Estimation of the hydraulic conditions in piezo-pilot-valves

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1 Abstract:

Usually in servo valves a low electrical power is converted into a differential pressure, which controls by a furthermore piston the effective power in the form of a hydraulic power flow. Due to the short reaction time, piezo valves are particularly suitable as pilot stage. A direct connection to the control spool has been avoided due to the small stroke of piezo actor. By a direct connection of the piezo actor and the control piston the tolerances in which the pilot valve must be produced are very small and this results in very expensive prototypes. To advance in the estimates of the yield pressure and the flow rate the possibility of simulation should be investigated on an existing prototype of a direct piezo pilot stage.

2 Introduction:

Servo valves are generally developed in 2 stages. The first stage converts an electrical signal into a pressure difference signal. This pressure difference affects a second pilot piston and in this way controls the hydraulic energy flow. The regulated servo-hydraulic plants should exhibit a linear frequency response if possible . Commercial 2-stage servo valves such as, for example, the MOOG 76-101, contain a linear frequency response up to approximately 60 Hz [1]. The maximum available power of a 2-stage servo valve should not exceed 22 KW with a system pressure of 280 bar, corresponding to a servo valve with max. 63 l/min. flow. If one would like to command larger energy quantities, either 3-stage or 4-stage servo valves [2] are used.

Different versions are used as amplifier systems (pilot control). Thus, for example, flapper nozzle amplifiers, jet pipe amplifiers or beam deflection amplifiers are operated by a torque - engine or a moving coil [3]. The main stage forms an oscillating system with the control piston as mass, the hydraulic oil as absorption and the re-conducting spring as the spring . The same is also valid for the pilot stage. The way in which the electrical control of the pilot occurs also affects the maximum possible switching time. A 2-stage servo valve with flapper-nozzles operation, such as for example the MOOG D685, contains a floating time of 35 ms.

For these reasons one tries to control the pilot stage via piezo mechanical actuators [4]. These are characterised by short response times, high setting sensitivity, production of high loads, loss-free maintenance of a position, and are practically without wear. A disadvantage is the low usability stroke of piezo mechanical actuators. Consequently, a direct control has been done without up till now. It has been attempted to reach the desired stroke either with mechanical or hydraulic transmission units [3]. The advantages of high rigidity and thereby also high resonant frequency are thereby however nullified again. In addition the temperature behaviour of the Piezo actuators must be considered. The temperature expansion can be larger than the actual utilisable stroke and must, therefore, be considered in the construction.

A possibility for direct control is via Piezo bending transducers, which represent a system in a Wheatstone bridge circuit [5].

Another possibility for direct control is the use of Piezo stack actuators, which directly affect the pilot piston. In this way all the advantages of the piezo mechanical actuator can be used. The construction must be arranged in such a way however that, despite the low stroke, the pressure yield in the pilot stage is large enough to accordingly influence the main stage.

As the costs for building prototypes is very high, the possibility of simulation is likewise to be examined in order to be able to assess at the outset whether it is worthwhile to manufacture a prototype.

3 Development of the pilot stage:

A pilot stage was developed for an existing main stage, the basic goal of which was the improvement of the dynamics without decreasing other capability characteristics. As a starting point the MOOG valve – series 72 was consulted - series 72, which is a flapper nozzle valve with mechanical feedback. Figure 1 shows that only the base with the pilot piston and the control sleeve was used. The structure of the pilot control was redesigned. A mechanical or hydraulic translation was dispensed with for the reasons specified above.

