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Pilot operated miniature valve with fast response and high flow capacity

Tapio Lantela*, Jyrki Kajaste, Jari Kostamo and Matti Pietola

Department of Engineering Design and Production, Aalto University School of Engineering, Sähkömiehentie 4, 02150 Espoo, Finland

This paper introduces a pilot operated miniature digital hydraulic valve with a high flow capacity, in comparison to the size of the valve, and a fast response. The valve is designed to be used as a part of a digital valve system, which consists of a large number of similar valves. This paper presents the structure of the valve as well as the response time, flow capacity and leakage measurements of the prototype. The presented valve has a volume of approximately 4 cm^3 and a flow capacity of 9 l/min with a 3.5 MPa pressure difference. Its response time is approximately 1 ms and the maximum operating pressure exceeds 30 MPa.

Keywords: digital hydraulics; fast acting

1. Introduction

A digital flow control unit consists of a number of on/off valves connected in parallel and it is the equivalent of one control edge of a proportional valve. Since the valves, that form the digital flow control unit (DFCU), have only two discrete states, open or closed, only a limited number of flow rates through the whole DFCU can be realised with a certain pressure differential. The number of different states for the DFCU depends on the number of valves and their orifice sizes. A DFCU should have approximately 200 different states in order to achieve control performance of a hydraulic system similar to that attainable with a good servo valve. This can be accomplished by using, for example, eight binary coded valves, i.e. valves with relative orifice sizes of 1:2:4:8:16:32:64:128. Alternatively, it is possible to compose the DFCU of 200 valves with identical orifice sizes. To form a four-way valve system, four DFCUs are required. Usually at least five valves per DCFU are used and thus there are at least twenty valves in a four-way digital valve system. (Linjama, 2003).

Even though replacing a spool type proportional valve with several on/off valves increases the number of parts in the component, it also brings about several benefits. First, all of the control edges in a digital valve system can be controlled independently, which improves the efficiency and the control performance of the hydraulic system. Second, a DFCU is inherently fault-tolerant since it consists of valves connected in parallel. The valve system can still operate with reduced or even unchanged performance if some of the valves fail. Third, the on/off valves have a simple structure and therefore they are more robust than proportional spool valves. Digital valve systems can usually tolerate higher temperatures and more contaminated hydraulic fluid than traditional valves. (Linjama, 2003) Finally, it is possible to implement several different functions with the same valve system. Depending on the situation and the program code controlling the valve system, the system can operate as a directional valve, a pressure valve or a flow valve. This also leads to less expensive valves due to mass production, since the different functions no more require specialised hardware.

Currently, most digital valve systems are built by mounting commercial on/off cartridge valves on a manifold. This results in a large and heavy valve system since the valves are not designed for compactness. Binary coded orifice plates are used to achieve the best possible resolution with the smallest number of valves. When using binary coding, the switching of the valves with the large orifices may cause large pressure peaks in the system. A fault in one of these valves also affects the performance of the valve system significantly. It is possible to increase the fault tolerance of the valve system and to reduce the pressure peaks by replacing the large-orifice valves with several smaller-orifice valves. However, with the currently used commercial on/off valves, increasing the number of valves often results in a too large valve system. The size of the valve system is important especially when the goal is to retrofit an existing hydraulic system with digital valve systems. In this case, the digital valve system has to be of the same standard size as the old valves; in practice it should be mountable to a CETOP subplate. Therefore, there is a need to develop smaller on/off valves that can be easily integrated into a compact digital valve system. In addition, the commercial valves usually have a slow response, which should be improved in order to apply digital valve systems to applications such as pressure control or vibration damping.

*Corresponding author. Email: tapio.lantela@aalto.fi

In recent years, some on/off valves for digital valve systems have been developed in research institutes. Uusitalo et al. developed a bistable solenoid actuated poppet valve with a 2 ms response time and a 3.3 l/min @ Δp = 1 MPa flow capacity (Uusitalo, 2010a). They also built a 16-valve DFCU with these valves (Uusitalo, 2010b). However, the valve was difficult to manufacture and thus the concept was not developed further. Subsequently, Karvonen et al. developed a small and simple direct operated solenoid actuated poppet valve with a 1.5 ms response time and a 0.3 l/min @ Δp = 1 MPa flow capacity (Karvonen, 2010, 2011). Its flow capacity, however, is very limited because the valve is directly controlled by a small solenoid actuator, which is not powerful enough to control a large pressure and a large flow rate simultaneously.

