A NOVEL PILOTED FAST SWITCHING MULTI POPPET VALVE

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Abstract

Switching and digital hydraulics require adequate switching valves for their practical application. A major criterion is the switching time which should in most cases be less than 5 ms. The demands on valve nominal flow rate are rather wide and range from less than 1 l/min (@ 5 bar) up to hundreds of litres. In this paper a new seat valve concept is presented. It is a piloted valve and employs a multi poppet concept. Its nominal flow rate is about 100 l/min and its switching time is about 1 ms for a pressure drop of 5 bar. With this valve energy efficient switching drives can be realized, for instance in the 10 kW to 20 kW range. Its seat type design makes the main stage a non leaking valve. Piloting is realized by a fast 3/2 solenoid actuated spool type switching valve with a nominal flow rate of 10 l/min @ 5 bar and a total switching time of about 2 ms. The paper presents the design of the poppet valve, simulation studies and experimental results concerning its static and dynamic characteristics.

Keywords: fast switching valve, seat valve, multi poppet valve

1 Introduction

In recent years numerous concepts of fast switching valves have been published. A fast switching valve, by definition, is a valve with a switching time less than 5 ms, making the valve suitable for switching technologies. Several of the available or published fast switching valves fulfill the demands on switching time but have a nominal flow rate less than 25 l/min @ 5 bar (Ploeckinger et al., 2009; Uusitalo et al., 2009 and Lumkes, 2010), which is insufficient for energy efficient applications in power ranges 10 kW and beyond. Such smaller switching valves are used as pilot valves or just for replacement of proportional or – in exceptional cases – servo valves. Energy efficient applications in ranges of 10 kW and more as well as emergency operations handled by a fast hydraulic actuation system, e.g. for a sudden machine stop, require valves with much higher flow rates of at least 50 l/min @ 5 bar (Winkler and Scheidl, 2006; Winkler, 2004 and Steiner et al., 2003).

Another crucial demand of switching control is a high fatigue-proof operating frequency. The aim is 100 Hz, and 50 Hz is an absolute minimum. Such high frequencies allow high precision and low ripples in the actuator position or speed control as well as a smaller size of switching converters (Scheidl and Riha, 1999).

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To reduce valve fatigue, adaptive frequencies related to the actual application requirements can be realized.

In this paper switching hydraulics is understood as continuous switching in PWM-mode whereas digital hydraulics employs a large number of parallel connected switching valves that are run in a pulse-coded mode to replace proportional valves in a quantized manner.

Hydraulic switching technology and digital hydraulics are also intended to reduce cost. This requires a valve design which facilitates low cost production and moderate electrical energy supply requirements to avoid expensive power electronic components, cabling, and connectors. One crucial factor is peak current which should be below 30 A to stay within certain industry standards.

The requirements on the new valve concept are summarized as follows:

- Nominal flow rate in case of energy efficient applications of about 100 l/min @ 5 bar
- Nominal flow rate in case of piloting and reduction of proportional valves of about 2 to 10 l/min
- Hydraulic switching time of 1 ms (or faster)
- Up to 100 Hz fatigue-proof
- Peak current < 30 A

- Compact design
- Low cost design

The paper presents the concept and some elementary dimensioning considerations in Section 2. Section 3 discusses the design around an existing 3/2 pilot valve and the poppet position measurement device which was an integral part of the valve design. Experimental findings and their interpretations are reported in Section 4. Section 5 draws conclusions and gives an outlook to further work on this valve concept.

2 The Basic Concept of the Multi Poppet Valve

As discussed in Winkler and Scheidl (2007) and Deimer (2009), the nominal flow rate of a valve can be increased either by a bigger stroke or by an increased diameter of the valve spool or the poppet. But since both measures tend to increase the switching time the flow passage area has to be made big with small diameter and stroke.

