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High-flow rate miniature digital valve system

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ABSTRACT
Digital hydraulic valve systems can offer several benefits compared to spool-type proportional valves, such as faster response, more flow capacity, better fault tolerance and individual metering. However, the digital valve systems currently available are large, heavy and slow and therefore there is a need to miniaturise them and simultaneously make them faster. This article describes the design and performance of a digital valve system which can be used to replace a proportional valve with ISO 03 (NFPA D03, CETOP 3, NG6) standard interface. The valve system consists of 32 pilot operated miniature on/off valves which form four independently controlled metering edges. The valve system has small dimensions (176 × 49 × 72 mm) even though it contains integrated control electronics, its response time is approximately 2 ms and flow capacity per metering edge is 30 l/min at Δp = 0.5 MPa. These properties are good, however, improvements are required to reduce the leakage caused by the pilot valves, which can reach 5 l/min per metering edge at 10 MPa pressure difference.

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Introduction

A digital flow control unit (DFCU) consists of several parallel connected on/off valves which can be utilised to throttle flow in a similar manner as one metering edge in a spool valve does. Because the on/off valves in a DFCU are either completely open or closed, the flow rate can only be adjusted in discrete steps, contrary to a proportional valve where the metering edge is continuously adjustable. However, as few as eight parallel connected on/off valves can give a similar flow control resolution as a good servo valve (Linjama *et al.* 2003). Another branch of digital valve research is fast switching valves, which control flow rate by continuously switching a single on/off valve with high frequency (Winkler *et al.* 2010, Kudzma *et al.* 2012, Garrity *et al.* 2016). Compared to these, the structure of a DFCU is more complex, however, a DFCU does not require continuous valve switching to maintain a steady flow rate, which reduces noise, wear and pressure pulsating problems. The functionality of a commonly used four-way proportional valve can be realised with a digital valve system consisting of four DFCUs, one for each metering edge. Compared to spool-type proportional valves, digital valve systems can offer several benefits, such as faster response time, larger flow rate, better fault tolerance and better energy efficiency enabled by independent metering (Linjama *et al.* 2003).

Digital valve systems have been researched for over a decade and they have been successfully utilised, for example, in paper machines and trains (Fischer *et al.* 2015, Valmet 2016). However, there are currently very few digital valve products available. This is partly because digital valve systems are commonly built by mounting commercially available cartridge-type on/off valves to machined manifolds. This leads to large and heavy valve systems because the cartridge valves are often unnecessarily large for this purpose. This kind of large digital valve systems can be utilised in heavy mobile machinery and stationary industrial applications, however, there is a need to develop more compact digital valve systems for applications like mobile robots which have stricter space and weight constraints.

The target of the authors' research has been to develop a small, high-flow rate digital hydraulic valve system with a response time of approximately 2 ms. Also, the aim has been to make the valve system mountable on a subplate with ISO 03 (NFPA D03, CETOP 3, NG6) standard interface. The standard interface makes it easy to replace proportional valves in an existing hydraulic system with digital valve systems in order to gain improvements in performance and energy efficiency. This research has been done partly in cooperation with Tampere University of Technology (TUT) where research has concentrated on digital valve systems consisting of direct operated miniature valves (Uusitalo 2010, Paloniitty *et al.* 2015). In order to pursue a larger flow capacity than what is possible with small direct operated valves, the presented valve system consists of pilot operated on/off valves.

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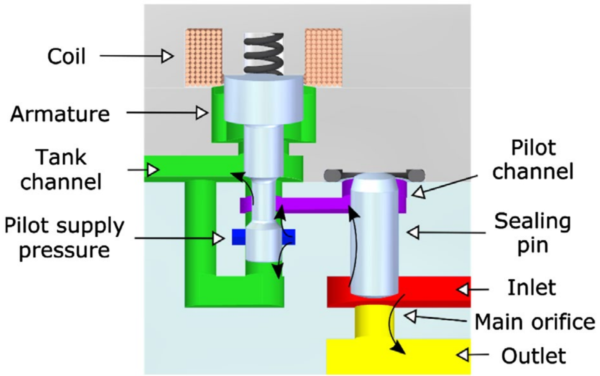

Figure 1. Cross section of a simplified model of one pilot operated on/off valve used in the valve system.
Note: Black arrows indicate leakage paths.