Previously, Lantela et al. developed a pilot operated miniature valve with a response time of approximately 2 ms and a flow capacity of 1.8 l/min @ Δp = 1 MPa. This valve contains a poppet type solenoid actuated 3/2 pilot valve, which requires roughly constant supply pressure. Constant pressure, however, is not available in many cases, for example in load-sensing systems. The valve's flow rate had also room for improvement and therefore the currently presented valve has a different structure. (Lantela, 2011).

2. Design

Figure 1 illustrates the structure of the designed valve. The pilot valve is a solenoid actuated 3/2 spool valve. The supply pressure for the pilot stage is taken from the high pressure side of the main flow channel and the pilot valve has a dedicated tank channel. The pilot valve can also operate with the tank line connected to a higher pressure, for example, the downstream side of the main flow channel. However, since the coil is connected to the tank channel of the pilot valve, a higher pressure in the tank channel would make it more difficult to seal the wire outlets of the coil and could lead to external leaking. When the pilot valve solenoid is energised, the spool moves upwards and connects the pilot channel to the tank channel. When it is not energised, the return spring pushes the spool down and connects the pilot channel to high pressure.

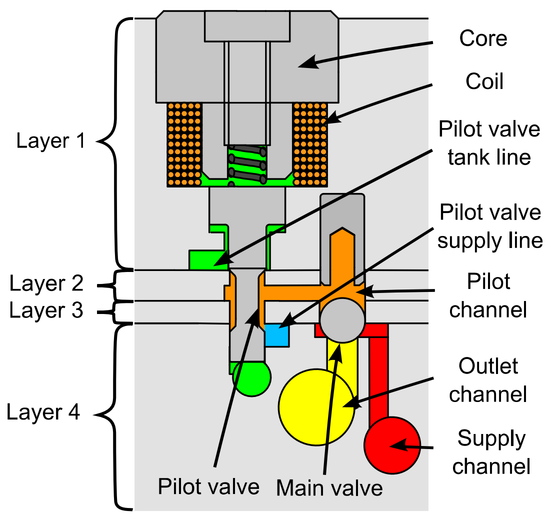

Figure 1. Cross section of the designed valve when the solenoid is not energised i.e. the main valve is in closed state. Figure not in scale.

The pilot valve was selected to be spool type because a spool can be pressure compensated and thus the force required from the actuator can be reduced. A spool type valve also provides a relatively large flow path with a small movement of the spool. The downside with a spool valve is that it always causes some leakage. To reduce the leakage, the upper end of the spool has a seat type seal, which closes the flow path between the pilot channel and the tank channel. The solenoid used as the actuator of the pilot valve has a similar plunger type design as in the previously designed pilot controlled valve (Lantela, 2011).

The structure of the main valve is similar to the fast switching multi poppet valve presented by Winkler and Plöckinger (Winkler, 2010). The sealing element of the main valve is a bearing ball or, after later explained modifications, a poppet ground from a bearing needle. Bearing roller elements are inexpensive and they have excellent dimensional tolerance, surface finish and durability. The bearing ball moves vertically in a bore, which is connected from the top to the pilot channel. When the pilot supply pressure is connected to the pilot channel, there is no pressure difference between the pilot channel and the main flow channel. However, the pressure difference between the pilot pressure and the outlet channel pressure at the main orifice pulls the ball downwards to seal the orifice. When the pilot valve connects the tank pressure to the pilot channel, the pressure difference between the supply pressure and the pressure in the pilot channel pushes the ball upwards and opens the main valve.

