Novel piezoelectrical drive mechanism for small valves

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Abstract

Piezo actuators show several advantageous properties for use in fluid power components. However, their application is a technological challenge. This paper highlights two major aspects of a research project, whose aim is the utilisation of commercially available piezoelectric stack actuators for small valves. A novel concept for a piezoelectrical drive mechanism is presented. It allows the proper operation despite of manufacturing variations by a tolerance permissive design and a concept for adjustment. The required miniaturized design is achieved by a load-specific design using finite element analysis. Another key challenge is the temperature stability. A compensation mechanism is presented, which suppresses the thermally induced deflections within the aspired temperature range.

KEYWORDS: piezo, stack actuator, valve, tolerances, adjustment, temperature compensation

1. Introduction

In the field of valve technology, the careful stewardship of the only to a limited extent existing resources yields a minimization of space, material and energy requirements. A possible starting point is the use of alternative drive technologies for valve actuation. Within the alternative drives the piezoelectric actuators bear the most prospects for the

realization of small valves. These actuators provide a high power density, high dynamics, and good energy efficiency.

Most of the previously known implementations use a piezoelectric bender actuator as a control element. They can be used without additional gearing. However, due to their low stiffness, only small actuating forces can be achieved. Furthermore, the dynamic transfer characteristics of these actuators are limited because of their low resonance frequency. Piezoelectric stack actuators are a possible alternative to these bender elements because they can afford high actuating forces and allow highly dynamic positioning. Nowadays, they are practically not used in fluid power technology, which is mainly due to the following reasons:

- The maximum stroke is only 1-2 ‰ of the actuator length,
- manufacturing deviations and deformations due to loads or temperature changes are sometimes even larger than the available stroke,
- for the deflection of the actuators a voltage above 100 V is typically required,
- piezoelectric actuators exhibit nonlinear properties such as hysteresis and drift,
- and the actuators are comparatively expensive.

So far the only known application of piezoelectric stack actuators in mass production is for common rail diesel injectors. The car control electronics provides the necessary driving power for the actuators and compensates their nonlinearities. Here they are used in cyclic operation and the actuator size is not restricted. This is not applicable for valve technology.

The above mentioned difficulties have to be overcome in order to take advantage of these actors for fluid power components. The Institut für Fluidtechnik at the TU Dresden together with an industrial partner tackled this task and developed a piezoelectric drive mechanism based on commercially available stack actuators. A small valve was chosen as an example. Its possible applications are in the field of measurement engineering, analytics, and environmental technology.

2. Valve Concept

As seen in **Figure 1**, the designed valve is composed of four subsystems: the actuator assembly, the lever system, the valve stage and the control electronics. The modular design emphasises the universal purpose of this drive technology for fluid power components. The actuator assembly provides the mechanical mounting and the preload of the actuator. The preload is used to protect the multilayer actuator against a tensile stress, e.g. during dynamic operation. Due to the small stroke of a few

micrometers, the thermal expansion of the structure has to be compensated to prevent a thermally induced actuation of the valve. This task is also fulfilled by the actuator assembly. In order to reach the required valve lift, a force/stroke conversion is implemented as a two-stage lever system. This allows a high gear ratio within a small installation space. The second stage is also the actuating element of the valve stage. The latter is a rocker design and therefore acts as proportional 3/2 directional valve. Additionally a control electronics has been developed, which allows the compensation of the nonlinearities. The functionality of the entire system shown in Figure 1 has been demonstrated by experimental studies.



Figure 1: Overview of the concept for a small valve

The working principle of the drive mechanism is illustrated in **Figure 2**. The fundamental aspect is the non-positive connection of the subsystems. As seen in Figure 2 the system holds two springs (A, B) which produce the required actuating force and generate a downforce that ensures the contact between the components. Due to his high stiffness, the piezoelectric actuator acts as a variable end stop for the system. The springs are biased during the assembly process. This guaranties the transfer of the actuator stroke over the kinematic chain to the valve lift above the seats of the pressure port (P) and the relief port (R).



