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STEADY STATE AND DYNAMIC CHARACTERISTICS OF WATER HYDRAULIC PROPORTIONAL CERAMIC SPOOL VALVE

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Abstract

Water hydraulics is both old and new technology area. The first fluid power applications were using water as pressure medium already in 18th-century, but the modern water hydraulics has been rapidly growing just recently in 1980's and 1990's. The main reasons for the comeback are environmental and safety aspects. Water hydraulics offers a significant alternative to establish motion control systems in environments, where possible oil leakage can cause serious problems.

One of the major tasks to solve in modern water hydraulics is to achieve more accurate control systems than today`s technology provides. The present water hydraulic control valves are not yet at the same technology level as that of oil hydraulics. This means that a lot of effort has to be put to develop better and more accurate valve constructions and on the other hand more intelligent control methods has to be developed to achieve reasonable valve characteristics.

This paper concentrates on a study of water hydraulic proportional valves, which are still very new developments in water hydraulics. The steady state and dynamic charateristics of water hydraulic proportional ceramic spool valve is studied both by computer simulation and laboratory tests. Some constructional improvements for the valves are studied and also different control methods are discussed and proposed.

Keywords: proportional valves, water hydraulics, fluid power

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1 Introduction

Proportional valves suitable for running with tap water are not widely available on the market. This paper concentrates on the study of a proportional ceramic spool valve, which is the one of the newest control valves in water hydraulics.

Ball seat valves have been quite commonly used as proportional tap water valves (Koskinen et al 1996). The ball seat principle has the advantage of low leakage and reliable operation. Manufacturing is also relatively easy, because the parts can be designed simply and most new materials can be used. All the parts in contact with pressure medium are made of corrosion resistant materials. The balls for example can be made of stainless steel or industrial ceramics, the latter being the usual type. When considering the steady state characteristics of the valve, the main problem is the actual control area, which located in the very narrow current range. This means that the controllability of the valve is quite poor, although the operation is relatively linear within that current range. However, hysteresis is over 30%, which makes the behaviour of the valve even more non-linear. In addition the basic flow characteristics of a ball seat valve have also non-linear nature.

The ceramic spool valve (Fig. 1) is the newest development in the range of water hydraulic proportional valves (Koskinen et al 1996, Trostman 1996). There are not many spool valves available that can be used with tap water, although spool valves have many advantages compared to other valve types. When we are considering the properties of water, the main problems in spool valves are high leakage (Koskinen et al 1994), high flow forces and risks of cavitation and erosion caused by the high flow velocity. The spool and the housing sleeve are made out of industrial ceramics and the outer housing is made of stainless steel. To achieve high-pressure levels, low leakage rates and long lifetime the manufacturing tolerances have to be very fine. The manufacturing technology of ceramic materials also sets different demands for designs compared to steel valves.

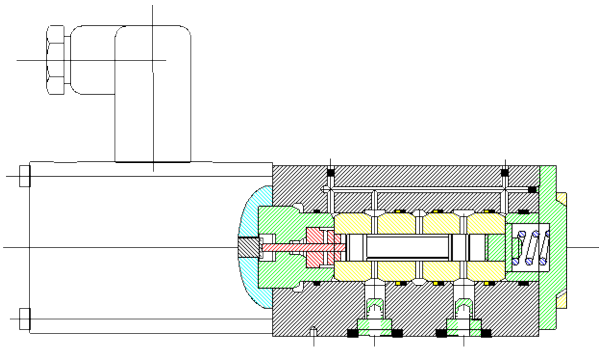

Fig. 1: *Schematic picture of water hydraulic ceramic spool valve*

The characteristics of water hydraulic valves are still slightly behind oil hydraulic valves like we can see from the Fig. 2. That is obvious, because the developmental history of proportional oil hydraulic components is much longer than it is with water hydraulic components. The movement of the spool is highly non-linear. The spool has to move almost a halve of its total movement, before flow from the port P to the port A occurs. After that the actual control range is very narrow and the total opening of the valve is achieved already with 50% of maximum current. When closing the valve, hysteresis is very large and closing happens very fast. When flow occurs through the valve the movement of the spool is not as controllable as when there is no flow from the port P to the port A. That means that flow forces have quite a big influence on spool movement.

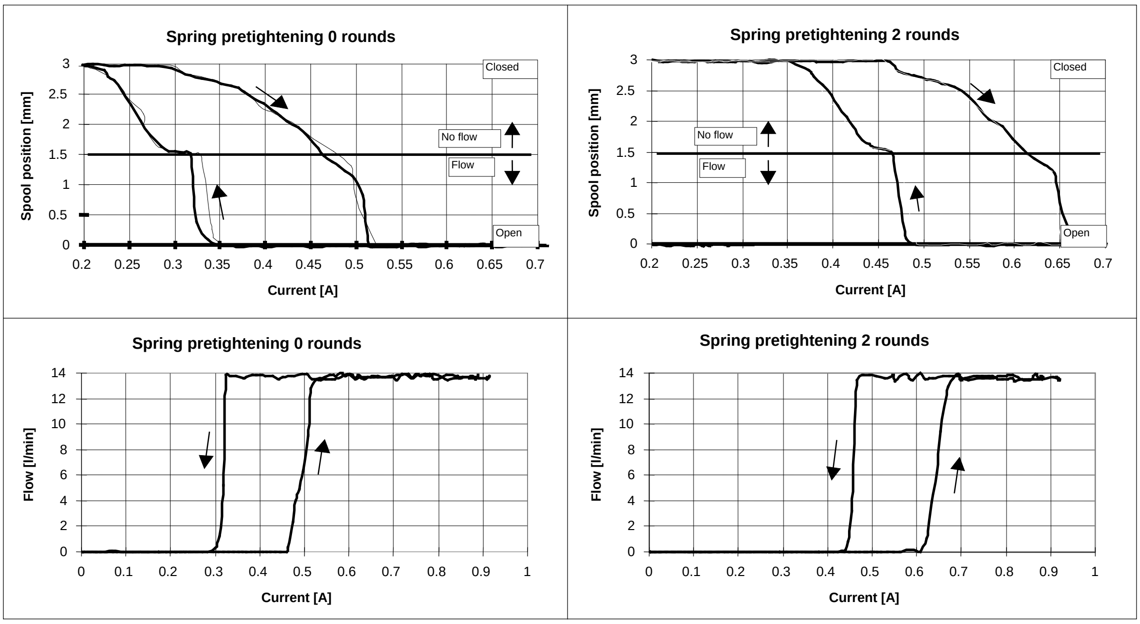

Fig. 2: *Measured spool position as function of command signal on a ceramic spool valve without spool position feedback*

