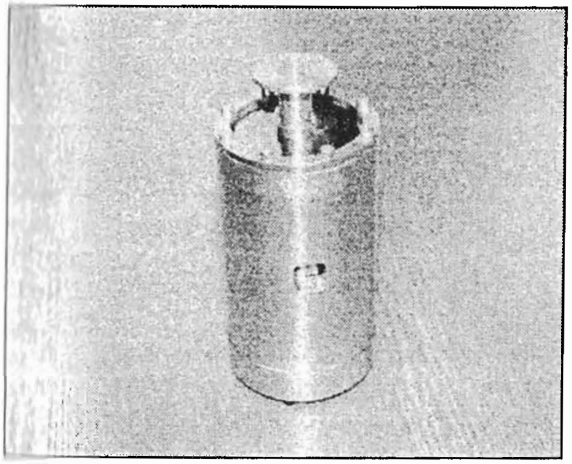

FIGURE 7: Mounted actuator device

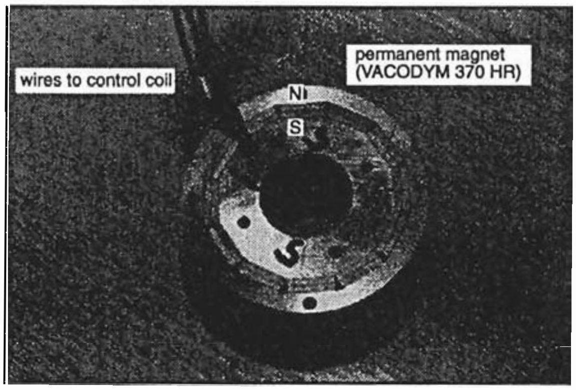

FIGURE 8: Top view of the actuator (upper membrane taken off)

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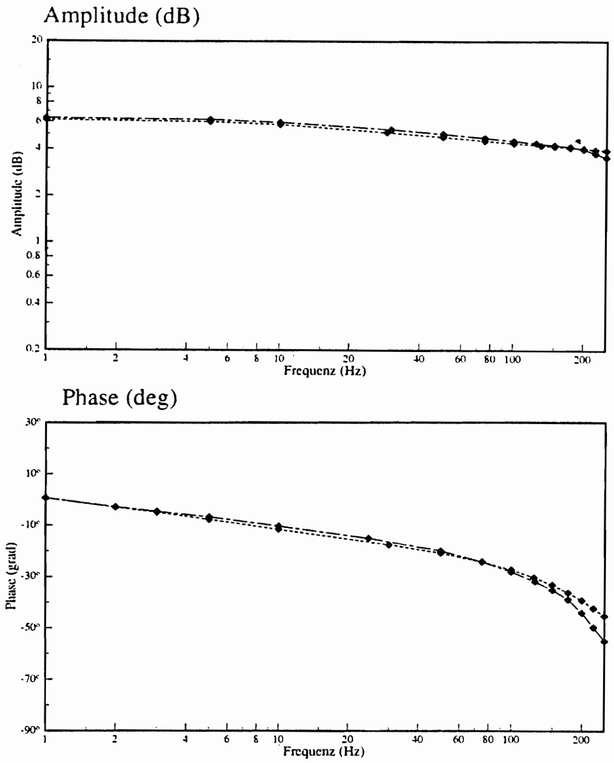
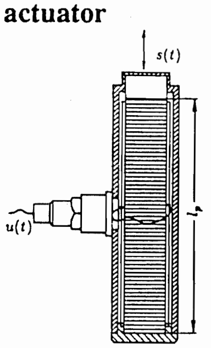
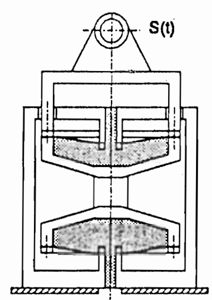
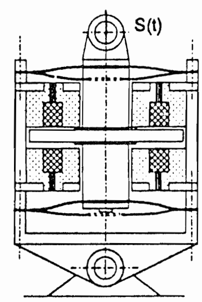


FIGURE 9: Frequency characteristics of the magnetic actuator with regard to its control force (air gap in neutral position: $x_0 = 1.0\ mm$, magnetic material only ordinary steel ST 37)
---- amplitude of *i(t)*: 1.5 A
---- amplitude of *i(t)*: 3.0 A

TABLE 1: Comparison of the different actuator concepts

piezo actuator	hydraulic actuator	magnetic actuator
		
$f_p(t) = l_p c_p k_p u(t) - k_p s(t)$	$f_H(t) = A^{*} \cdot \Delta p(t) - k_H \cdot s(t)$	$f_M(t) = k_i \cdot i(t) + (k_s - k_M) \cdot s(t)$
advantages: - simple transfer characteristic - average frequency range **disadvantages:** - large size - elektronics very expensive **regulating distances:** µm-range **regulating frequencies:** < 500 Hz	**advantages:** - compact - very large forces **disadvantages:** - transfer characteristic depends on fluiddynamics **regulating distances:** < 1.0 mm **regulating frequencies:** < 250 Hz	**advantages:** - simple transfer characteristic - large frequency range **disadvantages:** - large masses - relativly small forces **regulating distances:** < 1.0 mm **regulating frequencies:** < 1000 Hz

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