Figure 2: Working principle of the valve driven by a piezoelectric actuator

For the specific rocker design an actuation mutually serves both seats simultaneously. Thus, the opening of the seat on port P corresponds to the sealing of the seat on port R and vice versa.

3. Concept for adjustment

As mentioned in the introduction, the unavoidable manufacturing variations are sometimes larger than the available stroke of the stack Piezo actuator (nominal $12 \ \mu m @ 120 \ V$). To ensure nonetheless the proper operation, one would either have to restrict the tolerances or create appropriate compensation possibilities. The restriction of tolerances results in higher manufacturing costs and the required accuracy is not always technically feasible. Therefore, compensation possibilities have been included in the design of the system.

As a result of the non-positive connections, deviations appearing in manufacturing and assembling processes only influence the absolute rotation angle of the lever and the rocker. On the other hand certain rotation angles have to be applied to the rocker for the sealing and opening of the seats at the ports R and P. The purpose of the adjustment is to ensure the sealing and opening of both seats despite the mentioned variations.

3.1. Design

As described in /2/, an analysis of the transmission behavior of a technical system is helpful for the identification of suitable adjustments. Here a sensitivity analysis reveals parameters with a significant impact on the system output. The partial derivatives point out the high sensitivities to all variations in the effective direction of the actuator. Therefore these parameters should be avoided for adjustment.



Figure 3: a) Description of the rocker mechanics; b) valve lifts hs_P and hs_R as a function of the rockers rotation angle φ

Since the opening and sealing of the seats at ports R and P take place in the valve stage its transfer function needs to be analyzed. In **Figure 3 a)** the rocker mechanics is illustrated. The valve lifts hs_P and hs_R over the seats of the ports P and R can be

calculated according to equations (1) and (2) and are shown in Figure 3 b) as a function of the rocker's rotation angle φ .

$$hs_{P}=s_{y}-s_{x}\cdot tan(\varphi+\delta) \quad \text{with} \quad \vec{s}=(_{l}\vec{x}_{W}+_{l}R_{W}\cdot_{W}\vec{x}_{P3})-_{l}\vec{x}_{P}$$

$$hs_{R}=t_{y}+t_{x}\cdot tan(-\varphi+\delta) \quad \text{with} \quad \vec{t}=(_{l}\vec{x}_{W}+_{l}R_{W}\cdot_{W}\vec{x}_{P3})-_{l}\vec{x}_{P}$$

$$(1)$$

There are four boundary conditions to ensure proper operation of the valve, namely one for the opening and one for the sealing of each seat:

$$hs_P(\varphi_0) = hs_R(\varphi_1) = 80 \ \mu m$$
 (opening value lift) and (3)

$$hs_P(\varphi_1) = hs_R(\varphi_0) = -30 \ \mu m$$
 (sealing valve lift). (4)

Sealing. Like illustrated in Figure 3 b) a certain rotation angle $\varphi_{0,req.}$ has to be applied for the sealing of the seat at R and the angle $\varphi_{1,req.}$ for the sealing of the seat at P. These required rotational angles arise from the geometry parameters of the valve stage and are thus subject to variation. However, the actual rotation angle φ of the rocker is determined by the lever system, the actuator position, and the actuator stroke.

The transfer function between the actuator stroke and the resulting rotation angle depends on the actual geometry:

$$\varphi = f(\Delta x, geometry). \tag{5}$$

To seal the two seats of the valve stage the rotational angle has to be $\varphi = \varphi_{0,req.}$ for an actuator stroke of $\Delta x = 0$ and $\varphi = \varphi_{1,req.}$ for $\Delta x = \Delta x_{max}$, respectively,

$$\varphi_0 = f(\Delta x = 0, geometry) \stackrel{!}{=} \varphi_{0, req.}$$
(6)

$$\varphi_1 = f(\Delta x = \Delta x_{max}, geometry) \stackrel{!}{=} \varphi_{1, req.}$$
 (7)

Since the geometric dimensions of the lever system and the actuator assembly are parameters entering the transfer function, their variations influence the system behavior. However, this can be used to move operating points and thereby achieve the target values.