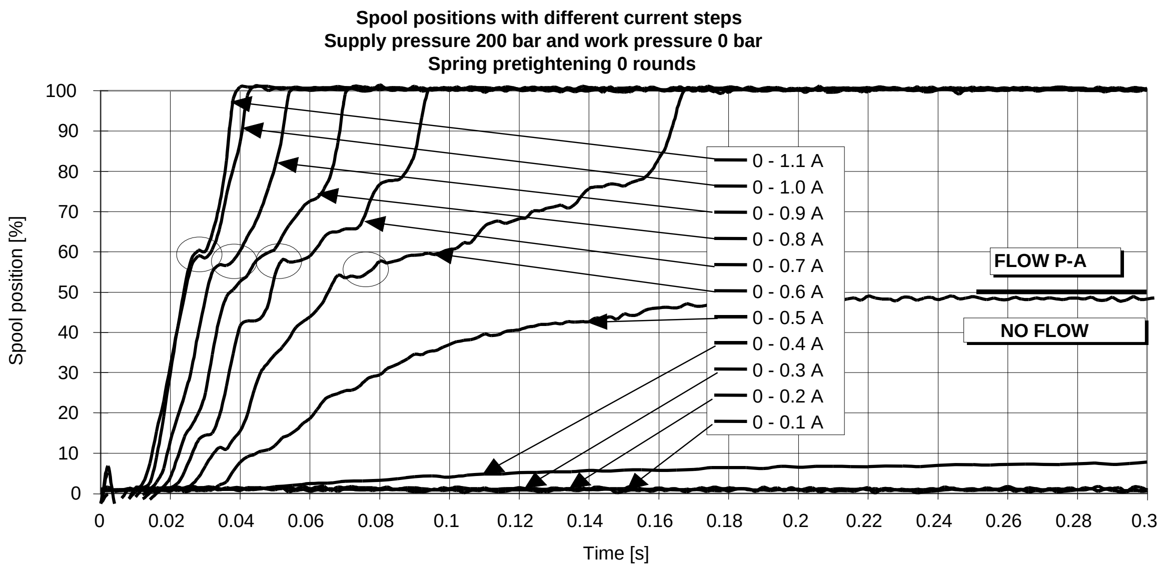

Fig. 3: *Series of measured current step responses with system pressure 200 bar and with no load pressure*

In Figure 3 series of measured current step responses for the ceramic spool valve are presented. It can be seen that very near to the value 0.5 A the spool remains in the middle position. When the P-port opens, the flow force tends to close the valve and bend the curves (circled points in Fig. 3). However, when the spool has moved to about 75-80% of its whole range, the effect of the flow force stops and the spool moves more quickly to the end of its travel. This is caused by the change in flow angle in the spool valve and the effect of the port orifices.

The spool position curve can be made quite linear by using spool position feedback. Also hysteresis can be reduced to quite small value. Spool position feedback can be established with analog or digital-control card and them the control algorithm used naturally effects control characteristics. Without spool position feedback the flow through the valve is very non-linear and hysteresis is relatively large. Therefore when using a ceramic spool valve in a servo system or as a pilot valve, spool position feedback is essential. The dynamic behaviour of the spool valve also depends on the control method used.

Two-stage flow control of large water flows in water hydraulics is traditionally established with a 2/2-way seat valve using separate pilot. Pilot control of the main valve can be established with two 2/2-way ball seat valves or with one 3/2-way ceramic spool valve (Koskinen et al 1996). The seat valve is controlled like an asymmetrical piston in a cylinder. Control pressure is continuously applied to the smaller control area and flow to the larger control area is controlled with pilot valves. Electrical control feedback is is established with position of the main spool. In the ceramic spool valve controlled system the electrical feedback loop is established using the position of the pilot spool too. The number of electrical control loop feedback used depends on the control card capability and the functions of the system. In a ball valve piloted system a second feedback loop can be obtained from the main line pressure, if pressure control is needed. Also feedback from main line flow transducer can be used, if very accurate flow control is required. In these systems the steady state and dynamic characteristics of the transducers have to be considered, when designing the systems.

2 Mathematical Model for the Ceramic Spool Valve

The schematic design of a 3/2-way proportional ceramic spool valve is illustrated in Fig. 3. The pressure port A and the tank port T are connected when the current of the proportional solenoid is zero. The spool is slightly overlapped and pre-setting of the return spring can be adjusted. The spool and the housing sleeve are made of industrial ceramics.

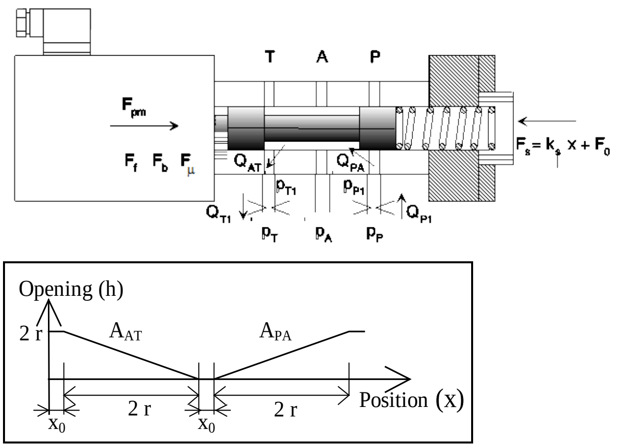

Fig. 4: *Schematic representation of 3/2-way ceramic spool valve and port opening*

When assuming turbulent flow from port P to A, the flow over the P-port control edge is

$$Q_{\mathrm{PA}} = C_{\mathrm{d}} A_{\mathrm{PA}} \sqrt{\frac{2(p_{\mathrm{P1}} - p_{\mathrm{A}})}{\rho}} \tag{1}$$

and when assuming turbulent flow from port A to T, the flow over the T-port control edge is

$$Q_{\mathrm{AT}} = C_{\mathrm{d}} A_{\mathrm{AT}} \sqrt{\frac{2(p_{\mathrm{A}} - p_{\mathrm{T1}})}{\rho}} \tag{2}$$

The flow areas A_{PA} and A_{AT} for circular ports can be calculated from the Eq. (3)

$$A_{\mathrm{PA}} = A_{\mathrm{AT}} = 4\left[r^2 \, a\cos\left(\frac{r-h}{r}\right) - \sqrt{r^2 - (r-h)^2}\,(r-h)\right] \tag{3}$$

where h for area A_{PA} is

$$\begin{aligned} &h = 0, \text{ when } x \le (2x_{\mathrm{o}} + 2r) \\ &h = x - 2x_{\mathrm{o}} - 2r, \text{ when } (2x_{\mathrm{o}} + 2r) < x < (2x_{\mathrm{o}} + 4r) \\ &h = 2r, \text{ when } x \ge (2x_{\mathrm{o}} + 4r) \end{aligned} \tag{4}$$

and for the area A_{AT}

$$\begin{aligned} &h = 2r, \text{ when } x \le x_{\mathrm{o}} \\ &h = 2r - x - x_{\mathrm{o}}, \text{ when } x_{\mathrm{o}} < x < (2r + x_{\mathrm{o}}) \\ &h = 0, \text{ when } x \ge (2r + x_{\mathrm{o}}) \end{aligned} \tag{5}$$