Design

The valves utilised in the presented valve system are similar to the ones previously introduced by the authors (Lantela *et al.* 2014). Figure 1 shows a simplified model of one on/off valve in a situation where the solenoid is energised and the main valve open.

Main valves

The seat-type main valve is opened by connecting the pilot channel to low pressure, which creates an upward force on the sealing pin since the inlet channel has a higher pressure level. The main valve is closed by connecting high pressure in the pilot channel. The sealing elements of the main valves are DIN6325 dowel pins with rounded ends. The pins were chosen as the sealing elements since they are available in many sizes at low cost and their diameter tolerance, surface roughness and hardness are excellent. When the main valves are opened by low pressure in the pilot channel, there is some leakage past the pins from the inlet channel to the pilot channel. The good diameter tolerance of the dowel pins enables a tight fit in their bore in order to minimise this leakage. When the main valves are closed by high pressure in the pilot channel, there is no pressure difference between the inlet channel and the pilot channels and thus no leakage.

The flow areas of the main valves are Fibonacci coded, i.e. the ratio of their theoretical flow capacities is 1:2:3:5. Fibonacci coding was chosen for the valve orifices since it reduces the pressure peaks caused by variation in the switching times of the valves and also improves the fault tolerance of the valve system. With Fibonacci coding, any flow rate besides maximum and minimum, can be realised with at least two different combinations of valves. Therefore, the number of difficult state changes is reduced and a single faulty valve does not necessarily affect the performance of the valve system. (Laamanen *et al.* 2007)

Table 1 shows the dimensions of the sealing pins and the orifices, which are sharp edged, in all the metering edges of the valve system. There are two valves of each size in each metering edge to test the variation in the flow rates. In theory, this valve configuration results in 23 different levels of flow rate per metering edge with a certain pressure difference. The flow resolution could be improved by sizing all the orifices differently. With Fibonacci coding, the ratios would then be 1:2:3:5:8:13:21:34 which would result in 88 distinct flow rates. This would require utilising valves with larger and/or smaller flow capacities than the valves in the current design and manufacturing those with the proposed design could require compromising either physical size or response time.

Table 1. Dimensions of the orifices and the sealing pins.

	Orifice diameter (mm)	Pin diameter (mm)	Valve opening (mm)
Valves 1 & 2	1.44	2	0.46
Valves 3 & 4	2	2.5	0.62
Valves 5 & 6	2.5	3	0.73
Valves 7 & 8	3.22	4	0.95

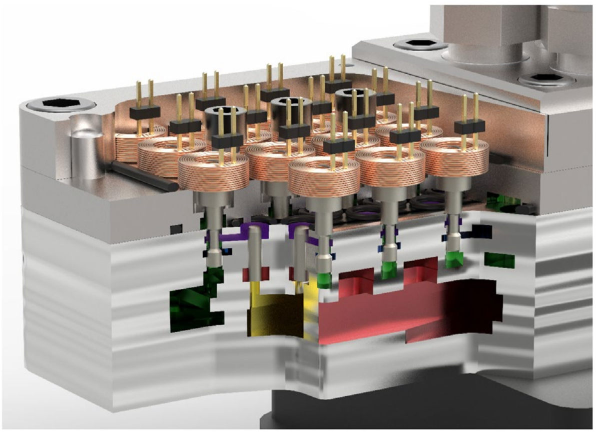

Figure 2. Cutaway view of the left half of the valve system.
Note: Flow channels highlighted with the same colours as in Figure 1.

Pilot valves

The pilot valves consist of a solenoid which actuates a spool connecting the pilot channel either to high pressure or to tank pressure. All of the 32 pilot valves are identical and they are placed in two horizontal 4 by 4 arrays on each side of the valve system. The internal structure of the left half of the valve system is shown in Figure 2. The left half contains the valves for metering edges P-A (supply channel – actuator channel A) and A-T (actuator channel A – tank channel) and the right half contains metering edges P-B and B-T. The coils and the armatures of the solenoids are housed between two machined layers which form the magnetic circuits of the actuators. The diameter of the coils is 10 mm and they consist of 140 turns of wire. During switching, the movement of the part which contains both the armature of the solenoid and the piloting spool is 0.4 mm.

