ADVANCEMENT AND DEMONSTRATION OF THE NEW GENERATION OF LCM'S FSVI4.1

Thomas Zehetbauer¹, Paul Foschum¹, Andreas Plöckinger¹, Bernd Winkler¹ ¹Linz Center of Mechatronics GmbH Linz, Austria Altenberger Straße 69, 4040 Linz, Austria E-mail: thomas.zehetbauer@lcm.at

ABSTRACT

The paper presents the latest switching valve technology designed and engineered by the Linz Center of Mechatronics GmbH (LCM). It will give an overview about the general advancement progress from generation 3 to 4, design rules, desired specifications and performance of the 2/2 way fast switching leak-free seat type valve.

Different control modes of the solenoid will show valve state behaviours like ballistic and ordinary switching. Due to consistent multi-criteria-optimization during the development process the maximum switching frequency of the valve is pushed to 500Hz. To point out the relevant characteristics and limits in a practical way, four of these valves were arranged in a valve block to realize fast 4/3 way switching valve functionality. It was integrated into a mostly existing hydraulic drive system with an accumulator charge system and a double rod cylinder with a certain dead load to demonstrate its performance and various control opportunities.

The conclusion confirmed our assumptions that digital hydraulics, especially by taking well-chosen components, provides the possibility to control a hydraulic cylinder by force respective pressure, position and even by velocity. An alternative control method the so called phase shifted PWM exciting the pulses with a frequency of about 1 kHz was tested in order to reduce noise emission and adverse mechanical vibrations.

KEYWORDS: fast switching valve technology, seat type valve, leak-free, 4/3 way fast switching, decoupled metering edges, valve control methods, digital hydraulics, PWMVCF-control (pulse width modulation with variable carrier frequency), phase shifted control

1. INTRODUCTION

The motivation of this paper's work was to point out the current limits, advantages and even disadvantages of the existing switching valve technology. One problem of our last version of the FSVi3.2 [1] was the valve spool which is very costly in manufacturing when low leakage has to be achieved. To reduce the price for series production a conventional pressure compensated seat-type poppet valve including the return spring and sealing system was used.

The challenge was to design a solenoid including the actuation in order to open and close the valve in the shortest possible time. After intensive testing under different physical conditions the operating limits of the valve were determined. These limits led to inspirations of new applications and control ideas, some of them will be presented in detail in the following chapters.

2. LATEST FAST SWITCHING VALVE

2.1. Basic development, components and modelling

As mentioned before, the initial idea was to open and close a poppet of a certain valve as fast as possible. To achieve the required solenoid forces the conventional poppet was mounted into a hydraulic block for measuring the quasi-static force-stroke hysteresis which is heavily influenced by the friction of the sealing system and the return spring (Figure 1). The peak forces have thereafter been determined to about 70N. Additionally a basic flow force model has been built. The goal was to get a hydraulic switching time in the range of 1ms with a minimum delay time between electric control signal and mechanical poppet movement. The function of the valve is shown in Figure 2. When the solenoid gets activated, the armature at its free starting position (with a free moving space to the upper end stop of about 0.1mm) touches an anti-wear ball which transmits the armature impulse towards the poppet. After switching off, the poppet will return in its original position by the return spring and flow forces. The poppet pushes the ball against the armature back into its initial state.



Figure 1: friction-lift hysteresis inclusive spring return



Figure 2: schematic illustration of the switching valve

Figure 3 shows the basic electromagnetic circuit which has been built via FEMM [2] such that the parameters can be changed within certain boundaries without leading to an invalid geometry. In a first attempt only rotational symmetric analysis has been performed. The armature of the solenoid will always be in the fluid space to guarantee the valve function in both directions. Therefore it was important to design the geometry such that physical influences like oil stiction and fluid displacement will be as low as possible. The cushioning grooves [3] on top and bottom side of the armature lower the velocities at the end stops, whereas no fluid stiction effects hinders the lifting of the armature out of its end stops in order to minimize the delay time of the valve After the rough geometry of the design has been chosen a dynamic model was built in MATLAB Simulink. The physical effects like oil stiction [4] at small gaps, pressure build ups on decreasing volumes, lift depending transient magnetic forces, flow forces [5], inertia and an end stop model at the hitting points were considered in the equation of motion.



