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Pilot Operated Cartridge Valve - Dynamic Characteristics Measurements for Energy Efficient Operation and Application

Master's thesis submitted for examination of the degree in Master of Science (Technology)

Espoo 10.09.2018
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Abstract of master's thesis

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Title of thesis Pilot operated Cartridge valve - dynamic characteristics measurements for energy efficient operation and application

Degree programme Master of Science in Mechanical Engineering

Major/Minor Machine Design/Mechatronics **Code** K3001

Thesis supervisor Professor Matti Pietola

Thesis advisor(s) Jyrki Kajaste

Date 10.09.2018 **Number of pages** 69+21 **Language** English

Abstract

The two-stage on/off valve scheme is studied in this thesis which is able to rapidly actuate within the fraction of a second for directional flow control applications. Also, it was able to replicate characteristics of direct operated solenoid on/off valves in the same category, with the addition of higher flow rate capacity up to 200 l/min and above with pressure drop under 5 bar. The valves utilized for piloting of the hydraulic operated cartridge valve are the normally closed and normally open direct solenoid operated on/off valve. In the experiments, the internal piloting structure is adopted to avoid external pressure source and keeping valve operation dependencies minimum to the electric power. The switching time was found to be lowest at 100 bar, that is approximately 12 ms for opening and 30 ms for closing from the point of activation signal. It is also observed that the flow rate has negligible effect on the switching duration and changing the poppet area ratio to 50% of A_x from 96% can reduce down the closing time. Further, it was observed that the internal piloting has its drawbacks which creates a closed loop between main valve metering edge and pilot chamber. This resulted in oscillations in form of poppet movement near the dead end on either side of the stroke, and also if the system is not stiff or due to pressure waves traversing in the long hoses. Another positive outcome from the measurements showed that different switching methods such as, intermittent, continuous and pulse switching can be performed in a controlled manner, but the study was limited to the capability of the valve scheme. Moreover, the simulation model is also built in the Matlab/Simscape environment to refine the valve model based on the experiment results for further measurements that were limited by the physical system. Additional simulations are conducted to reduce the marginal difference between the opening and closing duration by restricting the stroke length to 5 mm, contrary to the original 8.5 mm, at 200 l/min and the switching duration was considerably reduced by half of the original duration. The experiments were conducted for flow in one direction only, whereas in simulation the bi-direction flow capability is also carried out to displace an actuator with constant velocity, while lifting, holding, and lowering of the load to certain position.

Keywords Slip-in Cartridge valve, pilot operated, on/off valves, high flow rate, low-pressure drop, rapid switching measurement, Simscape model, bi-directional flow cartridge valve

Acknowledgments

The subject matter of the thesis is part of the Hydraulics Hybrid Project, being pursued at Aalto University and the Tampere University of Technology. The topic of study was offered in line with my interest in hydro-mechanical systems, covering numerical simulation, test setup and experiment work. The fluid power basics subject was my first aspiration as well as subsequent courses to advance my interest and career path towards hydraulics.

I would like to thank Prof. Matti Pietola as supervisor and advisor Jyrki Kajaste for accepting me to work in this area, particularly, the interesting mix of modeling and experiment work related to the on/off valves. Along with that, I would also like to extend my appreciations to Tapio Lantela for supplementing the valuable resource and information towards my thesis, as well as, Anttii Sinkkonen and Otto Gottberg for the support in the lab and practical areas. Also, I would like to acknowledge Heikki Kauranne for the valuable feedback on my report and other colleagues in fluid power group for creating a sense of compelling and conducive environment for the research work.

Espoo, 10.09.2018

Saad Saleem

Nomenclatures

A_A	Poppet surface area at port A	[m^2]
A_B	Poppet surface area at port B	[m^2]
A_o	Area of orifice	[m^2]
A_X	Poppet surface area at port X	[m^2]
$A_{(X)}$	Orifice area function of displacement	[m^2]
C_d	Discharge coefficient	[-]
C_v	Velocity coefficient	[-]
d_s	Stem diameter	[m]
D	Hydraulic diameter	[m]
E_{Tloss}	Total energy loss	[J]
f_D	Darcy friction factor	[-]
F_{flow}	Flow forces	[N]
F_{fi}	Flow forces closing	[N]
$F_{friction}$	Poppet friction forces	[N]
$F_{\mathrm{p}net}$	Net force on the poppet	[N]
F_{spring}	Valve's preload spring force	[N]
F_{seal}	Seal friction force	[N]
$F_{viscous}$	Viscous friction force	[N]
I_{NC}	Current to NC valve	[A]
I_{NO}	Current to NO valve	[A]
m	Mass	[kg]
L	Coil inductance	[H]
l_c	Length of cylindrical pipe	[m]
N_R	Reynold number	[-]
p_o	pre-charge gas pressure	[pa]
p_1	Minimum system pressure	[pa]
p_2	Maximum system pressure	[pa]
p_A	Pressure at port A	[pa]
$\mathrm{p_B}$	Pressure at port B	[pa]
p_{crack}	Cracking pressure	[pa]
$\mathrm{p_X}$	Pressure at port X	[pa]
q	Flow rate	[m^3/s]
q_p	Flow rate in pilot valve	[m^3/s]
q_v	Flow rate in valve	[m^3/s]
$\mathrm{q_{vm}}$	Flow rate through main valve	[m^3/s]
R	Solenoid resistance	[Ω]
s_{CV}	Stroke length of cartridge valve	[m]
T	Time constant	[N]
V	Volume	[m^3]
V_p	Pilot chamber volume	[m^3]
V_{pvs}	Voltage supply to pilot valve	[V]
$V_{o,g}$	Gas volume at pre-charge pressure	[m^3]
$V_{1,g}$	Gas volume at min. system pressure	[m^3]
$V_{2,g}$	Gas volume at max. system pressure	[m^3]

ν	Kinematic viscosity	[cSt]
v_{avg}	Average fluid velocity	[m/s^2]
X	Poppet displacement from seat	[m]
Δp	Pressure difference	[pa]
Δp_p	Pressure drop across pilot valve	[pa]
Δp_{CV}	Pressure drop across cartridge valve	[pa]
ΔV	Change in volume	[m^3]
ρ	Density of fluid	[kg/m^3]
Α	Poppet angle from seat	[degree]
Ø	Angle of flow jet	[degree]

Abbreviations and Acronyms

AI	Analogue input
AO	Analogue output
CAD	Computer assisted design
CV	Cartridge Valve
DAQ	Data acquisition
HPA	High-pressure accumulator
LPA	Low-pressure accumulator
NC	Normally closed
NO	Normally opened
PD	Pressure difference
PRV	Pressure relief valve
PSU	Power supply unit
PS1	Inlet pressure transmitter in valve block
PS2	Outlet pressure transmitter in valve block
PS3	Inlet pressure transmitter
PS4	Outlet pressure transmitter
PSP	Pilot pressure transmitter
PUV	Pilot unloading valve
TPS	Temperature sensor

Table of Contents

1 Introduction

This introductory chapter provides an overview about the general background of the hydraulic valves, the trend advancement in hydraulic valves systems based on the demand in conserving energy and the implementation of new technologies in the pursuit of efficient operations. Further, it covers the scope of the study undertaken and underlines the intended aim and objectives of the research. Also, the literature study is well covered to supplement this research and to support the experiments and simulation conducted.

1.1 Background

The hydraulic systems are extensively used in machinery ranging from hydraulic press, mobile machinery or renewable energy harvesting systems, where high power density is desired. All of the fluid power systems involve some form of hydraulic valves, be it flow control or directional valve, to block or direct the flow to the respective channel. Narrowing down to the directional flow control valve, which is an integral component in majority of the fluid power system is the subject of focus in this study, to replace the conventional proportional valve with more efficient, modular and on/off based valving scheme. Keeping in view with the increasing demand in stringent emission regulations, the energy efficient hydraulic powered machinery is increasingly being implemented to reduce the power consumption of machines and moving towards hybridization and electrification.

As most of the hydraulic system still relies on the proportional directional control valves, which are bulky in size, expensive, susceptible to the leakages, tight tolerances in manufacturing, limited flow channels and bigger the size for higher flow rate leads to lower response time. While, pressure loss during continuous flow is also a concern due to irregular or sharp corners and directional control of the flow through constrictions causes drastic losses in pressure and also flow loss through overlapping in channels, consequently excess energy consumption. Out of this concern, the two important parameters, power loss and response time is the criteria to be studied in the suggested on/off valve setup. As, the increase in flow rate proportionally increases the power losses and in applications where large transient flow rate requirement has to be met with lower switching time such as emergency kill switch to stop machines, hydraulically operated high voltage circuit breaker or hydraulic catapult system are one of the few examples of intermittent switching [1].

Based on the demanding performance characteristics, such as high flow rate capacity with low pressure drop, internal tightness to prevent leakages, modular build, and highly responsive, the potential use of on/off switching valves could address it. Another advantage is that, a set number of on/off valves can produce more number of channel path combinations than the multi-directional proportional valve [2].