The schematic diagram of the valve port openings is shown in Fig. 4. Turbulent flow through the orifice in port P is

$$Q_{\mathrm{P1}} = C_{\mathrm{d}} A_{\mathrm{P1}} \sqrt{\frac{2(p_{\mathrm{P}} - p_{\mathrm{p1}})}{\rho}} \tag{6}$$

and turbulent flow through the orifice in port T is

$$Q_{\mathrm{T1}} = C_{\mathrm{d}} A_{\mathrm{T1}} \sqrt{\frac{2(p_{\mathrm{T1}} - p_{\mathrm{T}})}{\rho}} \tag{7}$$

The force equation of the spool is

$$m_{\mathrm{red}} \ddot{x} + b_1 \dot{x} + k_{\mathrm{s}} x = F_{\mathrm{pm}} - F_{\mu} - F_{\mathrm{f}} - F_0 \tag{8}$$

The force of the proportional solenoid can be approximately modelled by

$$F_{\mathrm{pm}} = K_{\mathrm{I}} \cdot I \tag{9}$$

The dynamic characteristics of the proportional solenoid can be roughly approximated by the first order transfer function

$$\frac{I(s)}{U(s)} = \frac{K_{\mathrm{u}}}{1+\tau s} \tag{10}$$

The static and the Coulomb friction force are described by the equation

$$F_{\mu} = sign(\dot{x})\left[F_{\mathrm{co}} + \left(F_{\mathrm{st}} - F_{\mathrm{co}}\right)e^{-\beta|\dot{x}|}\right] \tag{11}$$

and the static and transient flow force for path P→A are

$$F_{\mathrm{fPA}} = 2C_{\mathrm{d}}A_{\mathrm{PA}}\left(p_{\mathrm{P}} - p_{\mathrm{A}}\right)\cos(\theta(x)) \tag{12}$$

and for the path A→T

$$F_{\mathrm{fAT}} = 2C_{\mathrm{d}}A_{\mathrm{AT}}\left(p_{\mathrm{A}} - p_{\mathrm{T}}\right)\cos(\theta(x)) \tag{13}$$

$\theta(x)$ is the spool orifice jet angle. It is approximated so that the angle is changing from value 69° (small openings) to 90° (full opening), when the circular port opens. The total flow force is

$$F_{\mathrm{f}} = F_{\mathrm{fPA}} + F_{\mathrm{fAT}} \tag{14}$$

The simulation model of the valve is based on the mathematical Eq. (1) … (14) presented above. The model has been implemented into the Simulink - simulation program package. The mathematical model of the ceramic spool valve has been verified by measuring the valve operation in laboratory test conditions (Koskinen et al 1996). The valve used in the mentioned measurements is a Hauhinco DN 3 Ceramic Spool Valve, Normally Closed. According to the verification, the developed simulation model can be used for simulation and design of water hydraulic systems, although the correspondence is not perfect. Because the valve is highly non-linear, modelling of all non-linearities is not possible nor is it generally necessary.

3 Steady State Characteristics of Ceramic Spool Valve

In this part the effect of three different parameters on the steady state characteristics of the valve are studied by computer simulation and compared with measurement results. The size of port orifice, friction characteristics and spring characteristics were chosen for the investigation. These parameters were chosen, because the first two are emphasised in water hydraulic components and the effect of spring characteristics is also easy to verify in practice with available components. In all simulations the supply pressure 100 bar is used and no load pressure is present.

3.1 Effect of Spring Characteristics on Valve Steady State Characteristics

The spring in the 3/2-way spool valve obviously has a significant effect on the valve behaviour. During opening the spring works against the force produced by the proportional solenoid and when closing the whole movement is caused by the spring. Increasing spring stiffness increases the damping and also therefore it increases the stability margin of the valve, but also effects the dynamic characteristics. The controllability of the valve can also be affected by using different springs and pre-setting vaues of the springs. In the following the spring constant has been varied and its effect on steady state and the dynamic behaviour of the valve has been studied based on simulation results. In these simulations the pre-setting displacement of the spring has been kept constant and thus the pre-setting force has varied. The original spring constant in the commercial valve was 4300 N/m.

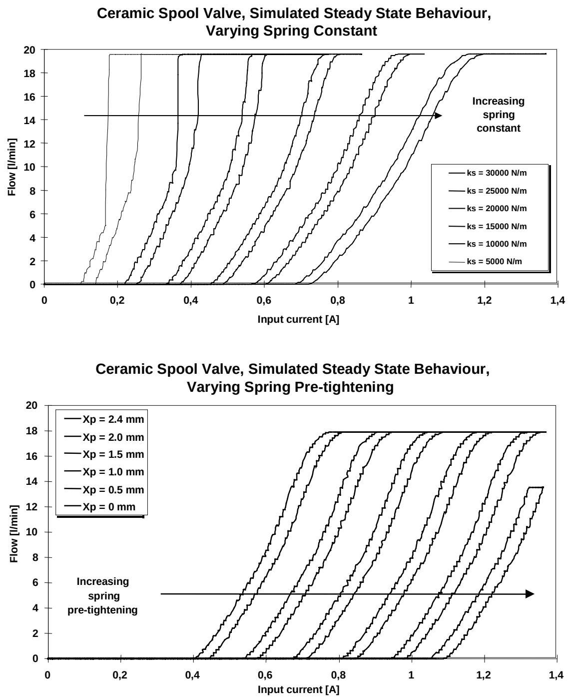

Fig. 5: *Flow as a function of input current with varying spring constant and variying spring pre-setting*

In Fig. 5 the simulated steady state behaviour of the valve is shown, when the spring constant value and spring pre-setting value are varied. With a small spring constant the controllability of the valve is poor, because the valve opens very fast to the fully open position. This is caused by the nature of the flow force, which is first at its maximum value with small openings, but it decreases when the opening increases. Because the flow force works against the proportional solenoid, a stiffer spring will decrease the effect of the flow force. It is clearly seen from Fig. 5 that the controllability of the valve improves with a stronger spring. With a spring constant of at least 20000 N/m, the steady state behaviour of the valve is acceptable. With a 30000 N/m spring, the valve utilises the whole available current range from 0 A to 1.2 A. The effect of the flow force is also decreasing when using a stronger spring. Also hysteresis is smaller because the spring has more force to control the spool in valve closing.