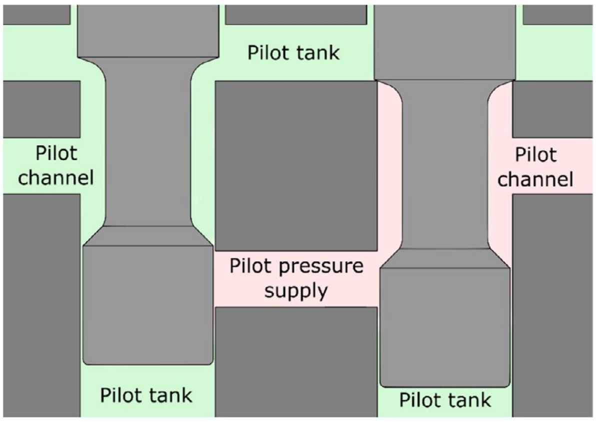

Figure 3. Position of the piloting spool when the solenoid is energised, i.e. main valve is opened (left side) and when a main valve is closed (right side).

The pilot valve was chosen to be a spool-type valve because it can be pressure compensated by connecting both ends of the spool to the low-pressure tank line. Thus, the pressure forces acting on the spool are in balance and a low-power solenoid can actuate the spool even at high operating pressure. A spool valve also creates a relatively large flow path with a small movement. The downside is that there is always some leakage past a spool. Figure 3 shows the position of the piloting spool when the main valve is closed or opened.

There are five leakage paths for each of the on/off valves, totalling in over 160 leakage paths in the whole valve system. These five leakage paths are indicated with black arrows in Figure 1. To reduce the leakage, most of which is caused by the pilot valves, the upper end of the spool has a seat-type seal between the pilot channel and tank channel. When the solenoid is de-energised and the return spring pushes the spool downwards, the coning of the spool closes this flow path and the contact also limits the spool's travel. The radial gap around the lower end of the spool causes small constant leakage from the pilot pressure supply channel to the tank channel but it is required by the pressure compensation. The main contributors to the leakages in the valve system are the metering edges of the pilot valve spools. When a main valve is opened by energising the solenoid, the metering edge creates a radial gap with approximately 0.1 mm overlap between the pilot pressure supply and tank channels. This very short gap causes a significant leakage flow to the tank when the main valves are opened.

The on/off valves in metering edges P-A and P-B should be controllable also when the pressure level in the actuator channel is higher than the supply pressure. This kind of situation occurs, for example, when holding a lifted mass high with a load sensing system. To make this possible, the pilot pressure supply for the valves in these metering edges is taken from the supply channel or the corresponding actuator channel, whichever has a higher pressure level. For both metering edges, the pilot pressure source is selected with two check valves integrated in the manifold. The tank channel is assumed to always have the lowest pressure level in the system and thus the pilot valves of A-T and B-T metering edges take their supply pressure directly from actuator lines A and B.

Manifold

The manifold is manufactured with a so called lamination method (Lasaar and Stoll 2010). The manifold consists of 14 layers of 1- and 2-mm thick laser cut stainless steel sheets. The sheets are vacuum brazed together to form a solid block with most of the flow channels inside. The flow channels can curve freely in the horizontal plane but vertically inclined channel walls have discrete vertical steps caused by the sheet thickness. Since laser cutting does not produce a sufficient surface quality for any sealing or sliding surfaces, features with this kind of functions were milled after the brazing.

The lamination method has been used also previously to produce a manifold for a digital a valve system (Paloniitty *et al.* 2015). In this study, the valves resembled cartridge-type valves housed inside the manifold and supported by the walls of the manifold. In the presented valve system, the manifold is an integral part of the structure of the valves. The manifold contains most of the features that make up the individual valves as well as all the flow channels for the 32 pilot and main valves. Consequently, its internal geometry becomes very complex and manufacturing it by drilling or casting would be infeasible. However, with the lamination method, the internal complexity of the manifold does not significantly affect the manufacturing process. Figure 4 shows an approximately 45 mm by 50 mm section of the internal structure of the valve manifold with individual channels in different colours.