The applications on/off valve scheme is not only limited to replicating conventional valve, but also in the scaling the performance of discrete fluid power systems, in which a number of on/off valves are installed in parallel that requires the similar properties for efficient operation as to the existing low capability system [3]. The extensive advancement in hardware control and electrification of machinery has made the discrete fluid power mechanisms to be a viable system for operation of the equipment, and the use of such technology is documented, for example Multi-pressure actuator system [3] and Wave energy converter [4].

Considering one case in the listed digital hydraulic system, in particular Multi-pressure actuator it employs solenoid valves with limited flow capacity and a particular actuation time, leading to certain power capacity available at the actuator, which in this case is 10 kW. As the size of solenoid valves increases so does the maximal flow rate capability along with it but leads to reduced response time in opening and closing, as well as increase in electrical energy consumption.

The factors affecting the switching time are not only valid to on/off solenoid valves but also for the hydraulic operated cartridge valves. Due to increase in area of metering edge, the pressure drop across the valve might be minimum at higher flow rates, which means large poppet diameter, stroke and pilot volume can slow down the opening/closing timing. The relationship between actuation timing and valve performance is well documented in the following papers [5], [6].

In order to scale the Multi-pressure actuator system for bigger machinery applications, the valves will have to be replaced ones with higher flow capacity, which leads to higher switching time. This creates a space to conduct research on high-performance digital valve scheme, while, replicating the similar characteristics and minimal switching time.

1.2 Research problem

Commercially, lots of switching valve options are available from bulky proportional spool type valves to small digital solenoid valves. The proportional directional control flow valve has been the long-running one and the capability to control the flow and position of the spool, even for the slightest of actuation. Since, the size, capacity, operation and flow channels of proportional valves varies depending on the requirement, the definite specifications are not quoted in here, but comparative study is presented in next section in line with the on/off valve. Based on the volume geometry, the digital valves on the lower end of the size scale, are capable of actuating the flow as low as 0.3 l/min with 1.5 ms response time, named Proto10 in the article [7] and another one recently developed has maximum flow capacity of up to 9 l/min and switching time within 1.0 ms [8], albeit still in the research phase.

The state of art digital valves mentioned are direct solenoid operated, former one developed in Tampere University of Technology (TUT) and latter one in Aalto University, and designed to operate as either on or off condition within mere part of a second. Moreover, digital valves in a combined set, installed in parallel can actuate a large flow within short switching duration. On the other hand, requires utilization of advance control hardware and digital electronics to operate, as well as, intricate manufacturing processes to manufacture the valves itself and the manifold block with flow channels.

Also considering the energy efficient switching operation as part of the research, besides pressure drop due to passing hydraulic fluid, the electric power consumption and leakages also has to be considered. So, basically bigger the size of solenoid valve more will be the current consumption in switching open the valve's spool/poppet and the holding current. Similarly, the higher pilot volume will be required for hydraulic operated cartridge valve.

Furthermore, the leakages during static and transient operation, that is the the flow lost from pilot volume or through main valve to the tank. The minimal leakages is an important factor to account for, in valve dynamics study for controlled directional flow control.

The previous paragraphs explain the proportional valve on bigger scale and digital valve in small size scale, which calls for mid-range size valves capable for higher performance application. It can also emulate the proportional valves characteristics and digital operation mechanism of switching, that is on/off, under the defined criteria in the Background section.

1.3 The scope of the study

In this thesis study, the suggested scheme is to utilize the two-stage valve construction having two switching positions and two flow directions utilizing hydraulic cartridge valve, capable of operating at pressure up to 350 bar and flow rate capacity of the main stage 325 l/min. The two-stage scheme has been utilized in previous researches [1], [4], [5] with the operating principle based on pressure control, through piloting of hydraulic operated cartridge valve (CV).

In the listed criteria range the existing commercially available valves for pilot operation and specifically build as screw-in solenoid models for mobile machinery are SAE sizes. Similarly, for the hydraulic operated 2/2 way valves are slip-in cartridge valves starting from size NG16 and higher based on the flow rate capacity and the dimensional volume. Both are available as stock components from the variety of manufacturers with various operating ranges and are similar in functional use, but comes with slight differences in the performance rating of flow and pressure drop. Usually, the valves in question, also dubbed as logic valves, can be either switched on or off to pass or stop the flow. Based on required performance rating and switching speed, the SAE size 5 for pilot operation is selected and NG16 for the main cartridge valve for the research purpose. The technical data is further explained in the Setup description chapter under components heading.

Aim and objectives of the research

The primary objective of this study is the successful operation of the proposed two-stage valve scheme in the hydraulic system. Moreover, the switching valves have to be leak free, able to actuate within the fraction of a second with the least possible delay, test-retest reliability and above of all, low-pressure drop across the valve. This characteristic has to be ensured in order to provide well-researched data for the use in digital actuation system in machinery such as off-road mobile machines, industrial automation, and manufacturing processes operating at high flow rate and high pressures systems.

Moreover, along with the performance rating of the valves, the dynamic characteristics are also covered in a transient state such as switching timing, an important factor for the fast operation of the system. This study presents the case setup, numerical modeling, experimental setup and the analysis of the combined performance of the two-stage valve configuration to create a basis for the future implementation and also to have a working model as a benchmark. This is done predominantly to analyze the dynamics of the valve to target high power operating system of more than 10 kW, high flow rate greater than 30 l/min up to 200 l/min, low pressure drops across the main stage valve, low switching time as well as possibility to capture the transient flow characteristics in the system with different operating pressure ranges.

1.4 Comparative literature study

In this section, the comparison between proportional directional control valve with two or more flow channels and on/off two-way valve is done. As directional control valves also encompass the fixed pressure and control function valves such as, shuttle and check valves that have been excluded and study is limited to externally adjustable ones during operation.

1.4.1 Conventional proportional and on/off type valves

The main difference between the conventional directional control valves and the on/off type valve is the design and build of the valve block and the actuating rod, i.e. spool or poppet itself. The conventional flow directional valve has been in used for long, and are significant for three ways or four ways channels (up to five ways used in mobile hydraulics) or also letting two flow channels open at once, contrary to two-way flow cartridge type valve which requires to be operated in the set. Simple schematic and channel path shown in Figure 1 and Figure 2.

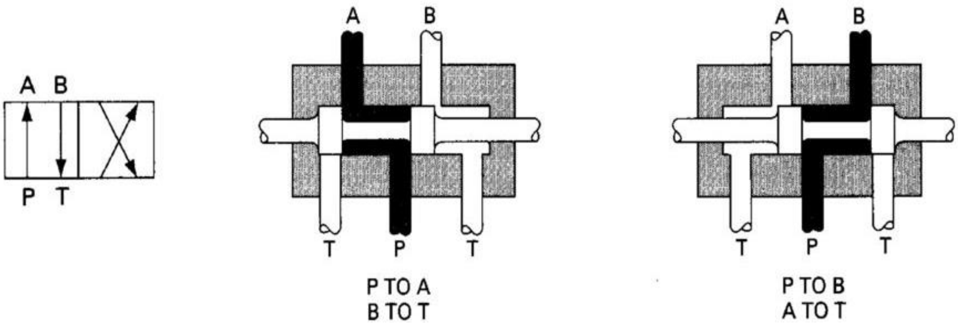


Figure 1. Schematic and graphics of conventional four-way spool valve [9]

Moreover, the proportional valve actuation mechanism can be activated through a manual lever, solenoid or pilot pressure with preset position equipped with spring, on the other hand on/off type flow directional valves are also solenoid or hydraulic pressure operated, combined with the bias preload spring force.

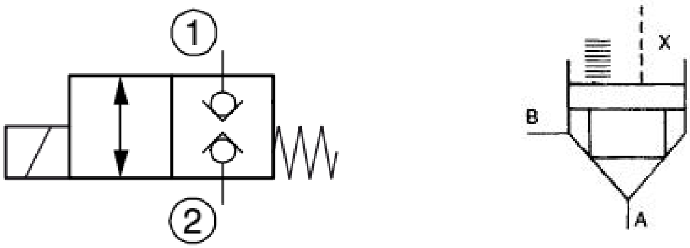


Figure 2. Schematic of on/off type solenoid valve and Cartridge logic valve

Although the servo/proportional valves are still known for good controllability, they are prone to cavitation, high power losses, sensitive to contamination and are costly [10]. Further, a ballpark figure can be taken from the paper [11], in which throttling losses have been identified up to 70 bar or 33% of 210 bar supply pressure at a nominal flow rate of 90 l/min. Some other shortfalls have been provided in the textbook [12], in which different center types (normal spool position), for example, causes high leakage flow at neutral center flow position and pressure peaks during switching with the closed center.