The concept presented in this paper realizes the required big flow passage area by a multitude of small poppets with diameter D opening a bore of diameter d (Fig. 1). Since a favorable d/D ratio turned out to be in the range of 0.9 these two diameters are not distinguished in the following discussions concerning the scaling properties of the multi-poppet design. Following the rule that the flow rate saturates roughly at a poppet stroke of a quarter diameter, the total flow passage area and so the nominal flow rate of an n-poppet valve are proportional to n times the bore cross sectional area

$$A_{\mathrm{F}} = n d^2 \pi / 4 \tag{1}$$

whereas the pilot volume is proportional to

$$V_{\mathrm{P}} = n d^3 \pi / 16 \tag{2}$$

The bigger the pilot volume the longer is the switching time for a given pilot valve. Thus, a large number of slim poppets can provide the same flow rate as a smaller number of thicker poppets at much smaller switching time (Eq. 3 and 4).

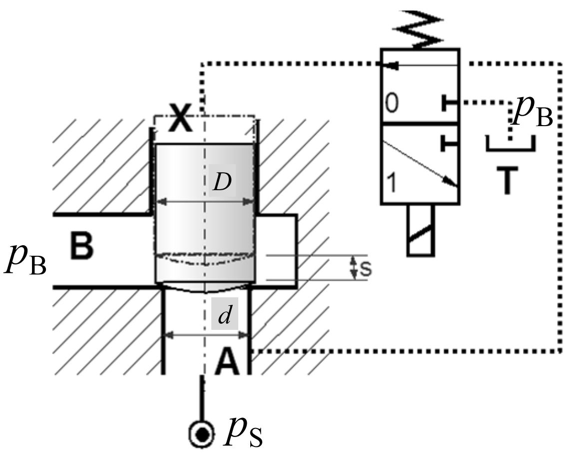

Fig. 1: *Scheme of the seat type valve (only one poppet shown) and its piloting with a 3/2 switching valve*

At first glance this idea seems to contradict the low cost demand, because low leakage from X-port to B-port requires precisely grinded-in poppets. But needle bearing rollers can be used as such poppets. They are very cheap despite a high accuracy of 1-2 microns. Grinding of poppet bores can be avoided by ballizing which is a quite powerful and cheap method in series production.

The basic sizing determines the number of poppets and their diameter. This is guided by the two functional requirements switching time T_{SW} and nominal flow rate Q_{N} which are related to the design parameters by the following equations

$$Q_{\mathrm{N}} = n \frac{d^2 \pi}{4} \alpha \sqrt{\frac{2 p_{\mathrm{N}}}{\rho}} \tag{3}$$

$$T_{\mathrm{SW}} = \frac{d}{4\alpha} \frac{Q_{\mathrm{N}}}{Q_{\mathrm{NV}}} \sqrt{\frac{\rho}{2\Delta p}} \tag{4}$$

where ρ is the fluid density, n the number of poppets, α the flow coefficient, Q_{NV} the pilot valve's nominal flow rate, p_{N} the nominal pressure drop, and Δp the pressure difference at the poppet's metering edge.

Equations 3 and 4 serve only as basic sizing rules and are only valid under the following assumptions:

- Instantaneously switching pilot valve
- Constant pressure drop Δp during the whole switching process
- Poppet acceleration forces are neglected
- Stroke of the poppet is *d/4*
- Neglecting effects like flow forces and fluid friction.

Models which take poppet acceleration and finite switching of the pilot valve into account have been established and used in the analysis of this concept. The most significant perturbation of the simple model leading to Eq. 2 and 3, however, is a variable pressure drop Δp. It depends strongly on the dynamics of the hydraulic system and its interaction with the valve dynamics. Since the system dynamics is application dependent it cannot be properly considered in the design of a valve for general use. Flow forces have a minor influence for this valve since the hydraulic pilot pressure forces exceed the flow forces by far.

Equation 4 shows that under the assumptions of this model the switching time depends linearly on the diameter of the poppet and on the desired nominal flow rate. Thus, the smaller the diameter the shorter is the switching time.

The switching time is also inversely proportional to the flow rate of the pilot stage. This simple model assumes infinite fast switching which may be too simple if the response time of the main stage comes close to the switching time of the pilot stage.

An excessive exploitation of the small diameter concept is limited by a cost efficient producibility of the poppet bores, by leakage constraints, by a minimum poppet stroke to avoid a clogging of the valve with large contamination particles, and by increased pressure losses due to increased fluid friction if the bores become too small since then the orifice equation is no longer valid.

