

# **Novel piezoelectrical drive mechanism for small valves**

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## **Abstract**

Piezo actuators show several advantageous properties which make them interesting as drives for fluid power components. However, using these actuators poses a technological challenge. In order to use commercially available piezoelectric stack actuators as drives in small valves, a new piezoelectrical drive mechanism was developed. This paper presents a concept to ensure proper valve operation, independent of manufacturing variations and temperature influences. The tolerance to manufacturing variations is realised by an adjustable design. For the temperature stability a compensation mechanism is shown, which helps suppress the thermally induced deflections. The required miniaturization is achieved by a load-specific design.

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

## **1. Introduction**

In the field of valve technology a trend of development is the minimization of space, material and energy requirements. The use of alternative drive technologies for valve actuation, serves as a possible starting point for this miniaturization. Beside the electromagnets the piezoelectric actuators bear the most promise for the realization of

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

Most of the previously known implementations use a piezoelectric bender actuator as a control element. The advantage of these is that 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 for highly dynamic positioning. Today, however, they are rarely 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 often 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.

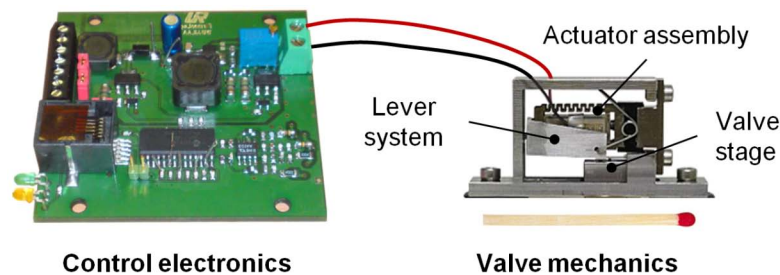
So far the only known application of piezoelectric stack actuators in mass production is for use in common rail direct fuel injections. The car's control electronics provide the necessary driving power for the actuators and compensates their nonlinearities. In this case they are used in cyclic operation and the actuator size is not restricted. This is not applicable, however, for valve technology.

The difficulties that are mentioned above have to be overcome in order to take advantage of these actuators in 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 a test case. Its possible applications are in the field of measurement engineering, analytics, and environmental technology.

## **2. Valve Concept**

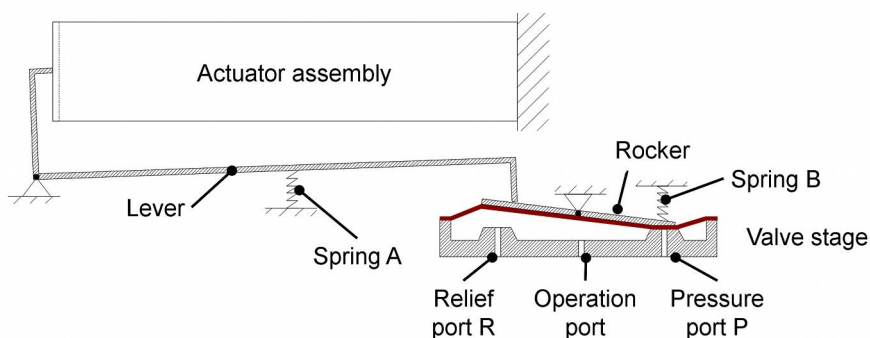
As seen in **Figure 1**, the designed valve is composed of four subsystems: the actuator assembly, the lever system, the valve stage and the control electronics. The modular design emphasises the universal purpose of this drive technology for fluid power components. The actuator assembly provides the 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 its high stiffness, the piezoelectric actuator acts as a variable end stop for the system. The springs are pre-loaded during the assembly process and guarantee 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 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.