With the previous information covered, the possibility of using combined two stage on/off valve does seem to be a prospective alternative option for applications discussed in the background section considering lower cost, smaller size, less prone to contamination and almost no leakage depending on poppet and seat design. In addition, the down sides as reported in the paper [10] published in the year 2003, and a known fact, that on/off or cartridge logic valve tend to generate noise during actuation, may induce pressure peaks in the system and controllability issues, however, since several controlling methods have long been devised. To support the latter statement, the experiments are documented within the same paper as well as in few others, published in recent years [3], [13], [14], to achieve energy efficient operation with the high-end position and velocity control.

Figure 3 has been taken from the paper [2], in which the logic valves have been used to build and replicate the functions of conventional four-port directional control. This attempt not only achieves the quality of being leak proof but is also capable of making a greater number of switching positions and flow path combinations. Such as the different central positions and flow channels, for example, tandem center, blocked ports, open center and so on.

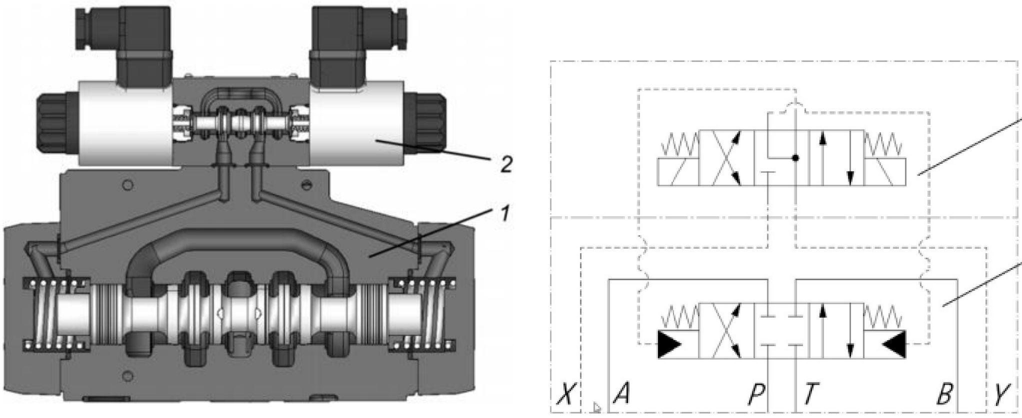


Figure 3. Schematic diagrams of conventional pilot operated flow directional control valve. 1 -main stage spool valve, 2- pilot stage solenoid operated valve [2]

Moreover, the combined set of logic valves block is comparable in size with the conventional proportional one, using six cartridge valves and internally piloted solenoid valves and can be mounted on ISO standard mounting plate, as shown in Figure 4.

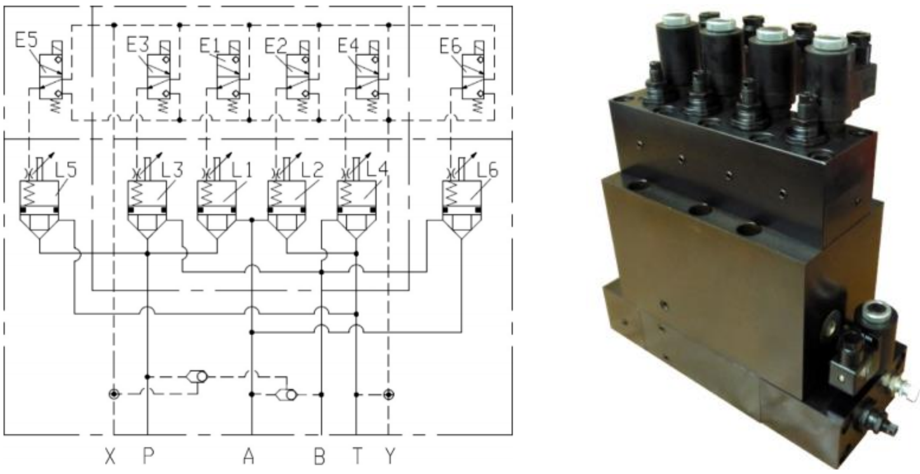


Figure 4. Directional control valve schematic and real block named L6-WEH22 [2]

In the same paper [2], a Bosch Rexroth two-stage pilot operated electro-hydraulic actuation proportional valve model WEH22 is used for comparative study. In this valve the solenoid operated pilot spool controls both sides of the main spool to shift to the desired position.

The trend of using slip-in cartridge valves gives the possibility for not only easy maintenance and upkeep, but also getting along with the digital controlling technologies. Another recorded benefit is the low-pressure drop, at least for AB and PT flow path at 300 l/min, which can be deduced from Figure 5, of around 4−5 bar (0.4−0.5 MPa). The pressure losses bar graph has been illustrated and compared, as shown in Figure 5, for both conventional WEH22 model spool valve and L6-WEH22 logic valve block and validated using both CFD analysis and practical measurements for different flow path conditions.

This paper greatly established the case and imperativeness to conduct measurements in this thesis study and present data to put forth the use of logic valves for applications in independent metering of flow and discrete fluid power systems.

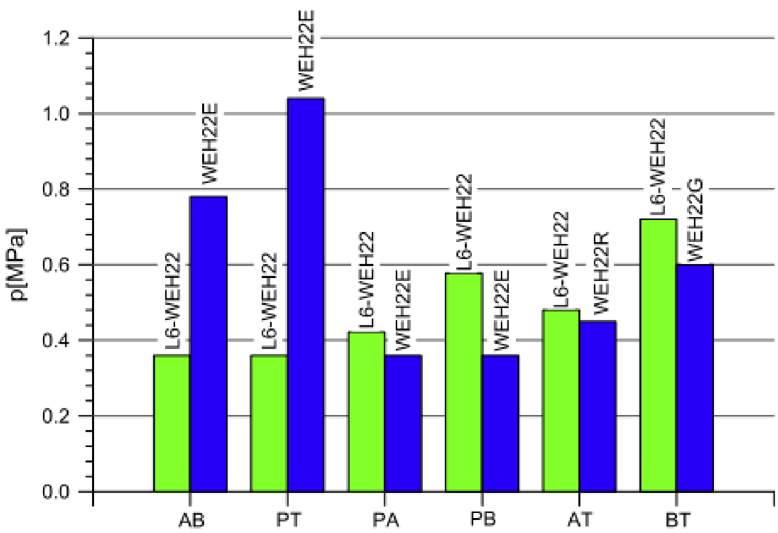


Figure 5. Pressure losses comparison at 300 l/min between L6-WEH22 valve block and conventional WEH22 block with different central channel models [2]

Overall, hydraulic operated cartridge valves can be utilized for different roles in a system with the piloting function, as follows:

- Pressure control: for pressure relief, unloading or sequence valve configuration
- Directional control: control of flow direction in either way
- Check valve: can imitate pilot operated check valve or the standard one
- Pulsating or discrete flow: with the use of the effective controlling system, the flow of fluid can be controlled depending on working frequency and actuation.

1.4.2 Comparative analysis of 2/2 directional valves

As the topic of study is mainly focused on the dynamic characteristics of on/off type two-way two-position (2/2) directional flow valves, both solenoid and hydraulic operated, it is important to do comparative analysis between both. The comparison is based on the performance and switching characteristic outcome and the capability to incorporate in previously mentioned systems, such as discrete fluid power applications related to digital hydraulics or one-time switch-on or cut-off actuation for further utilization.

The comparison analysis is based on the published articles and catalog parts for reference purpose. For example, already published researches related to 2/2 valves and/or multi-stage scheme involving on/off valve, be it custom designed based on a particular requirement or catalog parts, are compared in Figure 6. The figure gives a sense of available options and scheme for the potential setup of creating a multi-stage valve for faster switching and desired performance scenario.