Figure 3: schematic geometry of the solenoid during magnetisation

2.2. Design and objectives

In a next step the rough design had to be optimized. The solenoid had to be driven with the LCM's in house developed and manufactured valve electronics (Figure 4). Due to the high variable boost currents up to 70 Amps at 24 Volts the coil can be optimized in force and energy consumption without losing scope. The current control sequence can be divided into three segments namely boosting, holding and closing. During the boost phase the full voltage is applied onto the coil, whereas the current at the holding phase is PWM-controlled in order to reduce the energy consumption. At the closing period the coil is applied by the negative voltage without control for a certain time.



Figure 4: valve electronics (developed for FSVi)

The minimum requirements to the new fast switching valve are:

- switchable pressure drop up to 300bar
- time from signalling to valve opening below 3ms (@ dp =300bar)
- symmetric switching behaviour at 200bar (G1 \rightarrow G2 ~ G2 \rightarrow G1)
- symmetric opening and closing behaviour at 200 bar
- hydraulic switching time about 1 to 2 ms (5 to 95% valve opening) at 200bar
- about 5 l/min at 5 bar pressure drop (fixed due to the chosen poppet)
- easy manufacturing and assembling

The multi-criteria-optimization based on the requirements above has been executed via MagOpt, which is an LCM developed optimization tool and has already been used for some development procedures of the 3rd generation of LCM's fast switching valve technology (FSVi3.2 [6] and [7]). The basic models from the previous subchapter (FEMM-Model, Simulink-Model) were connected to MagOpt. After defining the boundary conditions (max. diameters, height, coil settings etc.) the geometry with the best fit to the specification below was chosen. In order to withstand the high pressure it was decided to weld the top cover and coil carrier onto the base body of the valve. A drastically increasing accuracy of the magnetic material model was given by the in house measurement system for BH-curves, though the model has still uncertainties caused by inadequacies on the mathematic model and the unconsidered high pressure depending dynamic forces of the sealings. So the model will always give "better" results (regarding the switching time) than in reality.

2.3. PERFORMANCE TESTINGS AND VERIFICATION

After manufacturing and assembling a small series of valves with standard CETOP 6 intermediate mounting plate connections (Figure 5), which are rotatable at 90° in each case, the tests were performed. The hydraulic scheme is shown in Figure 6. To verify symmetric switching, the valves were proven in both directions. The following results are based on the last state of development.



Figure 5: the valve design including the connection plate (CETOP 6)



Figure 6: hydraulic scheme of the valve test set up;

2.3.1. Ordinary switching mode

At first the function of the valve was verified for "ordinary"(non-ballistic) switching. The settings of the solenoid control electronics (boost and hold current and time) were determined by the system pressure at its maximum of 300bars. An eddy current sensor, directly mounted in the top cover, measured the armature position. Figure 7 shows the armature lift at different pressure drops for different valves according to the coil current. The measurements were made instantly after finishing the assembling. A run-in phase which has positive influence on the switching time behaviour can be observed. In our case the hydraulic opening time could still be improved by increasing the supply voltage and the maximum boost current, but since the requirements were already fulfilled with the present settings, those changes were not considered in order to achieve a higher maximum switching frequency. Figure 8 shows the switching behaviour in the two different directions $G1 \rightarrow G2$ and $G2 \rightarrow G1$ at a pressure drop of 200bar. It can be seen, that the switching time is highly symmetric.



FAST SWITCHING VALVE AFTER ASSEMBLING (0 cycles)

Figure 7: ordinary switching behaviour at different pressures (direction $G2 \rightarrow G1$)



Figure 8: ordinary switching, both directions in comparison

2.3.2. Ballistic mode

The advantages of the ballistic mode, which means the valve doesn't completely open during one cycle, are the anti-wear effect of the lower end stops and overall lower noise emission. The ballistic mode enables very small amounts of oil passing the valve per switching cycle. Possible applications could be the micro-positioning system by LCM [8]. Figure 9 shows the poppet lift due to different pressures drops and trigger times.



