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DEVELOPMENT OF A POSITION CONTROLLED DIGITAL HYDRAULIC VALVE

N.P. Sell *
N. Johnston
A.R. Plummer
S. Kudzma

Centre for Power Transmission and Motion Control
Department of Mechanical Engineering
University of Bath
Bath, UK
Email: n.p.sell@bath.ac.uk

ABSTRACT

Digital hydraulics is a key part of the continuing applicability of fluid power in the modern world. In order to realize the potential of digital hydraulic circuits, valves which are able to switch at high frequencies whilst retaining high flows are first required. This paper details the development of a valve which is capable of switching in 0.5ms whilst providing a flow rate of over 50L/min at a 10bar pressure drop. Unlike most of the other valves currently in development, position control is used as opposed to the more common bang-bang actuation. This has obvious benefits for valve robustness and offers the possibility of a hybrid control approach which utilizes both throttling and switching control. This paper will detail the design and empirical testing of the valve and benchmark it against published and commercial valves before proceeding to discuss the challenges present in developing the valve further.

Introduction

In 2010 an intentionally inflammatory paper, 'Convicted to Innovation in Fluid Power' [1], was published. In it Achten decried the fluid power industry for relying on the, previous, monopoly that the hydraulic cylinder had upon linear actuation. It was his contention that this unassailable position would soon be undone due to two growing problems. Firstly, cost. Hydraulic components cost more than twice as much per kilogram as their mechanical counterparts [1]. This is in part due to the lack of standardization in components and in part because of the inherent, but growing, complexities of fluid power systems. Secondly, efficiency, during their conception in the late 18th and early 19th centuries, hydraulics were predominantly used for metal working, cranes and other heavy industry, including raising and lowering London's Tower Bridge. As was their wont engineers and what passed for OEM's had no concern for the efficiency of the systems they employed, partly due to the novelty, partly due to the low cost of energy and partly due to the inefficiencies elsewhere. This is clearly no longer the case. In every application of hydraulics, whether mobile machinery, passenger vehicles [2] or renewable energy [3], efficiency is now a measure of fitness. Increasing energy costs and tighter emissions regulations serve only to compound the problem. Classically, hydraulic components have had good efficiencies with pumps and motors capable of efficiencies in the range of 90% to 95% and cylinders having low losses due to friction. Sadly, this potential is not carried forward into efficient systems and losses of more than 50% are not uncommon [4]. The main cause of this is the metering orifice. It is for this reason that in their reply to 'Convicted to Innovation in Fluid Power' the authors of 'Is the Future of Fluid Power Digital?' [5] suggest that it is the replacing of metering orifices

* Address all correspondence to this author.

with switching elements which will prove the salvation of the fluid power industry. A good survey of some ways in which it is proposed to do this is given in [6]. They suggest that the use of switching elements has not only the ability to improve the system efficiencies of fluid power but also to reduce the cost by standardizing what are arguably the most expensive components in most hydraulic systems, the valves. By using multiple valves with relatively low flow capacities it is possible to achieve a reasonably good range of flow rates whilst maintaining the fast switching speed desired in digital hydraulics. This approach also allows a single valve design to be used across a large selection of applications simply by changing the manifold configuration rather than the designs of the switching elements themselves. It is the development of these switching elements that has formed the bottleneck to the uptake of digital hydraulic elements.