The pre-setting of the spring also affects the characteristics of the valve. The right pre-setting depends on the spring constant and the valve behaviour required by the user. In Fig. 5 the effect of different pre-settings on the steady state and dynamic behaviour of the valve are studied using 23700 N/m as the spring constant value. It can be seen that the increasing pre-setting of the spring is only transferring the curves to the larger current range and that the shape of the curve remains almost the same. With the biggest pre-setting of the spring the spool does not reach its end position, because the current rises above the limit of the used solenoid.

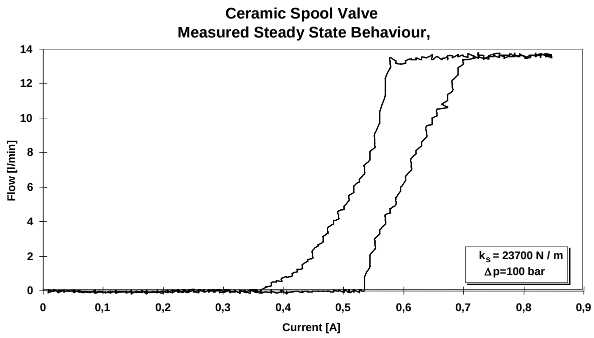

Fig. 6: *Flow as function of input current with new spring*

The effect of using a stiffer spring has been also verified in practice by measuring the behaviour of the valve with a new stiffer spring. The original value for the valve spring constant was 4300 N/m. In the following the new spring with a spring coefficient of 23700 N/m has been installed into the existing valve and the behaviour of the valve has been measured. In Fig. 6 the measured steady state behaviour of the valve with the new spring is presented. With supply pressure the effect of flow force is seen, but with the new spring the effect is not so big as with the original spring. The actual flow control range with the new spring is shown in Fig. 6.

3.2 Effect of Port Orifices on Valve Steady State Characteristics

The purpose of the port orifices in a spool valve is to decrease the pressure drop acting over the spool control edge. This decreases the risks of erosion and cavitation in spool and sleeve. To use a geometry of the valve body that reduces the pressure in two or more steps is a generally known and used method in water hydraulics. This is clear, because water systems normally have larger flow velocities due to very low viscosity, causing erosion and cavitation more easily than in oil systems. Port orifices also affect the steady state and the dynamic characteristics of the valve, which have to be considered in the design of a valve. Port orifices also enables to design the valve for two different flow areas, when considering the use of the valve with or without the orifices. In that case the change of characteristics is useful to know to avoid malfunctions.

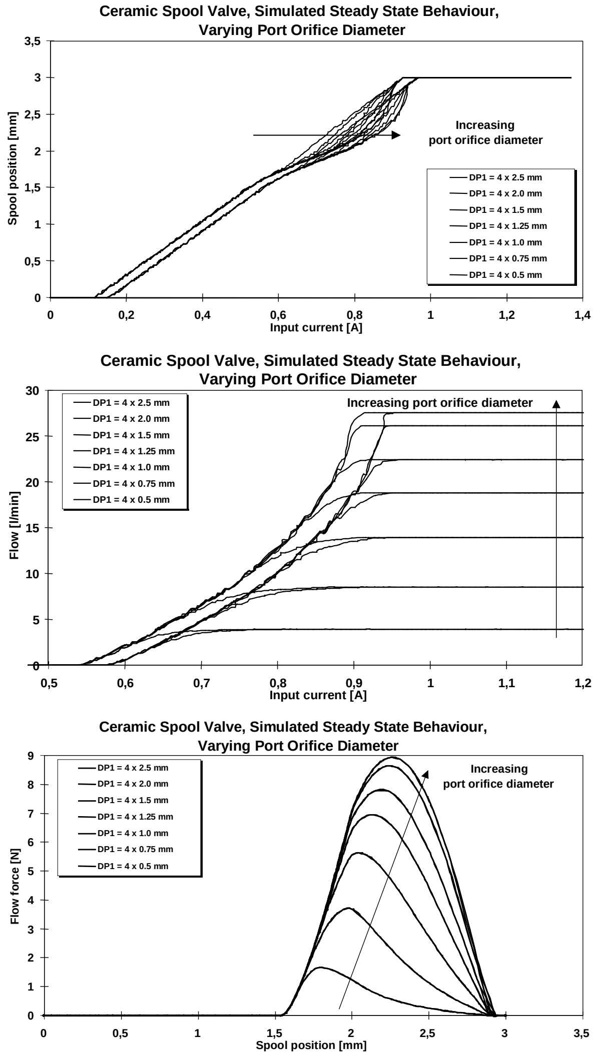

Fig. 7: *Spool position, flow and flow force as a function of input current with varying port orifice diameter*

In this context the port orifice consists of a plug with four same size orifice holes. Therefore the diameter of the P-port orifice is represented as D_{P1} = 4 x D_o in Fig. 7. The plugs are connected with threads into the port bores of the valve. This study concentrates on the effect of the port orifice in P-port. The original size of the port orifice of existing valve is D_{P1} = 4 x 1.4 mm.

In Fig. 7 the simulated steady state behaviour of the valve is presented, where the size of the P-port orifice has been varied. The supply pressure is 100 bar and no load pressure is present. We can see that increasing the port orifice bends the spool position curve. This is caused by the flow force, which is increasing quite strongly along with the orifice diameter. On the other hand, the maximum flow is quite small with under 1 mm diameters, so from the point of view of steady state characteristics, the right values for the port orifice diameter are in range 1 mm … 2 mm.

3.3 Effect of Friction Characteristics on Valve Steady State Characteristics

Friction characteristics are closely related to the materials used and fluid characteristics. The manufacturing tolerances and impurities of the fluid also affect the static and the Coulomb friction. Viscous friction depends on the speed of the sliding parts. Friction behaviour in a spool valve is a very non-linear phenomena, is difficult to describe completely with a mathematical model and also difficult to measure in practice. However, simulation can give guidelines of the effect of different friction behaviour on valve static and dynamic characteristics.

Static and Coulomb friction vary substantially according to the materials used and the lubricants. In the water hydraulic ceramic spool valve the spool and sleeve are made of industrial ceramics, with relatively good gliding properties. On the other hand adhesion forces are possible if the sliding surfaces are very smooth. Also water has poorer characteristics as a lubricant than oil, and the clearances between the spool and the sleeve are smaller for water hydraulics. These factors can cause sticking and very variable friction forces, which are depending on the operation conditions and the state of wear of the valve. In the following the static and the Coulomb friction have been varied and steady state and dynamic behaviour of the valve studied. The ratio of the static and the Coulomb friction has been assumed to be F_s / F_c = 2, justified considering the materials and lubricant used.

In Fig. 8 the effect of static and Coulomb friction on the steady state behaviour of the valve is shown. It is clearly seen that the increase of the friction also increases the hysteresis and causes a step-shaped profile on the flow curves.