The direction of the fluid flow is normally from the supply channel to the outlet channel, i.e. downwards through the orifice in Figure 1. Therefore, when the valve is closed, there is no pressure difference between the supply channel and the pilot channel and consequently no leakage between them. The flow can also be reversed, but in that case there is leakage from the pilot channel to the outlet channel, when the valve is closed. Reversing the flow direction also requires taking the supply pressure for the pilot valve from the new higher pressure side. This can be accomplished with a shuttle valve.

3. Prototype

A DFCU prototype consisting of four on/off valves was built to validate the functionality of the design. Figure 2 shows the prototype mounted on a subplate.

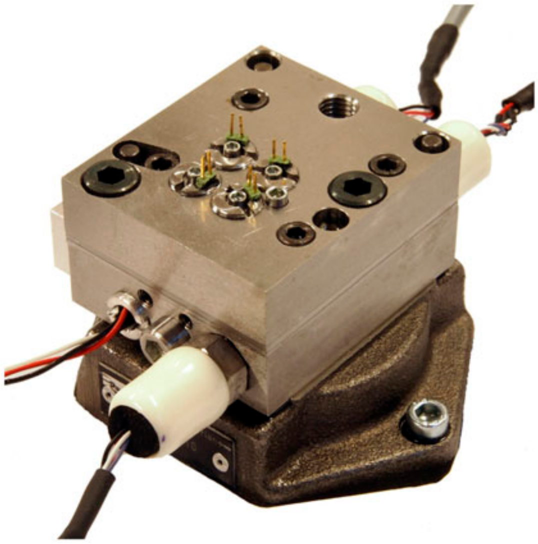

Figure 2. The four-valve prototype mounted on a CETOP 3 subplate.

The prototype consists of four layers, displayed in Figure 1, which form the body of the valve. Most of the valves' features are machined on these layers in order to minimize the number of individual parts. The valves are arranged in the manifold to a 2 by 2 array with a 12.5 mm spacing between the rows and the columns. Thus, for example 16 valves could be fitted into a square area of 5 by 5 cm.

Layer 1 on the top contains half of the magnetic circuit of each of the solenoid actuators. The other half for each valve is formed by the core of the solenoid, which is attached to layer 1 with a thread. The core and the armature of the solenoid are made of Uddeholm Stavax ESR mould steel, which is a modified AISI 420 martensitic stainless steel. Stavax ESR contains 13.6 % of chromium, which reduces its electrical conductivity significantly, therefore reducing also eddy currents.

The return spring of the solenoid is housed in the core. The force of the returns spring is approximately 8 N, but it can also be adjusted individually for each valve with a screw. Each solenoid in the prototype has a coil with a different number of turns so that also the effect of coil inductance can be studied. Table 1 shows the number of turns for each valve.

Most of the flow channels are drilled into layer 4 on the bottom. Small flow channels are also milled on the surfaces of layers 1, 2 and 4. Layer 3 separates the channels milled on layers 2 and 4. Because the distance between some of the milled flow channels on the same layer is only 1 mm, an O-ring is not suitable for separating them. Therefore, the three bottom layers are glued together with epoxy glue to prevent external leaking and leaking between the flow channels.

The movement of the solenoid's combined armature/spool is 0.2 mm. This was estimated to be the smallest opening for the pilot valve which would not cause blocking up by the larger contaminants in the hydraulic fluid. There are two different sizes of main valves in the prototype. The dimensions of the orifices and the sealing elements for each valve are displayed in Table 1. The opening of all of the main valves was 0.8 mm.

The four layers of the prototype are made of AISI 12L14 low carbon steel because of its magnetic properties and good machinability. AISI 12L14 has a relatively high permeability and saturation magnetic flux density while also having approximately double the electrical resistivity of pure iron. The hardness of AISI 12L14 is, however, only 84 at Rockwell B scale which proved to be a problem. After some switching of the valve, the seats of the main valves were deformed significantly and the balls acting as the sealing elements of the valves were able to move too far downwards and partially out of their bores. This increased leakage past the balls from the supply channel to the pilot channel. Therefore, the balls were replaced with poppets, which were ground from bearing needle rollers. This increased the length of the throttle between the sealing element and the bore.