Various proposals for switching elements of multiple occurrences of the same size [7] or different sizes [8] have been presented. One of the first was presented by Uusitalo *et al* [9]. This used permanent magnets in order to create a bi-stable electromagnetic actuator. Unlike the more conventional electromagnetic spring combination which requires the electromagnet to provide the opening and holding forces, the bistable actuator requires the solenoid to only provide the transient forces as it is 'stable' at both the open and closed position requiring no force input. This reduces the maximum period for which the actuator must be powered allowing significantly higher current density to be used before heat becomes a problem. This provided a valve with a response time of <3.5ms and a nominal flow rate (at 10bar pressure drop) of 4.4L/min or 0.21(L/min)/cm^3. The design was modified in [10] to provide a response time of around 2ms but for a nominal flow of 3.3L/min or 0.47(L/min)/cm^3. Karvonen *et al* proposed a needle valve design with a solenoid and spring return which gave a flow 0.5L/min for a 20bar drop with a response time of 1.2ms [11] but no information on the valve's size has been published. A slightly different approach was taken in [7] where multiple poppets were built into a single unit. This meant that unlike the other designs mentioned thus far the number of control elements could not be changed after manufacture. With 14 separate poppets a flow rate of 85L/min, or 0.17 (L/min)/cm^3 was achieved for a 5bar pressure drop and an estimated switching time of 1ms, though this was difficult to confirm due to the deformation of various components. The latest published results in the field come from Lantela *et al* [8] who designed a pilot operated valve which should be mountable on a CETOP 3 sub-plate, provide 4.7L/min at 10bar, equating to 1.18(L/min)/cm^3 and switch in between 0.9 and 1.3ms. There were, however, problems encountered with leakage in the pilot stage of the valve and the flow rate was lower than modelling predicted due mainly to the trade-off between opening speed and size. As well as the belief that the solution to the problem of digital valves is the use of multiple small orifices, all the papers discussed above share another similarity: all the valves operate in a bang-bang configuration. This results in very simple control systems but, given the switching rates often required in digital systems (>50Hz), this produces very stringent wear and longevity criteria on the material selection and depending on the valve design can have interesting implications for failure modes. Table 1 is modified from one presented in [8] and gives the performance data for a range of digital hydraulic valves in development and one which is available commercially for comparison.

TABLE 1: COMPARISON OF DIGITAL HYDRAULIC VALVES (MODIFIED FROM [8])

	Valve from [8]	Valve from [11]	Valve from [10]	Valve from [12]	Parker [8]
Response Time (ms)	0.9-1.3	1.2-1.5	2	1.5-2	3.5
Max. Pressure (bar)	>300	200	21	300	35
Flow($L/min@10bar$)	4.7	0.3	3.3	15	21
Volume (cm^3)	4	2.4	7	88	559
Specific Flow $(L/min)/cm^3$	1.18	0.13	0.47	0.17	0.15

The rest of this paper is dedicated to presenting the design and performance of a valve which seeks to break this mould. By exchanging bang-bang control for position control a considerable number of questions surrounding the robustness and longevity of switching valves can be elegantly side stepped as well as reducing vibrations in the system. Furthermore the task of retrofitting existing systems with digital ones is simplified significantly as, if problems occur in the digital system, the valve can easily act as a proportional valve giving safe failure modes. Ease of comparison and extended capabilities are also advantages offered in exchange for this added complexity. It also departs from the common trend of creating self contained switching elements which are then mounted on a single manifold, instead being closer to Winkler *et al* [7] in having a common actuation system for multiple switching elements.

Design

Perhaps the biggest difference between the valves presented above and that given below is not in the physical design but rather the design philosophy. Rather than attempting to design a digital hydraulic valve, it was the author's intention to design a valve specifically for use in a Switched Inertance Hydraulic Transformer (SIHS) [13]. It is the belief of the authors that the future of fluid power is closely tied to digital hydraulics and that the emergence of a standardized switching element is therefore of critical importance. However by using bespoke components as a stepping stone it is possible to advance digital hydraulics without needing to standardize on a single switching element as the development and validation of digital hydraulic circuits are fast approaching the limit imposed by commercially available valves. The effect of bespoke valve design can clearly be seen by comparing the results of [14] and [13], both of which doc-