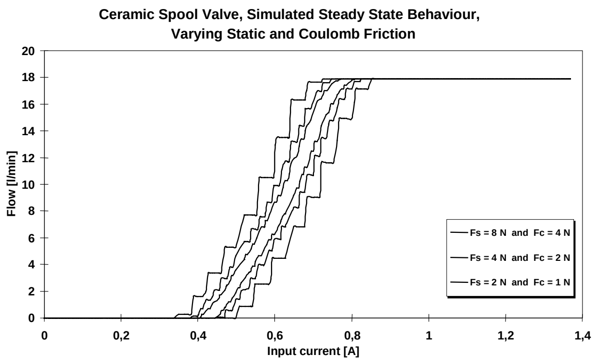

Fig. 8: *Flow as a function of input current with varying friction characteristics*

One method that could decrease the the effect of the static friction is to use a dither-signal. The frequency of the dither-signal is normally selected to be a factor 4 … 5 of the time constant of the valve and then the amplitude is is adjusted to the right level to obtain a satisfactory performance. In Fig. 9 the measured steady state behaviour is presented, where a dither-signal is used. Comparing the Fig. 9 to Fig. 6 it can be noticed, that the use of the dither improves the steady state behaviour by decreasing the hysteresis.

4 Dynamic Charateristics of Ceramic Spool Valve

4.1 Effect of Spring Characteristics on Valve Dynamic Characteristics

In Fig. 10 the simulated dynamic behaviour of the valve is shown, where value of the spring constant was varied. The input voltage was adapted in all curves so that the valve spool moves to the same position at every step input. That means that the current consumption of the solenoid is different at every step. The opening step is given from 0 to 2.25 mm. At the end position P-port is open and flow occurs from P-port to A-port. The closing step is given backwards from 2.25 mm to 0 mm

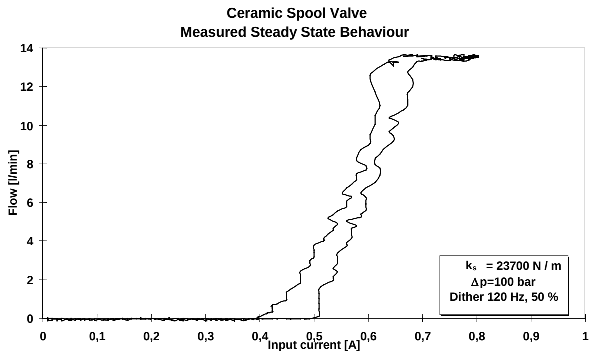

Fig. 9: *Flow as a function of input current using dither-signal*

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Fig. 10: *Spool position responses in valve opening and closing with varying spring constant*

Figure 10 shows that in valve opening the response is faster with stiffer springs even though the spring force is an opposing force to the solenoid force. This is obvious, because the current taken by the solenoid is higher and so the force of the solenoid is also higher. However, the effect of the spring constant in valve opening is not so significant as in the closing. The closing of the valve is the result only of the spring force. Therefore the closing speed is increasing considerably with stiffer springs. However, the speed increase is larger between spring constant values 5000 N/m and 15000 N/m than between values 20000 N/m and 30000 N/m. This means, that there is no use to increase the spring constant much over 20000 N/m. Therefore the spring constant for the new spring is selected to be 23700 N/m, because according to above results of the study, it gives reasonable steady state and dynamic characteristics, and it is also possible to install such a spring into the present valve. Thus the operation of the valve with a new spring can be verified with measurements.

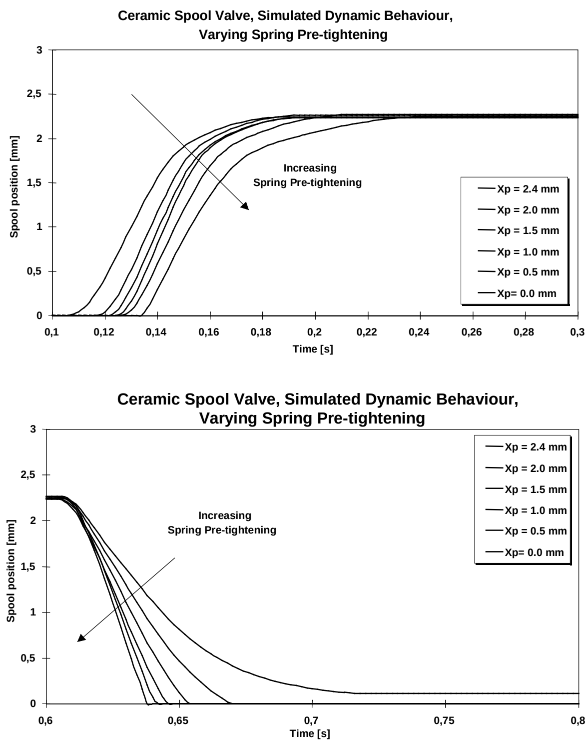

Fig. 11: *Spool position responses in valve opening and closing with varying spring pre-setting*

In Fig. 11 the simulated dynamic behaviour of the valve is shown where the spring pre-setting value was varied. The input voltage was adapted in all curves so that the valve spool moves to the same position at every step. This means that the current taken by the solenoid is different at every step input. The opening step input is given from 0 to 2.25 mm, where the P-port is open and flow occurs from the P-port to the A-port. The closing step input is given from 2.25 mm to 0 mm, where the P-port is closed and flow occurs from the A-port to the T-port. Figure 11 shows that a larger pre-setting value gives a slightly slower opening and considerably faster closing. To obtain the best operation both in opening and closing a compromise has to be made and the pre-setting should be selected from range 0.5 mm … 1.5 mm.

In Fig. 12 shows a measured response of the valve with a new spring, of which pre-setting is 0.5 mm. In this example the opening and closing times are 60 ms and 90 ms, repectively.