3 Design of the Multi Poppet Valve

Crucial for the final valve design are, first, number, diameter and arrangement of the poppets and, second, the pilot valve.

3.1 Pilot Valve

An adequate valve for the utilized main stage is a 3/2-switching valve developed by LCM and published in (Winkler et al., 2008 and Ploeckinger et al., 2009).

This valve allows only small pressures at its tank port (< 20 bar) which poses some constraints on the piloting circuit. The constraint is satisfied by connecting the tank port of the pilot valve directly to a separated tank line, as shown in Fig. 1, and not with port B of the poppet valve which would be the simplest way since no extra piloting ports are required.

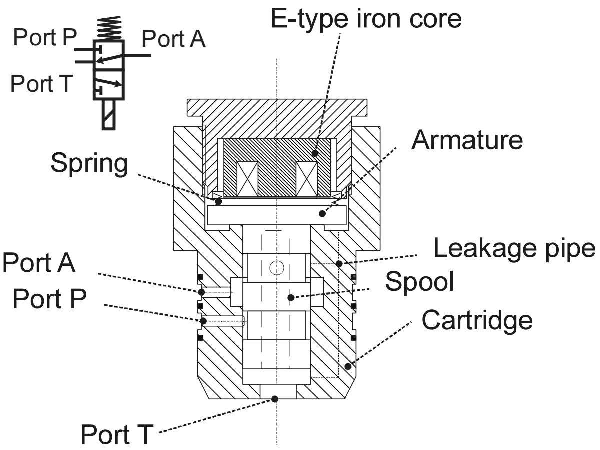

Fig. 2: *Schematic of the pilot valve concept*

Figure 2 depicts the schematic of the pilot valve. It is a spool valve with two metering edges. Because of its superior force capacity a flat armature E-type solenoid is used in order to achieve the low switching times. The iron core of the solenoid is manufactured from a proper laminated package of magnet steel. To get a really compact design a wave spring (Smalley, 2008) is used. Detailed information on this valve is published in Winkler et al., 2008 and Ploeckinger et al., 2008. It's main performance data are:

Nominal flow (@ 5 bar):	10 l/min
Switching time (hydraulic):	1-2 ms
Leakage (@ 200 bar):	<0.05 l/min

To get fast switching, a high starting current of up to 50 A has to be applied. (In a further optimization procedure this value should be reduced to 30 A). After one ms the current is reduced to a hold current of only 5 A at approx. 2 V. In Fig. 3: measurements of the optimized solenoid for equal on and off switching times at different supply pressures can be seen. The solenoid's optimization targeted low manufacturing costs, acceptable electric power requirements, fast and reliable switching, and the easy integration of the solenoid into the valve.

With oil in the armature chamber the switching time is in the range of 1.5 to 2 ms and, compared to other valve concepts, there is a low dependency on port pressure levels.