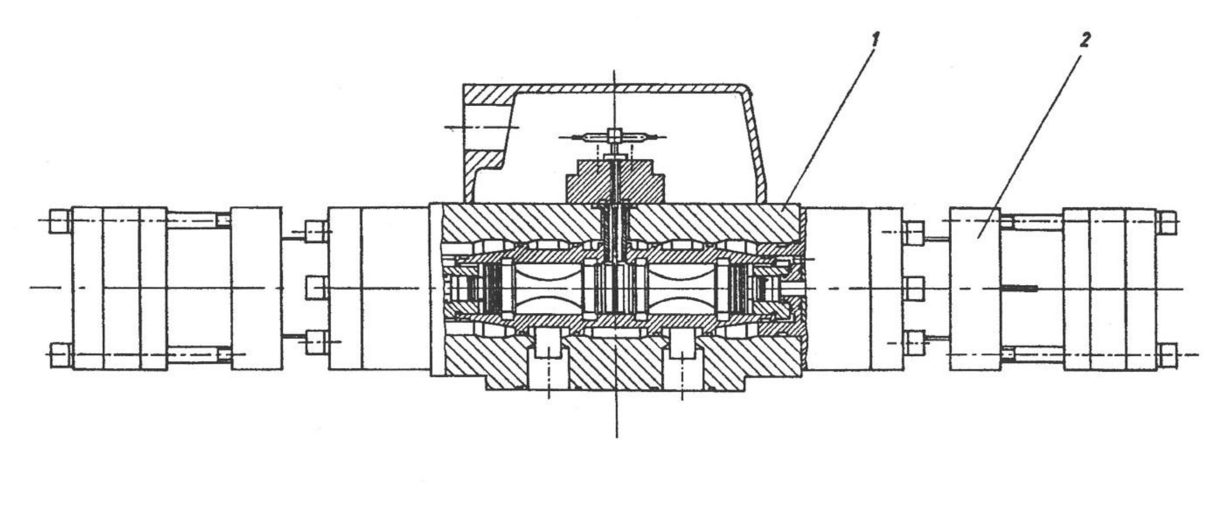

Figure 1: Main stage

From the coupling diameter Piezos from the 239.xx series of the Physik instruments company could be used. The maximum position movements of this series reach Piezo lengths from 5 bis 180 μm with resonance frequency from 12 to 2 kHz. Thus a compromise between the stroke and resonant frequency must be found.

Technical data of the Piezo actuators used:

Max. regulating distance	(with -1000V) 40 μm
Stiffness	44 N/ μm
Electrical capacity	0,7 μF
Resonant frequency	6,5 kHz
Temperature expansion	1,1 μm/K
Overall length	63,5 mm
Max. pressure force	3500 N
Max. tensile force (by internal pre-loading)	500 N

The pilot control occurs via a pilot piston, which is operated by the Piezo actuator. Several constructional designs of the pilot piston were sketched and manufactured. So that a large number of prototypes which can be expensive to manufacture could be dispensed with, the simulated hydraulic conditions were compared with the hydraulic results from tests on the manufactured prototypes. Two implemented variants were examined:

• Pilot piston with a control edge and a fixed throttle (differential piston A)

• Pilot piston with two control edges (differential piston B)

Due to the manufacturing tolerances a direct coupling through screwing the pilot piston to the Piezo actuator was dispensed with. So that the piston follows the movement of the Piezo actuator, this is pressed on the actuator by the pilot pressure.

It was attempted to combat any arising bending forces (misalignments) by an oil cushion between pistons and actuator. By diagonally arranged inlet drillings the pilot piston would complete a rotation. This rotation causes a better lubricating film with the micro movements in the dynamic operation. In Figure 2 it can be seen that the case is preloaded over the diaphragm springs and therefore counteracts a preloading which is too high. The possibility of also siphoning off the leakage oil off between Piezo actuator and diaphragm springs is also foreseen.