Two free variables are necessary to solve the equation system (6)-(7). Since all geometry parameters are present in both equations, an adjustment by altering two of them is not free from interactions. To decouple these two equations, only one geometry parameter is used as a variable. Therefore, the vertical position of the valve stage

within the chassis serves as adjustment element. This location is easily accessible during assembly and has a comparably low sensitivity. Hence, equation (6) can be satisfied, which corresponds to the sealing at port R. In order to fulfill equation (7) (sealing at P), the adaptation of the maximum actuator stroke Δx_{max} is used as second variable. The corresponding maximum electric drive value can be adjusted in the drive electronics, since the needed actuator stroke is smaller than the nominal value.

Opening. The valve lifts over the two seats are obtained for each rotation angle of the rocker according to the equations (1) and (2) from the geometry of the valve stage. While the sealing on both ports is ensured by adjusting the rocker's rotation angle φ , the adaption to the desired opening valve lift would need two additional variables. Because of the symmetry of the structure the vertical position of the rocker axis effects both equations. In order to reduce the adjustment effort, it is tempting to use solely this geometry parameter as variable. Although the system is then mathematically overdetermined, there is a Pareto-optimal solution for this problem. In the latter, both conditions are not fulfilled exactly, but are fulfilled at the same time as best as possible. As long as the resulting deviations are small, one adjustment variable is enough.

The concept for adjustment is derived from the transfer function of the system with the help of a sensitivity analysis. It considers the four boundary conditions resulting from the technical requirements by three variables: the vertical position of the valve stage y_C , the maximum actuator stroke Δx_{max} , and the vertical position of the rocker axis y_W .

3.2. Analysis

The proof of the concept for adjustment based upon a single prototype is not possible. A plurality of sample parts would be necessary to reproduce the manufacturing variation sufficiently. Alternatively, a virtual verification can be realized by utilizing probabilistic simulation and numerical optimization as illustrated in **Figure 4**.

This method requires a simulation model which represents the transfer function. Therefore a MBS model was set up and validated by analytical calculations. The simulation offers the advantage that, in addition to kinematic observations, statements on the kinetics of the system are available. The latter are required for the structural design.

Beside a general proof of the concept the adjustment sensitivity has to be estimated. Finally detailed information about the required adjustment range is necessary for the engineering of the valve.



Figure 4: Coupling of optimization and probabilistic simulation for analysing the adjustment concept

Proof of concept. The adjustment is treated as an optimization problem. Therefore, the objective criteria are the sealing and opening valve lifts. The adjustment elements (vertical position of the valve stage y_c , maximum actuator stroke Δx_{max} and the vertical position of the rocker shaft y_W) are considered as optimization variables. A Hooke-Jeeves algorithm serves for the optimization, since it is robust and efficient. The "virtual adjustment" determines an appropriate value for each adjustment element, which fits the sealing and opening valve lifts best. In Figure 4 a) the procedure is shown schematically, and **Figure 5** shows an exemplary result of such an optimization run. For the real valve, this corresponds to setting the respective adjustment screws for one particular set of geometry parameters. As seen in Figure 5, the chosen three adjustments enable the adoption to all four boundary conditions. Therefore the proper function of the valve is guaranteed.

Figure 5: Proof of the adjustment concept by numerical optimization

Adjustment sensitivity. The adjustment sensitivity measures how improperly set adjustments affect the valve function. Using the probabilistic simulation illustrated in Figure 4 b), it is possible to determine the variations of the systems output variables as a function of the variations of the input variables. Hence, the adjustment sensitivity can be evaluated by considering the adjustments as varying input variables of the valve, while all other geometry parameters remain constant. The resulting variation of the output variables (valve lifts above the seats at ports R and P) is determined by repeating the simulation for the input values randomized within an estimated variation range. The frequencies of the results are counted and distribution parameters (e.g. mean, variance for normal distribution) are fitted to the data. As seen in **Table 1** the precision achieved with conventional technical means (e.g. micrometer screw) does not significantly limit the valve function.

