FEASIBILITY STUDY OF A NOVEL SLEEVE TYPE SWITCHING VALVE

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Abstract

Digital hydraulics became an area of high interest in the last years. Its success depends heavily on cheap and fast switching valves which also feature compact design, low leakage, low electrical current and power requirements, and easy integration into the hydraulic system. This paper presents a feasibility study of a novel magnetically actuated, two coil, bistable 3/2 way sleeve type seat valve. The applicability of the valve was analysed by analytical models, a prototypal design, by measurements of the prototype, and by CFD simulation of flow forces. The sleeve concept turned out to have several inherent problems: insufficient closing-capability due to pressure forces and a pressure dependent clamping of the sleeve; some leakage because of manufacturing tolerances of the contact surfaces, and a complex design with many precise and costly parts to obtain compactness.

Keywords: fast switching valve, bistable valve, seat type valve, sleeve valve

1 Introduction

The current trend to realise digital hydraulic systems (Linjama and Laamanen, 2008; Scheidl and Winkler, 2009; Laamanen and Linjama, 2010; Linjama and Scheidl, 2010) stimulates the development of fast switching valves with switching times in the milliseconds range. In the last decade different new switching valves have been developed and evaluated. They combine state of the art or novel spool or seat type concepts with existing or new actuation principles to better fulfil the manifold requirements on such valves. These different attempts to develop better valves are going into quite different directions of the design space. It seems to be a widely accepted conjecture that better switching valve configurations than those already known still can be found.

The main requirements on advanced switching valves are:

- Fast switching: is the key for the realisation of switching control, since switching converters become more compact and more efficient with higher switching frequencies (Manhartsgruber, 2006); but also digital valve systems benefit from a fast valve to control fast response actuators and to avoid pressure ripples due to asynchronous switching of different valves (Laamanen, 2007); required switching times are in the range of 1ms.
- Low cost: digital valve systems require multiple valves and will be competitive to proportional valves only if cost of switching valves can be kept low.
- Compact design: in switching control compact arrangements of all components are essential for efficiency, low pressure pulsation and noise; in digital valve systems the size depends on compact valves and the possibility to arrange them appropriately.
- Electrical thresholds concerning power, voltage, or current: the thresholds stem from cost barriers of microelectronic components, or actual standards of control hardware's frontends.
- Low parasitic hydraulic capacitance and inductance of internal flow channels where high flow acceleration occurs.
- Easy integration into the hydraulic system: integration concerns the mechanical (e.g., cartridge design) and hydraulic connection, (minimize parasitic effects of connection lines), the acoustic impacts, protection of sensitive valve components against mechanical destruction, and electrical aspects.

The following research results constitute the main cornerstones of switching valve research and development of the last fifteen years.

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(Sturman, 1994) developed a so called "latching valve", a magnetically actuated, two coil, bistable spool valve. The bistable behaviour provided by magnetic remanence forces allows avoidance of a holding current.

(Garstenauer, 1999; Garstenauer, 2001) investigated a spool type switching valve actuated by a buckling beam aiming at conservation of the kinetic energy of the spool motion by the beam's bending energy in the buckled state. For realisation of the buckling of the beam and, hence, the actuation of the valve piezoelectric actuators have been proposed, although the prototypes exploited a hydraulic loading of the beam.

(Winkler, 2004) proposes multiple metering edges for high flow rate spool type valves to keep the required valve stroke small, a prerequisite to provide an efficient and fast electromagnetic actuation by flat armature magnets.

(Lauttamus, 2006) presents a seat valve with a double metering edge to reduce the hydrostatic forces of seat type valves and provide a bistable hydraulic behaviour and enable in this way a direct magnetic actuation. This concept is extended in (Uusitalo, 2009) by a magnetic actuator that has some backlash of its motion relative to the poppet. In an initial motion phase the armature gets some momentum by means of which it jerks the poppet free against the high hydrostatic forces active in the closed valve state.

In (Winkler, 2007) the concept of multiple metering edges is adopted for a piloted plate type seat valve very similar to the plate type check valves employed in compressors.

(Ouyang, 2008) studies a piezo actuated poppet valve where piezoelectric stack actuators "shoot" the poppet from its one end position to the other by accelerating the poppet to the travelling speed in a short initial section of its full stroke only.

(Ploeckinger, 2008) outlines the design of a solenoid actuated, 3/2 way spool valve for moderate flow rates, with onboard electronics and a minimum number of parts to keep costs low.

(Platzer, 2009) employs the pressure distribution of a fluid flowing in a narrow gap the height distribution of which is the essential control input. With a slight variation of this height distribution in the range of several microns the valve motion can be controlled. The flow control element in this valve is a sleeve.

(Winkler, 2009) studies a multiple hydraulically piloted poppet valve to achieve fast switching for high nominal flow rates.

(Mahrenholz, 2010) proposes a 3/2 way spool valve with two opposing coils acting on an annular armature. Eddy current problems are restricted by a very thin armature plate and iron part despite a fast current build-up and a non laminated design. The design leads to a rather compact valve.

The use of sleeves as the flow control element instead of spools, poppets, or balls is not very common in hydraulic valve technology. There have been trials in engine valve technology (Autocar Handbook, 1919) but this technology did not prevail despite several repeated attempts in the combustion engine's history.