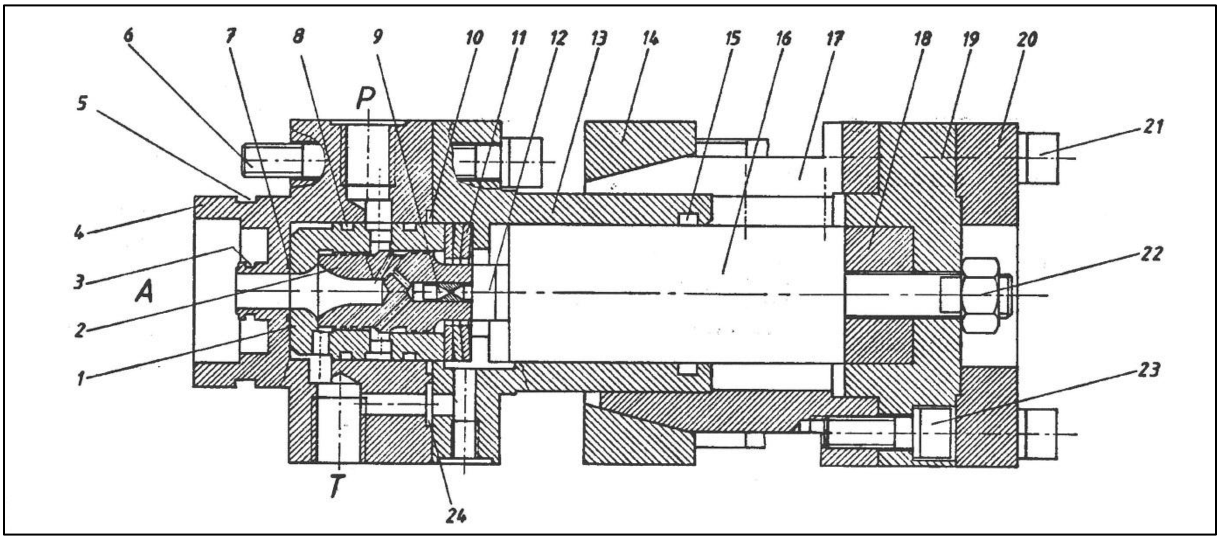

Figure 2: Pilot stage, pilot piston with two control edges

In order to consider the different thermal expansions, another compensating element in aluminium is inserted between the Piezo actuator and Piezo actuator profile. Thus the different temperature expansion coefficient between steel bodies and translator is balanced.

3.1 Pilot piston with a control edge and fixed throttle:

The differential piston is shown in Figure 3 . Over the fixed throttle in the supply pipe the control oil arrives in the pilot stage. On the face side a radial clearance is released by the movement of the Piezo actuators which causes the pressure difference for the main stage together with the fixed throttle. The force difference between loaded and unloaded differential piston is larger than with the version described in chapter 3.2, since the piston area is likewise subjected to the pressure difference. Here the possibility exists that the pilot piston of the contraction of the Piezo actuator can no longer follow without lifting.

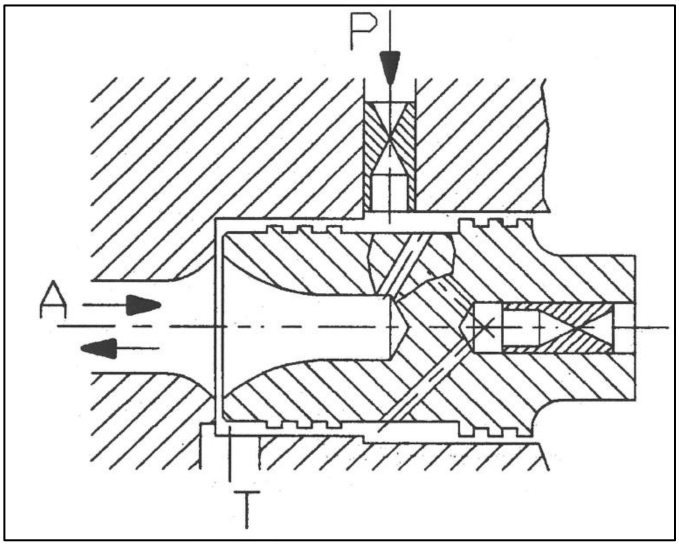

Figure 3: Pilot Piston with one control edge and a fixed throttle (version A)

3.2 Steuerkolben mit zwei Steuerkanten:

In Figure 4 it can be seen that, instead of the fixed throttle in the inlet drilling, a further control edge was added. Now the supply stream is also affected. The geometry was selected in such a way that with different diameters the cross sections are the same size. A positive overlap is given to 200 bar supply pressure on the control edge 2. Over 200 bar supply pressure outweighs the load by the impinged ring surface and impinged frontal circle. There is a deflection of the piezo, so that the control edge opens 2 and a negative overlap arises. Thus the leakage oil quantity and also the maximum attainable pressure are affected.

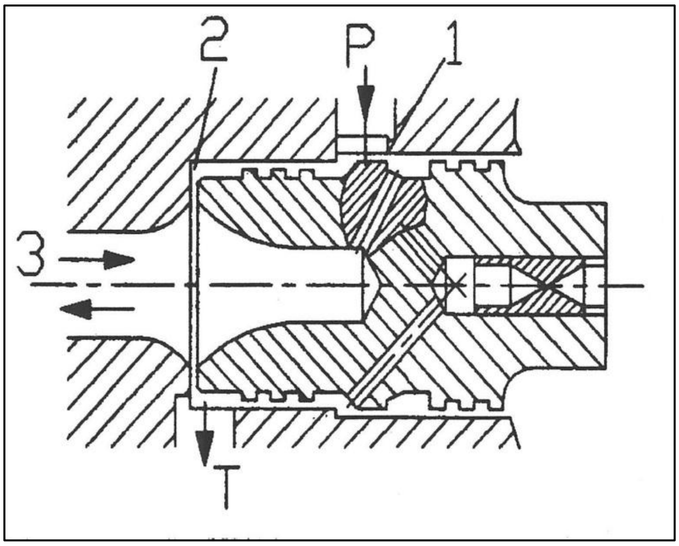

Figure 4: Pilot piston with two control edges (version B).