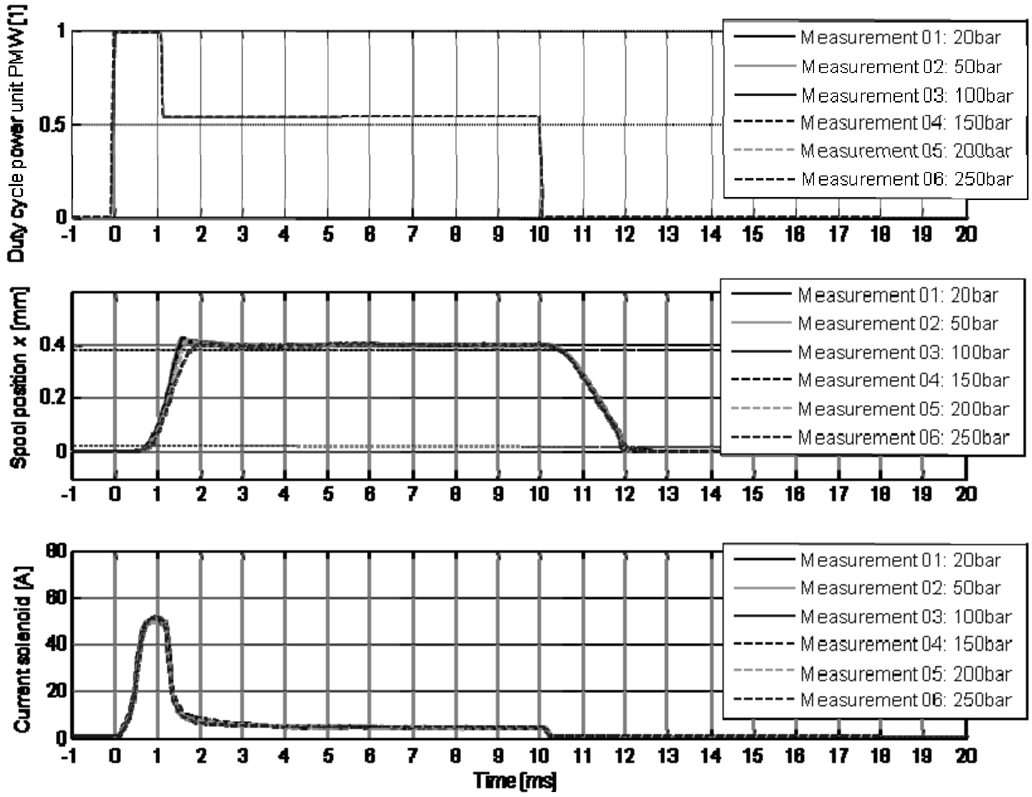

Fig. 3: *Switching behaviour of the pilot valve for various system pressure levels; Top: duty cycle of the PWM-signal; middle: spool position; botttom: solenoid current*

The onboard electronic device is shown in Fig. 4. The device is equipped with a microcontroller and an integrated CAN interface. Besides control of the solenoid current, it also performs several condition monitoring tasks (protection against overheating, reporting too high currents and too low power supply voltage). The valve can be accessed via a TTL signal or a CAN interface.

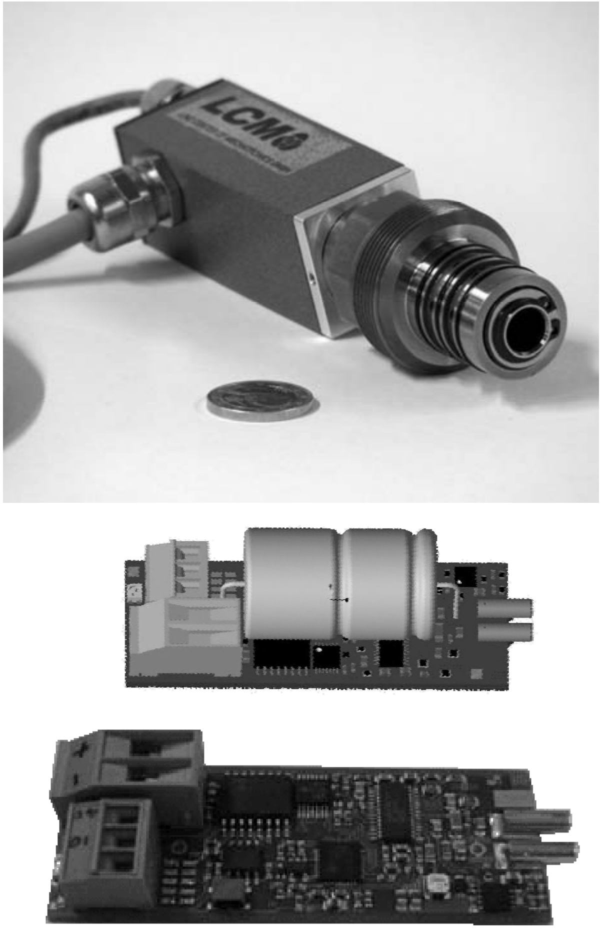

Fig. 4: *Assembled prototype valve with integrated electronics (top) and 3D model (middle) and photo of the electronics (bottom)*

In Table 1 and Table 2 the hydraulic and electric specifications are shown.