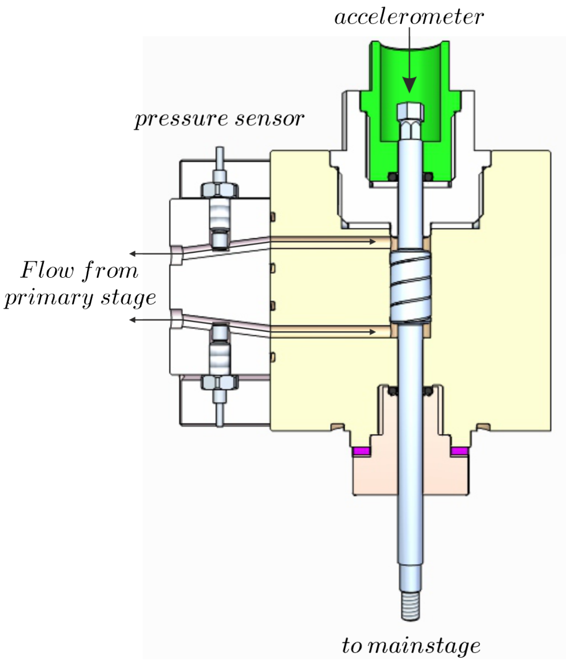


FIGURE 1: CUT-OUT OF ACTUATION MODULE

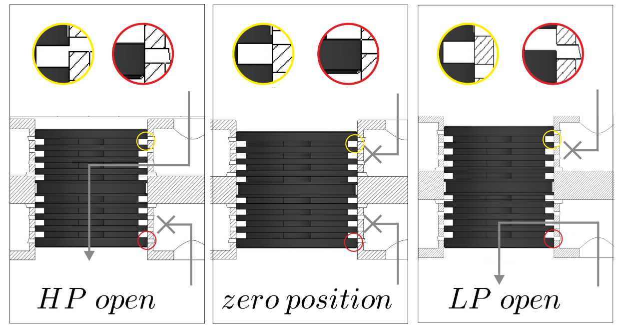


FIGURE 2: MULTIPLE GROOVES ON SPOOL AND SLEEVE

ument research into the SIHS. The former used an off-the-shelf valve, significantly limiting the scope of research as alluded to in the conclusion, whereas the latter had a custom designed valve which met the desires expressed by the former, significantly expanding the horizon of research that could be, and was, undertaken.

The fast switching linear valve presented in this paper is a three stage design with two inlet ports (high pressure and low pressure) and a common outlet port, rather than a simple on-off action. The same net outcome could have been achieved with two on-off valves but by building a three-port valve the difficulties of timing the opening of multiple valves, component count, instrumentation needs and sources of error are all reduced. The valve could still be used as an on-off valve if desired by simply blocking an inlet port. The valve can be divided into two main areas, the actuation module and the main stage. The actuation module is composed of a Moog E050-899 servovalve, the first stage, and an equal area actuator that forms the second stage. A sectioned view of this actuation module can be found in Fig. 1.

For instrumentation purposes there is a pair of pressure tappings on the inlet to the second stage and for control and instrumentation purposes an accelerometer is screwed into the top of the actuator. It can also be seen that all external seals are o-rings. Although these are designed for static sealing, due to the small expected displacements of the piston (fractions of a millimetre), it is believed that rather than needing to provide a sliding seal the o-rings can deform sufficiently to accommodate the piston's movement. The use of o-ring seals for small hydraulic components is supported by [15] who concluded that for cylinders of diameters smaller than 10mm o-rings are convenient and essentially leak free, even when providing a sliding seal.

The threaded end of the actuation piston is screwed into the main stage to provide actuation to the multiple switching elements. In this case, grooves on a spool. This is similar to the design proposed in [16] which suffered from unstable positioning of the central spool at high pressures. This problem is largely mitigated by the use of zero lapping and pressure balancing in the main spool. Having four metering edges per port, see Fig. 2, gives a very high flow gain. For a displacement of 0.1mm a flow area of 37.7mm^2 is exposed. Calculations conducted on an earlier iteration of the fast switching linear valve found that a flow of 65L/min could be expected for a pressure drop of 10bar, based on a discharge coefficient of 0.6 and density of 870kgm^{-3} [17]. The geometry of the main spool remained unchanged during the reiteration of the valve design meaning similar figures are expected.