Figure 9: ballistic mode with different duty cycles and pressures

2.3.3. Duration

Long-term tests should show switching time dispersions and an estimation of lifetime. The first prototype of the new valve was set at a constant pressure drop and constant frequency of 50Hz. Each ten-thousandth cycle was saved and plotted in the following Figure 10. The darker the colour of a trace, the higher is the amount of lifecycles. After 55 million cycles the measurement failed due to malfunction of the positioning sensor over the weekend, so the long-term test was stopped. The sensor cable caused some problems and the rise and fall of the lift curve is not representative due to the non-linearity of the eddy current sensor. There might be a kind of run-in period that lasts about 14 million cycles, which is probably caused by the friction change of the centering and pressure equalising sealing between poppet housing and valve plug.

The hydraulic switching times have, except some statistical outliers, no abnormalities. The valve closing lasts a little bit longer than in the specification above, but this is caused by the low flow forces due to the pressure drop of only 3bars.



Figure 10: long term test with first FSVi4 prototype

After a test time of 100 million cycles the valve was still working, but one phenomenon which could not be clarified, was the apparently arbitrary alternating different states of the hold time. Due to the fact, that the maximum pressure was set to 4 bar and cushioning grooves were applied, hydro-mechanic effects were excluded as a cause. A possible reason could be an inhomogeneous current control, which should have been stable [9]. The impact of the current control fluctuations was higher than expected. A

boost time change of about 0.2 milliseconds led to a hold time variation up to 1.4 milliseconds. Our presumption was some magnetic short circuit due to the cushioning grooves. The magnetic holding force highly depends on the magnetic remanence, which is not controlled and therefore hard to reproduce. Increasing boost times due to the incorrect current control has a vast influence on magnetisation¹ and also the magnetisation history has some influence, which could be an explanation of the variation of the open state period. Figure 11 shows a bar-chart, the black one stands for hold times above 2.5ms and the grey one for those below 2.5ms. The height of each bar represents the frequency of each current curse with the specific boost time which results in one of the two hold time ranges. As can be seen, the hold time range below 2.5ms (grey) has more often longer boost periods than those above, what means that the historical magnetisation could come into effect here. Temperature changes do have an influence to the hydro-mechanical system of the oil too. Especially the smooth rise of switching time between 21.5 and 22.5 million cycles resp. 5 hours could be temperature caused. But to reach exactly two different hold time states in that short time as it is at 14 million and 42 million it would have been necessary to control the temperature by heating and cooling, so it was excluded from our point of view. Besides the two switching states, it seems that the variation of the switching time decreases over lifetime.



Figure 11: statistic frequency of switching time depending on coil current course

Possible causes for the abruptly change of the hold time: Differing boost times and unstable magnetic circuit, temperature changes or electromagnetic inferences.

At this point the roots of this problem could not be explained clearly, but one potentially solution to avoid the variation of the hold time was to get a more stable magnetic circuit, which leads to less sensitivity towards the possible causes above. This was realized by manufacturing the cushioning grooves via build-up welding out of amagnetic material on the top side of the solenoid. The duration of the endurance test with the new valve was shorter as before, but long enough to show the reproducibility of the switching time (Figure 12). As can be seen, the peak boost current was decreased and the hold time was enlarged in order to get a more stable process. The run-in period

¹ The level of magnetisation depends on the coil current as well as the gap between armature and magnet. If the boost mode lasts longer, the current is higher and the gap is smaller; also the negative voltage, which should demagnetise the armature, has different influences depending on the gap size. Due to the high sensitivity of the magnetic circuit, it is obvious that the varying magnetisation and magnetic hold forces result in different opened valve states.

only lasted about 3 million cycles and the variation of the total switching time, which starts at 5% valve opening and ends at 95% valve closing, was in a more than satisfactory range of 0.4ms, so that the initial problem can be considered as solved. The requirements were proofed and can be seen in Table 1.