Two interesting aspects of a sleeve valve are (i) the reduction of the axial flow forces by its small abutting face area, provided the sleeve is rather thin walled or tapered at its ends, and (ii) an easy hydraulic realisation of a bistable behaviour. These aspects have been the main motivation to develop a sleeve type valve actuated by two solenoids, one for each direction, in order to avoid any holding current as is the case in the valve designs of (Sturman, 1994) and (Lauttamus, 2006). For a seat type valve the leak tightness is essentially determined by the contact pressure. The higher the pressure to be sealed the higher must this contact pressure be. In view of the high pressures in hydraulics the high specific force capability of hydraulic principles provides a decisive advantage over magnetic or spring forces for realising this contact pressure.

This paper presents a study on a 3/2 way sleeve type seat valve actuated by two solenoids aiming at an assessment of this concept with respect to its suitability for fast switching valve technology. The concept is outlined in Section 2, models of its hydraulic and dynamical performance in Section 3. Section 4 and 5 present a prototypal design and experimental results. The experimental evaluation of this concept is presented in Section 6.

2 Basic Concept

Figure 1 provides the basic concept of the sleeve type seat valve. The sleeve is guided by a fixed pin and actuated by two solenoids, one for each switching direction. The abutting faces of the sleeve are nearly sharp edges and form seats with adjacent faces of the valve housing. The bistable property of this valve is achieved by two small interior shoulders (A_{IS}, A_{IT}) which generate hydrostatic forces in conjunction with the pressures of the P-port (high pressure) and the T-port (low pressure). In A-T position of the valve the A port has pressure p_T, thus all abuttal faces of the sleeve see low pressure p_T with exception of A_{IS} which faces high pressure p_S. Hence, the sleeve is held in its A-T position by the hydrostatic force $(p_S - p_T)A_{IS}$. In position P-A all abuttal faces of the sleeve have pressure p_S, only A_{IT} has pressure p_T. This creates a hydrostatic force of size $(p_S - p_T)A_{IT}$ to hold the sleeve in position P-A. The combination of these forces and the sharp edges of the sleeve give high contact pressure and should lead to a nearly leak free valve. The A-port is the exit port that is alternately connected to either the P or the T-port.

A small clearance between the pin and the sleeve provides a sealing of these two ports, yet with some small leakage.

Major design parameters of this valve are the sleeve diameter d_{SI}, the shoulders' gauge Δd_S, the valve stroke x_{max}, and the solenoid's length l_M, iron core width s, winding count N, and initial air gap h_{SP0}. Two designs of the iron core have been considered: (i) a non-laminated, preferentially grooved ring and (ii) four laminated straight segments, the design which was finally realised by the prototype.

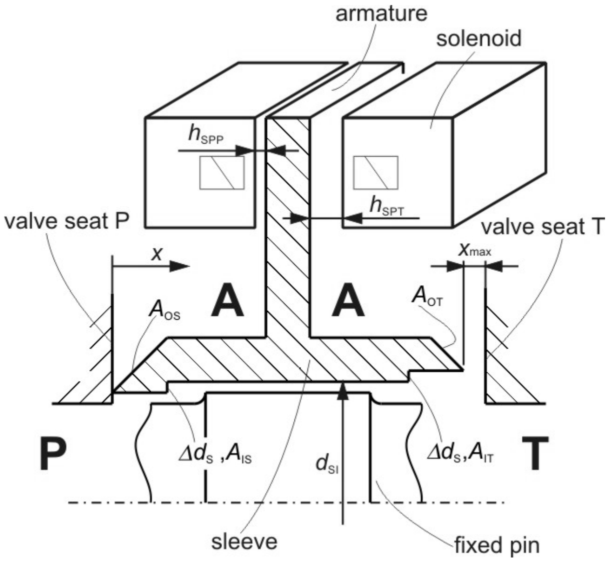

Fig. 1: *Basic concept of the valve*

The two main functional requirements (FRs) are

- the nominal flow rate Q_N shall meet the desired value $Q_{N,des}$
- the total switching times $t_{sw,PT}$ and $t_{sw,TP}$ to switch from P to T and from T to P, respectively, shall be $t_{sw,des}$.

The total switching time is measured from voltage signal onset till full valve opening. Throughout the paper switching time means total switching time according to this definition.

In addition to the two FRs the demands on low leakage are formulated as constraints:

- the leakage flows Q_{PA}, Q_{PT}, and Q_{AT}, shall be smaller than the respective tolerated values $Q_{PA,tol}$, $Q_{PT,tol}$, and $Q_{AT,tol}$.

The general demands on the valve are:

- Low complexity design
- Low manufacturing costs
- Robust design
- Low oil cleanliness requirements

3 Mathematical Modelling, Simulation and Design Optimization

3.1 Nominal Flow Rate

The nominal flow rate at a certain nominal pressure loss Δp_N = 5 bar is determined by the orifice equation (Eq. 1). The relevant sleeve dimensions are defined in Fig. 1, α_D is the flow coefficient and ρ the density of the fluid (Merrit, 1967):

$$Q_N = \alpha_D d_{SI} \pi x_{max} \sqrt{\frac{2}{\rho}} \sqrt{\Delta p_N} \tag{1}$$

α_D = 0.6 is a typical value for sharp-formed metering edges.

3.2 Fluid Pressure Force

The different fluid pressure levels in the valve yield a pressure force acting on the sleeve.