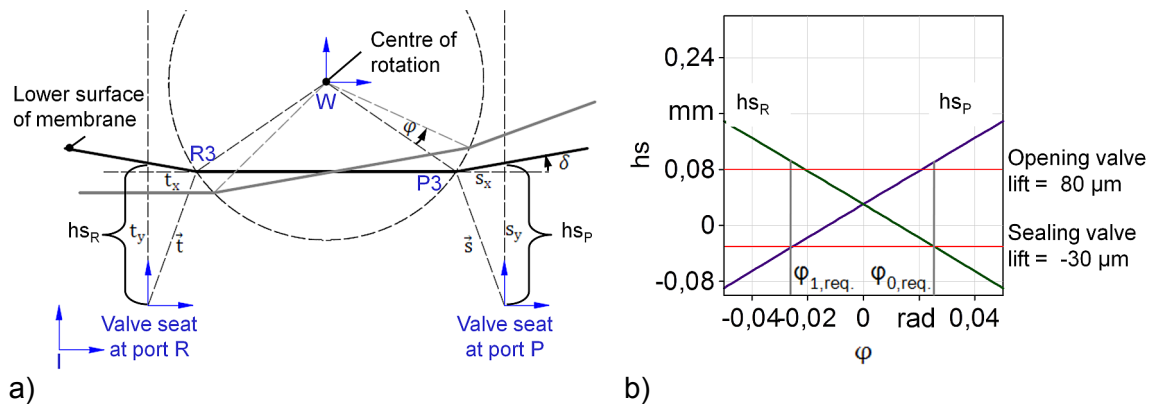
### 3. Concept for adjustment

As mentioned in the introduction, the unavoidable manufacturing variations are often larger than the available stroke of the stack Piezo actuator (nominal  $12\text{ }\mu\text{m}$  @  $120\text{ V}$ ). 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. 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  $\varphi$

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  $\varphi$ .

$$hs_P = s_y - s_x \cdot \tan(\varphi + \delta) \quad \text{with} \quad \vec{s} = ({}_I\vec{x}_W + {}_I\vec{R}_W \cdot {}_W\vec{x}_{P3}) - {}_I\vec{x}_P \quad (1)$$

$$hs_R = t_y + t_x \cdot \tan(-\varphi + \delta) \quad \text{with} \quad \vec{t} = ({}_I\vec{x}_W + {}_I\vec{R}_W \cdot {}_W\vec{x}_{R3}) - {}_I\vec{x}_R \quad (2)$$

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:

$$hs_P(\varphi_0) = hs_R(\varphi_1) = 80 \mu m \quad (\text{opening valve lift}) \quad \text{and} \quad (3)$$

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

**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  $\varphi$  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:

$$\varphi = f(\Delta x, geometry). \quad (5)$$

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,

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

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

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  $\Delta x_{max}$  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  $\varphi$ , 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.

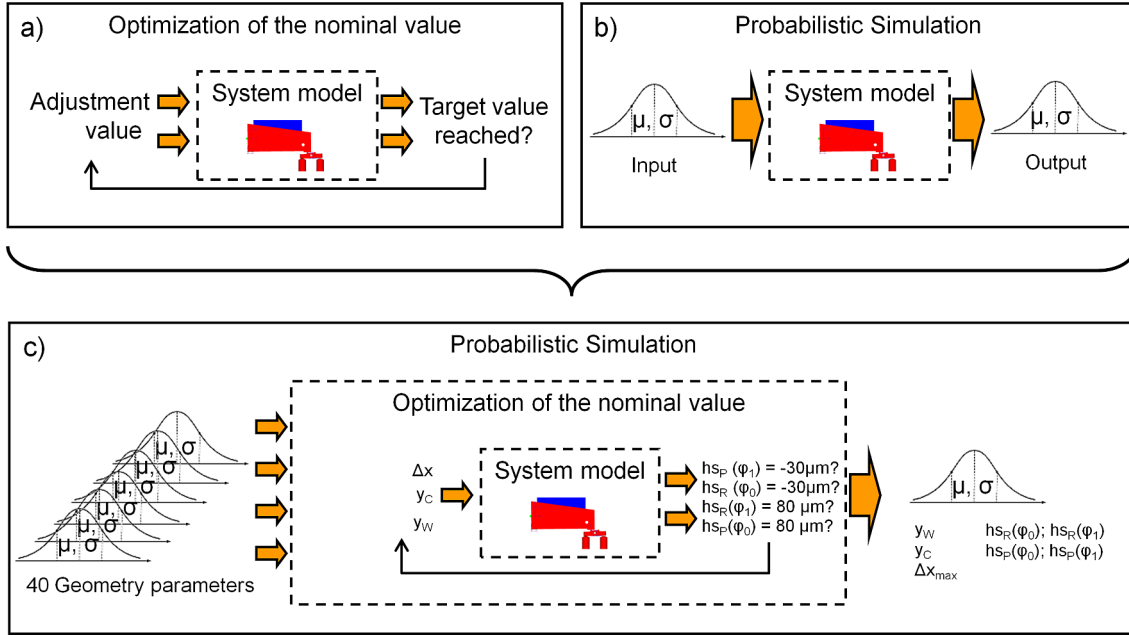
The concept for adjustment is derived from the transfer function of the system with the help of a sensitivity analysis. It considers the four boundary conditions resulting from the technical requirements by three variables: the vertical position of the valve stage  $y_C$ , the maximum actuator stroke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