This spool is designed with zero lap and a maximum allowable clearance of 6μm between spool and sleeve, giving an anticipated 0.013 (L/min)/$\sqrt{bar}$ leakage flow for a 10bar pressure drop at 0mm. At +0.1mm the estimated 65L/min @ 10bar should flow from HP to common port A and at -0.1mm flow should be from LP to A. This can be seen in Fig. 3. The assembled valve can be seen in Fig. 4 below. It is immediately apparent that the valve is significantly bulkier than the proposed universal designs. This is mainly a result of machining the valve body from four rounds of solid stainless steel, one of the design compromises made in order to allow prototype manufacture of the valve. As well as making the valve considerably larger than need be this presented the risk of misalignment within the valve, particularly between the main and actuation stages. Therefore bosses were used between each of the four pieces with bolts providing the fastenings. The other key compromise made in deference to prototyping was the selection of the Moog servovalve used for the primary stage. This has a measured flow of 29.5L/min at 69bar, less than half the intended operating pilot pressure. This flow

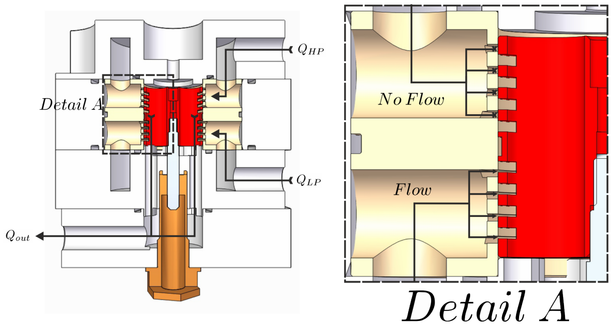


FIGURE 3: FLOW PATHS THROUGH THE VALVES MAIN STAGE

FIGURE 4: PICTURE OF ASSEMBLED VALVE

rate is significantly larger than even the highest flow rate likely to be required based upon the research of De Negri *et al* [14], Eq. 1.

$$
\begin{aligned}
Q_{max} &= \frac{Displacement}{Switching\ Period} \cdot Area \\
Q_{max} &= \frac{0.2 \times 10^{-3}}{0.5 \times 10^{-3}} \cdot 5.02 \times 10^{-5} \\
&= 2 \times 10^{-5} m^3 s^{-1} \\
&= 1.2 L/min \qquad (1)
\end{aligned}
$$

A more closely matched valve would likely gain in bandwidth what it gave up in flow rate. The frequency response of the Moog valve, Fig. 5, shows a clear resonant peak around 1kHz. Generally it is considered that for good tracking the closed loop bandwidth of the control element should be an order of magnitude higher than the plant. This suggests that achieving switching

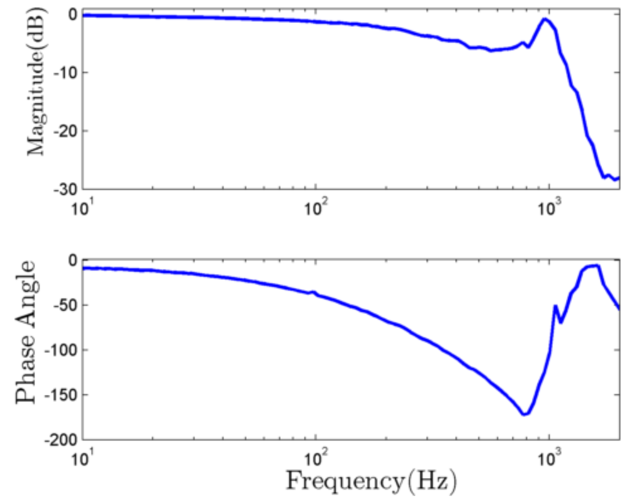


FIGURE 5: FREQUENCY RESPONSE OF MOOG PILOT VALVE

rates above 100Hz will be challenging.