Table 1: Hydraulic specifications

hydraulic specifications	value	comment
valve type	3/2 spool valve with flat armature	alternatively there is a 2/2 valve normally open or normally closed available
nominal flow rate	Q_N = 10 l/min @ 5 bar	over one metering edge
switching time	< 2 ms in symmetric mode	possibility to tune - eg., valve opening faster and closing slower
maximal repeating frequency	f_{max} = 100 Hz duration 100% f_{max} = 200 Hz duration approx. 10%	
maximal pressure	p_{max} = 200 bar	
maximal flow	Q_{max} = 40 l/min	
kinematic viscosity range	23 to 50 mm^2/s	restricted to fulfil load cycles and switching time
maximum load cycles	> 100 million	
dimension	max diameter 40 mm x length 70 mm	without electronic
	max diameter 40 mm x length 155 mm	with electronic

Table 2: Electric specifications

electric specifications	value	comment
power supply	24 V DC	
actuation	TTL-signal or via CAN	CAN galvanic separated (24 V fault protected)
mode	current control	
condition monitoring	voltage, temperature on electronic, peak current	
max. current	I_max=70 A	
	full bridge	
electric max. repeating frequency	200 Hz	depends on currents and temperatures of housing
temperature housing	-20 °C up to 85 °C	when using automotive version of the parts

The valve was tuned to run durably at 100 Hz and it ran 100 million switching cycles without any problem. More details are given in Ploeckinger et al., 2009.

3.2 Main Stage

The prototypal design of the main stage is shown in Fig. 5. The poppets (1) are arranged around the pilot valve (6). For a nominal flow rate of 100 l/min at 5 bar pressure drop 14 poppets are needed (considering only the pressure drop at the metering edge). The poppet housing (2) guides the poppets (1) and also includes the poppet seat to provide their precise coaxial arrangement by a drilling in one set. The ring (5) centers housing (3) and poppet housing (2) and also limits the poppet stroke.

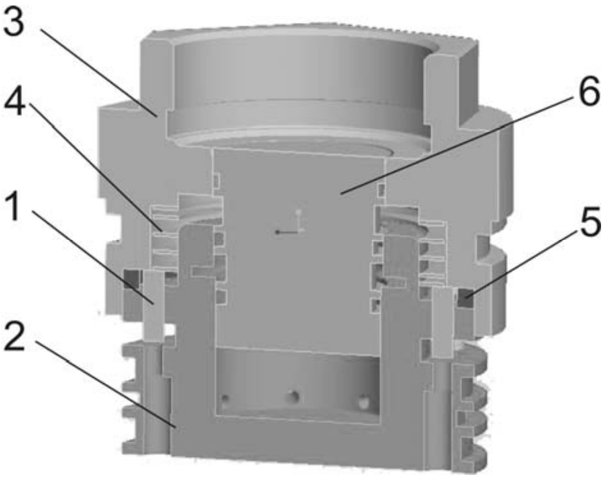

Fig. 5: *Mechanical design of the multi poppet valve*

To avoid tiny springs for each poppet one wave spring (4) common to all poppets is used. Synchronization of the poppets is provided by a shim ring placed between the poppets and the wave spring. Poppet housing (2) design is strongly determined by the spatial arrangement of the diverse pilot flow channels. The pilot valve's (6) ports are not optimally placed for this use as a pilot valve. A much simpler housing design would be possible with a modified pilot valve port placement.

3.3 Poppet Position Measurement

Measurements of the prototype must also encompass poppet position to get direct information on switching times. To measure every poppet is far too complex and costly and hardly realizable at all due to the integrated design of the valve.

Instead of a poppet the shim ring position is measured. To do so, an eddy current sensor is placed in a separate bore. The measurement device and its arrangement in the valve are depicted in Fig. 6. The eddy current sensor (type micro epsilon EU05) (8) is fixed via a tube (9) in the valve. This tube (9) realizes also the cable routing via port A (11) out of the valve. Figure 6 shows also the arrangement of the valve in the valve block (7). Port B (10) is realized as relatively big undercut in the valve block (7). One shortcoming of this method is that the positions of the shim ring and the poppet may differ since the ring is quite flexible and may get out of contact with the poppets.