In an ISO 03-sized proportional valve, the metering edges are located at the centre of the manifold. In a similar-sized digital valve system, there is not enough space in the centre for the valves and they have to be located outside the mounting pattern of the manifold. This makes it difficult to design low flow resistance channels in the manifold since the 50-mm width limitation and the locations of the mounting points restrict the placement of the channels. The flow channels inside the subplate are also quite restrictive. Due to the pressure losses in the subplate and in order to best measure the dynamics of the valve system, the pressure sensors of the digital valve system are mounted at the centre of the prototype manifold.

Control electronics

Since a fast switching response is required from the valves, the solenoids must be provided with a sufficiently high voltage to increase the coil current rapidly when

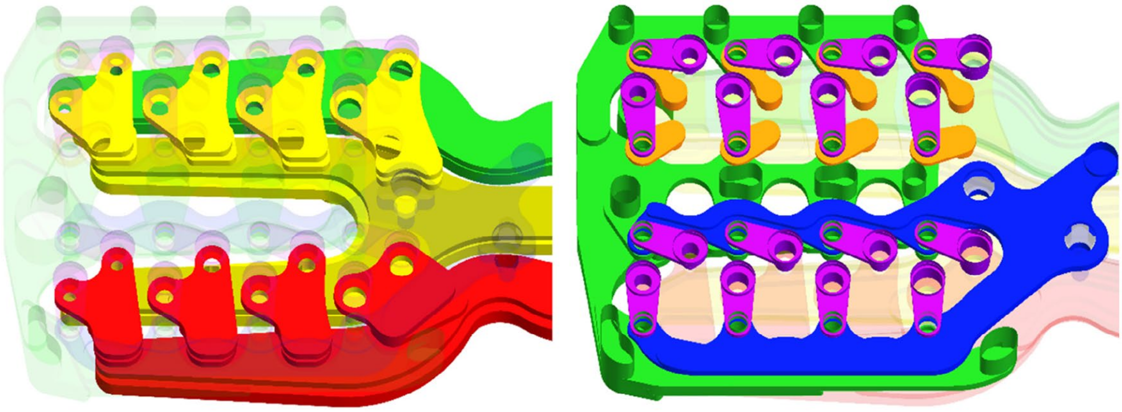

Figure 4. Flow channels inside the left half of the laminated manifold.
Notes: On the left side, the channels related to pilot stages are semitransparent and main flow channels are opaque. Right side shows the pilot stage related channels as opaque. Channels are in same colours as in Figure 1.

the solenoid is energised. On the other hand, the valve system is compact and thus excessive heating of the coils and control electronics must be prevented by keeping the holding current of the solenoids low. This functionality is realised with pulse width modulation-based booster electronics designed by the authors.

To reduce the required cabling and number of external components, the booster and control electronics are integrated in the valve system. The control electronics on each side of the valve system consist of a microcontroller board which controls a separate 16-channel booster board. The boosting voltage is 24 V which results in a maximum current of approximately 9.5 A during the 0.6 ms boosting pulse. Energy required by the boosting pulses is stored in capacitors and consequently high current power supply is not required. When a valve is held open, coil current is limited to approximately 0.5 A by reducing the pulse width of the input voltage to the solenoid actuator. The compact footprint and lack of wiring between the booster circuits and the coils should reduce any electromagnetic interference caused by PWM based current control. The only external connectors to the electronics are a power connector supplying the average current at 24 volts and an RS485 bus connector.