4. Measuring and results

4.1. Setup

Figure 3 shows the hydraulic circuit of the test setup and Table 2 lists its most important components. One of the four pilot valves (component 9) of the prototype and one main valve (10) are displayed on the right side of Figure 3. There are four pressure transducers mounted on layer 4 of the prototype. Therefore, any pressure losses in the pipes or the subplate do not affect the results when measuring the pressure difference over the valve.

The current to the solenoids is controlled with a self-designed booster circuit. The booster is supplied with 24 volts for the boost phase and a holding voltage, which is different for each valve. The booster is also able to supply current in the negative direction, to quickly dissipate the remaining magnetisation in the solenoid when the valve is switched off.

Table 1. The specifications of the prototype.

	Valve 1	Valve 2	Valve 3	Valve 4
Coil turns	78	73	152	130
Orifice diameter	2.58 mm	2 mm	2.58 mm	2 mm
Sealing element diameter	3.5 mm	3 mm	3.5 mm	3 mm
Prototype dimensions	Width 70 mm, depth 72 mm, height 38 mm			

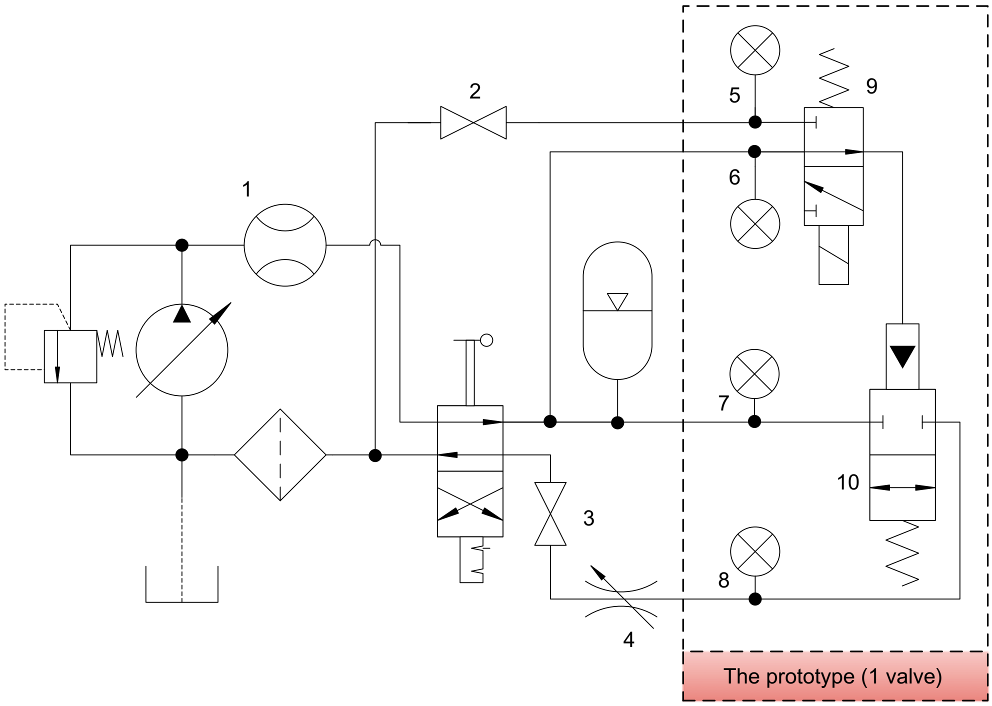

Figure 3. The hydraulic circuit of the test setup.

Table 2. The most important components of the test setup.