The interior shoulders of the sleeve (labelled by A_{IS} and A_{IT} in Fig. 1) lead to an axial pressure force which should keep the valve remaining in the actual end position without any magnetic force.

In a real application the pressure at port A is determined by the specific static and dynamic properties of the hydraulic system. Since no specific application is chosen for this study a specific model is constructed for the further analysis, namely that port A pressure is a weighted sum of port P and port T pressures with the valve position x as weighing function:

$$p_A(x) = p_T + \frac{p_S - p_T}{x_{max}} x \tag{2}$$

With this assumption, the resulting axial pressure force is

$$F_P(x) = (A_{OS} - A_{OT}) p_A - p_S A_{IS} - p_T A_{IT} \tag{3}$$

A_{IS} and A_{IT} are the active areas of the interior shoulders, A_{OS} and A_{OT} the active surfaces outside of the metering edge (see Fig. 1).

3.3 Flow Force

Steady state flow forces are exerted on the spool at the metering edges. These flow forces shall be minimized. It is an essential part of the design concept of this valve to reduce flow forces by a rather sharp edged sleeve with only tiny planar abutting surfaces. The pressure drop should just happen in this small annular volume between these surfaces. This assumption is confirmed by CFD computations of (Falkner, 2009) - briefly addressed in Section 6. But a sufficiently sharp edge can hardly be realized practically as will be discussed in Section 6 too. A model for the flow forces caused by the not perfectly sharp edges is presented in Section 6. It condenses CFD simulation results into a simple analytical model.

3.4 Fluid Friction Force

Fluid friction arises in the sealing gap between sleeve and fixed pin. The friction force according to Eq. 4. depends on the area in contact, the dynamic viscosity η, the clearance h_D and the velocity of the sleeve $\dot{x}$ (Merrit, 1967).

$$F_R(\dot{x}) = \frac{d_{SI} \pi b_D \eta}{h_D} \dot{x} \tag{4}$$

3.5 Squeeze Film Damping Force

Due to the movement of the armature between the two solenoids, oil is squeezed out of the narrowing gap and on the other side the same amount of oil is fed into the widening gap. The pressure distribution in the gap can be computed by the Reynolds-Equation (Szeri, 1998). To simplify the calculation, only the onedimensional form of this differential equation is used:

$$\frac{\partial}{\partial x}\left(\frac{h_S{}^3}{\eta}\frac{\partial p}{\partial z}\right)=12\dot{h}_S \tag{5}$$

The solution of this differential equation yields the fluid pressure in the gap h_S between solenoid and armature (see Fig. 2).

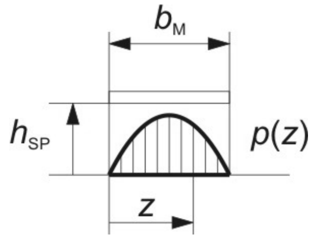

Fig. 2: *Parabolic pressure distribution*

It is assumed that only a positive absolute pressure is possible. Therefore, $p(z)$ is limited to $\tilde{p}(z)$ in a way that the condition $p_A+\tilde{p}(z)>0$ is fulfilled everywhere in the gap.

By integrating the pressure over the rectangular area, the resulting squeeze force for the P-sided solenoid is given by

$$F_{DP}\left(h_{SP},\dot{h}_{SP}\right)=\int_0^{b_M} l_M\,\tilde{p}_P(z)dz \tag{6}$$

and for the T-sided solenoid by

$$F_{DT}\left(h_{ST},\dot{h}_{ST}\right)=\int_0^{b_M} l_M\,\tilde{p}_T(z)dz \tag{7}$$

3.6 Magnetic Force

Figure 3 displays the magnetic circuit with its main geometric dimensions. A linear magnetic circuit model (Kallenbach, 2003) is used.

Further modelling assumptions are:

- Immediate current rise after voltage signal onset
- Negligence of magnetic saturation and eddy currents

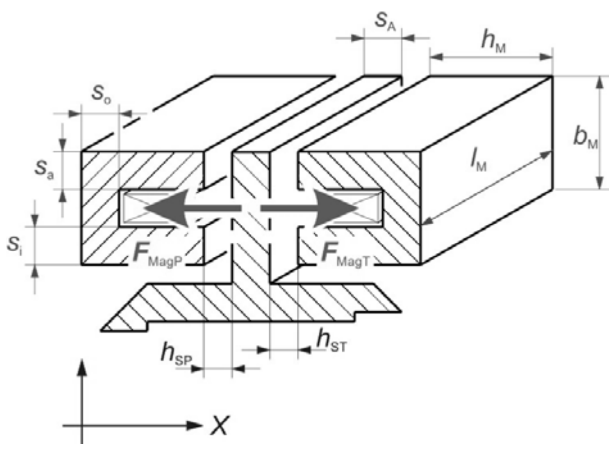

Fig. 3: *Magnetic circuit (solenoid and armature)*

The magnetic resistance R_{MP} and R_{MT} of the two solenoids (see Fig. 3) consists of the resistance of the iron part and the air gap:

$$R_{MP}\left(h_{SP}\right)=\underbrace{\sum_{j=1}^{4}\frac{l_j}{\mu_0\mu_r A_{I,j}}}_{\text{iron part}}+\underbrace{\frac{h_{SP}}{\mu_0}\left(\frac{1}{A_{I,2}}+\frac{1}{A_{I,4}}\right)}_{\text{air gap}} \tag{8}$$