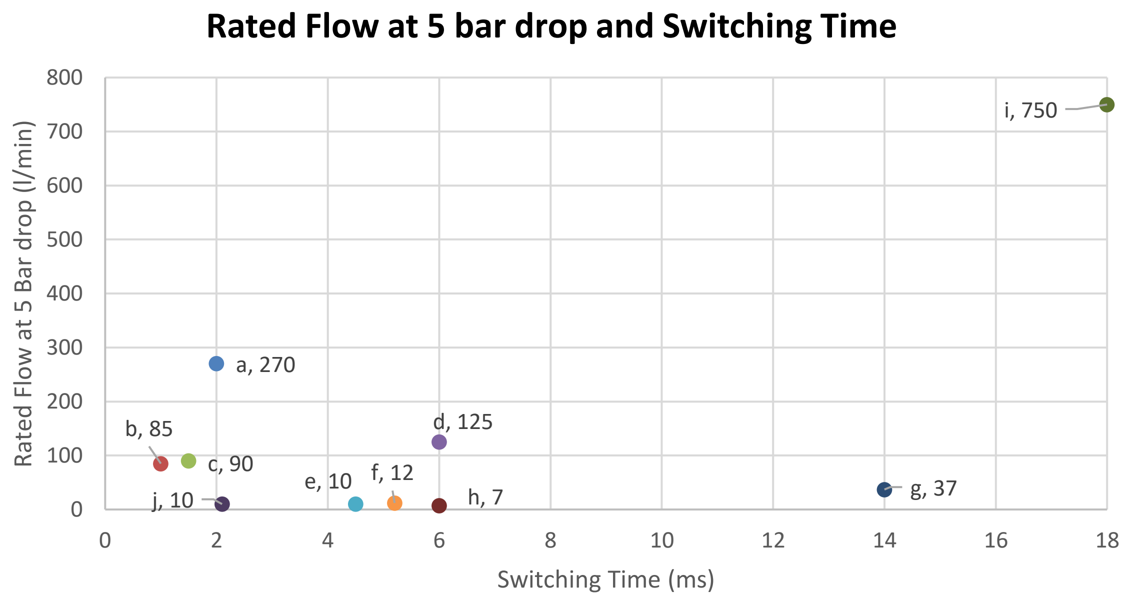


Figure 6. Switching time comparison of hydraulic cartridge and on/off solenoid valves plotted as standalone or multi-stage valve scheme for rated flow at 5 bar pressure drop. Each valve is denoted with alphabet has been summarized with references in section below

The Figure 6 presents the scatter plot of various two-way flow direction valves with flow rate at y-axis at 5 bar pressure drop and switching time on x-axis and the adjoined label denotes to the explanation below and rated flow values. The flow capacity at the quoted pressure drop across the valve has become the benchmark to compare the listed valves.

Although switching time data has been plotted for each valve, this characteristic is influenced by several system conditions; such as pressure difference across the valve, oil viscosity, and flow rate. Moreover, only the opening time has been plotted at certain specific conditions to give a holistic overview of the currently available options. Therefore, various switching times at different system pressure and flow rate conditions can be found, for both opening and closing, in the researches and also explained in the later text.

There are also valves with better switching capacity of less than 2−3 ms, reported in the dissertation [15] table number 3, but excluded here due to reduced maximum flow capacity (under 5 l/min) per one valve or other limitations. Second thing to note that some valve timings are quoted for boosted actuation, which defines as a higher voltage applied than the holding voltage for rapid switching until the full stroke takes place.

Plotted valves summary

The plotted valves have been further explained below. The list sets precedence and provides valuable information in conducting this thesis with regards to experimental setup, performance analysis, benchmarking and concluding the results.

a. Parker CE016 cartridge valve, with a stated pressure drop of 5 bar in the technical catalog at 250 l/min [16] an approximated 2 ms of opening time has been recorded through simulation in Simscape. The boundary conditions were set similar to valves in the plotted group, whereby pilot pressure was set to be zero bar, and 200 bar supply pressure at port A. The quoted time duration is until when the flow rate saturates at the partial stroke. Further, a detailed analysis is done in the latter part of the thesis.

b. Multi-poppet valve by Winkler, employs a multi-poppet design to obtain the desired flow requirement, 85 l/min at 5 bar was achieved and piloted using a 3/2 fast switching valve. The simulated switching time is shown to be 1 ms at 5 bar pressure difference [5], even though the timings were measured at several levels of pressure difference ranges, but the response was not accurately recorded. Also to note, the switching time is the main stage's (poppet) opening time from position 0 to 1.

c. Hörbiger plate valve by Winkler, utilizes the metering edge (fluid port) design based on Hörbiger plate to achieve small stroke and ultimately short switching time. At 200 bar pressure difference, 1.5 ms of opening time of the main stage valve was recorded [17]. The opening curve of the valve is severely influenced by the pressure difference after the valve starts to displace from its seat, the further valve motion timing is affected due to drop in pressure difference.

d. Bi-directional valve for wave energy converters, which also uses multi-poppet technology to reduce pilot volume and have poppets with smaller diameter for better switching capabilities used in the discrete fluid power system. In the paper [4], the intended valve design is for 1000 l/min at 5 bar pressure drop, but the functionality test has been performed at 125 l/min. However, several switching times have been recorded at active and passive switching conditions during opening and closing. The main stage valve is piloted by 3/2 directional solenoid valve and a switching time of bi-directional valve for closing was found to be 15 ms, active switching opening time to be about 6 ms.

e. Bucher WS22 Size 5, is a solenoid operated on/off valve, which is utilized in this thesis study, with listed actuation time for opening and closing to be 6 ms and 10 ms, respectively. One thing to note is that the actuation time is based on the excitation voltage, rather than the supply voltage. The rated flow capacity is around 10−5 l/min, which makes it suitable to use it as a piloting valve.

f. Hydac WS08W-01, is an alternative solenoid on/off valve, but with higher actuation time of 5.2 ms and 12 l/min rated flow.

g. Bucher WS22GDA-10, is a bigger version of size 5 valve with higher rated flow of about 37 l/min and boosted actuation response time of 14 ms. For reference purpose, it is important to document the alternative option, but due to small pilot volume requirement and expected lesser combined switching time, the use of this valve may exceed the objectives defined in the thesis.

h. Bosch Rexroth SEC6-3/3, is basically a solenoid operated valve with two poppets making it a three-way, three-position valve with a rated flow of about 7 l/min and 6 ms boosted response time. This particular valve is available with its own boosting device and potentially can be used for piloting pressure between high and low-pressure lines. It is not being considered for lower flow rate than Bucher valve and flow density.

i. Three stage valve by Bing Xu, is a three-stage valving scheme, that allows rapid switching of comparably high flow rate flow between the load and pressure line. The reason of including this scheme provides a valuable case study which itself includes multi-level switching to have low actuation time and high flow rate [1]. In reverse order, the main stage spool valve requires a larger pilot volume to move the main spool due to its size, for high-speed actuation to connect the pressure line to load and load line to low pressure or tank. The piloting is done using cartridge poppet valve for high flow rate and low-pressure line, keeping in view the least pressure losses even during switching. The schematic is shown in Figure 7,

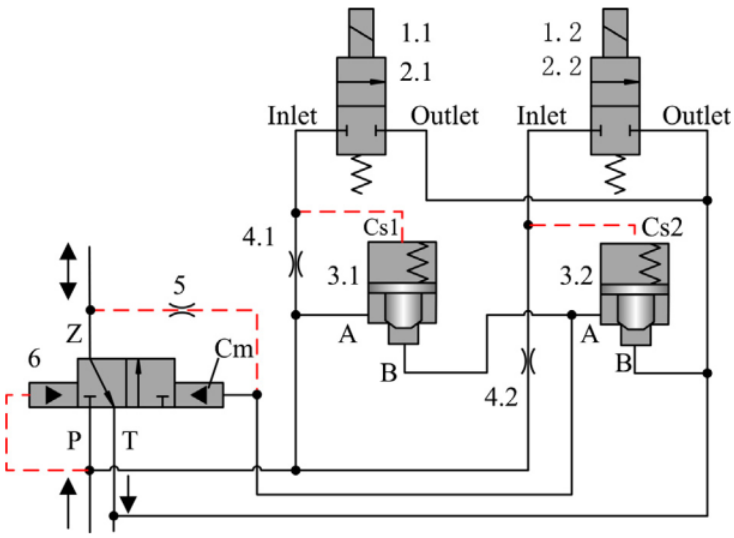


Figure 7. Three-stage valve scheme, 1) High-speed solenoids, 2) Pilot valves, 3) Secondary valves, 4) Secondary valve control orifices, 5) Main valve control orifice, and 6) Main valve [1]

Continuing with 'i' valve scheme, the need for this kind of high-performance valve meeting high flow capacity, pressure and instantaneous supply and the cut-off is required in applications as mentioned in the article [1]; such as high voltage circuit breaker. From previous valve listings, it is evident that the higher the rated flow, the longer the switching time because of the larger size of the sub-components. The main stage spool valve is a 3/2 way valve which can be switched in either direction by the combined pilot and secondary valve for each side. Figure 8 shows the supply flow rate to the measured switching time and the graph includes the start of switching from the point of activation command until maximum flow rate is reached.