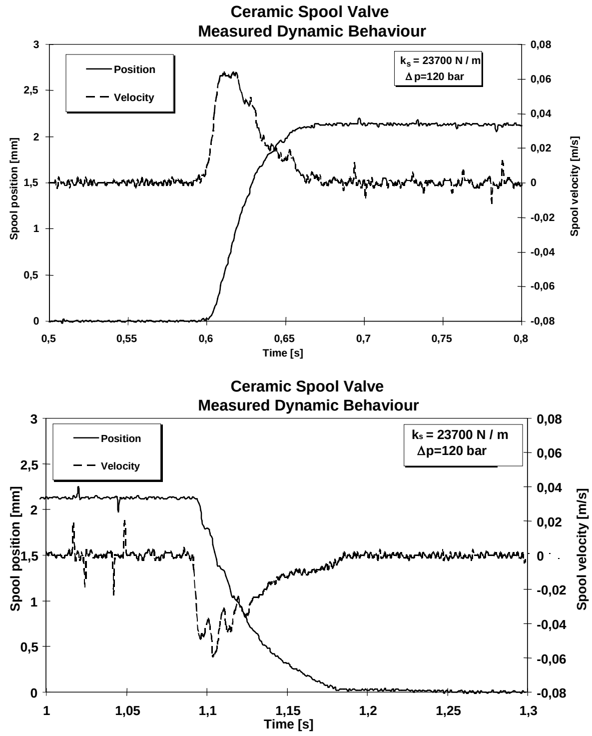

Fig. 12: *Measured spool position and velocity responses in valve opening and closing with new spring*

4.2 Effect of Port Orifices on Valve Dynamic Characteristics

In Fig. 13 the simulated dynamic behaviour of the valve is presented, where the size of the P-port orifice has been varied. The same voltage step input has been given to all simulations. Pressure p_{P1} represents the pressure decrease at the port orifice when the supply pressure is 100 bar. We can see that with the diameter value 0.5 mm almost the whole pressure drop occurs in the port orifice and with value 2.5 mm in spool control edge. That means that the port orifice D_{P1} = 4 x 2.5 mm gives almost the same flow area as the port boring itself. From spool position response we can see, that the port orifice does not affect much the response time. Instead the end position of the spool changes a little, caused by the bigger flow force level. Dynamically the flow force behaves slightly differently with small orifices. The force first rises above a steady state value and after that settles down. With bigger orifices the 'overshoot' is not so remarkable. This means that with small orifices the flow force has more effect on step response, which in high frequency work cycles can lead to unstable operation.

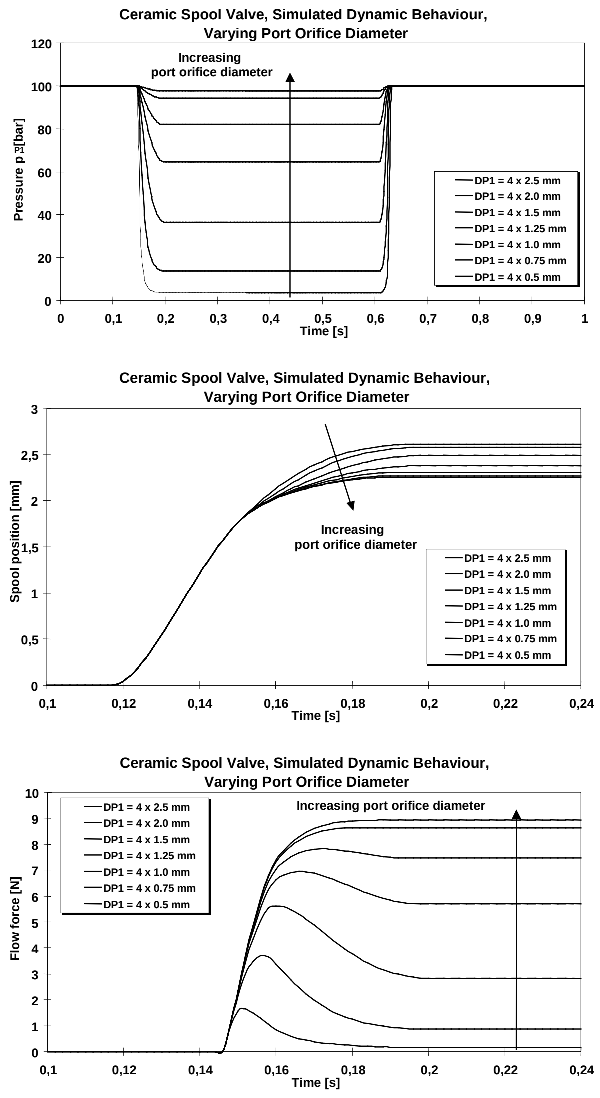

Fig. 13: *Pressure p_{P1} , spool position and flow force with varying port orifice diameter*

4.3 Effect of Friction Characteristics on Valve Dynamic Characteristics

In Fig. 14 the effect of the static and the Coulomb friction on the dynamic behaviour of the valve is shown. In all the simulations the input was a voltage step to the solenoid which causes the current step. Friction increase clearly, delays the spool position responses and therefore affects the end position during valve opening. After a small delay the velocity of the spool increases very rapidly, when the static friction transfers to Coulomb friction. This in practice easily causes instability during valve operation. The effect of friction is emphasised in the valve closing, because the whole movement is caused only by the spring.

Viscous friction also affects the valve spool dynamics behaviour. Because viscous friction by its nature is a velocity dependent variable, it is natural in this context to focus on the dynamic characteristics of the valve. In Fig. 15 the simulated dynamic behaviour of the valve is shown, where the viscous friction value is varied. Figure 15 clearly shows that viscous friction affects valve damping. The viscous friction coefficient

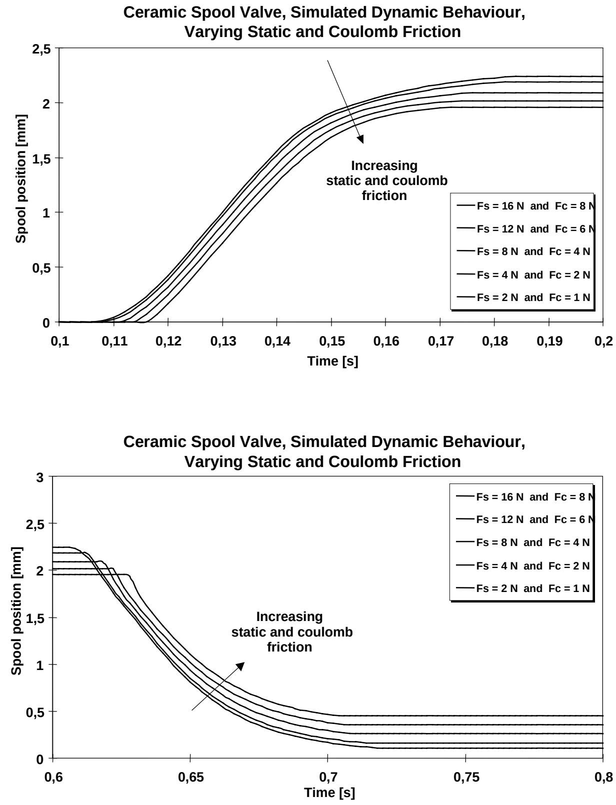

Fig. 14*: Spool position responses in valve opening and closing with varying friction characteristics*

is varied in the range 20 Ns/m to 1000 Ns/m. Generally, the viscous friction value depends on the used fluid. Normally the value 4000 Ns/m is used in oil hydraulic servo system design for a hydraulic cylinder.