$$R_{MT}\left(h_{ST}\right)=\underbrace{\sum_{j=1}^{4}\frac{l_j}{\mu_0\mu_r A_{I,j}}}_{\text{iron part}}+\underbrace{\frac{h_{ST}}{\mu_0}\left(\frac{1}{A_{I,2}}+\frac{1}{A_{I,4}}\right)}_{\text{air gap}} \tag{9}$$

$$\begin{aligned}&A_{I,1}=s_A l_M\,;\;A_{I,2}=s_a l_M\\&A_{I,3}=s_o l_M\,;\;A_{I,4}=s_i l_M\\&l_1=l_3=b_M-\tfrac{1}{2}\left(s_a+s_i\right)\\&l_2=l_4=h_M-\tfrac{1}{2}s_o\end{aligned} \tag{10}$$

The magnetic flux is given by

$$\Phi_P\left(h_{SP}\right)=\frac{N I_P}{R_{MP}} \tag{11}$$

$$\text{and }\Phi_T\left(h_{ST}\right)=\frac{N I_T}{R_{MT}}\,. \tag{12}$$

For this linear magnetic model, magnetic co-energy equals magnetic energy which reads:

$$W_{MagP}\left(h_{SP}\right)=\frac{N^2 I_P{}^2}{2R_{MP}} \tag{13}$$

$$W_{MagT}\left(h_{ST}\right)=\frac{N^2 I_T{}^2}{2R_{MT}} \tag{14}$$

The magnetic forces, which in general are the partial derivative of the magnetic co-energy with respect to the air-gap distance are given by the following relations:

$$F_{MagP}\left(h_{SP}\right)=\frac{1}{2}\Phi_P{}^2\frac{\partial R_{MP}}{\partial h_{SP}} \tag{15}$$

$$F_{MagT}\left(h_{ST}\right)=\frac{1}{2}\Phi_T{}^2\frac{\partial R_{MT}}{\partial h_{ST}} \tag{16}$$

3.7 Equation of Motion

The equation of motion of the sleeve's axial displacement derived from the linear momentum balance principle reads

$$\begin{aligned}m\,\ddot{x}=\sum F&=F_{MagT}-F_{MagP}+F_P\\&-F_R-F_{DP}-F_{DT}\end{aligned} \tag{17}$$

3.8 Simulation and Design Optimization

These previous mathematical models for fluid pressure force, fluid friction, squeeze film damping, magnetic force, and linear momentum, are combined to a numerical simulation model established in MATLAB/Simulink® (Falkner, 2009). To determine the optimal geometric dimensions of the sleeve, several simulations with different design parameter combinations were run to find an optimal solution with respect to low consumption and switching time.

Figure 4 is a geometric representation of the results of this parametric study. It shows that a large-scale sleeve diameter in combination with a short stroke is optimal.

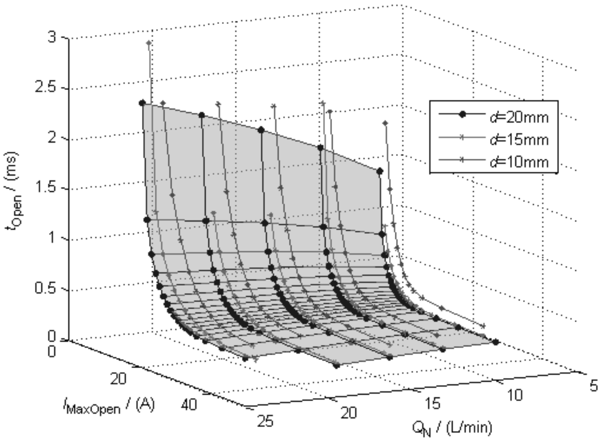

Fig. 4: *Simulation results for the design optimization*

A short stroke of the sleeve requires a small maximum air gap between solenoid and armature which leads to high magnetic forces for a certain impressed current and, in turn, to a fast switching.

On the other hand, a small stroke means a small flow passage width at the metering edges which endangers valve robustness with respect to oil contamination. It was decided not to go below 0.2 mm stroke.

With this fixation of the stroke to x_{max} = 0.2 mm the sleeve diameter is determined from the nominal flow rate condition Eq. 1 and was fixed to d_{SI} = 10 mm.

4 Prototype

4.1 Requirements

The requirements on the valve prototype are:

- Compact and fast pilot sleeve valve with a nominal flow rate of about 5…10 l/min at a pressure drop of 5 bar and a switching time $t_{sw,des}$
- Robust design with respect to manufacturing tolerances and oil cleanness

4.2 Valve Design

The contact surface for the first metering edge of the sleeve is located at the bottom of the housing. This metering pair provides the P-A port connection.

The sleeve element exhibits four separated armatures around the sleeve (see Fig. 5). The fixed pin and the sleeve's inner bore form a sliding guide and sealing gap with narrow tolerances of only a few microns to keep leakage very small. Furthermore, it is crucial to have a quite precise orthogonal alignment of the pin and the seat in the housing.

The two shoulders in the bore of the sleeve (areas A_{IS}, A_{IT} in Fig. 1), cause a resulting pressure force, which is utilized for a bistable operating mode of the valve (for a detailed explanation see Section 3.2). The metering edges of the valve are located at the two front faces of the sleeve. The metering edges are relatively sharp in order to reduce the flow force (see Section 3.3).