Figure 12: longterm test with the final design of the new FSVi4

Parameter	Value
max. tested switchable pressure differnece	300 bar
max. hydr. switiching time (5-95% vavle opening)	1.5 ms
Time from signalling to 95% opening	< 3 ms
max. hydr. switiching time (95-5% vavle closing)	1.5 ms
Time from signalling to 5% valve closing	< 2.5 ms
Signalling time for max. ballistic mode @ pmax	2.2 ms
Signalling time for min. ballistic mode @ pmax	1ms
Max. ordinary switching frequency @ pmax	200 Hz
Nominal flow rate	5 l/min @ 5 bar
Max. ballistic frequency @ pmax (10% opening)	500Hz

Table 1: measured and verified valve specifications

3. APPLICATIONS

For testing the new valve design in an application, LCM built a new demonstrator which will be explained in the following subchapter.

3.1. DIGI-ACTUATOR

The demonstrator consists of a synchronised cylinder which is driven by a 4/3-way valve² with decoupled metering edges. The power supply is realized by a small accumulator charge system.

3.1.1. 4/3-way block

The 4/3-way valve block was designed in order to create a 4/3-way valve with decoupled metering edges. Figure 13 shows the CAD drilling pattern of the block and the manufactured prototype with the mounted 2/2-fast switching valves.



Figure 13: left: drilling pattern of the 4/3-valve block colour code; black: system pressure line, white: tank, dark grey: side A, light grey: side B, grey: holes for mounting screws; right: prototype of the 4/3-valve with decoupled metering edges

3.1.2. Mechanical design

The mechanical design is based on an existing demonstrator [10]. It was upgraded with a new positioning measurement and the 4/3-way switching valve as described above. Figure 14 shows the hydraulic scheme [11] and the CAD-Model.



Figure 14: left: hydraulic scheme, right: CAD-design of the demonstrator

 $^{^{2}}$ Actually the 4/3 way switching valve with decoupled metering edges is a 4/16 way switching valve, but only 7 of those 16 switching possibilities are reasonable in this case.

3.1.3. Measurement technique and control unit

The pressure will be measured at 3 different positions, namely directly at the accumulator side and on the two cylinder ports. The position will be measured via incremental encoder on the top cylinder side. The control unit is a power panel (touch screen with CPU connected to an IO-system) which is basically an ordinary industrial used PLC.

3.1.4. Control methods

One challenge of digital hydraulics is the noise emission due to the switching between high and low pressure. Depending on the hydro-mechanical system the loudness can reach values that are not acceptable anymore. The switching valves can basically be driven up to 200Hz (ordinary switching mode, 500Hz max. ballistic mode). The idea is to phase shift the valves of the opposite sides, during control. This was already published in connection with pneumatics [12], but with way lower frequencies and also in hydraulics [13], [15]. That means e.g. if the basic frequency is 200Hz (5ms period duration) and the nominal duty cycle is set to 50% the specific valve on side "a" will be in "on-state" for 2.5ms, after that, the valve on side "b" will be switched (Figure 16). This leads to a frequency doubling effect in excitation of the system, which can be heard and even seen on the pressure oscillation. If one of the 2 active valves is closed during the opposite is opened then the movement of the cylinder is forced by the compressibility of oil³. The advantage with frequency doubling is not the octavo band shift of the noises, but the reduced amplitude of the pressure difference between the chambers A and B, which directly influences the cylinder position due to the equation of motion. As a result the smoother movement of the cylinder leads to lower noise. Taking into account that the maximum sensitivity of a human ear lies between two and five kHz [14], those noises with the same amplitude as ones in another frequencyspectrum will be recognized drastically louder. Therefor the frequency due to the doubling effect should not reach this range. The controller was implemented in such a way that both duty cycle and frequency can be changed simultaneously, what means that it can be called "pulse width modulated control with variable carrier frequency" (PWMVCF). An Advantage of this control type is that the generated noise can be transformed into a harmonic melody or at least in the spectrum of ambient noises (microphone \rightarrow FFT of ambient noises \rightarrow first frequency of FFT is the carrier frequency). This was already tested⁴⁾ in the laboratory of LCM.