3.3 Modeling:

In order to be able to simulate the hydraulic characteristic data, the construction was simplified. Only those parts through which the oil flows were undertaken. In and outlets were extended, in order to create an intake and a discharge distance. Thereby eddies beyond the vortex shedding edges are avoided.

In Figure 5 one can see the refined net in the area of the fixed throttle. Likewise the areas of the control edges were also provided with a finer net.

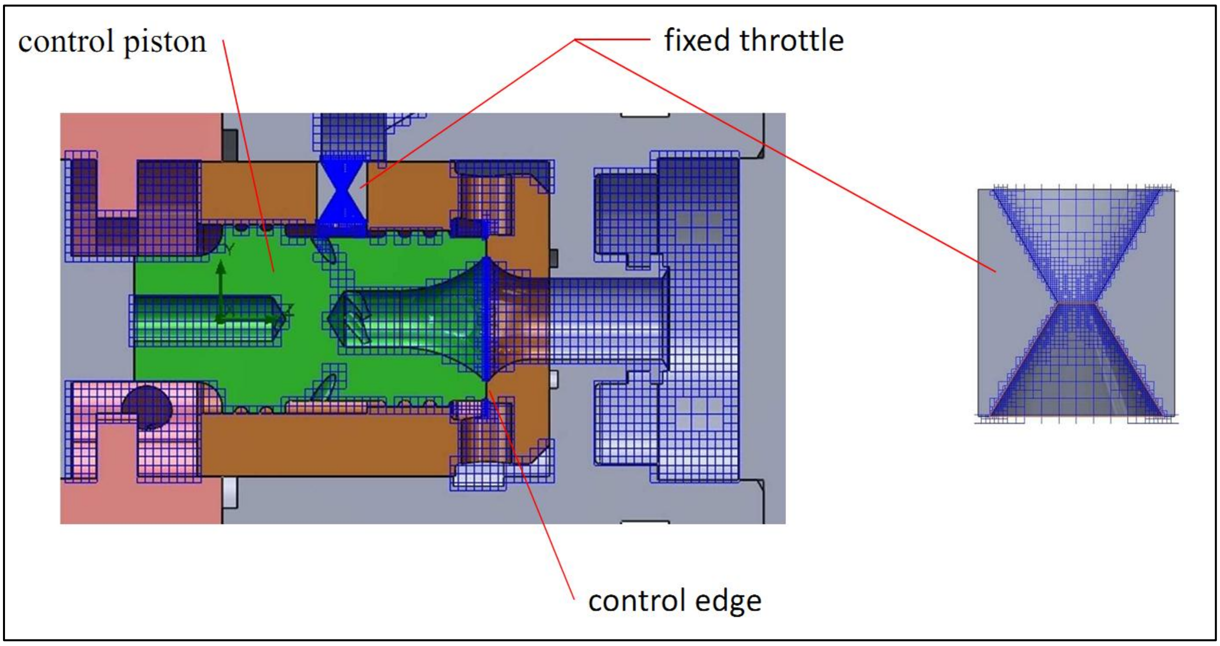

Figure 5: Net in the area oft the fixed throole

The technical data of the hydraulic fluid were, under consultation with Aral (Aral Vitam AP 46, DIN 51524 T 2), included in the calculation, so that the data of the oil in the simulation and in the hydraulic system of the test correspond.

The following was calculated:

- Flow with fully opened control edge depending on the supply pressure
- Characteristic curves with 200 bar supply pressure

The calculations as well as the pertinent measurements were statically implemented. Dynamic calculations such as, for example, the computation of a pressure increase and pressure drop curve with the switching of the Piezo actuator, were not implemented.

From the static computations an evaluation of whether a construction is suitable should be possible.

4 Results:

For the measurement of the flow rate and the characteristic curves, the pilot stages were installed on a test block (see Figure 6). On the test block a pressure sensor by Senso - control SCPT - 400 -0 -02 is assembled in relation to the pilot unit. With this sensor both the static values (pressure) as well as the step responses are measured with different Piezo settings and noted.