The second metering surface is located at the front face of the cap. In combination with the metering edge of the sleeve it controls the T-A connection.

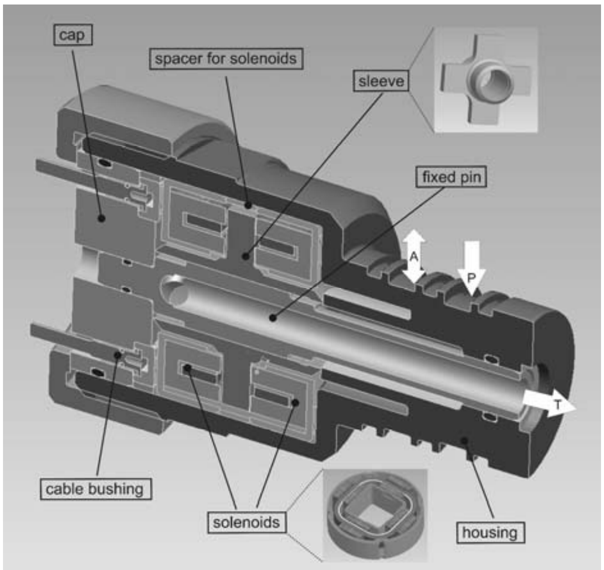

Fig. 5: *Basic design of the valve*

The solenoid consists of four separate magnet cores. The carrier of these cores is made of non-ferromagnetic material. The cores and the coil are backfilled by an epoxy resin.

The two solenoids are held in a certain distance via the spacer. They provide also a means to adjust the opening of both valve metering edges, and thus of the nominal flow rate.

The solenoids are in a highly pressurized area. Thus, the electrical connections of the magnets have to be conveyed to the outside in a leak proof way. For the leak proof cable bushings a combination of an o-ring and a plastic socket (using the high strength plastic PEEK) is applied.

A photo of the prototype is given in Fig. 6.

Fig. 6: *Prototype of sleeve valves*

5 Experimental Results

The fulfilment of the functional requirements, which constitute the main characteristics of a switching valve, was tested by measurements of the prototype.

To analyse the influence of the sleeves' tapered abutting faces on the valve's performance, three different sleeve versions were manufactured and tested.

5.1 Switching Time

The switching time is determined from the sleeve position which was measured by a pressure resistant eddy-current proximity sensor. Measurements were taken for three different sleeve edge dimensions. The geometries and accompanying switching times as a function of system pressure are shown in Fig. 7. For a comparison, the computed switching times based on the simple models of Section 3 are shown.

The duration to switch the A-port from T to P remains nearly constant at rising system pressure. The period to switch from P to T, however, is strongly increasing with system pressure.

This effect is quite strong and for system pressures beyond 50 bar the valve cannot be closed by the solenoid.

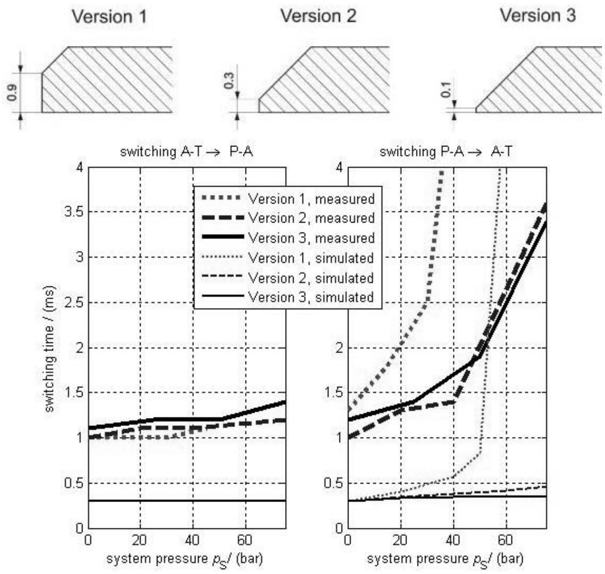

Fig. 7: *Switching times for the different sleeve versions; simulated and measured results*

The problem of the strongly pressure dependent closing time originates from the edgeless metering edges. As analysed in more detail in Section 6, that shape causes a strong pressure force. The radial extent of the sleeves' abutting faces is directly related to the quantity of the pressure force acting on the sleeve.

Based on this hypothesis, the radial extent of the abutting faces is reduced from 0.9 mm in version 1 to 0.3 mm in version 2. The tolerances for the radial extent is ± 0.02 mm. This `sharpening` of the metering edges yields an improvement in the closing dynamics (see Fig. 7).

But a further reduction of the radial face width to 0.1 mm in version 3 results only in a minor further reduction of the switching time of switching from P to T.

The computed switching times are much smaller throughout the whole parameter range. The main reason is the simple model of the solenoid, foremost the assumption of a constant current from the start of the switching command. Current rise up to a level where the magnetic force exceeds the fluid pressure force takes approximately 0.5 to 0.75 ms. A further shortcoming of this model is the negligence of magnetic saturation and of eddy currents. Both effects cause a further delay in switching, the quantification would require more elaborate models. A great problem of accurate modelling of such solenoids is also the variation of magnetic properties. The armature as part of the sleeve is made of heat treated steel. Its magnetic properties are not properly documented. A precise simulation requires measurement of the magnetic properties (magnetic flux versus field- strength curves), for which adequate equipment was not available.