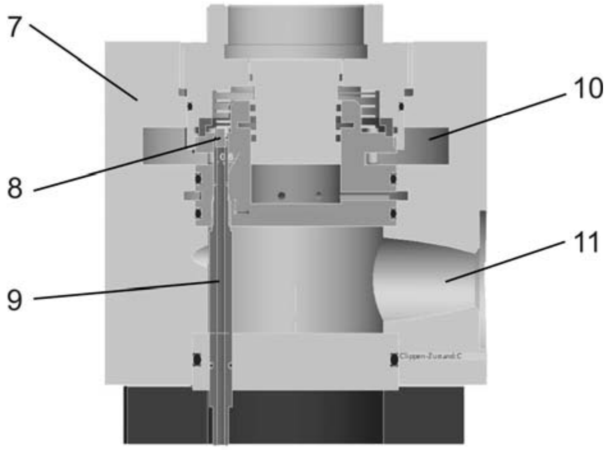

Fig. 6: *Valve with block and sensor*

4 Measurements of the Multi Poppet Valve

4.1 Stationary Flow Characteristic

As stated in Section 1 the specified nominal flow rate is 100 l/min.

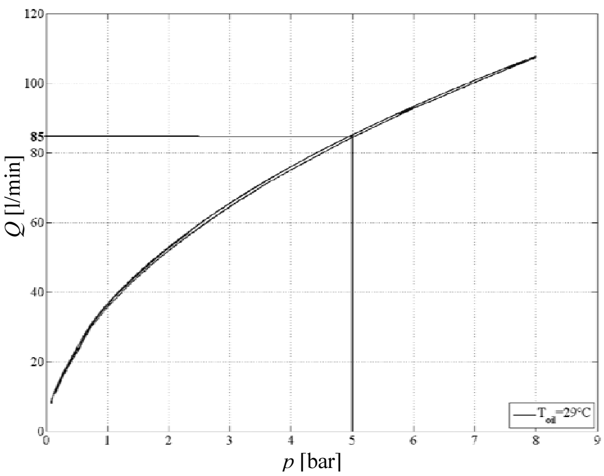

Fig. 7: *Flow characteristic*

Figure 7 shows that a nominal flow rate of 85 l/min at 5 bar is achieved. The difference to the specified value results from additional flow resistances in the flow channels system or from a smaller flow coefficient of the metering geometry than assumed. Oil temperature of these measurements was 29 °C (HLP32).

The deficiency in flow rate can easily be corrected by two or three more poppets (every poppet contributes 6 l/min at 5 bar).

4.2 Switching Time

Figure 9 shows the experimental valve opening results; the dash dotted line depicts the pilot signal, the continuous line the movement of the pilot stage. All other lines show the measured position of the main stage's shim ring. For technical reasons the pilot stage spool position is measured separately under comparable conditions (the switching time of the pilot valve is basically not pressure dependent). This measurement has been inserted in the main stage measurement plots to provide a better understanding of the main stage responses' relation to the pilot stage motion. The switching time of the used pilot valve is about 2.1 ms which is more than the results shown in Fig. 3. The reason is that the results are from different pilot valves. Due to the sensitivity of the valve to oil sticking effects resulting from slightly different armature-solenoid distances or oil temperatures, the switching time can vary in a range of 0.5 ms.

It is particularly noticeable that the main stage is much faster than the pilot stage. For an initial pressure difference of 25 bar between A- and T-port the main stage opens within 0.44 ms (5 % - 100 % valve opening).

The used pilot valve is not symmetric and switches from its 0-position to 1-position at approximately 20% relative spool position. Applying this assumption to the valve opening (Fig. 9) the pilot valve would switch from P to T-port at about 1 ms time. According to the properties of the valve system and the results of Fig. 11 (valve closure) the main stage should respond instantaneously to pilot stage switching. In Fig. 9 this is indicated by the bold dashed line which shows the assumed poppet motion. But the measured shim ring motion starts not before 1.3 ms. This discrepancy is explained as follows (also Fig. 8): The shim ring stretches over a large portion of the annular control (X-) chamber. The cross sectional area of all 14 pistons is about 17 % of the annular cross section. Thus, the fluid displacement of the pistons is significantly smaller than that of the ring and must be compensated by a fluid flow between shim ring and the housing. When the poppets move upwards during the valve's opening motion they force the ring to do the same at all contact points. The difference of oil displaced by the poppets and the ring has to flow around the ring to the cavity area below the ring. This flow is resisted by the small clearance between the ring and annular cavity and causes a higher pressure in the upper part of the cavity. That pressure deflects the ring downwards, particularly at the place of the sensor where a relatively large sector of the ring is unsupported by poppets. The overshoot of the shim ring motion that can be seen in Fig. 9 indicates clearly the shim ring's high flexibility.