	1 "yw"	2 "yc"	3 ,,Δx _{max} "
	= 0,845mm ± 1µm	= 0,256mm ± 2µm	= 6,8 μm ± 0,1 μm
hs _P (φ ₁)	-30 µm	-29,98 ± 5,7 μm	-29,99 ± 3,8 μm
hs _P (φ₀)	80 ± 2,3 μm	80,1 ± 5,4 μm	80,1 µm
hs _R (φ ₁)	80 ± 2,3 μm	80,1 ± 5,6 μm	80,1 ±3,8 μm
hs _R (φ₀)	-30 µm	-30 ± 5,5 μm	-30 µm

Table 1: Calculated sensitivity of the valve lifts with respect to the adjustment values

Adjustment ranges. The calculation of the necessary adjustment ranges requires a combination of the two above mentioned approaches, as demonstrated in Figure 4 c). In that case all the geometry parameters vary according to DIN ISO 2768-f and the above presented optimization run is performed for each sample out of all the parameters.

Figure 6: Determined operating range for adjustments and variation of valve lifts

As a result, the values for the adjustments and the corresponding valve lifts are determined as output values for each individual experiment. The frequencies are obtained from all experiments and plotted in a histogram. This indicates the necessary adjustment range for the respective elements.

The results of the analysis are shown in **Figure 6**. With reference to the variations of the valve lifts, the functional capability of the drive mechanism is guaranteed for the selected tolerance range.

4. Load-specific design

In addition to manufacturing variations the deformations due to static and dynamic loads have to be considered. The small stroke of the actuator must not be lost due to the deflection of the lever or the rocker as a result of the impressed forces. The maximum deflection s_{max} of a beam is calculated from equation (8).

$$s_{max} = F \cdot L^3 /_{3 \cdot E \cdot I}$$
(8)

Accordingly, for a small deflection the length L of the lever arm has to be as short as possible. On the other hand the necessary gear ratio requires a long lever arm. To resolve this conflict, the force F on the lever was kept low while maximizing the geometrical moment of inertia *I*. If the point of load incidence from the spring (A) is as close as possible to the point of load incidence from the coupling point between the lever and the rocker (cf. Figure 2) the resulting moments partially compensate each other. Then only the difference between both loads leads to a deformation. To enlarge the geometrical moment of inertia *I* a mass distribution at large distances from the centroid is chosen. This is realized by a u-shaped design of the lever. In this design the lever can be placed around the actor, and so requires only a little installation space. **Figure 7** illustrates the design found and shows the calculated static deflection s_{max} under load.

By the non-positive connections, an additional advantage arises in regard to the permissible deflection s_{max} . As already mentioned, the piezoelectric actuator acts as an end stop for the system and the springs are biased during the assembly process. Due to the slight stiffness of the springs, the impressed forces change little over the operation range. This keeps the deflection s_{max} , resulting from the spring forces, constant and therefore has little impact on the available valve lift.

Figure 7: a) Lever design; b) Determined displacement

5. Compensation of thermally induced deflections

The aspired application conditions include at least a temperature range between 0 °C and 60 °C. In this range the thermal strains within the drive mechanism are about the size of the available actuator stroke. In order to ensure the functionality of the drive mechanism, a compensation of the thermal strains is required. To design a compensation, exact knowledge of the thermal-mechanical behavior is needed.

Heat enters the drive system from various sources (e.g. electrical energy, environment). Inhomogeneous temperature fields arise from their superposition. Simple analytical calculations of the thermal expansion based on a homogeneous temperature distribution do not achieve the required accuracy. However, a calculation of the inhomogeneous thermal field is possible by means of finite element analysis (FEA). By applying the resulting temperature distribution as a load in a structural-mechanical FEA, the thermal strains can be calculated. An analogous issue is described in /4/ for industrial robots.

The temperature distribution depends on the ratio of the heat storage to the heat dissipation capacity of the components. The relevant parameters are the specific heat capacity, thermal conductivity and specific heat transfer coefficient for convection and radiation. The parameterization was done using tabular data and similarities discussed in the literature /1,3/. The stationary and transient behavior of the model was verified by punctual measurements (P1...P4) and by comparison of the resulting temperature distribution with thermal images. As shown in **Figure 8** the simulation results are in good agreement with the measurements.