The position control was implemented via a velocity feed forward and positioning PIcontroller [15]. The velocity feed forward is a look up table which calculates the nominal duty cycle by the given inputs PWM-frequency and nominal velocity. The measurement of the LU-Tables was realized in the opposite way. A sinusoidal duty cycle setting at different PWM-frequencies (100, 200, 300, 400 and 500Hz) lead to the velocities as output. Due to the fact that the feed forward already includes the different control behaviour caused by the time between signalling and valve movement, it does

³ Frequencies above 200Hz lead to an overlapping valve opening, so the movement is given by oil flow, while the peak pressures are still phase shifted.

⁴ The first test of the demonstrator with this idea, by taking the Eigen frequency of the CNC milling as carrier frequency for the control, resulted in a non-audible piston movement.

not have to be considered in the PI-Controller anymore and keeps the controller as simple as possible. Figure 15 shows the recorded velocity feed forward LU-Table at different frequencies. It can be seen, that the inertia and sealing friction (stick slip effect) lead to a hysteresis effect. The light grey dots show the movement during acceleration and the dark grey dots stand for deceleration. In order to have a stable feed forward, the minimum rising and falling curves were taken (black dots) and implemented into the LU-Table.



Figure 15: measured and implemented velocity and frequency depending look up table



Figure 16: illustration of the 2 different control methods at 200Hz and 50% dc; left: synchronous valve control, right: 180° phase shifted control

3.1.5. Results

Figure 17 and Figure 18 show the measurement results of the digi-actuator during different control settings. The black lines show the synchronous valve control and the light grey lines show the PWM shifted valve control. The top table including the first subplot illustrates the predefined carrier frequency (grey boxes) and the sinusoidal position trajectory courses. The measured positions, the duty cycles and the pressure difference between chamber A and B are also shown in this figure. Figure 18 illustrates the target velocity, the duty cycles based on the control, the actual measured velocity (at sampling rate 1 kHz) and the filtered velocity.

The following statements are related on the pressure difference between chamber A and B (Figure 17, lower diagram) as well as the unfiltered piston velocity (Figure 18, third diagram). Obviously, the pressure pulsations are reduced at rising carrier frequency. But the more satisfying result is, as we mentioned in our presumption, that the shifted control mode has way lower pulsations as the synchronous control at the same frequencies. In [15] it was mentioned, that PWM shifted control does not influence the seesaw movement. But in this case the PWM-shifted control reduces the peak to peak values of one oscillation of the pressure difference p_B-p_A to less than 50% compared to those with synchronous control.

4. CONCLUSION

Fast switching valve technology is still not at its limits. Reasonable challenges are prizing and life time duration. The one and only way to get one step further is collaboration between manufacturers, R&D labs and industrial partners. High quantities in production would decrease the prize of components dramatically.

The digi-actuator is an example which shows that the components and algorithms are ready for digital transformation. As already mentioned, the velocity control has a duty cycle feed forward which depends on frequency. If the automation system would save the measured data and recalculate the look up tables for the feed forward, the drive would be some sort of learning by conditioning. It would be also possible to feed the data to some sort of neuronal network for deep learning. Therefor temperature and acceleration sensors could be applied in order achieve better results.

A practical application would be a large digital piloted 2/2 way seat type logical valve (e.g. NG32) where the valve poppet can be controlled the same way as in the system above. Thus, such a valve can be operated proportionally in a simple and low cost way.. In addition it is possible to open the proportional valve like a switching valve, by actuating the FSVi4.1 logic in ordinary switching mode.



Figure 17: illustration of nominal frequencies (grey boxes), nominal position trajectory; measurement results like cylinder position, pressure difference, positioning error for the two different control types (synchronised control \rightarrow black and phase shifted control \rightarrow grey)



Figure 18: target, measured and filtered velocities for the two different control types including the duty cycle (synchronised control \rightarrow *black and phase shifted control* \rightarrow *grey)*

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