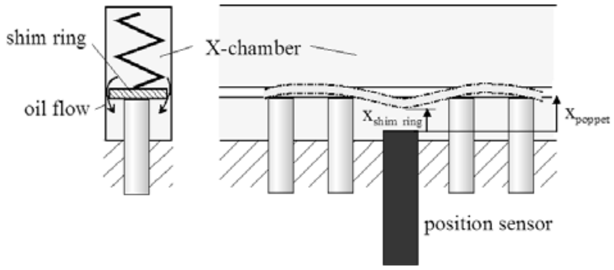

Fig. 8: *Measured shim ring position may differ from poppet position due to shim ring deformation*

The authors considered to test their conjecture on the reasons for the difference between the ring and poppet position by a simulation model. This model would have to include the ring's deformation dynamics, its contact with the pistons and wave spring, resistances for the flow bypassing the ring, and the flow through the pilot valve. Since this model would possibly require additional measurements the authors decided to spend the effort for a direct measurement of the poppet position.

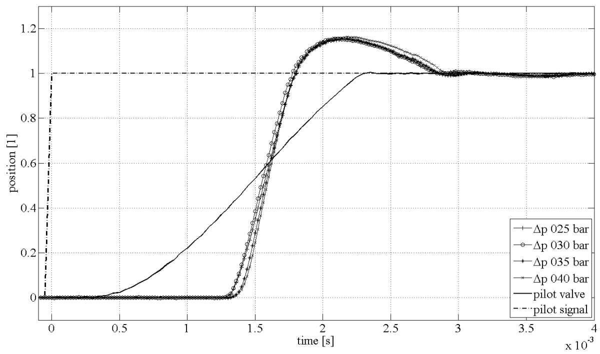

Fig. 9: *Valve opening signals: pilot stage and main stage shim ring position signals at 25 - 40 bar pressure difference between A- and T-port and supposed poppet motion (bold dashed line)*

No significant change in the valve response can be observed between 25 bar and 40 bar A-T ports pressure difference. This is in conflict with the basic result of Eq. 2 which predicts a clear influence of the pressure difference but a further confirmation of the assumption about the true nature of the shim ring motion. It does not follow immediately the poppet as originally expected but is superposed by a strong deflection of the shim ring at the point of measurement as a result of the oil displacement processes as discussed above and a damped free elastic swing when the poppets have reached the fully open position.

Figure 10 shows the switching times for various A-T ports pressure differences ranging from 95 bar up to 220 bar. The measured ring motion becomes faster for these higher pressures and also the overshoot is more intense.

Figures 11 and 12 depict the measurements of the closing motion. Due to the piloting circuitry, the A and B port pressure difference determines the response dynamics of closing. Since this pressure difference was smaller for closing than for opening, closing lasted longer than opening. Furthermore, measured shim ring position probably differs from poppet position. If the model sketched above for the opening motion is true the poppets should move faster than the shim ring and switching time for closing is shorter than shown in the figures.

The pressures in the legends of Fig. 9 to 12 are the pressure differences before valve opening. The valve is held open for 10 ms before it is closed again. In such short time dynamical effects in the hydraulic system may not fully decayed so that the actual pressure difference at moment of closure can differ considerably form the pressure in the legend. These dynamical effects cause a sign reversal of pressure difference some time after valve closure which reopens the valve for a short time (Fig. 11 and 12).