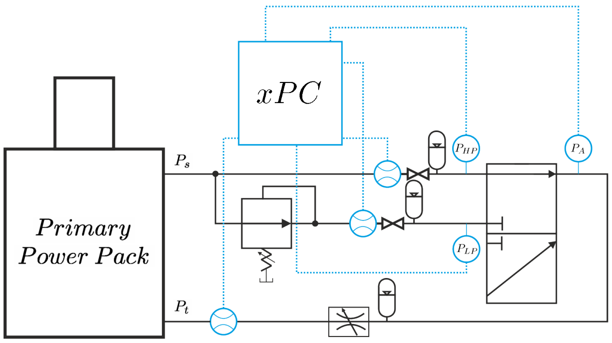
Measurement Methods and Results

Setup

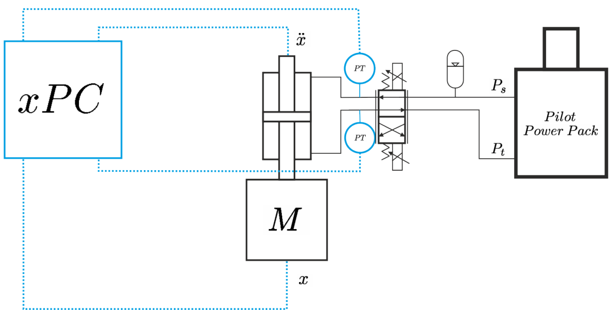
The valve can be considered as two separate hydraulic circuits, one for the actuation stage and another for the main stage. For testing purposes flow to the actuation stage was supplied by a separate power pack to that which supplied the main stage. Figure 6 shows these two hydraulic circuits as they were set-up for testing. Unlike the hydraulic circuits of the different sections which were handled separately, the instrumentation was centralized utilizing an xPC system, with a theoretical maximum sample rate of 100kHz. In practice the instrumentation and control needs meant this was reduced to 10kHz, an order of magnitude larger than the 1kHz which is likely to provide the upper limit of the control bandwidth due to the frequency response of the pilot valve. The instrumentation utilized in the test set-up can be seen in Table 2. A pair of 4^{th} order low pass filters with a cut-off frequency of 5kHz were used as anti-aliasing filters on both the position and accelerometer signals and a hand held temperature probe was used for to measure fluid temperatures.

Sensor Fusion

In order to increase the bandwidth of position measurement over and above that provided from the inductive position sensor a pair of complementary filters was designed using the methodology outlined in [18]. These enabled the position sensor to be used to provide lower frequency information whilst the twice integrated acceleration signal provided the high frequency information. The cross over location (where the contribution from the



(a) DIAGRAM OF HYDRAULIC CIRCUIT FOR MAIN STAGE



(b) DIAGRAM OF HYDRAULIC CIRCUIT FOR PILOT STAGE

FIGURE 6: Experimental test set-up

TABLE 2: INSTRUMENTATION

Device	**Location**	**Model**
Position sensor	1	Keyence EX-110
Accelerometer	2	Kistler 8730AE500
Flow meter 1	3	AW Gears JVS-20KG
Flow meter 2	4	AW Gears JVS-60KG
Flow meter 3	5	Hydac EV3100
Pressure Sensor 1	6,7,8,9	Meas XP5
Pressure Sensor 2	10, 11 ,12	Meas EPX350

two signals is equal) was selected by applying a filtered chirp signal to the pilot valve, plotting the frequency response of the two sensors and finding where they most closely agreed. Figure 7 shows the position response of both the position sensor and accelerometer to a chirp signal. It is clear that both phase and magnitude are in close agreement between around 40Hz and 80Hz. A

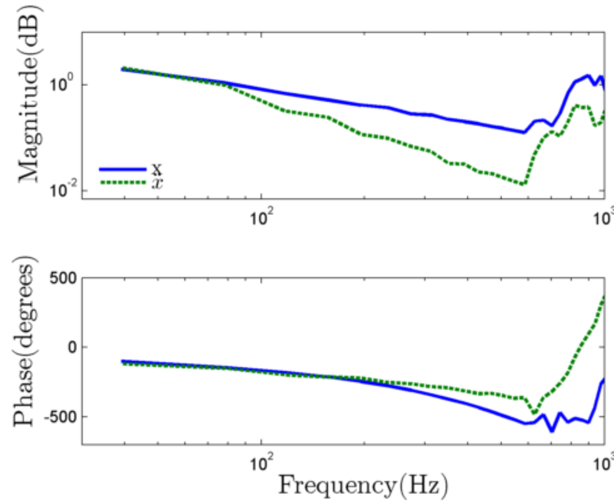


FIGURE 7: FREQUENCY RESPONSE OF POSITION AND ACCELERATION SENSORS

cross over at 50Hz was selected as this showed the closest agreement when examining the signal traces. The filters can also be used to stop drift in the acceleration signal by using double differentiated position data and the accelerometer signal. Velocity measurements can also be provided by combining acceleration and position signals.