5.2 Stationary Flow Characteristic

The flow measurements for the two switching positions P-A and A-T showed the curves illustrated in Fig. 8.

For comparison, the theoretical flow curve according to Eq. 1 with a flow coefficient $\alpha = 0.6$ is also indicated.

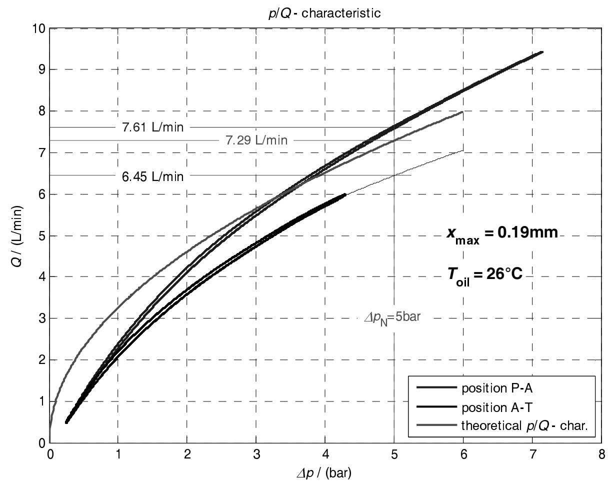

Fig. 8: *Measured p/Q-characteristics of the valve compared with the prediction by the orifice equation with a flow coefficient $\alpha = 0.6$*

The deviation between the two measured flow curves can be explained by slight differences between the two metering edge geometries and the differences from the theoretical values by additional losses in the flow channels inside the valve which are not covered by the orifice equation.

5.3 Leakage

According to Fig. 9, the total leakage can be divided into two constituents:

- Leakage over the sealing gap between sleeve and fixed pin
- Leakage over the metering edge P/A or A/T

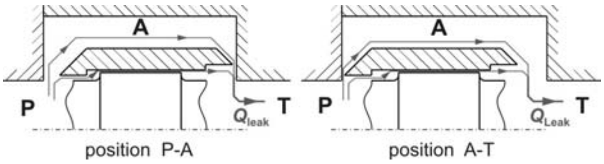

Fig. 9: *Leakage paths in both positions*

The total leakage was determined by volumetric metering of the leaking oil. The measurements at different system pressures showed a strong leakage up to 16 % of the nominal flow rate.

One reason for the strong leakage of the valve can be a weak sealing effect of sleeve and seat due to machining tolerances of the involved components. This may concern the evenness of the contacting surfaces and the right angularity of fixed pin and metering surfaces. The latter is considered to be the main reason for the high leakage. Another possible reason for the leaking is insufficient contact force between poppet and seat.

Table 1: Leakage Q_{Leak} at different system pressures

p_S	Position P-A		Position A-T	
bar	(1/min)	(% Q_N)	(1/min)	(% Q_N)
50	0.16	≈ 5%	0.49	≈ 17%
100	0.39	≈ 2%	1.28	≈ 6%

6 Evaluation of the Concept of a Sleeve Type Valve

The abutting faces of the sleeve cannot be realised perfectly sharp. If the cone angle is small, such an edge would not resist high contact and radial pressure forces. The blunt-shaped edges form an annular flow passage area. If the gap height is sufficiently small, Reynolds theory could be employed. For a parallel gap cross section the pressure distribution would be linear, for other shapes of the gap the distribution could be easily derived from an integration of the Reynolds equation. For larger gaps, however, flow and pressure fields are essentially influenced by effects not included in the Reynolds equation, primarily by vortexes. The fluid pressure distribution across the metering edges for larger gaps were studied by a CFD-Analysis with the CFD program Fluent. The details of this axisymmetric analysis are documented in (Falkner, 2009).

The radial distribution of the hydrostatic pressure derived from this CFD analysis, relevant for the pressure forces, is depicted in Fig. 10.

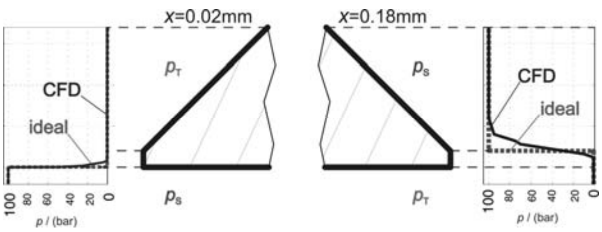

Fig. 10: *Pressure progress across the metering edges*

Assuming the ideal pressure distribution of Fig. 10, the active surface for the pressure force in position P-A (in Fig. 11) is strongly increasing, dependent on the radial extent of the sleeves' front faces.