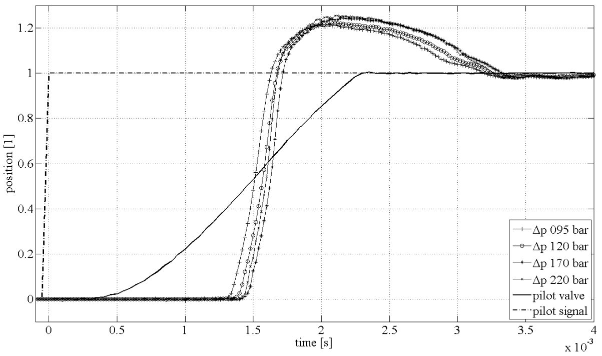

Fig. 10: *Valve opening of pilot stage and main stage at 95 – 220 bar pressure drop*

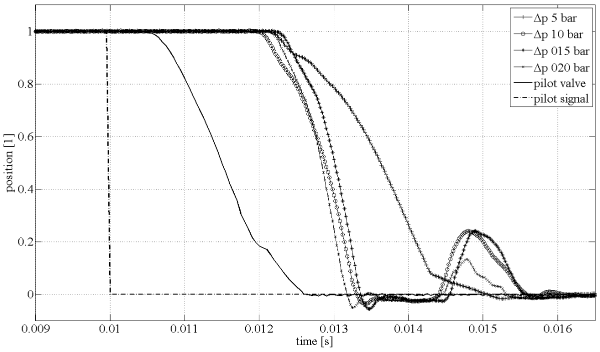

Fig. 11: *Valve closing of pilot stage and main stage at 5 - 20 bar pressure drop (A-B ports)*

In Fig. 11 and 12 a short reopening of the poppets can be observed at about 0.015 s in case of low pressure drops and 0.016 s in case of high pressure drops. This results from a short negative pressure difference at the valve due to pressure waves provoked by the fast valve closure.

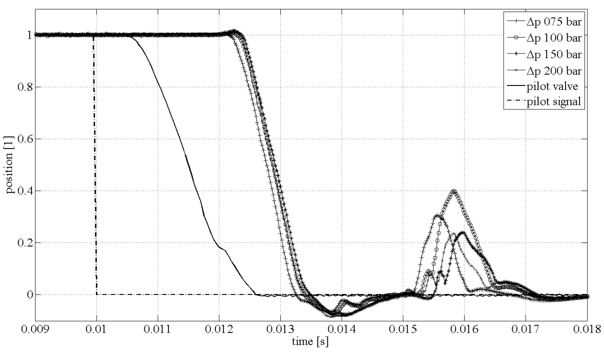

Fig. 12: *Valve closing of pilot stage and main stage at 75 – 200 bar pressure drop (A-B ports)*

4.3 Valve leakage

First measurements showed that the presented valve is basically not leaking. Only in case of raising the pressure in combination with some air in the oil a certain leakage that fades away after some seconds can be observed. This phenomenon has not been studied sufficiently so far. Thus, complete information on the leakage cannot be given yet.

Conclusions

A new concept of a piloted valve is presented, which is particularly encouraging in many respects. For a valve of 85 l/min nominal flow rate (@ 5 bar) a switching time of about 1 ms at a pressure difference of 5 bar was simulated. The measured switching times which have been significantly lower than the simulated ones are probably a measurement artifact. For realistic switching time measurements the spring system and the measurement method have to be improved.

The valve design facilitates low cost manufacturing by

- using needle bearing rollers as poppets and
- employing the ballizing process for producing the poppet bores and
- a simple overall design

The used pilot valve fulfills the demands on low switching time and a high fatigue-endurable operating frequency. But the applied pilot valve has too large flow rate for this poppet valve. A smaller nominal flow rate but faster pilot valve could reduce the switching times. With a modification of its ports arrangement the complexity of the design of the multi poppet valve can be drastically reduced.

Future work will be devoted to improved measurements of poppet position and leakage, an optimized pilot stage, and the optimized design for cheap and reliable manufacturing in view of powerful manufacturing processes.

Nomenclature

ρ	fluid density	[kg/m^3]
A_F	fluid passage area	[mm²]
d	bore diameter at metering edges	[mm]
D	poppet diameter	[mm]
n	number of poppets	-
Δp_N	nominal pressure drop	[bar]
Δp	pressure difference	[bar]
p_B	pressure at port B	[bar]
p_S	supply pressure	[bar]
p_T	tank pressure	[bar]
Q_N	main stage nominal flow at a pressure drop Δp_N	[l/min]
Q_N	Pilot valves nominal flow rate at a pressure drop Δp_N	[l/min]
s	poppet stroke ~D/4	[mm]
V_P	pilot volume	[mm³]

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