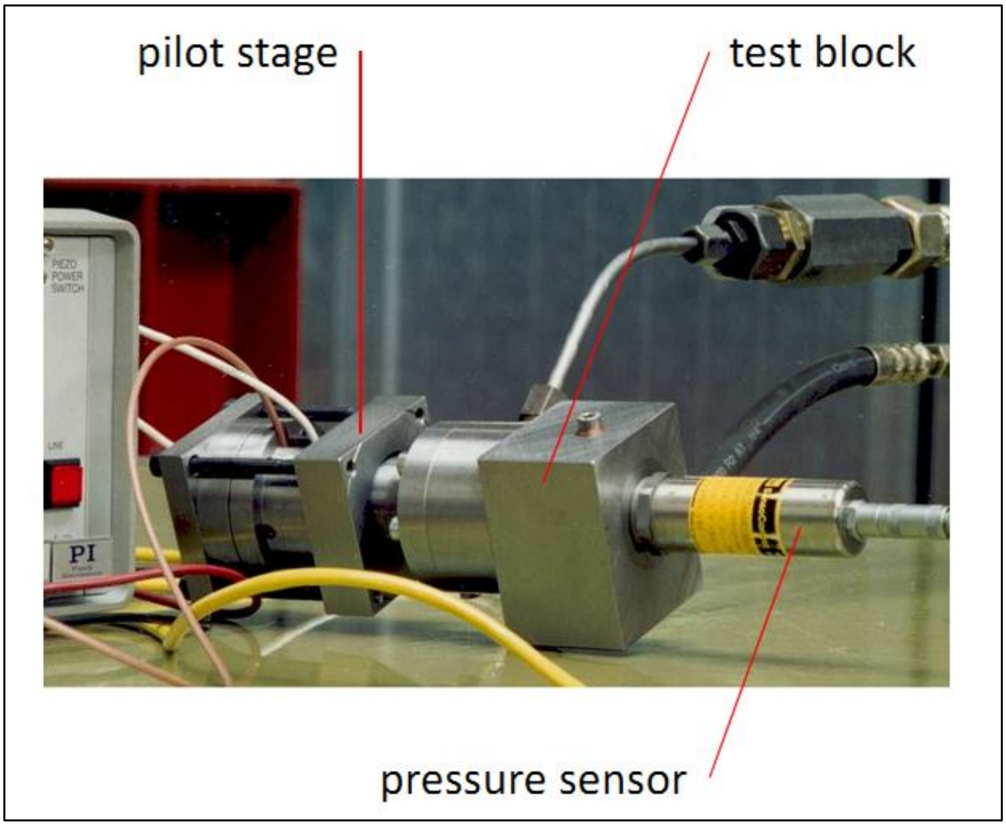

Figure 6: Pilot stage with test block

4.1 Differential piston A:

In Figure 7 the flow rate is spread depending on the supply pressure. The deviation between computed and measured flow rate can be attributed to the manufacturing tolerances and to the change of temperature with the flow over the control edge (see deviation in chapter 4.2, Figure 10).

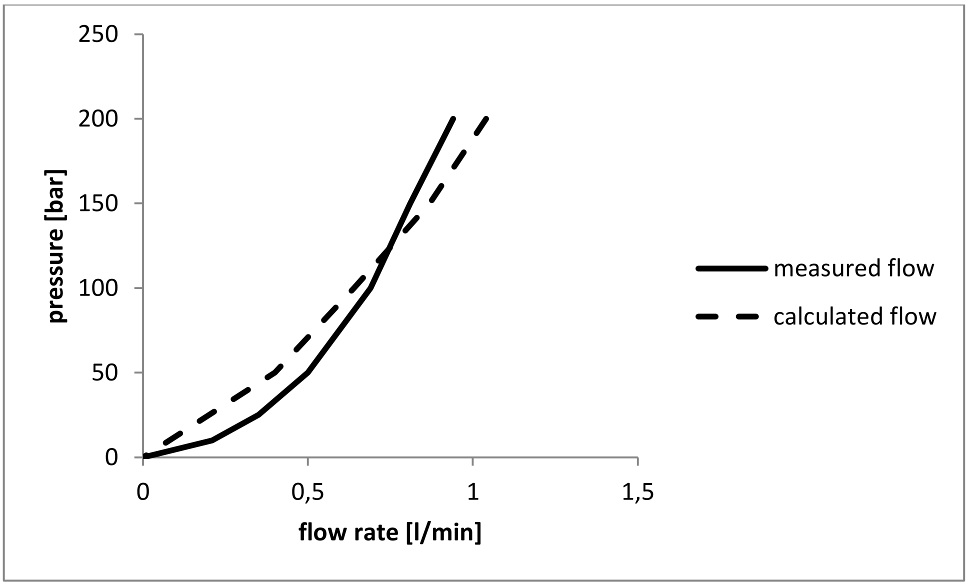

Figure 7: Flow rate

The computed characteristic curve, to be seen in Figure 8, deviates substantially from the measured characteristic curve with constant pressure and different actuator positions. The Piezo actuator is no longer able to fully close the control edge. The differential piston is pressed against the piezo actuator due to the diameter difference (see Figure 3). Thus the actuator is compressed due to the pressure build-up so far that the control edge cannot be closed any longer. The pressure in the pilot stage with fully opened control edge can be computed however and thus an estimation of the pressure yield is possible. In Figure 9 the pressure distribution in the pilot stage is shown with fully opened control edge; the decrease in the pressure at the fixed throttle can be easily seen.