Device	Number in Figure 3	Model
Flow meter	1	Kracht VC5
Pilot tank pres. sensor	5	Keller 4LC
Pilot pressure sensor	6	Wika TTF-1
Supply pres. sensor	7	Wika TTF-1
Tank pressure sensor	8	Wika TTF-1
Current probe		Fluke 80i-110s
Data acquisition device		National Instruments USB-6215

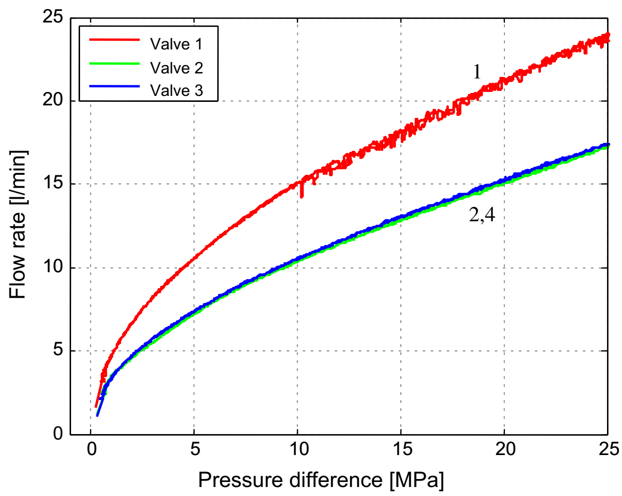

Figure 4. The flow rates through the valves as a function of the pressure difference.

Valve number 3 of the prototype functioned properly only sporadically and therefore its results are not presented here. No apparent reason for the malfunction was found when the prototype was disassembled.

4.2. Flow capacity

The flow capacity of the valves was measured one valve at a time by keeping the valve open and changing the supply pressure slowly from approximately 0.8 MPa to 25 MPa. The fluid used in the test system was Mobil DTE Excel 68 and its temperature was kept at approximately 40 °C. Figure 4 displays the flow rates of valves 1, 2 and 4 as a function of the pressure difference over the valves.

Valves 2 and 4, with a 2 mm orifice diameter, have a flow capacity of approximately 6 l/min @ Δp = 3.5 MPa. Valve 1, with an orifice of twice larger area, has a larger flow capacity, approximately 9 l/min. The full advantage of the larger orifice is not gained because the vertical movement of the sealing element is the same for all the valves. These measurements contain also the leakage from the supply channel to the pilot stage. The flow capacity with a reversed flow direction is approximately the same for all the valves.

4.3. Leakage

The leakage from the supply channel to the outlet channel was measured by closing the outlet channel with a ball valve (component 3 in Figure 3), removing the

Table 3. Leakage from the supply channel to the pilot channel when each of the main valves is open.

	Valve 1	Valve 2	Valve 3	Valve 4
Leakage at Δp = 3.5 MPa	240 g/min	330 g/min	252 g/min	375 g/min

pressure transducer (8) and weighting any fluid leaking from the mounting port of the transducer. The leakage was 2 g/min with a 3.5 MPa pressure difference and 6.2 g/min with a 10 MPa pressure difference. It is also possible that some of the fluid leaked through the ball valve (3) since the tank line was slightly pressurised. Also the leakage past the lower end of the pilot valve spool, from the pilot supply pressure channel to the tank channel, was measured with a similar procedure. The flow rate from the pilot stage to the tank channel, when all the miniature valves were closed, was 19 g/min with a 3.5 MPa supply pressure and 76 g/min with a 10 MPa supply pressure. This corresponds to approximately 0.02 l/min flow rate past one pilot valve, with a supply pressure of 10 MPa.

A real problem with the prototype is the leakage from the supply pressure to the tank channel of the pilot stage, when the main valves are open. When the valves are open, the pilot channels are connected to the tank pressure and there is a pressure difference over the sealing elements of the main valves and consequently there is some leakage past them. Additionally, fluid is leaking past the pilot valve spool from the supply pressure channel of the pilot stage to the tank channel. The flow rates to the tank channel of the pilot stage were measured when each valve was open one at a time and the results are displayed in Table 3. All the leakage measurements were done with approximately 35 °C fluid temperature.

4.4. Response time

The response time of the solenoid actuator was measured separately before attaching the upper layer of the valve to the three lower layers. The position of the armature was measured with NAIS LM300 laser distance sensor. The sampling time of the laser sensor is 0.1 ms which only enough for a rough estimation of the response time. The response time of the solenoid consists of a delay, when the magnetic force is still building up, and the movement which starts when the magnetic force exceeds the force of the return spring. When energising the solenoid, the delay was measured to be between 0.2–0.3 ms and the movement took approximately 0.2 ms. Therefore, the response time to fully open the pilot valve is approximately 0.4–0.5 ms. The solenoid was measured dry. Therefore, in normal operation the movement time is longer due to the fluid around the armature.