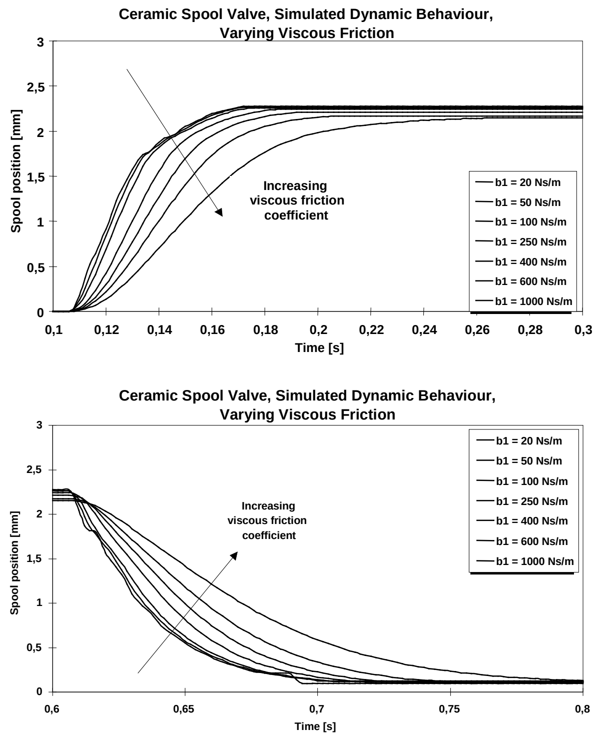

Fig. 15: *Spool position responses in valve opening and closing with varying viscous friction*

However, the viscosity of water is essentially lower than the viscosity of oil and therefore it is natural to assume that the viscosity coefficient in water hydraulics is also lower. We can conclude that a good estimate value in this case is in the range between 100 Ns/m and 500 Ns/m. Viscous friction varies also very much according to the actual application. In order to obtain reliable reference values for viscous friction in different water hydraulic components and systems further research work is needed in the future.

5 Operation of the Valve with Spool Position Feedback

When the spool position feedback can be utilised, the the valve characteristics can be improved with the aid of control technology. In our case a commercial digital control card was used for controlling the valve. The behaviour of the controller-valve combination was measured under laboratory conditions. The used control card enables the use of advanced control methods. In this context the study was concentrated on the dynamic behaviour of the valve, but also the steady state error was considered. Different control algorithms were studied and the effect of some special characteristics of the control card are also utilised. The used supply pressure in the tests is about 200 bar and with no load pressure. The step input was in the range 50% to 75% so that flow occurs through the valve in the end of the step.

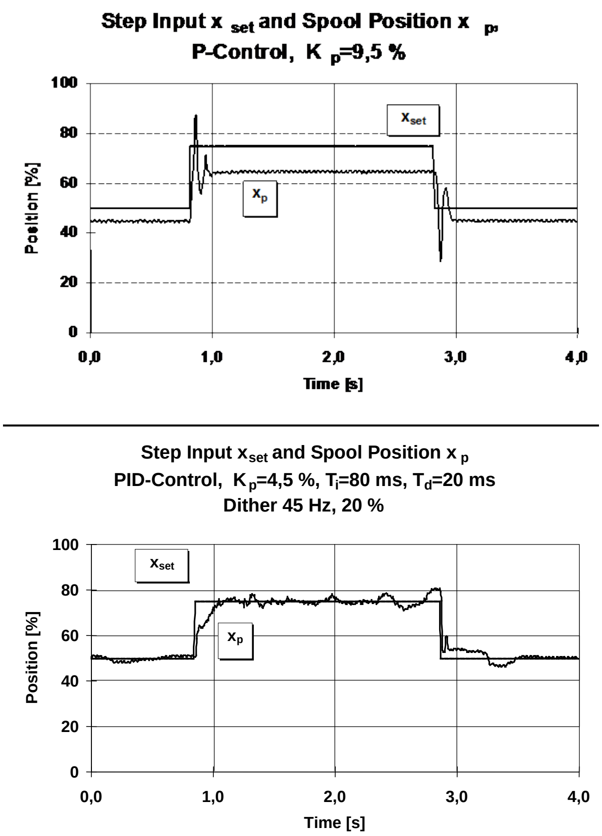

Fig. 16: *Operation of the valve with normal P-control, and PID-control and dither*

In Fig. 16 the step response with P-control when K_p = 9.5% = 0.89 is shown. The opening and closing times are both around 40 ms. It is shown that the steady state error can be reduced by increasing the gain, but the valve begins to oscillate with low damping in the opening and the closing. In fact the valve soon becomes unstable if the gain is increased further. Using the PID-controller, smaller gain K_p must be used. This is caused by the D-term, which makes the response faster but also increases the risk of instability of the system. The I-term is used for correction of steady state error. Figure 16 also shows the response with PID-controller and dither signal. It can be seen that the steady state error is smaller than with the P-controller. The opening time is about 250 ms which is much longer than with the P-controller. The dither signal causes some uncertainty to the response and increases the instability, because the D-term used and the dither frequency is not optimal considering the time constant of the valve.
In Fig. 17 the responses with the PD-controller with controller output limiters are shown. With the parameters O*Hilim* and O*Lolim* the controller output can be

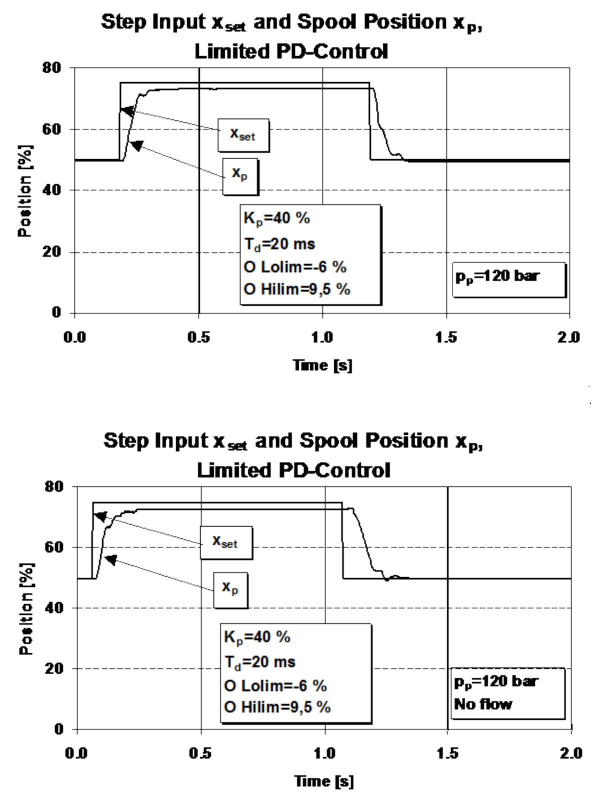

Fig. 17: *Operation of the valve with limited PD-controller with and without flow through the valve*

restricted to a certain maximum or minimum value despite the error signal. For a ceramic spool valve this is a very efficient way to accomplish fast control and obtain a reasonable stability margin. Using the controller's output limits, larger gain values can be used without making the system unstable. The values for the limits to be used correspond to the controller output proportional to the open loop critical gain of the system. Figure 17 the upper curves the pressure difference over the valve is 120 bar and flow occurs, but for the lower curvers no flow occurs through the valve. It is clearly seen that the PD-controller with controller output limiters is not so sensitive to flow forces and other valve non-linearities. The steady state error is slightly greater with PD-control, because the correcting I-term is not present. However the steady state error is reasonable and appropriate cylinder control applications for example, where good repeatability and fast operation are important properties for a valve.