Steady State Measurements

The position sensor was located by means of a grub screw, so there is no fixed reference from which to calculate the valve's zero position. This was therefore done by sweeping the valve from one extreme to the other using a simple PI controller, firstly with only HP connected and then with LP connected. Figure 8 shows the flow position relationships found for the two ports. It was determined that the spool was closed at 0.4mm with maximum flow occurring around ±0.1mm either side of this as expected. A slight increase in flow was measured for openings greater than ±0.1mm but this is possibly a result of inertia in the flow meter and inertance tube or charging/discharging of accumulators. By repeating the sweep test with both LP and HP connected but outlet blocked, the leakage between the two ports can be clearly seen. Figure 9 below shows the relationship between position and leakage flow with a 50bar pressure differential between the HP and LP ports. A clear hysteresis effect can be seen in the results. However this likely a result of the accumulators recharging rather than leakage. The actual leakage is therefore believed to be the internal curve. There is no leakage at -0.1mm and 0.1mm and the maximum leakage occurs at the zero position. This suggests that the spool has a slight under lap rather than zero lap designed, but this is unlikely to cause a problem given the anticipated switching time. In fact, this effect may

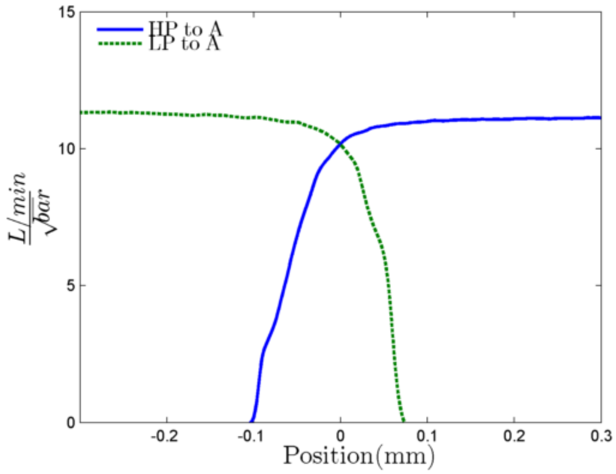


FIGURE 8: OUTLET FLOW AGAINST POSITION

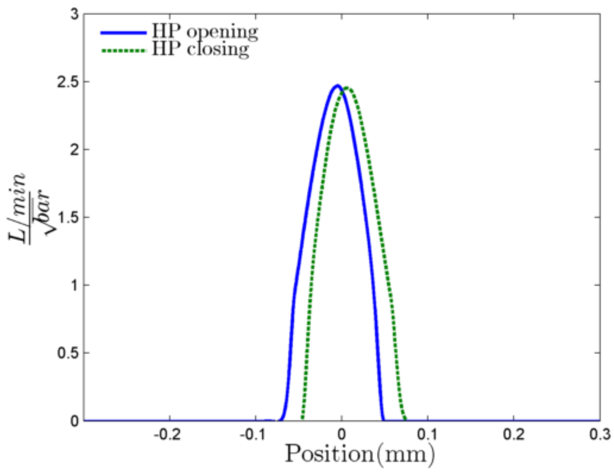


FIGURE 9: LEAKAGE FROM HP TO LP

help to reduce switching vibration. These measured leakages and their locations supports the calculation of port openings and zero position. It is also worth noting that the intended switching time is ≤1ms meaning the flows will likely to negligible. When operating at 200Hz (the fastest measured frequency) 20% of the cycle is spent switching, in this case if there were a 10bar drop from HP to outlet and 200bar drop between HP and LP it is expected that leakage flow would be around 4% of the flow through the valve.