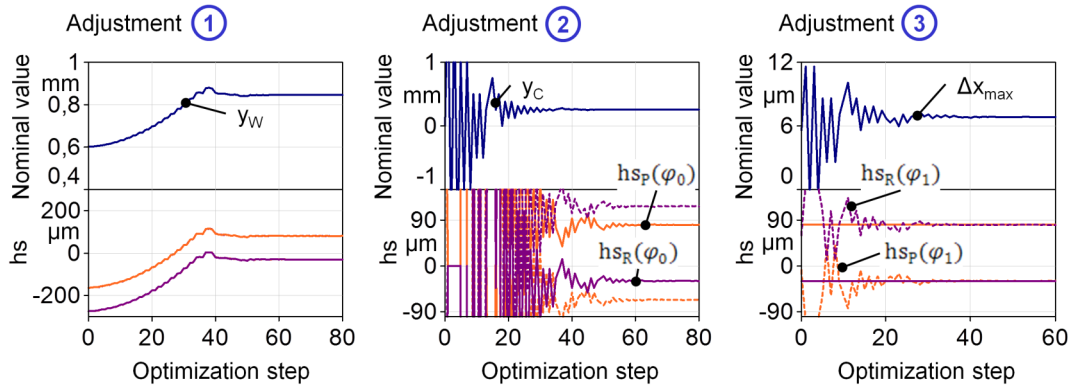
This method requires a simulation model which represents the transfer function. Therefore a multi-body simulation model was set up and validated by analytical calculations. The simulation offers the advantage that, in addition to kinematic observations, statements on the kinetics of the system are available. 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  $\Delta x_{max}$  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.