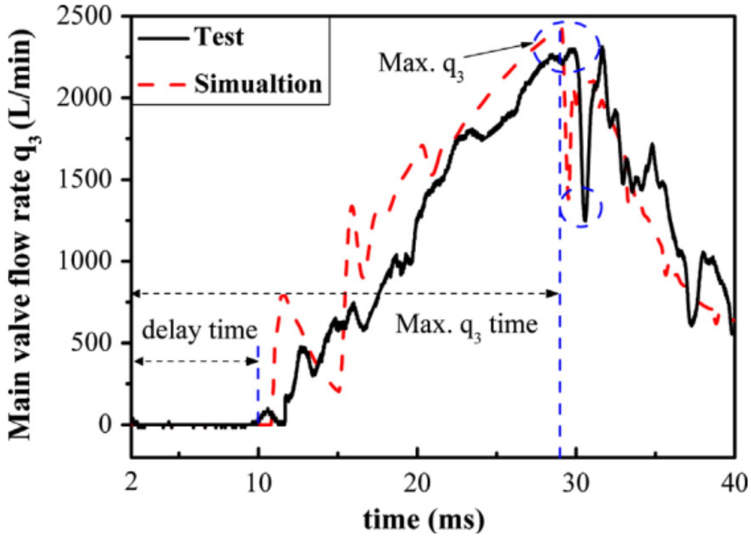


Figure 8. Main stage flow rate, q_3, of three stage valve, to switching time duration [1].

j. LCM 3/2 by Winkler, is a poppet feedback type valve which can output signal to its real time position. This valve can replace two 2/2 valves (one normally closed and another normally open) at pilot as one combined valve, with switching time less than 2 ms and 10 l/min at 5 bar pressure drop. The only constraint specified with this valve in the article [5] is the limited tank port pressure (< 20 bar), which could be limiting factor during internal piloting for bi-directional capabilities, if conceptually acquired, to reduce piloting time to be used in the thesis study.

Further technical specifications for the valve f and g listed in Appendix 1. From listed review on different valve capabilities and characteristics, the following valve parameters have to be taken in consideration for 2/2 valves for analysis and measurement aspects:

- Maximum flow capacity
- Maximum operating pressure
- Nominal flow rate at 5 bar pressure drop at main valve*
- Pilot volume requirement
- Actuation method
- Switching time
- Switching frequency
- Leakages – Pilot valve, Main valve and overlapping leakages in pilot valves.
- Flow density**
- Power consumption (Electrical/Hydraulic Pressure or loss of flow)
- Fluid cleanliness

*The 5 bar pressure drop Δp at nominal flow has been used as the reference size and performance of the 2/2 cartridge on/off valve in researches and adopted in this thesis too.

**Flow density attribute is an important sizing factor in defining the maximum flow rate capability over the volume dimension for a compact system design, but not considered in this study.

The above-mentioned parameters are mostly provided in the technical data sheet of the products available commercially. However, few characteristics such as switching time or leakages are measured in static conditions, while pressure-inducing effects on the system has to be studied because of fast actuation. Keeping in the view of high-performance requirement to have a higher flow rate in shortest possible switching time and least pressure drop, the multi-stage valving scheme has to be analyzed.

The use of multi-stage valve scheme brings flexibility in using piloting valve with different specifications, depending on the application. But with added complexity such as, its own addition of switching times, limited maximum switching frequency, pressure losses, overlapping and limited flow rate capacity.

1.5 Objective and requirements

The thesis topic has been pursued on a sub-system level, limited to the measurement and analysis of the dynamic characteristics of the pilot operated logic valve. From the reviewed literature, the requirements can be clearly defined based on project objectives, the order of testing, expected issues and occurrences which may affect the results and optimization.

Currently, the operating requirements have been formed by making the Multi-pressure actuator system [3] as a targeted application and existing one as a reference, but also applicable for stand-alone intermittent switching besides discrete fluid power system. The thesis study explores the performance results conducted on the multi-stage on/off valve setup to achieve operating conditions listed below:

- Higher flow rate capability (0−200 l/min)
- Operating at high-pressure difference across the valve (8 bar−80 bar)
- Combined switching time (< 15 ms [opening/closing] for full stroke - 0−100%)
- Main flow pressure loss across CV (≤ 5 bar up to a maximum of 200 l/min)
- Leakages should be negligible in main valve (Port A <−> Port B)

Above parameters have been defined based on currently available system component capability, available technical information and literature studies. Based on the objectives, further static and dynamic characteristics results can be obtained and have been analyzed in the measurements and results in the section related to individual component functionality and behavior, the effect on the system such as transient flow, pressure peaks, switching time dependency, losses, leakages, and delays. However, non-dynamic characteristics such as valve size compactness, contamination resistance, have not been considered.

First of all, the case of uni-directional flow setup, depicted in Figure 9 shown without bypassing, safety valves and additional components, has been studied for both cases of intermittent (one-time) switching and continuous switching (also known as frequency switching), which could potentially be viable for the high-performance hydraulic switching system.

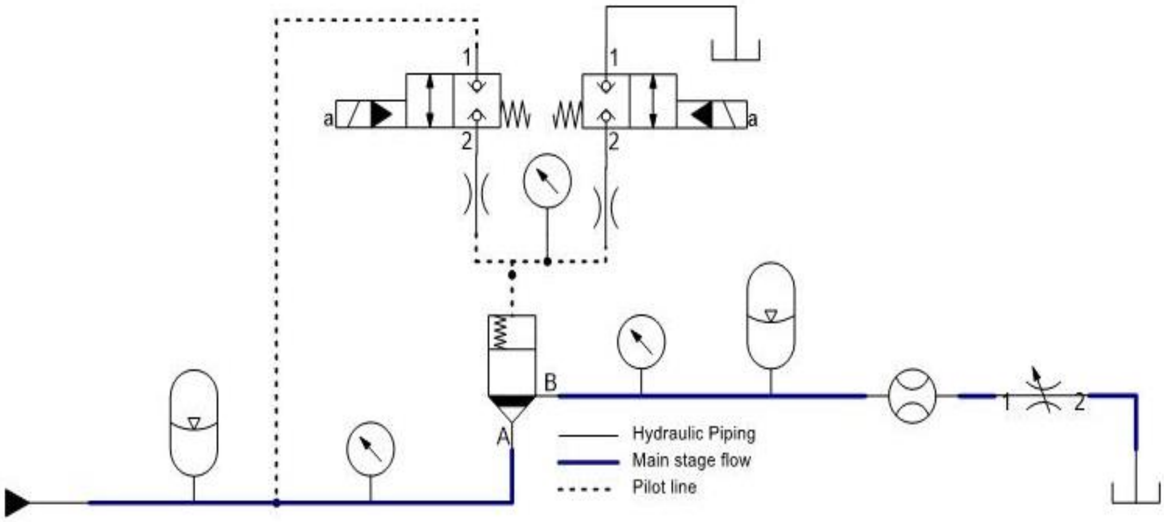


Figure 9. Simple uni-directional flow scheme

Second, the case of a bi-directional valve setup is performed, but only limited to simulation based on the outcome of the first setup, for the potential application use, has been studied. The flow direction is from high-pressure line (port A) to low-pressure line (port B) and to the actuator, and reversible with the change in pressure levels. Simple scheme setup is shown in Figure 10, and further explained in the Chapter 5.

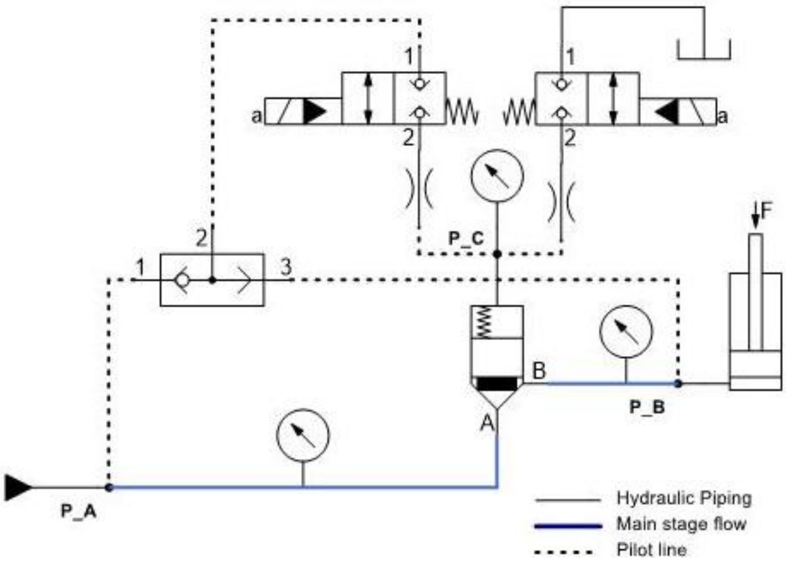


Figure 10. Simple bi-directional flow scheme

1.6 Outline of the thesis

The thesis report has been structured in a way, from presenting the case study to relevant articles review, defining objectives, and methodology. Further it documents the building of system in the laboratory for the experiment purpose and the simulation model to be able to correlate with the experimental results and comply with the objectives.

- Chapter 2 explains the intended basic scheme of work as well as the physical components and sub-systems involved in it to perform the desired experiment.

- Chapter 3 includes the Matlab/Simscape simulation model to emulate the actual system and also act as reference mark for the measurements acquired from the experiments. It also provides the detailed parameters required for modeling and that can be run at higher operating conditions than the actual setup.

- Chapter 4 presents the analysis of measurements and results from both numerical simulation and the experiment, to showcase the outcome of the uni-directional flow valve switching characteristics.