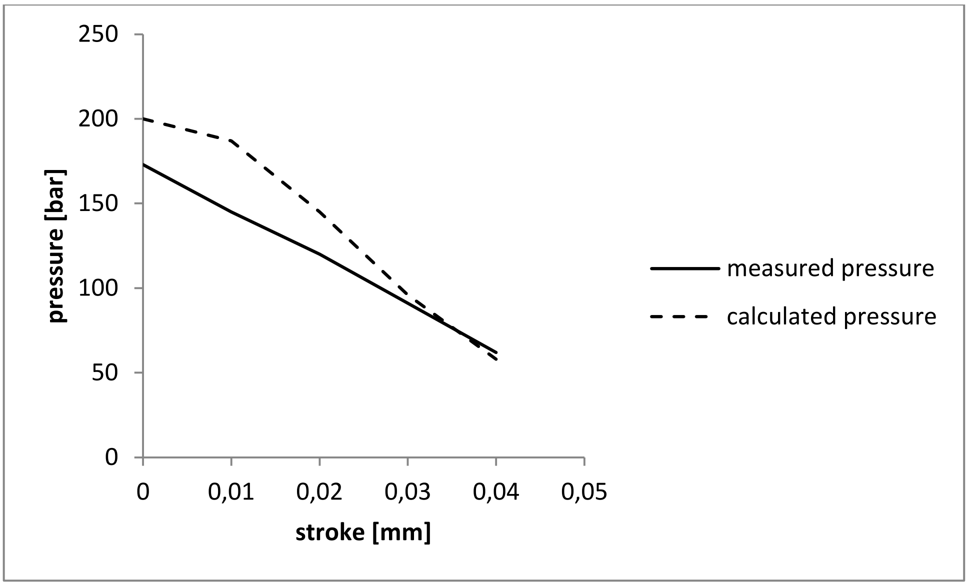

Figure 8: Pressure characteristic

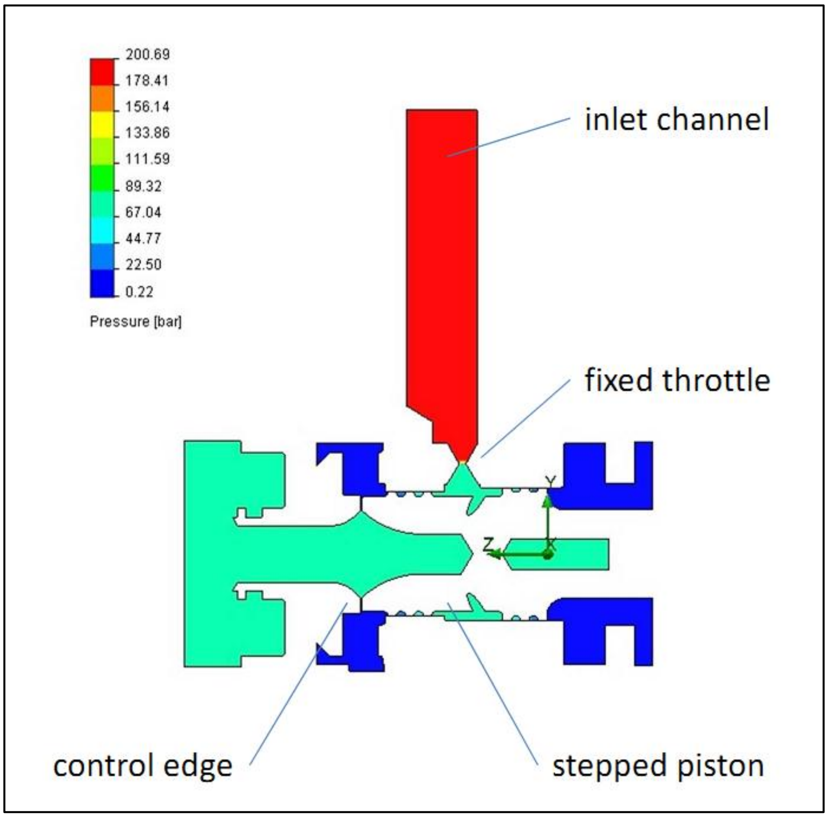

Figure 9: Pressure distribution in the pilot stage at fully open control edge (40 μm)

4.2 Differential piston B:

In Figure 10 the flow rate depending on the supply pressure is again shown. This time there were two measurements with two identically manufactured pilot pistons. The influence of the manufacturing tolerances should be determined. The tolerances alone in the manufacturing lead to substantial differences in the flow rate. If one compares the measurement with the A2 piston and the computed flow rate behaviour similar to that in chapter 4.1 Figure 7 occurs.