Because it is difficult to measure the position of the main valve sealing elements, the response times of the valves were determined from the pressure measurements according to ISO 6403 standard. (International Organization for Standardization, 1988) A throttle (component 4 in Figure 3), which creates approximately the same pressure drop as one of the main valves, was placed in the outlet channel close to the prototype and the pressure transients in the small volume between the valve and the throttle were studied. During the opening phase, the valve was determined to be fully open, when the pressure in the volume had risen to 90 % of the steady state pressure. Conversely, during the closing cycle, the valve was determined to be closed when the pressure had dropped to 10% of the initial pressure. The pressures and the coil current during one opening and one closing cycle of valve 4 are displayed in Figure 5 and Figure 6. The determined beginnings and ends of the responses are presented with vertical lines. The fast switching of the valves caused large pressure peaks in the small volume. Therefore, the outlet pressure measurements were filtered and the response times were determined from the filtered signals.

The response times presented in Figure 7 and Figure 8 were measured one valve at a time by setting the valve to change state at 2 Hz and changing the pressure difference over the valve slowly from 0.8 MPa to 30 MPa. The results show that the opening and closing response times of the valves are between 0.9 ms and 1.3 ms for most of the operating pressure range, however, below 2 MPa supply pressure the response starts slowing down to approximately 2 ms. The apparent reduction in the opening response times with all the valves below 5 MPa is caused by difficulties in determining the responses times because of increased pulsation in the supply pressure below the preload pressure of

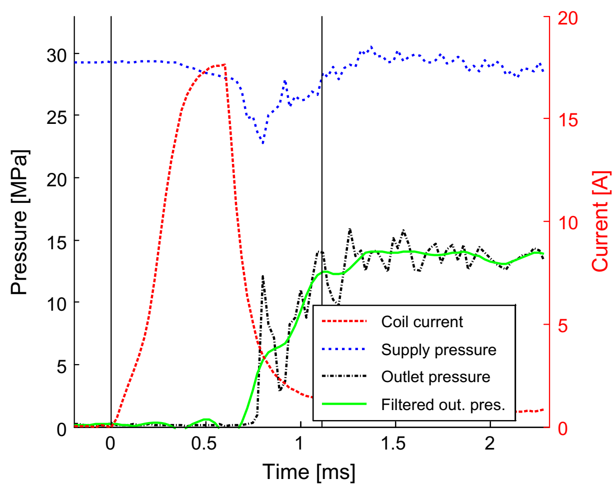

Figure 5. The pressures and the coil current of valve 4 during an opening cycle.

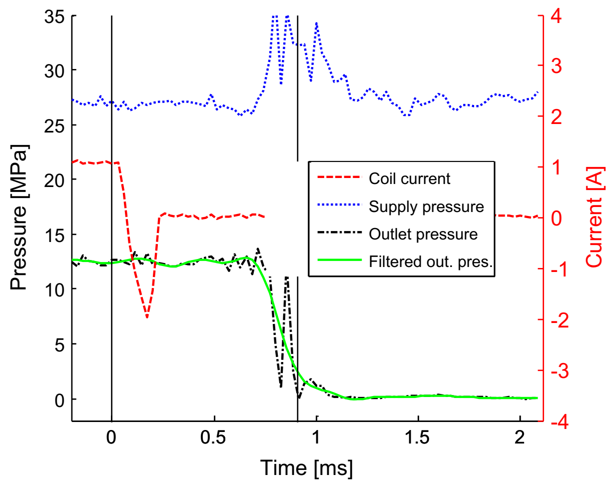

Figure 6. The pressures and the coil current of valve 4 during a closing cycle.