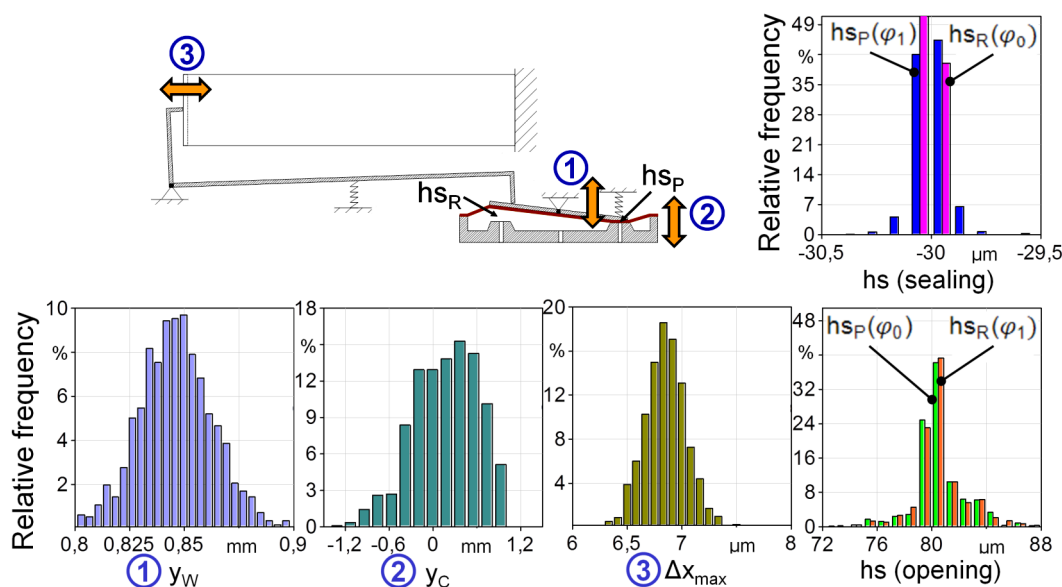
**Figure 5:** Proof of the adjustment concept by numerical optimization

**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 „ $y_W$ “ = $0,845\text{mm} \pm 1\mu\text{m}$	② „ $y_C$ “ = $0,256\text{mm} \pm 2\mu\text{m}$	③ „ $\Delta x_{\max}$ “ = $6,8\mu\text{m} \pm 0,1\mu\text{m}$
$hs_P(\varphi_1)$	$-30\mu\text{m}$	$-29,98 \pm 5,7\mu\text{m}$	$-29,99 \pm 3,8\mu\text{m}$
$hs_P(\varphi_0)$	$80 \pm 2,3\mu\text{m}$	$80,1 \pm 5,4\mu\text{m}$	$80,1\mu\text{m}$
$hs_R(\varphi_1)$	$80 \pm 2,3\mu\text{m}$	$80,1 \pm 5,6\mu\text{m}$	$80,1 \pm 3,8\mu\text{m}$
$hs_R(\varphi_0)$	$-30\mu\text{m}$	$-30 \pm 5,5\mu\text{m}$	$-30\mu\text{m}$

**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.

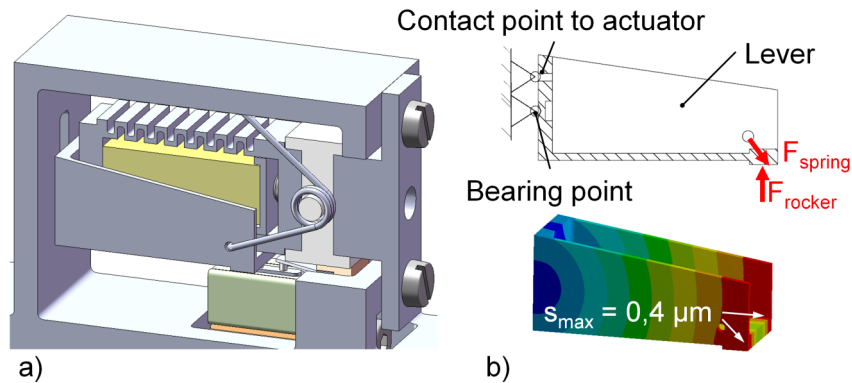
The results of the analysis are shown in **Figure 6**. With reference to the variations of the valve lifts, the functional capability of the drive mechanism is guaranteed for the selected tolerance range.

#### 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).

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

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.



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

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.