6 Conclusions

Spool position feedback can be used in the valve by applying a position transducer and a control card. This rises the price of the valve considerably and therefore it is justifiable to try to improve the basic operation of the valve without using spool position feedback and control. Spring characteristics, size of the port orifice and friction characteristics were selected as variable parameters during the computer simulations. Increasing the spring constant was found to be a very efficient and simple way to improve the characteristics of the present valve design.

The port orifice was found to be very efficient in defining the flow velocity and flow force in the valve. The use of the port orifice makes the stepped pressure decrease over the valve possible. Thus, flow velocity can be kept to a reasonable level. When low flow velocity can be contained, risk of cavitation and erosion effects can also be minimised. The static and the Coulomb friction were varied together and the ratio $F_s / F_c = 2$ was used. It can be noticed that quite small increase in the static and the Coulomb friction shows clearly in steady state curves and therefore the value 4 N for the static friction is near the maximum value, which can be allowed. A main conclusion is, that to obtain good steady state and dynamic characteristics with this type a ceramic spool valve, the static friction must be around 2 N. The recommended range for the viscous friction coefficient in this case is in the range 100 Ns/m – 500 Ns/m.

Based on the study of the present valve with spool position feedback and control, it can be concluded that the controlling of the present valve design needs advanced control strategies to obtain good dynamic characteristics over the whole valve operation area. This is caused by the highly non-linear operation of the valve. The main non-linearities affecting valve operation are flow forces and friction forces. The use of limited PD-control was found to give the fastest response with reasonable steady state error and a good stability margin for the valve. In addition the operation of the valve was similar regardless of the pressure conditions.

Nomenclature

A_A	A-port orifice area	[m^2]
A_{AT}, A_{PA}	flow areas in ceramic spool valve	[m^2]
A_{P1},A_{T1}	port orifice areas in ceramic spool valve	[m^2]
A_P	P-port orifice area	[m^2]
A_{P1}, A_{P2}	other effective pressure areas in P-port	[m^2]
A_v	flow area of valve	[m^2]
B	bulk modulus	[Pa]
C_d	discharge coefficient	dimensionless
D_A	diameter of port A	[m]
DP	pressure difference	[Pa]
D_0	port orifice hole diameter	[m]
D_{P1}	diameter of port orifice flow area	[m]
F_{ex}	external force	[N]
F_p	pressure force	[N]
F_{pm}	force of proportional solenoid	[N]
F_f	flow force	[N]
F_{fPA}	flow force for path P-A	[N]
F_{fAT}	flow force for path A-T	[N]
F_{co}	Coulomb friction force	[N]
F_0	spring pre-setting force	[N]
F_{02}	spring pre-setting force	[N]
F_{st}	static friction	[N]
F_μ	combined friction force	[N]
I	current	[A]
K_I	current coefficient	[N/A]
K_p	gain	[%]
K_u	voltage-current coefficient	[A/V]
$L_{1,2}$	length	[m]
P	circumference	[m]
Q	flow	[m^3/s], [l/min]
$Q_{A,P}$	A-port/P-port orifice flow	[m^3/s], [l/min]
Q_v	flow through the valve	[m^3/s], [l/min]
T_I	integration time	[s]
T_d	derivation time	[s]
U	voltage	[V]
V	volume	[m^3]
b_1	viscous friction coefficient	[kg/s]
d_h	hydraulic diameter	[m]
h	actual opening of circular port	[m]
k_s	spring constant	[N/m]
$k_{s1,2}$	spring constant	[N/m]
m	mass	[kg]
m_{red}	reduced mass	[kg]
p	pressure	[Pa], [bar]
p_s	supply pressure	[Pa], [bar]
$p_{A,P}$	pressure in A-port/P-port	[Pa], [bar]
$p_{A1,P1}$	pressure between A-port/P-port orifice and spool	[Pa], [bar]
r	radius	[m]
x	position	[m]
x_0	overlap position	[m]
x_{op}	circular port opening	[m]
x_{set}	reference value	[%]
x_p	piston/spool position	[%]
Δp	pressure difference	[Pa]

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References

Koskinen, K. T., Vilenius, M., Hollingworth, B. 1996. The Characteristics of Two-Stage Water Hydraulic Proportional Valves. *12. Aachener Fluidtechnisches Kolloquium,* Aachen Germany, 12.-13.3. 1996. Aachen. pp. 115-132.

Koskinen, K. T., Uusi-Heikkilä, J. K. & Vilenius, M. J. 1996. Simulation and Control of Proportional Water Hydraulic Ceramic Spool Valve. *9th Bath International Fluid Power Workshop,* Bath, England, 9-11.9. 1996, Preprints. Bath.

Koskinen, K. T. 1996. *Proposals for Improving the Characteristics of Water Hydraulic Proportional Valves Using Simulation and Measurement.* Dissertation. Tampere University of Technology, Publications, No 189.

Koskinen, K. T., Uusi-Heikkilä, J. K. & Vilenius, M. J. 1996. The Characteristics of Ceramic Spool Valve Piloted Water Hydraulic Proportional Valve. In: *Nair, S. & Chandran, R. (ed). Fluid power systems and technology 1996, Collected Papers. The ASME international mechanical engineering congress and exposition,* November 17-22, 1996 Atlanta, Georgia. 3. New York, New York, USA. pp. 71-76.

Koskinen, K. T., Uusi-Heikkilä, J. & Vilenius, M. J. 1997. Improving the Characteristics of Water Hydraulic Proportional Valves. *The Fifth Scandinavian International Conference on Fluid Power, SICFP 97,* Linköping, Sweden, 28-30 May 1997. pp. 277-290.

Koskinen, K. T. & Vilenius, M. J. 1994. Water as a pressure medium in fluid power systems. *IFAC Workshop on Trends in Hydraulic and Pneumatic Components and Systems,* Chicago II, 8-9 November 1994. 13 s.

Trostman E. 1996. *Water hydraulics control technology*. Marcel Dekker, New York.

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