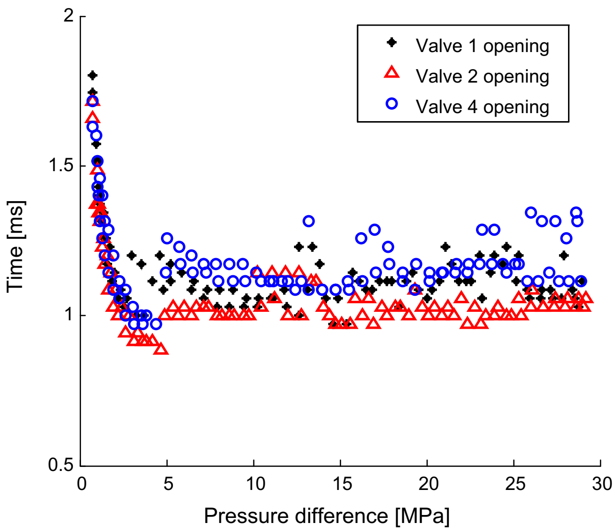

Figure 7. The opening response times of valves 1, 2 and 4 as a function of the pressure difference.

the supply line accumulator. It seems that there are no clear differences between the valves in the opening and closing response times, even though they have differently sized orifices, sealing elements and coil turns.

The response times are affected by the temperature of the hydraulic fluid. The response times in Figure 7 and Figure 8 were measured with approximately 40 °C fluid temperature. With 20 °C fluid temperature, the average closing response time increases to approximately 1.5 ms while the opening response time is less affected.

The maximum operating pressure for the valves exceeds 30 MPa. The testing pressure was limited by the maximum operating pressure of the flow meter. All of the valves operated with a 0.2 MPa pressure difference, which was the minimum pressure available from the power unit. Valve number 4 also switched with full amplitude momentarily up to a switching frequency of 450 Hz with a 10 MPa pressure difference.

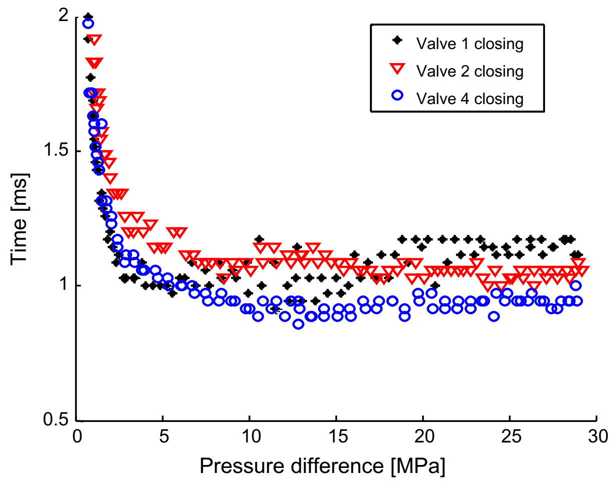

Figure 8. The closing response times of valves 1, 2 and 4 as a function of the pressure difference.

4.5. *Electric power*

During the 0.6 ms boost pulse with 24 V, the maximum current was approximately 39 A with the 73 turn coil and 18 A with the 152 turn coil. The electrical power required to keep a valve open varies depending on the preload of the return spring of the solenoid, the number of coil turns and the supply pressure level. The valves require approximately 50 to 80 mW of power to stay open at any pressure in their operating range.

5. Discussion

Even though the flow capacity of the designed valve is very good compared to its size, the measured flow rates of the valves are significantly smaller than what can be expected from the equation of turbulent orifice. It is possible to increase the flow rate of at least valve 1, with a larger orifice diameter, by increasing the opening of the valve. This, however, will also increase the response time of the valve.

The outlet channel of the valve is in practise leak free, however, the leakage from the supply pressure to the tank channel of the pilot valve is a major problem with the current prototype. This leakage does not affect the actuator connected to the DFCU, but it causes energy loss. The leakage can be reduced by increasing the length of the throttle between the main flow channel and the pilot channel by utilizing longer bearing rollers as the sealing elements of the main valve. The overlap of the pilot valve spool can also be increased to reduce leakage, but it will require increasing the movement of the solenoid armature. This will increase the response time of the actuator.