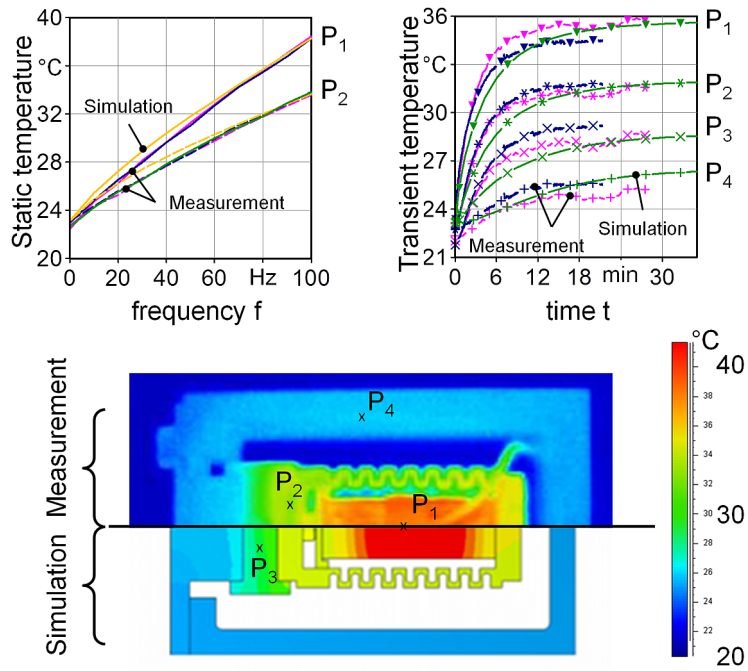
## 5. Compensation of thermally induced deflections

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.

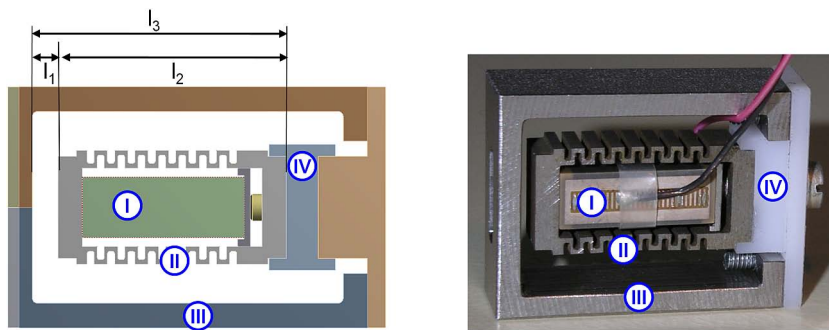
The temperature distribution depends on the ratio of the heat storage to the heat dissipation capacity of the components. The relevant parameters are the specific heat capacity, thermal conductivity and specific heat transfer coefficient for convection and radiation. The parameterization was done using 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.

Valve actuation occurs when the relative distance between the driving side of the actuator assembly and the bearing point of the lever (distance  $l_1$  in **Figure 9**) changes.



**Figure 8:** Validation of the finite element model by comparison to thermal images

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  $l_1$  constant within the given temperature range.



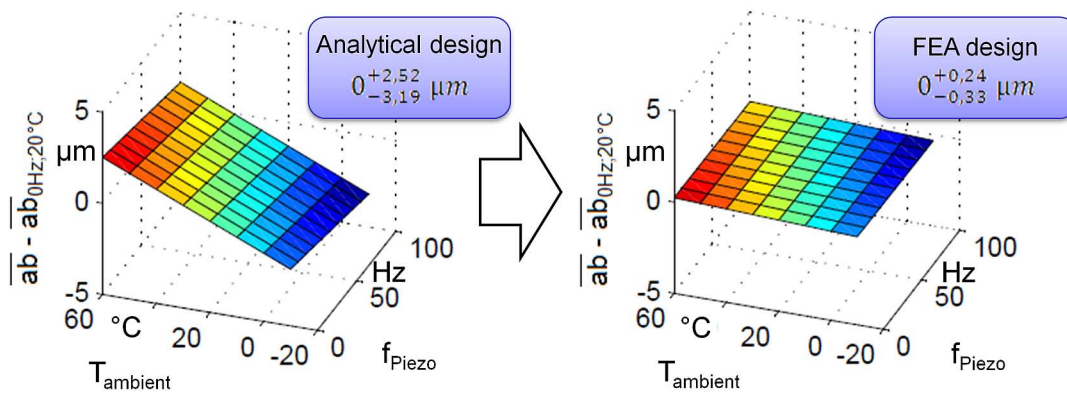
**Figure 9:** Structure of the drive mechanism (I-actuator, II-preload spring, III-chassis, IV-compensation element)

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  $l_3$ . This is avoided by a compensation element IV, which moves the actuator assembly in the opposite direction and keeps the distance  $l_3$  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.