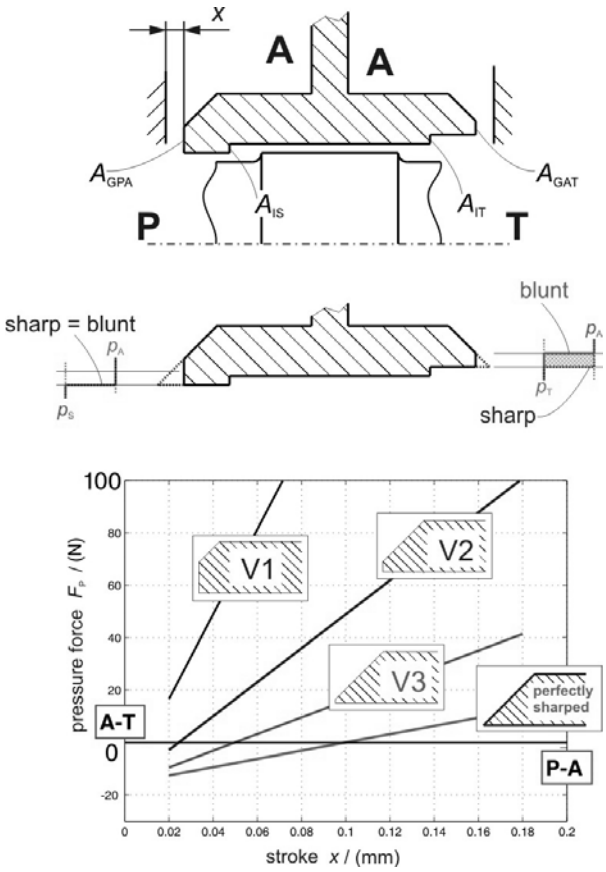

Fig. 11: *pressure forces for the blunt shaped sleeve versions*

Port A pressure is assumed to be dependent on the valve opening, according to Eq. 2. The change of the total hydrostatic force (compared to Eq. 3) based on the blunt shaped sleeve edges can be expressed as the following flow force:

$$F_{\text{Flow}} = \left(p_{\text{A}} - p_{\text{T}}\right) A_{\text{GAT}} \tag{18}$$

Figure 11 illustrates the total pressure force as a function of the sleeve position at a system pressure of 100 bar for the 3 different sleeve versions used for the measurements in section 5. Additionally, the pressure force for perfectly sharp sleeve edges is shown.

The metering edge geometries of version 1 and 2 show a significantly higher total pressure force than originally assumed in the dimensioning of the valve. The solenoid cannot generate such high magnetic forces, needed for closing the valve.

For version 3, however, the force required to overcome the pressure force should be in the range of the solenoids' magnetic force.

Thus, there must be further causes for the insufficient closing-capability of the valve, since the valve failed to switch at that system pressure level.

One possible cause is a pressure dependent clamping of the sleeve. In open position of the valve (P connected to A) the sleeve is compressed. This deformation consumes the clearance between pin and sleeve if system pressure exceeds a certain threshold value.

7 Conclusion

The main reasons of this study were the following prospects:

- a sharp edged sleeve exhibits smaller flow forces than a spool,
- it has small leakage since a seat type valve is formed at the metering edges
- the integration of the magnet into the valves is more compact than in spool valve,
- a bistable behavior can be accomplished by a simple hydrostatic principle.

The analysis of these speculations was done by simple analytical models, construction and testing of a prototype valve, and by a CFD analysis.

The final analysis by a CFD model showed that the flow forces are proportional to the size of the abutting surface at the metering edge of such a sleeve. Though this corresponds with the original speculation, the opposing requirement that this area must not become too small to avoid high contact pressure prohibits reducing flow forces below typical values of a comparable spool valve.

The opposite of the small leakage expectation turned out to be true. The main reason is that it is very challenging to manufacture the contact surfaces of the contact pairs sufficiently accurate. The right conclusion of this finding is not that small leakage is totally impossible but too costly.

The expectation of an easy integration of the magnet in the valve that was motivated by first sketches of the valve similar to Fig. 1 was too optimistic. The design and manufacturing lead to a valve design that consists of too many precise parts and a difficult assembly process which contradicts the requirements on a cheap production.

Nomenclature

A_{GAP}	Abuttal face P - A	[mm^2]
A_{GAT}	Abuttal face A - T	[mm^2]
A_{IS}	Surface of inner shoulder S	[mm^2]
A_{IT}	Surface of inner shoulder T	[mm^2]
A_{OS}	Active outer surface S	[mm^2]
A_{OT}	Active outer surface T	[mm^2]
b_D	Guided length of sleeve	[mm]
b_M	Magnet segment width	[mm]
d_{SI}	Inner sleeve diameter	[mm]
F_{DP}	Fluid damping force (P-side)	[N]
F_{DT}	Fluid damping force (T-side)	[N]
F_{Flow}	Flow force	[N]
F_{MagP}	Magnetic force of P-sided solenoid	[N]
F_{MagT}	Magnetic force of T-sided solenoid	[N]
F_P	Fluid pressure force	[N]
F_R	Fluid friction force	[N]
h_D	Clearance pin-sleeve	[mm]
h_S	General air gap of solenoid - armature	[mm]
h_{SP}	Air gap, P-sided solenoid	[mm]
h_{ST}	Air gap, T-sided solenoid	[mm]
h_{SP0}	Initial air gap	[mm]
I_P	Solenoid current (P-side)	[A]
I_T	Solenoid current (T-side)	[A]
l_M	Magnet segment length	[mm]
m	Sleeve mass	[kg]
N	Number of windings	[1]
P	Gap pressure	[bar]
$\tilde{p}$	Limited gap pressure(P-side)	[bar]
$\tilde{p}$	Limited gap pressure(T-side)	[bar]
p_A	Exit port pressure	[bar]
p_S	System pressure	[bar]
p_T	Tank pressure	[bar]
Q_{Leak}	Leakage flow rate	[1/min]
Q_N	Nominal flow rate	[1/min]
R_{MP}	Magnetic resistance of P-sided solenoid	[A/(Vs)]
R_{MT}	Magnetic resistance of T-sided solenoid	[A/(Vs)]
s	Iron core width	[mm]
T_{OIL}	Oil temperature	[°C]
t_{SW}	Switching time	[ms]
W_{MagP}	Magnetic energy, P-sided solenoid	[J]
W_{MagT}	Magnetic energy, T-sided solenoid	[J]
x	Actual sleeve position	[mm]
$\dot{x}$	Actual sleeve velocity	[mm/s]
x_{max}	Valve stroke	[mm]
α_D	Flow coefficient	[1]
Δd_S	Inner shoulder's gauge	[mm]
Δp	Pressure drop	[bar]
Δp_N	Nominal pressure drop	[bar]
η	Dynamic fluid viscosity	[Pa s]
ρ	Fluid density	[kg/m^3]
Φ_P	Magnetic flux, P-sided solenoid	[Vs]
Φ_T	Magnetic flux, T-sided solenoid	[Vs]