Table 4 illustrates a comparison between the presented valve, two recent prototype valves from Tampere University of Technology and a fast switching valve prototype from Linz Center of Mechatronics. Since the four valves in the prototype have slightly different properties, valve number 1 represents the designed valve in the comparison. Table 4 shows also the specifications for

Table 4. A comparison of the presented valve with three other prototype valves; ProtoØ10 (Karvonen, 2010, 2011), Hammer valve (Uusitalo, 2010a) and Fast 3/2 switching valve (Plöckinger, 2009) as well as two commercial valves Hydac WS08W-1 (Hydac) and Parker D1FP (Parker, 2013).

	Presented valve (#1)	ProtoØ10	Hammer valve	3/2 switching valve	Hydac	Parker D1FP
Response time [ms]	**0.9–1.3** *1	1.2-1.5	~2	1.5-2	~12	<3.5
Maximum pressure [MPa]	>30	20	21	30	25	35
Flow rate q @ 1 MPa [l/min]	4.7	0.3	3.3	15	17	~21 *2
Volume V [cm^3]	4	2.4	7	88 *3	73	559 *3*4
q/V [l/min/cm^3]	**1.18**	0.13	0.47	0.17	0.23	0.15

*1 With Δp > 2 MPa.
*2 Per control edge, calculated from 40 l/min @ Δp=3.5 MPa.
*3 Without control electronics.
*4 4 control edges.

two commercial valves; Hydac WS08W-01, a direct operated solenoid actuated seat valve, and Parker D1FP high response servo valve. The Hydac WS08W-1 is a commercial cartridge valve, which has been used in building DFCUs in some previous projects at Aalto University. The response time of the Hydac valve was measured in Helsinki University of Technology with a 24 V boost voltage. Parker D1FP is a very fast spool type servo valve mountable on a CETOP 3 subplate. D1FP could be considered a reference to which a four way digital valve system is compared.

The results show that the presented pilot operated miniature valve is very competitive. Its response time is the lowest and its flow density, i.e. flow rate compared to the size of the valve, is the highest of these six valves.

Future work on valve development includes reducing the leakage to the pilot stage to an acceptable level. Valves with a larger flow rate can also be built by simply increasing the size of the main orifice and its sealing element.

Now that the valves required for a compact, fast response and high flow capacity digital valve system are available, the future goal is to implement a miniature digital valve system with a performance exceeding that of a high performance servo valve. Already with the currently measured valves it is possible to assemble a four way valve system with a flow capacity of 70 l/min $\Delta p = 3.5$ MPa per control edge, 1.5 ms full amplitude response time and dimensions similar to those of a CETOP 3 subplate mountable proportional valve.

6. Conclusions

This paper introduced a pilot operated miniature on/off valve, which can be used to build compact digital valve systems with a high flow capacity and a fast response. The flow capacity of the valve is 9 l/min @ $\Delta p = 3.5$ MPa, the response time is 0.9–1.3 ms with Δp > 2 MPa and the maximum operating pressure exceeds 30 MPa.

There is still development required especially in minimizing the leaking of the pilot valve. However, the presented valve concept combines excellent response time with a very small physical size, a high flow capacity, a high maximum operating pressure and the possibility to control flow in both directions.

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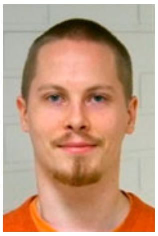
Notes on contributors



Tapio Lantela received his M.Sc. in technology at Aalto University in 2012. Currently working as a PhD student in Aalto University School of Engineering. His main research interests are miniature hydraulic valves and fast electromagnetic valve actuators.

Jyrki Kajaste received his DSc in technology at Aalto University in 1999. Currently working as a university teacher in Aalto University School of Engineering. His main research interests are dynamics and energy of hydraulic systems.

Jari Kostamo received his M.Sc. in technology at Aalto University in 2007. Currently working as a PhD student in Aalto University School of Engineering. His main research interests are fast magnetorheological valves and actuators.

Matti Pietola received his DSc in 1989 at Helsinki University of Technology (later Aalto University) and has been there Professor of Mechatronics (Fluid Power Systems) since 1997.

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