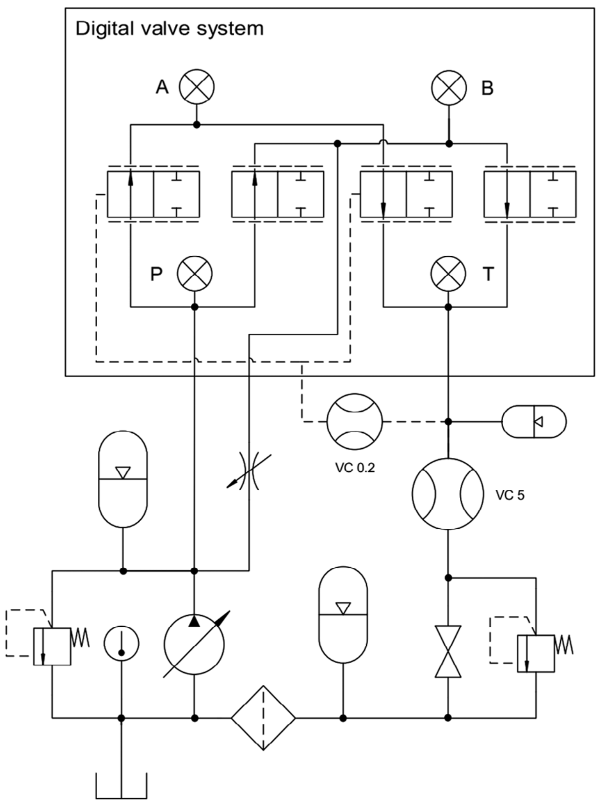

Figure 5. Hydraulic circuit of the valve test set-up.

Measurements

Figure 5 shows the hydraulic circuit of the test bench and Figure 6 shows the valve system prototype mounted on a subplate. On each side of the prototype can be seen the control electronics and in the middle of it six Wika TTF-1 pressure sensors. The additional port on the bottom left of the manifold is an outlet for the tank line of the pilot valves of metering edges P-A and A-T. The valve system was tested with operating pressures up to 25 MPa. All the tests were made with the oil and the valve system at approximately 40 °C, where the viscosity of the hydraulic fluid is 46 cSt.

Since direct measuring of the positions of the sealing pins inside the manifold would be difficult, response time measurements were made according to ISO 6403 standard by plugging the actuator channel outlets and using the valves to pressurise or depressurise the small fluid volumes in the sealed channels (ISO 1988). A rising edge in a voltage signal received by the microcontroller in the valve control electronics triggers the valve switching and the switching is determined to be complete when the outlet pressure level reaches 90% of its final steady state value. Full amplitude response times were measured only for metering edge P-B, since an additional restrictor valve was required to create some pressure in B channel also when metering edge P-B was completely closed. This was required to provide a sufficient pilot pressure for metering edge B-T to hold a specified state.

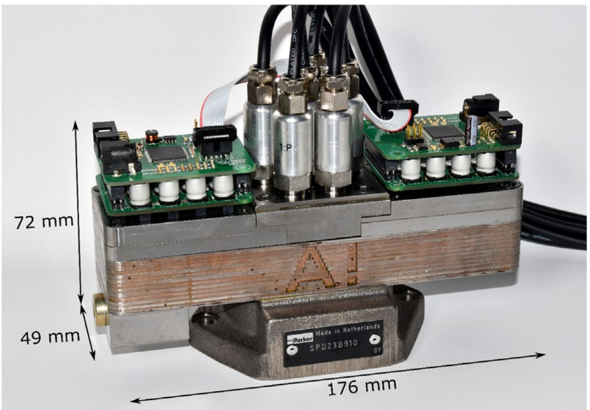

Figure 6. Valve system prototype mounted on an ISO 03 subplate.

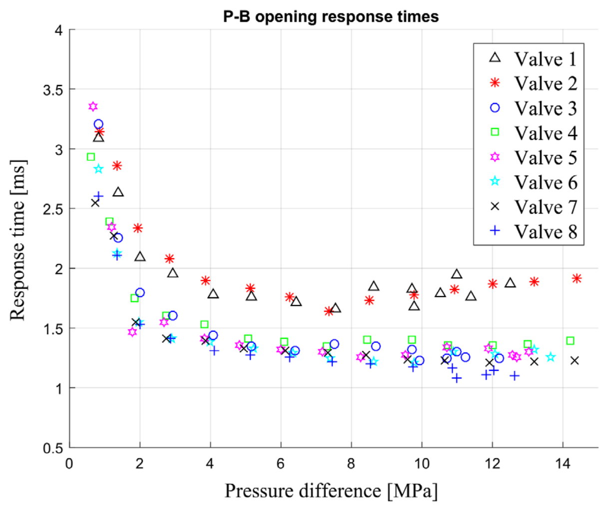

Figure 7. Opening response times of valves in metering edge P-B.