As stated above the design flow rate of the valve was 65L/min at 10bar. In order to validate this flow rate the valve was held at +0.1mm and the pressure at the HP port increased until a drop of 10bar was registered. The results of this, seen in Fig. 10, show a lower than anticipated flow, around 50L/min. This is likely to be in part due to the oil temperature being fairly low (30^{o}C) but is probably mostly a result of the prototype man-

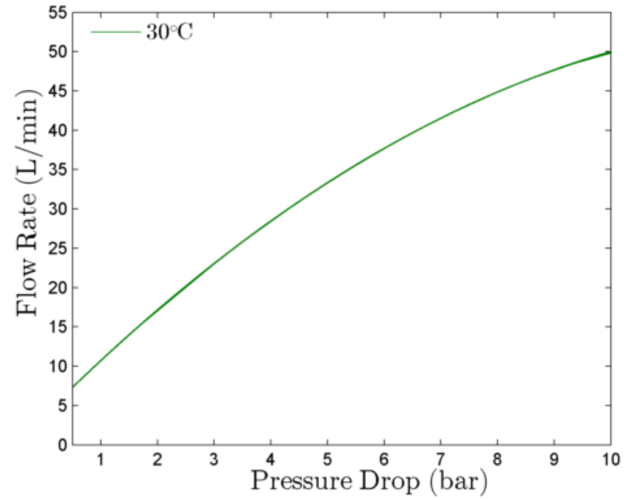


FIGURE 10: FLOW AGAINST PRESSURE DROP

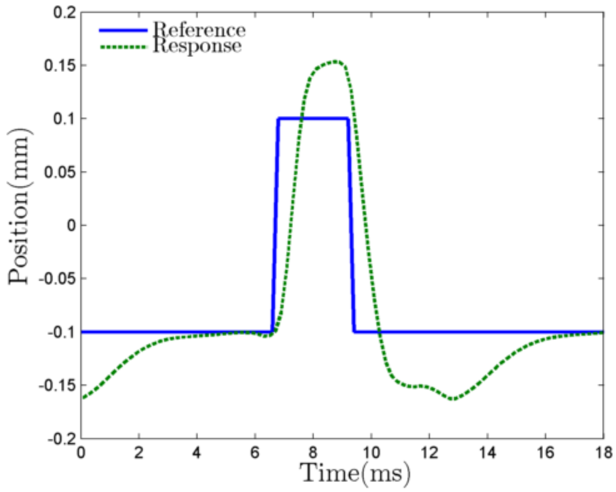
ufacture, which introduced a lot of right angled turns and sharp edges into the flow path.

Dynamic Tests

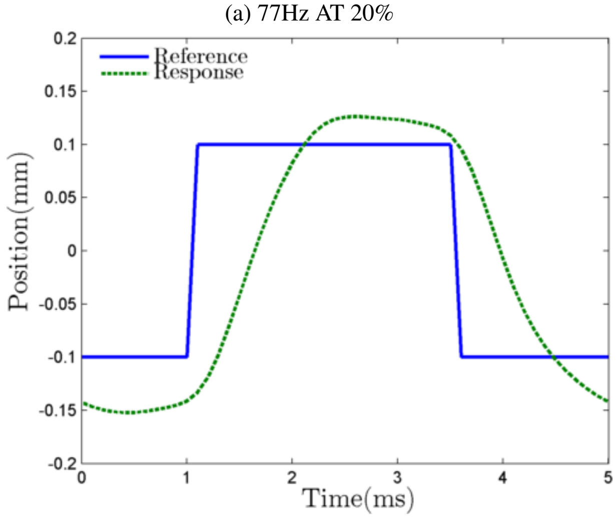
In [14] it was that concluded, from modeling, that in order to create an effective SIHS the switching valve should settle in 1ms. Given the particular design of the valve in question the requirement for settling in 1ms can be exchanged for switching, transitioning through the change over region of ± 0.1mm, in 1ms. As long as the valve does not overshoot by more than 0.2mm there should be no real difference between the two conditions. In order to control the valve a state variable feedback (SVF) system was used. This made use of the availability of position, velocity and acceleration measurements. This was cascaded with an Iterative Learning Controller (ILC) to provide the switching rate required. An article containing a more detailed description of the control system is currently being written. The response in Fig. 11 below shows that the valve is capable of achieving the 1ms switching time required and provides good tracking up to 200Hz. It is possible to achieve an even faster switching rate if required. This can be seen in Fig. 12, where switching occurred in 0.5ms, though this did require a greater input energy and also approached the saturation limit of the accelerometer.