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References

Falkner, J. 2009. *Entwicklung eines schnell schaltenden 3/2-Wege Sitzventils*. Master Thesis, Johannes Kepler University Linz.

Garstenauer, M. 2001. *Theoretical and Experimental Investigations of a Novel Concept for Hydraulic, High-speed, Switching Valves.* PhD Thesis, Johannes Kepler University Linz, Austria.

Garstenauer, M. and **Scheidl, R.** 1999. High-Speed Switching Valves Actuated by Parametrically Excited Structures. *Bath Workshop on Power Transmission and Motion Control (PTMC 99)*, pp. 137-150.

Kallenbach, E. et al. 2003. *Elektromagnete Grundlagen, Berechnung Entwurf und Anwendung*, Teubner Verlag.

Laamanen, A. and **Linjama, M.** (Editors) 2010. *Proceedings of the Third Workshop on Digital Fluid Power*, Tampere, Finland.

Laamanen, A., **Linjama, M.** and **Vilenius, M.** 2007. On the pressure peak minimization in digital hydraulics, *The 10th Scandinavian International Conference on Fluid Power, SICFP'07*, May 21-23, 2007, Tampere, Finland.

Lauttamus, T., **Linjama, M.**, **Nurmia, M.** and **Vilenius, M.** 2006. A Novel Seat Valve with Reduced Axial Forces. *Proc. Symposium on Power Transmission and Motion Control (PTMC 2006)*, Bath University, England.

Linjama, M. and **Laamanen, A.** (Editors) 2008. *The First Workshop on Digital Fluid Power*, Vol. 3, Tampere, Finland.

Linjama, M. and **Scheidl, R.** (Editors) 2010. *Special Issue on Digital Fluid Power, International Journal of Fluid Power*.

Mahrenholz, J. and **Lumkes, J.** 2010. Analytical Coupled Modeling and Model Validation of Hydraulic On/Off Valves. *Journal of Dynamic Systems Measurement and Control*, Vol. 132, Issue 1.

Manhartsgruber, B. 2006. A hydraulic control valve for PWM actuation at 400Hz. *Proc. Symposium on Power Transmission and Motion Control (PTMC 2006)*, Bath University, England.

Merritt, H. 1967. *Hydraulic Control Systems*. John Wiley & Sons.

Ouyang, X. P., **Yang, H. Y.**, **Jiang, H. Y.**, and **Xu, B.**, Simulation of the piezoelectric high-speed on/off valve. *Chinese Science Bulletin*, Vol. 53, No. 17, pp. 2706 - 2711.

Platzer, T., **Vael, G.** and **Achten, P.** 2009. The flexible seat valve. *www.innas.com*, Innas BV, NL.

Ploeckinger. 2008. Fast Development and Rapid Prototyping of a Compact Fast 3/2 Way Switching Valve. *Proc. 20th International Conference on Hydraulics and Pneumatics,* ISBN 978-80-02-02074-5, pp. 137 - 143, Prag, Czech Republic.

Scheidl, R. and **Winkler, B.** (Editors) 2009. *The Second Workshop on Digital Fluid Power*, Linz, Austria.

Sturman, O. E. and **Masey, S.** 1994. *Static and Dynamic Pressure Balance Double Flow Three-Way Control Valve.* U.S. Patent 5,598,871.

Szeri, A. 1998. *Fluid Film Lubrication Theory and Design.* Cambridge University Press.

The Autocar. 1919. *Autocar Handbook* (Ninth ed.). pp. 36 - 38.

Uusitalo, J. P., **Soederlung, L.**, **Kettunen, L.**, **Ahola, V.** and **Linjama, M.** 2009. Novel Bistable Hammer Valve for Digital Hydraulics. *Proc. 2nd Workshop on Digital Fluid Power*, Linz, Austria.

Winkler, B. Development of a Fast Low-cost Switching Valve for Big Flow Rates. *Proc. 3rd FPNI-PhD Symposium on Fluid Power*, Terrassa, Spain.

Winkler, B. 2009. Ploeckinger, A. and Scheidl, R. 2009. A Novel Piloted Fast Switching Multi Poppet Valve. *2nd Workshop on Digital Fluid Power (DFP'09)*, Linz, Austria.

Winkler, B. and **Scheidl, R.**, Development of a Fast Seat Type Switching Valve for Big Flow Rates. *Proc. Tenth Scandinavian International Conference on Fluid Power, SICFP'07*, Tampere, Finland.

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