Flow capacity measurements were conducted by holding a single valve from the inspected metering edge open at a time while holding all of the valves in the opposing metering edge open. The flow rate through the valve system and the pressure difference over the valve in question were recorded continuously over a slow supply pressure sweep. To prevent cavitation, tank pressure during the flow rate and response time measurements was approximately 2 MPa. Kracht VC5 gear-type flow metre was used to measure flow rates.

To distinguish between the flow rates through the pilot stages and the main valves, the tank line of the pilot valves in metering edges P-A and A-T was routed outside the manifold and back to the tank channel through a Kracht VC0.2 flow metre. This flow rate has been compensated from the flow capacity measurements of the main valves. Also the flow rate through the pilot valves of P-B and B-T metering edges was estimated and removed from the flow capacity measurements utilising a Simulink model representing all the flow paths in the valve system. The model was based partly on geometry measurements with a coordinate-measuring machine but contains also some assumptions and thus it does reduce the accuracy of the flow capacity measurements since the simulated leakages are significant.

Results and discussion

Response time

The response times of the pilot valves are 0.6–0.7 ms when the main valves are opened, i.e. when the solenoids are energised. The opening of a pilot valve can be seen as a small decrease in the pressure level of the pilot pressure supply. The response times of the pilot valves when the solenoids are de-energised are approximately 1 ms. The opening and closing response times of the main valves depend on the pressure difference between the pilot channel and the controlled channels.

Figure 7 shows the total opening response times for the valves in metering edge P-B as a function of the initial pressure difference between the P and B channel pressures. The opening response times are less than 2.5 ms with pressure differences of over 1.4 MPa. With decreasing supply pressure level, the response times increase up to 4 ms with the measured pressure levels because the pressure difference moving the main needle decreases. The figure also shows that the smallest valves have a slower response than the larger ones and their response time slightly increases with increasing pressure difference. This apparent increase in the response time of the small valves could be explained by their smaller capacity to provide fluid to pressurise the B channel. During these measurements, B channel pressure was the same as tank pressure, approximately 2 MPa. Increasing B channel pressure in relation to tank pressure would make the response slightly faster.

Figure 8 shows an example of the pressures during an opening cycle of valve number 3 in P-B metering edge. The determined end pressure for the switching is circled. The beginning of the movement of the piloting spool can be seen as a decrease in the pressure levels of pilot pressure supply and P channels at approximately 0.3 ms after the switching is triggered.

Figure 9 shows that also the closing response times for the valves in metering edge P-B stay below 2.5 ms down to a pressure difference of 1 MPa between P and B channels. During these measurements, supply pressure was varied and B channel pressure was approximately half of the supply pressure. If the pressure level in B channel is close to the supply pressure, i.e. close to the pilot pressure closing the main valves, especially the larger valves close significantly slower. The reason for the slower response of the larger valves is that their sealing pins displace more fluid during their movement, which causes larger flow rate and pressure loss in the pilot valve.

All of the valves in metering edge P-B switch also simultaneously with a response time of approximately 2.5 ms down to a pressure difference of 2 MPa. Since

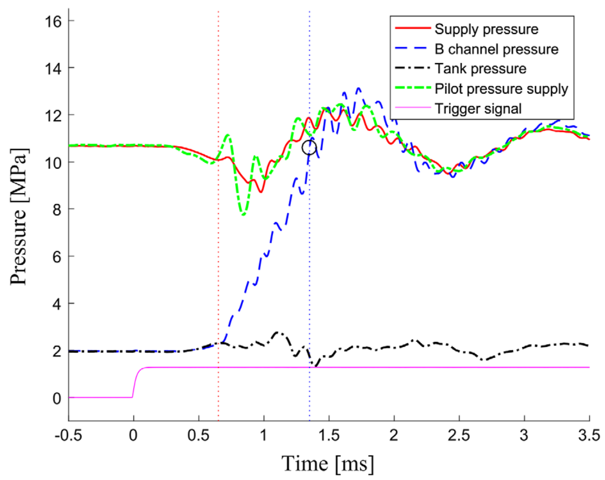

Figure 8. Opening response of valve number 3 in metering edge P-B.
Note: Vertical lines show the response times of the pilot valve and the main valve.