- Chapter 5 is the second phase of this study, in which the simulation model is built to simulate the bi-directional flow capabilities based on the results from the experiment results on uni-directional flow setup.

- Chapter 6 includes the conclusion of the thesis and future recommendations with respect to space for further improvement and applicability.

2 Setup Description and Operating Principle

This chapter contains the details of the individual physical components used, their technical specifications, working principle, sizing and adjustments which were done for the particular experiment. The experiment setup is prepared specifically to measure the dynamic characteristics of the valve in question.

The components selected, inclusive of sensors, hydraulic piping, support structure, have been used to withstand the intended operating conditions. Additionally, solenoid valves have been tuned to operate within shortest possible switching time and the CV is a stock part, acquired based on the required performance rating, with maximum flow rate capability of up to 325 l/min and further justification on selection of components are covered in later headings.

The setup shown in the Figure 11 has been assembled in laboratory close to the power pack (flow pumps) with firm steel structure built in to resist possible vibrations and inertial effects to keep up with the high flow rate and pressure conditions and also to stabilize the CV's poppet transient movement during the switching.

Figure 11. Laboratory experiment setup. Schematic in Figure 12

The physical setup is represented in schematic form in Figure 12, and the corresponding components in Table 1.

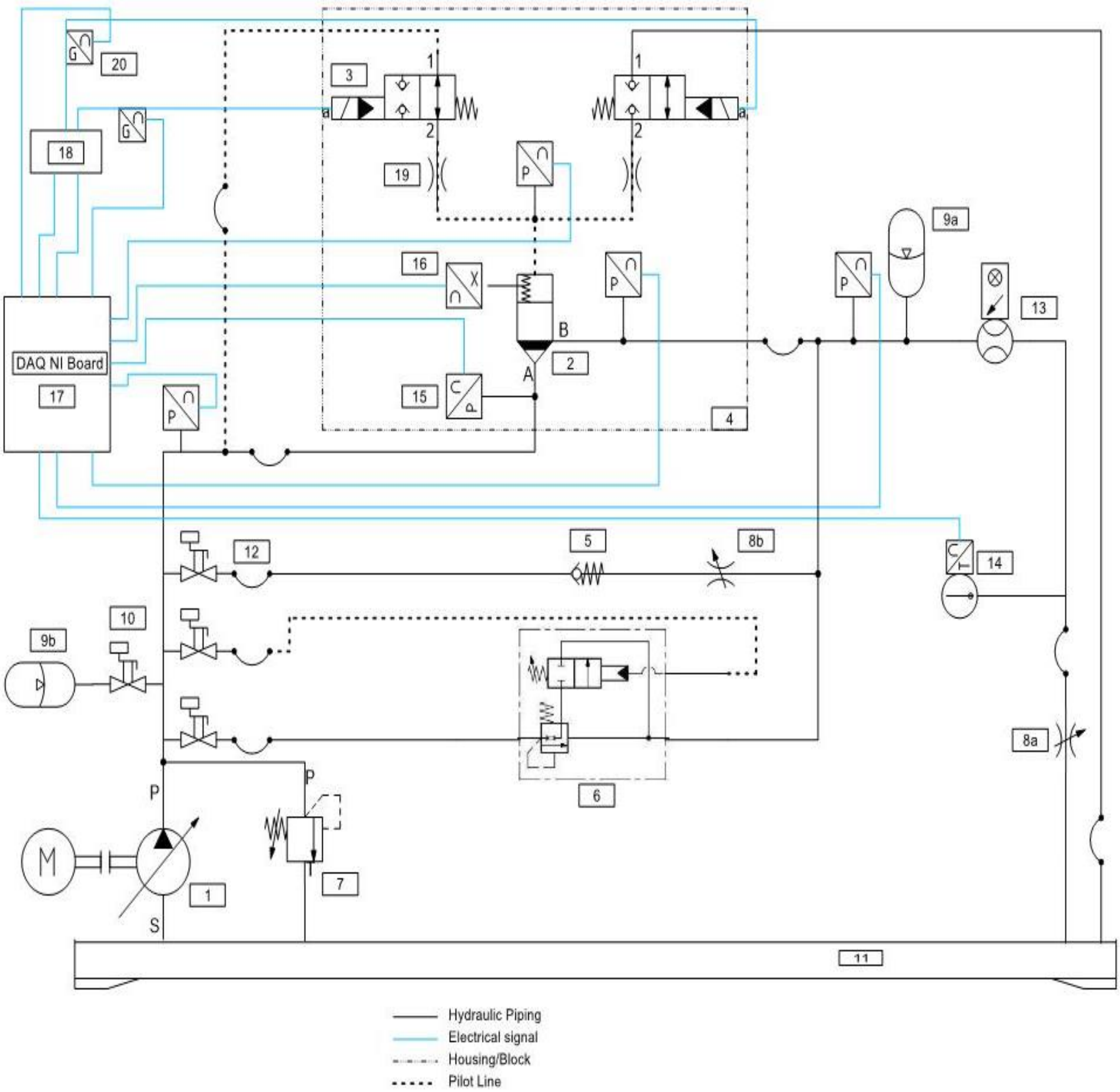


Figure 12. Detailed physical test setup for the uni-direction flow experiments. Parts are listed in Table 1

In the schematic, the solid black line shows the oil flow, the dotted lines represent the pilot line connections, broken dash lines represent valve cover or housing block and the blue lines, data signal connections for the electrical components. However, for the solenoid valves, the DAQ board provides the command signal for actuation at the programmed time, whereas the valve controller box supplies the controlled voltage to the valve.

The setup has been built for one-way flow, from inlet port A of the CV (2) to the outlet port B, and towards the low-pressure side and pressure load created with variable throttle (8a). The main stage flow lines have been built in combination with steel tubes and threaded joints for easier modification in future for succeeding experiments, as well as, to make the structure rigid. Heavy duty flexible hoses, suitable for the high pressure, have been used to connect piloting lines, main stage high pressure line to valve block (4) inlet and parallel attaching components for bypassing the flow through check valve (5) and pilot-unloading valve (6).

Table 1. Physical parts listing from the schematic in Figure 12

No.	Component	Description
1	Motor/Pump	For the main stage hydraulic flow
2	Cartridge valve	2/2 way logic main stage flow valve
3	Solenoid valve	2/2 way solenoid on/off valve for pilot operation
4	Valve block	Solid block with cavities for valves, sensors and inlet/outlet cavity
5	Check valve	Fixed at 8 bar cracking pressure for keeping pressure difference across CV and bypassing the flow
6	Pilot unloading valve	Hydraulically pilot operated bypassing valve for unloading excess pressurized flow from the system
7	Pressure relief valve	To limit maximum pump pressure and safety valve
8	Throttle	To maintain and vary system pressure and a second one for creating a pressure difference
9	Accumulator	Pre-charged accumulators for dampening pressure spikes
10	Ball valve	Manual operated valve to shut off the connected line
11	Reservoir	Main tank for oil supply and return
12	Conduits	Flexible hoses to connect the flow components
13	Flowmeter	Main stage flow rate sensor
14	Temperature sensor	Outlet oil temperature sensor
15	Pressure sensor	Pressure measurements at various points
16	Position sensor	Laser displacement sensor to measure CV switching duration
17	Data acquisition board	Sensors signal acquisition and valve switching
18	Valve controller	Power supply and boosted voltage to on/off valve
19	Orifice	Pilot valves orifice flow channels
20	Current sensors	For voltage signals proportional to current flow

The hydraulic and electrical components have been individually briefed in following sections to establish the sizing of components, settings and technical properties.

2.1 Hydraulics components description

The hydraulic components have been described in detail in subheadings, while few elements such as flow pump and safety components are just briefed about the operational capacity, because the target study is of valve dynamics. Flow source is set to provide a constant flow of hydraulic oil for intended measurements.

2.1.1 Motor/Pump

The Fluid Power laboratory at Aalto University has three axial piston variable displacement pumps coupled with three phase ABB 75 kW rated electric motors with which the flow rate can be adjusted in infinite steps. Each pump has the capacity to deliver approximately 140 l/min maximum flow rate at operating pressure of 315 bar, which accounts for 73.5 kW of power at the outlet.

In this study, the power consumption or any other related losses are not covered, as the power pack has been only taken as the only source of main flow from the tank to the system and producing the system pressure as needed. However, for the higher flow rate in parallel, the additional pump has been added to achieve total flow of up to 200 l/min to meet the set operating conditions.

Figure 13. Main pumps for the hydraulic power source in the laboratory

2.1.2 Pressure relief valve (PRV)

The PRV (7 in Figure 12) installed with each main pump, is a proportional pressure relief valve with pilot operated function with maximum set pressure at 315 bar. However, this valve will not be used to restrict system pressure at the high-pressure line (the inlet of CV), because of its slow dynamic response upon reaching the set pressure, but as a primary safety valve.