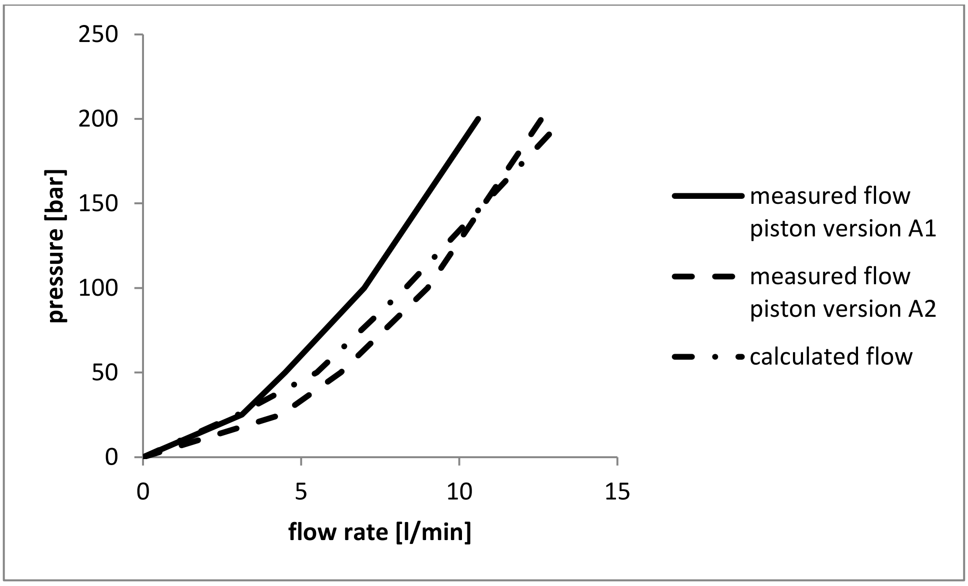

Figure 10: Flow rate

The connection between pressure and piston position is better shown during the execution with two control edges. That is above all due to the fact that the control edge 2 can be fully closed and thereby also the entire supply pressure is reached. With measurements with higher pressures the deflection of the actuator occurs again. Up to the deviation with small opening (0.01 mm), the computed pressure pattern corresponds with the measured pressure pattern.