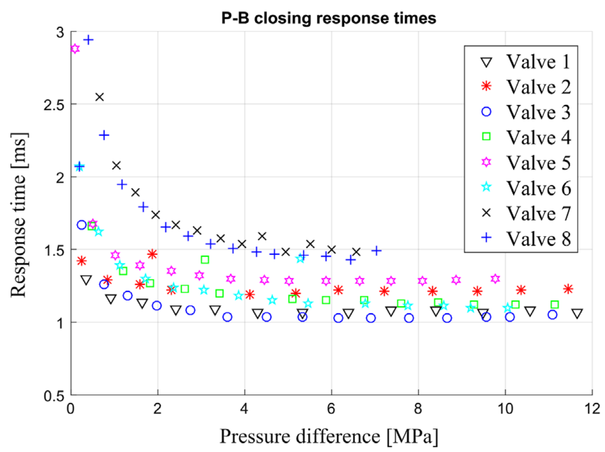

Figure 9. Closing response times of the valves in metering edge P-B.

simultaneous switching of the valves results in a relatively large momentary flow rate through the pilot stages, with low pressure levels the pressure loss in the small supply channels of the pilot stages becomes more significant and limits the full amplitude response time. The opening and closing response times for the valves in the other metering edges are similar to metering edge P-B with small individual variation.

Leakage

The combined leakage through all the pilot valves of metering edges P-A and A-T is approximately 0.7 l/min when all the valves are closed, supply pressure is 10 MPa and A channel pressure is low. In this situation, most of the leaking is caused by the radial gaps around the bottom ends of the piloting spools in metering edge P-A. Maximum leakage of approximately 5.5 l/min was measured when all the valves in metering edge P-A were open and pressure levels in P and A channels were 10 MPa. These results show that most of the leakage, approximately 0.6 l/min per piloting spool at 10 MPa pressure difference, is caused by the metering edges of the pilot spools.

To further investigate the reason for such a large leakage, the dimensions of the spools were measured with a coordinate-measuring machine. The machining quality of the spools was found out to be low as the diameter and the roundness of the spools varied significantly. The average height of the radial gap around a spool was approximately 16 μm. By selecting the best available spools to metering edge P-A, the leakage per spool was reduced to an average of 0.46 l/min. If the leakage flow is assumed to be laminar, the flow rate through the radial gap is proportional to the third power of the gap height. Thus, decreasing the height to for example 7 μm by improving manufacturing quality would reduce the leakage to approximately 0.05 l/min for a single valve.

These leakages have little effect on the control performance of the valve system but they do cause energy loss in the hydraulic system. This is a problem especially in modern high-efficiency hydraulic systems which often rely on storing energy in pressure accumulators. It is possible to reduce the leakages significantly with the proposed design, however, future research should also investigate the possibility of using seat type pilot valves.

Leakage of the main valves was not measured since in the current prototype it would have been difficult to separate their leakage from the leakage of the pilot valves. Leakage of the main valves is assumed to be negligible due to them being seat type.

Flow capacity

Figure 10 shows the flow rate curves for the valves in metering edge P-A and Table 2 shows the flow rates with a 0.5 MPa pressure difference. The flow rates of the other metering edges are similar to these with some individual variation. The flow rate through metering edge P-A with all the valves open and a pressure difference of 0.5 MPa is approximately 30 l/min.

Despite the Fibonacci coded flow areas, the flow capacities of the larger valves do not follow the intended coding. Also, the individual variation in flow capacities is stronger in the larger valves. Furthermore, the total flow capacity of the completely opened metering edge is significantly smaller than the summed flow capacities of individual valves. These results are likely caused by pressure losses in the flow channels leading to the orifices of the valves. At higher flow rates, the relatively small flow channels inside the manifold are a significant pressure loss source in addition to the valve orifices. Further analysis of these pressure losses would require computational fluid dynamics modelling. These results, however, suggest that either improving the geometry of the flow channels or increasing their size is required.