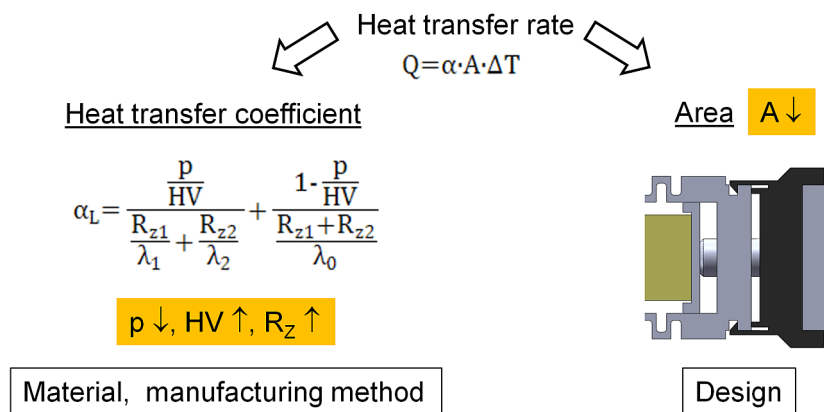
The used piezoelectric actuator 1 itself has a negative coefficient of thermal expansion. This is compensated by the positive extension of the preload spring frame II. So the length of actuator assembly (distance  $l_2$ ) remains constant within the temperature range.



**Figure 10:** Thermal induced stroke against ambient temperature and driving frequency

Altogether it is possible to minimize the change in the distance  $l_1$  to about  $\pm 0.3$  micron.

**Figure 10** shows simulation results for the thermally induced stroke and the achieved improvement by FEA design in comparison to the analytical design.



**Figure 11:** Realisation of the thermal insulation between actuator assembly / chassis

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  $\alpha_L$  and the contact area  $A$ . The heat

transfer coefficient  $\alpha_L$  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  $\alpha_L$  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.

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 new and innovative piezoelectric drive mechanism for small valves was developed. The key new aspect of the research is a design tolerant to manufacturing variations. This is achieved by the non-positive connection of the components and requires an adjustment to the four boundary conditions resulting from the required valve lifts. An appropriate concept was derived from the transfer function of the system using sensitivity analysis. An approach for the virtual analysis of the adjustment concept was presented. In addition to the proof of concept, the adjustment sensitivity and the required adjustment range were calculated. 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. Verifying this will be the subject of future research on this topic. In conclusion, the valve concept is scalable and the drive mechanism can be used for other applications, e.g. pumps. The research shows that a well-thought-out design overcomes challenges associated with miniaturization, and thereby even high precision systems are achievable using conventional and already-available production technologies.

## 7. Acknowledgement

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## 9. Nomenclature

$A$	area	$m^2$	$\lambda$	thermal conductivity	W/mK
$\alpha_L$	heat transfer coefficient	W/m <sup>2</sup> K	$p$	pressure	N/m <sup>2</sup>
$E$	Young's modulus	kN/mm <sup>2</sup>	$\varphi$	rotation angle	degree
$f$	frequency	Hz	$Q$	Heat transfer rate	W
$F$	force	N	$R_z$	surface roughness	$\mu m$
$h_s$	valve lift	$\mu m$	$s$	displacement	mm
$HV$	Vickers hardness	HV	$T$	temperature	°C
$I$	geometric moment of inertia	m <sup>4</sup>	$\vec{x}$	position vector	m
$L$	length	m	$\Delta x$	actuator stroke	$\mu m$

## Subscripts

$req.$	required	$P$	Pressure port
$max$	maximum	$R$	Relief port