Discussion

The valve has been shown to have not only a high flow rate (>50L/min at 10bar) but also a very fast switching time (0.5ms) which can be considered the equivalent of settling time in a bang-bang valve. If taken on their own these numbers would allow the valve to be favorably compared to other prototype and commercially available valves, particularly if you considered the valve



(a) 77Hz AT 20%



(b) 200Hz AT 50%

FIGURE 11: VALVE RESPONSE TO PWM SIGNALS

as a pair of connected on-off valves. However, this simplistic view overlooks that the valve in its present incarnation is, intentionally, bulky and over-complicated. The main issue with the current design is that actuation power is supplied from a power pack running at constant pressure, if the losses in generating this pilot flow are neglected then the actuation power would be ¡1% of that passing through the valve. For the purposes of experimentation this was expedient given the desire to provide a test bed for SIHS. However it does mean that, realistically, any SIHS system utilizing the valve in this form will have a heavily penalized energy efficiency. Of secondary concern is the issues which arose with manufacturing. It was necessary to re-manufacture the central spool and sleeve due to problems with meeting the zero-lap criteria, as was shown above this perfect zero-lap was still not achieved though the results were deemed acceptable. This

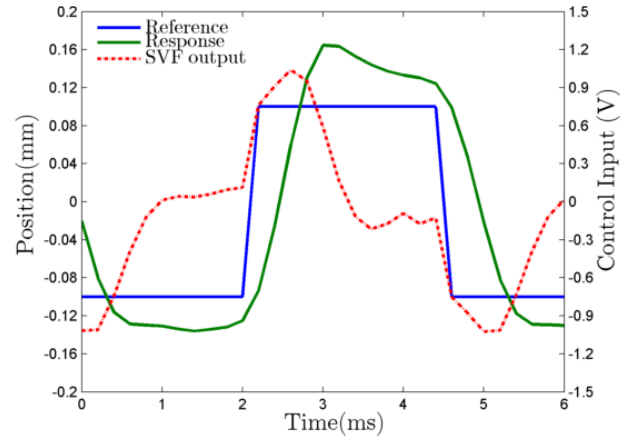


FIGURE 12: FASTER SWITCHING OF VALVE

problem is likely to be compounded if more grooves were added either for higher flow rates or in order to create a four-port valve.

The limitation of the current incarnation of the valve must be overcome if it is to be of use on anything other than a test bed. One proposed solution to the problem of power supply to the pilot stage is to develop a recursive SIHS circuit. This could take flow from the HP line and output a constant pressure supply. The system would take a similar form to that presented in [19] though with a significantly smaller flow rate than their 20L/min at 20bar. Thus a significantly smaller valve and inertance, or inductance, pipe could also be used. This would make the valve less efficient, with some portion of the power provided to it being used as actuation energy, but would remove the need for a separate power supply circuit as, given the small size of the SIHS, a standard piezo actuated valve should be usable.

Due to the repetitive nature of the testing to be conducted on the SIHS the use of an ILC provided a good candidate for gaging the maximum performance of the valve. For ease of implementation the ILC was cascaded with a feedback controller which in itself could achieve switching times of around 1ms. However, it would be possible to instead use the ILC to tune the feedback controller over a series of conditions. The learning could then be turned off, giving a feedback system that would not require a cyclic input profile.

Conclusion

This paper has presented the design and performance of a fast switching valve designed with the expressed desire of furthering research into Switched Inertance Hydraulic Systems (SIHS) rather than the more common goal of creating generic on-off valves. The reasons for doing this were given in along with the implications this had on the design. The outcome was a valve with a flow of >50L/min at 10bar pressure drop, a minimum recorded switching time (shown to be analogous to settling time) of 0.5ms and specific flow of 0.48(L/min)/cm^3 for the actual switching section of the valve. This last measurement was included in order to make the figure comparable with the table found in [8]. This could be improved were attention paid to the space efficiency in the design stage. These figures, when taken in isolation, suggest a real hope of fulfilling the dream of Scheidl, Linjama and Schmidt [5]; a single common switching element. However, there are some key limitations that need overcoming. Firstly, a more efficient and convenient method of actuation, a control system not dependent on a cyclic input and finally validation of the manufacturing suitability. However the valve does meet the specifications given in [14] for furthering the testing of SIHS.

ACKNOWLEDGMENT

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