From the system schematic, it can be seen that the bypass valve is used to create pressure difference across CV, instead of using PRV. The reason it is left out, is due to rapid switching of CV the flow should be able to pass from high resistance path (after closing of CV) to least one (bypass valve) without delay. The delay in opening of the bypass valve can cause pressure peaks and retard the CV's switching time. The transient response of PRV to overcome the solenoid force has been documented well in [18] and an approximate response time at certain flow conditions for the installed PRV, the step response from 100% to 0% takes about 150 ms [19].

2.1.3 Reservoir

The reservoir in the lab with the power pack is adequate enough to supply the set flow at the required operating conditions. It has an inbuilt heat exchanger to maintain the temperature of the fluid and a filter for removing contamination particles.

2.1.4 Cartridge valve

The main stage valve is a slip-in hydraulic operated valve (and also called seat type, screw in or slip in cartridge valve, based on external installation design). It can be utilized for efficient and high-power applications, with a low-pressure drop across the valve and comes in different available sizes and brands. The one being used in the experiment is from the brand 'Parker', it is compact in size with a large flow rate capacity, good sealing to prevent leakage from port A or B to port X (pilot chamber) and easy to integrate in to the system and is hydraulic operated.

Cartridge valves are a type of poppet valve with an effective surface area for pressure forces, their large metering edge requires comparatively shorter stroke, they are reliable and have good precision of repetition or return to original position upon actuation in comparison with the spool type valves [20]. The logic valves are also at an advantage over strong anti-contamination ability and open enough to let the full flow pass, depending on stroke limit [21].

Even though, considering the mentioned advantages, the overall size including valve block, piloting valve cover and controllability still needs to be taken in view, depending upon the application. For the experiment, there are two available cartridge valves of the size NG16, with the main difference being the poppet shape on metering edge side.

The basic function and usage of the cartridge valves are defined by the control areas formed on the poppet. The poppet A in Figure 14, has primarily two control areas A and X, and is used for pressure function such as pressure relief or one-way flow, while on B side it has almost no control surface due to its contact with the conical seat and therefore the pressure on the B side is not effective enough to lift the poppet.

As for poppet B in Figure 14, it serves for a directional valve with the two-way flow in either direction, A to B or vice versa, depending on the higher-pressure gradient across the valve enough to lift the poppet in the sleeve. The area ratios of the poppet A and poppet B are given in Table 2. The control areas are usually flat, chamfered or step shape formed on the poppet's face, and as for pilot chamber X, pilot pressure along with the preloaded spring acts against the main stage flow.

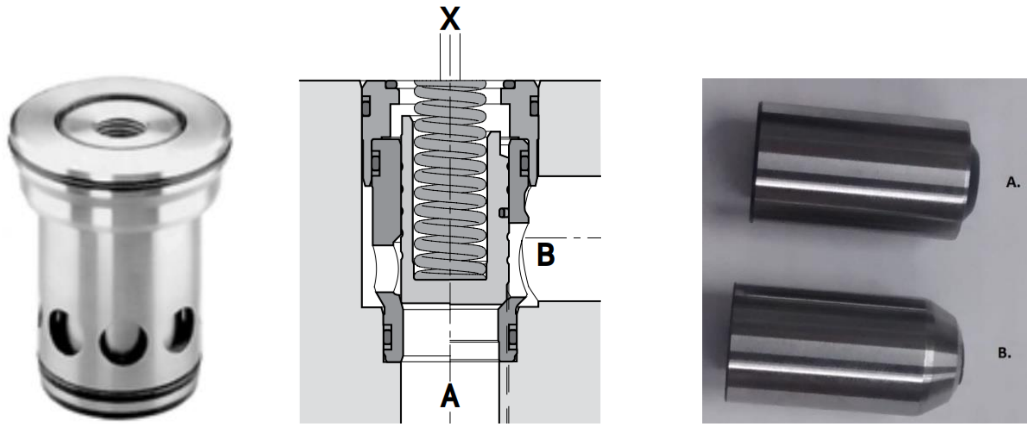


Figure 14. Parker cartridge valve [22] and its poppet alternatives

Basically, the forces acting on each control surface due to pressure determine the net force direction either to open the CV or keep the poppet seated and closed, combined with the spring force. The net force is defined as,

$$F_{pnet} = A_A p_A + A_B p_B - A_X p_X - F_{spring} - F_{flow} - F_{friction} \quad (1)$$

Where,
F_{pnet} defines the direction of force acting
A_A, A_B and A_X are poppet port areas respectively
p_A, p_B, and p_X are pressures acting on each control area
F_{spring} is the pre-compression of spring change multiplied by the stiffness
F_{flow} is the flow forces of hydraulic fluid when the poppet starts to move
$F_{friction}$ is the sum of viscous friction $F_{viscous}$ and seal friction F_{seal}
Viscous force is the friction between poppet and sleeve due to shear stress applied by the hydraulic fluid. Seal friction is due to rubbing against the sleeve when in static and motion.

For now, the fluid forces are considered to be small due to exceeding pressure at piloting side and also internal piloting [5], when the valve is closed but accounted for during closing motion in simulation, the friction forces are not taken into the consideration.

Table 2. Technical data for Parker Cartridge valve [16]

Attribute	Specifications	Description
Brand	Parker	CE016S07S99N / CE016S04S99N
Size	NG16	-
Nominal flow at 5 bar drop	250 l/min	Full flow to pressure difference graph in chapter 3
Pilot volume	2.0 ml	Flow consumption for the complete closing stroke of the poppet
Maximum operating pressure	350 bar	-
Pre-loaded pressure*	1.6 bar	After adding a rod to measure displacement through Laser sensor, due to pre-compression of the spring, the new initial cracking pressure is 1.91 bar
Flow density	5550 1/min	Nominal flow at 5 bar drop to geometric volume
Poppet	Poppet A	Areas A_A=0.96A_X and A_B=0.04A_X
	Poppet B	Areas A_A=0.60A_X and A_B=0.40A_X
Poppet dimensions*	Ø18mm, Ø14mm	Outer diameter, inner diameter
Dimensions*	Ø32mm x 56mm	Detailed in Appendix 4
Stroke*	8.50 mm	-
Spring stiffness*	2717 N/m	-
Viscosity	20−400 mm^2/s	-
Cleanliness level (required)	18/16/13	ISO 4406
Leakage to Pilot	With oil seal	Seal between sleeve and poppet to prevent leakages to and from pilot chamber

The Table 2 consists lists of attributes of the cartridge valve taken from the technical data sheet, the marked (*) ones were measured or calculated, since they were not provided in the data sheet. The switching time is measured in the latter part of the report, as part of experiments. For the first experiment, poppet A with area ratio 'A_A=0.96A_X' was used and is referred as CV096 in this thesis.

2.1.5 On/off solenoid valve

For piloting the CV, two 2/2 solenoid operated valves have been used, one normally closed and one normally open. The pilot valves are set up, as shown in Figure 12, to either set the px equal to p_A or $p_{T,}$ hereby setting the flow direction. Figure 15, shows the actual valve and its symbol, where WS22GD is normally closed (NC) installed at tank line and WS22OD is normally open (NO) installed at the high-pressure line. In the event of switch opening the CV, the NO and NC valves activate in order to take off the pressure on port X of CV and allowing the pilot volume to flow to the tank. For closing of the CV, both valves are deactivated to restore pilot pressure on CV's port X.

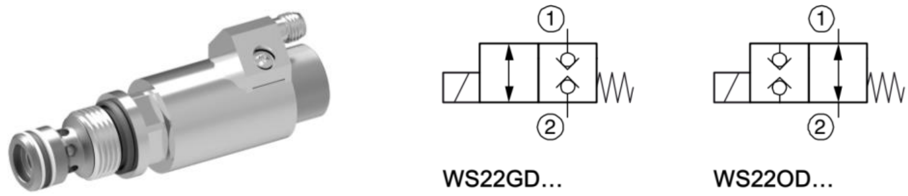


Figure 15. Bucher 2/2 solenoid operated valve size NG5 and schematic of WS22GD(NC) and WS22OD (NO) [23]

The switching performance of the CV is highly dependent on the piloting stage's ability to fill up the CV's pilot volume and on the poppet movement of the pilot valve, so the dynamic performance such as switching time and operating pressure of the pilot valves itself highly influences the actuation [1].

The Bucher 2/2 size NG5 valves are compact on/off valves with a limited flow capacity of up to 30 l/min and approximate boosted actuation time within 5 ms from the activation signal, more of the technical specifications given in Appendix 1. However, the boost timing is highly dependent on the fluid pressure, flow rate, dwell time under pressure and viscosity of the fluid [25]. Another important characteristic for the pilot stage is to be leak free to prevent loss of pressure in the CV's pilot chamber, though, this has not been quoted in technical data sheet but mentioned in [24] to be 5 drops or 0.2 ml per minute at undefined pressure difference.