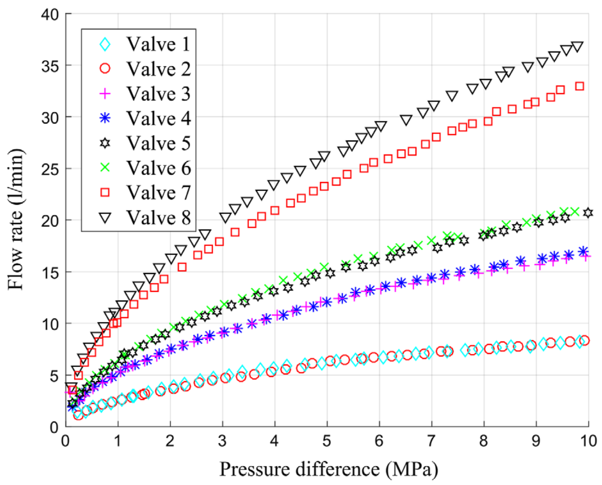

Figure 10. Flow rates of the valves in metering edge P-A.

Table 2. Flow rates of the valves in metering edge P-A with a 0.5 MPa pressure difference.

Valve #	1	2	3	4	5	6	7	8
Flow rate (l/min)	1.7	1.8	3.7	3.7	4.4	4.6	7.2	8.0

Other

The operation of metering edges P-A and P-B with a reversed pressure difference was verified, however, a more detailed response time, flow capacity or leakage tests with reverse pressure were not conducted.

The main valve seating surfaces in the manifold were significantly deformed by the impacts from the sealing pins and thus they should be hardened in future prototypes.

The power consumption of the control electronics is approximately 1 W per valve when a valve is held open. Only about 50% of this energy consumption is caused by the resistive losses in the coil. The rest is caused by voltage losses in the capacitors, switches and flyback diodes in the booster electronics. Thus, there is room for improvement in the energy efficiency of the electronics. The switching energy during the initial boost phase when opening a valve is approximately 80 mJ per switching event.

Conclusions

The structure and measurements of a digital valve system consisting of 32 pilot operated valves was presented. Since the valve system can be mounted on a standard subplate and contains integrated control electronics, it can be easily utilised as a replacement for a 4/3 proportional valve. Table 3 shows the main properties of the presented valve system as well as the properties of Parker D1FP proportional valve and a digital valve system prototype built in TUT. The digital valve system from TUT consists of 64 direct operated miniature valves similar to the ones used in previous studies (Linjama *et al.* 2015).

Table 3. Properties of the presented digital valve system, a 64-valve system from Tampere university of Technology and Parker D1FP proportional valve (Parker Hannifin 2016).

	Presented valve system	TUT 64-valve system	Parker D1FP
Step response at 100 % step, Δp = 10 MPa (ms)	<2	<3.4	<3.5
Flow rate at Δp = 3.5 MPa per control edge (l/min)	78	22.4	40
Max. operating pressure (MPa)	25	21	35
Dimensions l × w × h (mm)	176 × 49 × 72	295 × 49 × 67*	222 × 48 × 147
Flow density [l/min/cm^3]**	0.5	0.093	0.1

*Dimensions received from Miika Paloniitty, TUT. Dimensions include control electronics.
**Flow capacity of four metering edges divided by approximate volume of valve system.

The response times of the individual valves as well as the full amplitude response times of the metering edges of the valve system are approximately 2 ms with a sufficient pressure difference. This is faster than any servo valve with a similar flow capacity known to the authors. The fast response is a consequence of utilising several small valves which move only a small distance during switching. Since the valves are hydraulically operated and use pilot pressure taken from the main flow channels, their response time increases significantly below a pressure difference of 1 MPa. This is an inherent property of pilot operated valves and may be a problem in some applications.

The measured prototype suffers from leakage caused by the spool-type pilot valves. This leakage could be reduced significantly with minor design changes and improved manufacturing quality.

The flow capacity of the valve system is very good, especially considering the dimensions of the valve system and the limitations caused by the subplate mounting. The high flow capacity was enabled by using the lamination method for manufacturing the manifold which made it possible to integrate most of the features of the valves compactly into the manifold. To fully benefit from the high flow capacity and small size of the on/off valves, the valve system could also be redesigned to an application specific form in which wasted space, such as the centre part of the current manifold, is minimised and flow channels could be properly optimised. An interesting option would be to integrate the valves into an actuator, for example into the end cap of a hydraulic cylinder.

Disclosure statement

No potential conflict of interest was reported by the authors.

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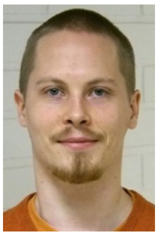
Notes on contributors



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