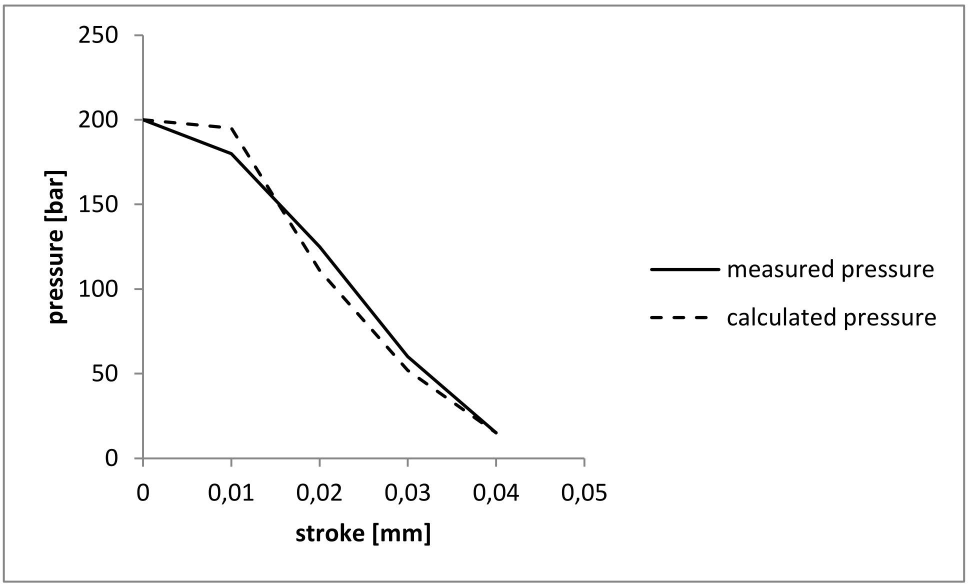

Figure 11: pressure characteristic at 200 bar supply pressure

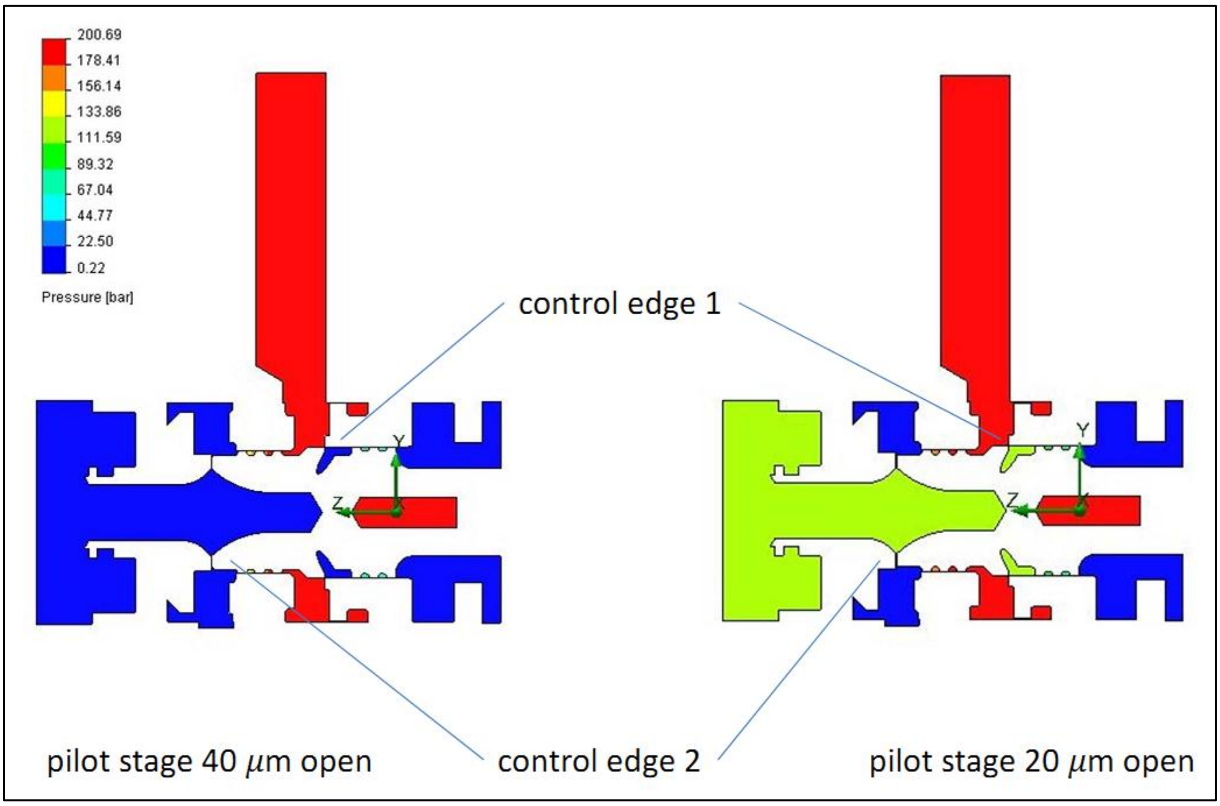

Figure 12: Pressure distribution in the pilot stage at different piston positions

5 Conclusion:

With the assigned numeric method it is possible to make statements regarding the hydraulic conditions in the pilot stage in the static load case. As it has been shown however, deformations lead to deviations due to the pressure load . Likewise the ability of the Piezo actuators cannot be considered with this method as the control edge can no longer be completely closed with high pressures. In addition, the computation reflects the ideal manufacturing condition again and again. Comparisons with several types in an implementation showed in the experiment that the manufacturing tolerances can lead to substantial differences in measurements.

6 References:

[1]

S. Helduser, Einfluss der Elastizität mechanischer Übertragungselemente auf das dynamische Verhalten hydraulischer Servoantriebe. Dissertation Aachen, 1977

[2]

Rudolf Schwall. Schenck Druckschrift PAB 101 -10. 1080 md, Carl Schenk AG

[3]

Kord Witte. Antriebe für einstufige elektrohydraulische Servoventile. Dissertation Braunschweig 1988.

[4]

Lutz Pickelmann. Einführung in die Piezomechanik. *Piezomechanik · Dr. Lutz Pickelmann GmbH, München, 2010*

[5]

Maxim Reichert. Piezoaktoren in der Ventilvorsteuerung. Hochdynamisches hydraulisches Servoventil. *O + P. Ölhydraulik und Pneumatik,*51: 22-26, 2007