Figure 8: Validation of the finite element model by comparison to thermal images Valve actuation occurs when the relative distance between the driving side of the actuator assembly and the bearing point of the lever (distance ab in **Figure 9**) changes. In proper operation this distance is set by the actuator stroke. Any change due to thermal strain is superimposed to the actuator stroke, and undesired valve actuation occurs. The aim of the compensation is to keep the distance ab constant within the given temperature rage.

Figure 9: Structure of the drive mechanism (I-actuator, II-preload spring, III-chassis, IV-compensation element)

There are two main heat sources causing thermal strains: self heating through electrical losses within the piezoelectric actuator and ambient temperatures. This results in an inhomogeneous temperature field. By introducing an insulating layer a thermal separation between the chassis III and actuator assembly I+II was created, which leads to an homogenization of the temperature field. Thus it is possible to compensate the thermal strain of the chassis and the actuator independently.

Since the actuator assembly is mounted oppositely to the lever bearing the thermal strain of the chassis would increase the distance \overline{ac} . This is avoided by a compensation element IV, which moves the actuator assembly in the opposite direction and keeps the distance \overline{ac} constant. To realize the same elongation within a smaller installation space the compensation element has a much higher coefficient of thermal expansion than the chassis.

The used piezoelectric actuator 1 itself has a negative coefficient of thermal expansion. This is compensated by the positive extension of the preload spring frame II. So the length of actuator assembly (distance \overline{bc}) remains constant within the temperature range.

The above mentioned thermal separation requires a reduction of the heat transfer rate between the actuator assembly and a chassis. As shown in **Figure 11**, there are two important parameters: the heat transfer coefficient α_L and the contact area *A*. The heat transfer coefficient α_L is mainly determined by the material as well as the selected manufacturing and joining processes. For this purpose, the largest possible surface roughness R_Z combined with a high hardness *HV* and low contact pressure *p* is needed. For the intermediate medium air the thermal conductivity α_L is already very low. To minimize the contact area *A*, a snap-action connection with a gap between the spring frame II and the compensation element IV is used. The design is shown in Figure 11.

Since the measurement of the thermal strain within the given temperature range for a complete assembly demands a high measuring precision the validation is still pending.

6. Conclusion

A novel piezoelectric drive mechanism for small valves was developed. The fundamental aspect is the non-positive connection of the components. This requires an adjustment. An appropriate concept was derived from the transfer function of the system with the help of a sensitivity analysis. It considers the four boundary conditions resulting from the technical requirements by three variable elements. An approach for the virtual analysis of the adjustment concept was presented. Beside a general proof of the concept the adjustment sensitivity and the required adjustment range were estimated. The deformations due to static and dynamic loads have been considered by a load-specific design using finite element analysis. The thermal strains resulting of self heating and ambient temperatures are compensated passively. This has to be validated as the next step. The valve concept is scalable and the drive mechanism can be used for other applications, e.g. pumps. The research shows that a thorough design overcomes challenges associated to miniaturization. Thereby even high precision systems are fabricable with conventional production technologies.

7. Acknowledgement

The presented research was mainly part of the project KF2452601DF9 – "Entwicklung einer piezoelektrischen Antriebstechnik für fluidtechnische Komponenten" which was funded by the Federal Ministry of Economics and Technology within the ZIM programme. The scientific processing was carried out in cooperation with the company Bürkert Fluid Control Systems.

Supported by:

on the basis of a decision by the German Bundestag

The authors would like to thank all parties involved for their helpful assistance.

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9. Nomenclature

A	area	m²	λ	thermal conductivity	W/mK		
α _L	heat transfer coefficient	W/m²K	p	pressure	N/m ²		
E	Young's modulus	kN/mm ²	φ	rotation angle	degree		
f	frequency	Hz	Q	Heat transfer rate	W		
F	force	Ν	Rz	surface roughness	μm		
hs	valve lift	μm	S	displacement	mm		
HV	Vickers hardness	HV	Т	temperature	°C		
Ι	geometric moment of inertia	m⁴	X	position vector	m		
L	length	m	Δx	actuator stroke	μm		
Subscribts							
req.	required		Ρ	Pressure port			
max	maximum		R	Relief port			