Figure 16 shows the time taken for boosted opening and closing of the both NC and NO valves along with the stroke, measured using a laser displacement sensor without external hydraulic pressure. The full displacement was found to be 0.69 mm for both models and total duration from signal activation for opening and closing of NC/ NO valves to be about 5 ms / 5.5 ms and 6 ms / 5.5 ms respectively. The current profile graph acquired through oscilloscope plotted with displacement can be found in Appendix 5.

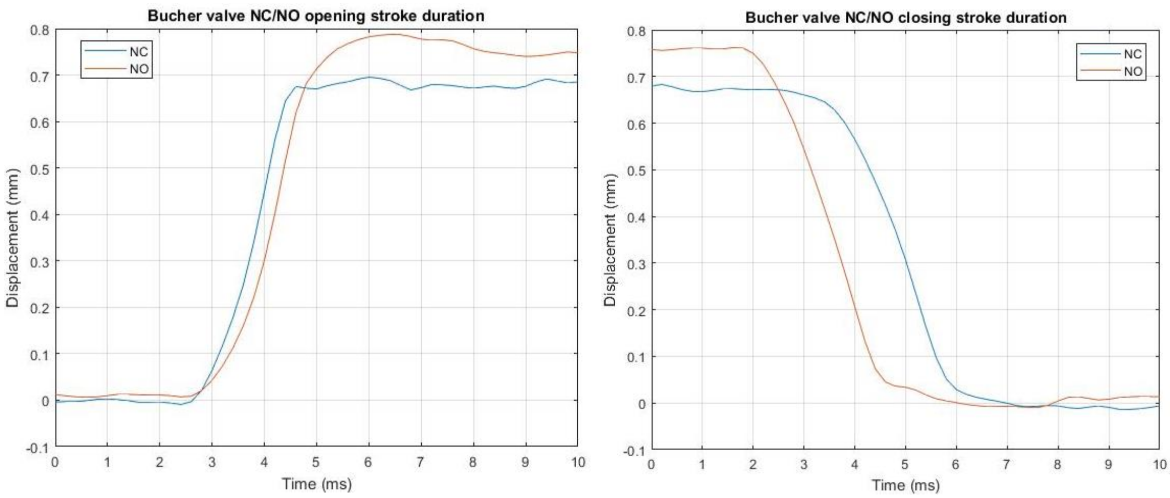


Figure 16. NC and NO valve's opening (left) and closing (right) duration measured with the laser sensor. laser sensor signals have been offset negatively by 0.44 ms due to delay in transmission of signal (further information given under laser displacement sensor heading)

These measured values are of importance for the simulation and the analysis of the results, to estimate the dynamics of the pilot valve and combined valve switching for possible optimization of the valve system later on. The solenoid valve dynamics shown in Figure 17, is an important aspect to be considered, since it largely affects the poppet position of CV.

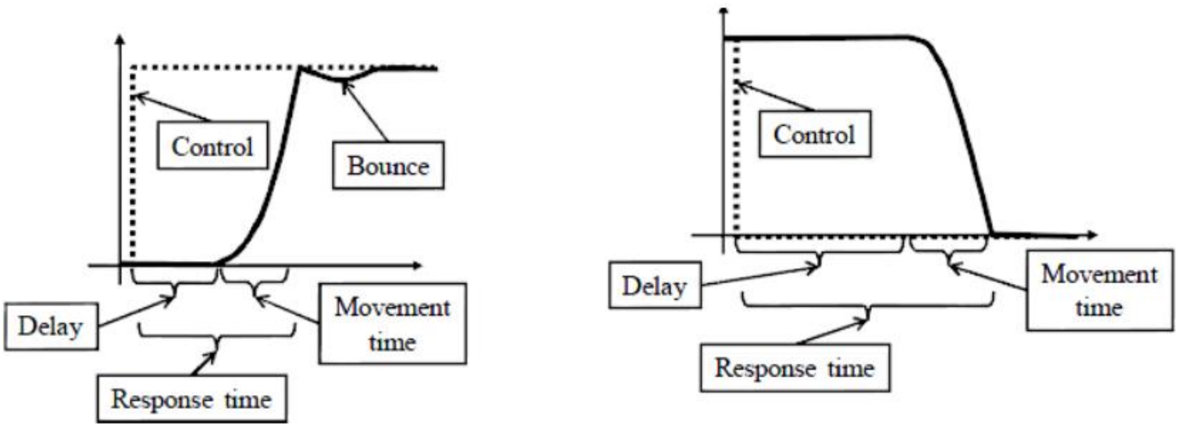


Figure 17. Solenoid valve dynamics during opening (left) and closing (right) [24]

When the control signal, that is zero or one, is applied to activate the current of the solenoid valve through valve controller box, there is a delay between the point of command and the start of poppet movement. That delay is due to coil resistance before the poppet movement starts defined by the time constant,

$$T = \frac{L}{R} \tag{2}$$

Where T is the time constant in seconds, L is the inductance value in Henry and R is the resistance in ohms. The approximate delay, from the current profile graph in Appendix 5, before the start of the movement, was found to be 2.5 ms. The second phase of the movement time is influenced by magnetic force, spring force, flow force and external pressure drop across the valve. Moreover, movement time is the duration from a seated position to fully open, and bounce represents the drop due to back EMF, albeit, not prominent in the current profile graph from oscilloscope in Appendix 5, but can be picked up from the sensitive current measuring device, given in [25].

The total measured time is an approximate due to the unknown value of L and no position measurement for pilot valves is installed during wet measurement, but current measurements are taken to approximate the full stroke opening/closing based on the current signals over time at each operating condition.

2.1.6 Valve block

The two-stage valve system is machined into a valve block, comprising of cavities for the cartridge valve, pilot valves, sensors and hose clamps for the experiment. The valve block is a prototype, not intended to be compact, for the purpose of dynamic characteristics measurements of CV, designed by the former doctoral candidate Tapio Lantela, has been mounted in the setup and the labeled CAD model and sectioned view is shown in Figure 18. While each component in the valve block assembly is listed in Table 3.

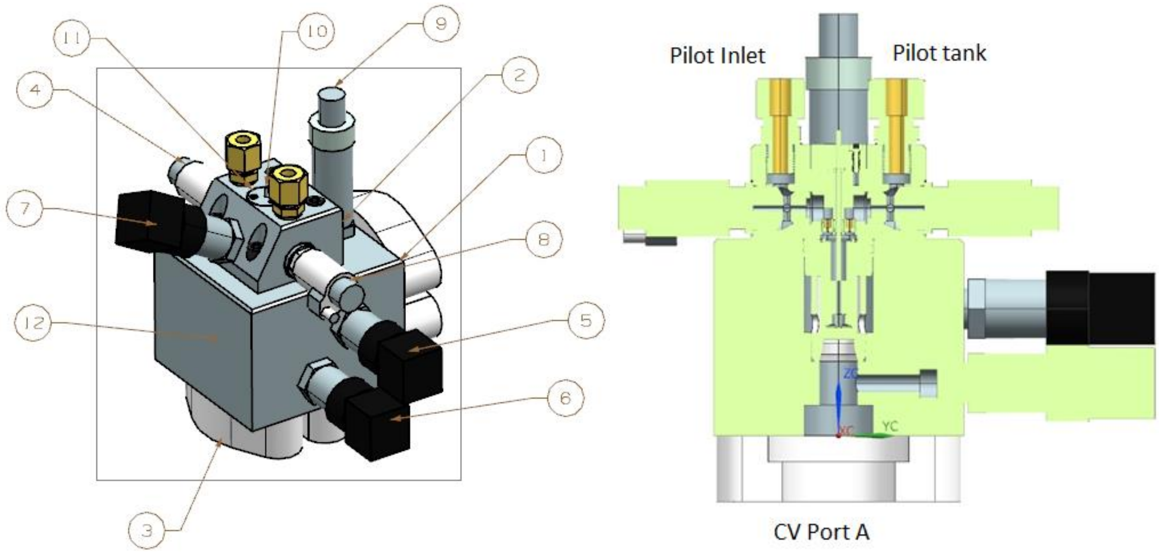


Figure 18. Valve block CAD model and sectioned view of flow channels

Table 3. Valve block components

Label	Component	Description
1	Main block	For CV, hose clamps and pressure transmitters
2	Pilot cover	Holds pilot valves, pressure sensor, oil seal and timing rod
3	Hose clamps	Inlet and Outlet clamp
4	NO valve	Solenoid valve connected to the high-pressure line
5	PS2	Outlet pressure transmitter
6	PS1	Inlet pressure transmitter
7	PSP	Pilot pressure transmitter
8	NC valve	Solenoid valve connected to the low-pressure line
9	TPS	Temperature sensor
10	Position sensing rod	To measure position using laser sensor of Ø2.0 mm
11	Oil seal and Stopper	Oil seal NBR Ø1.78mm x 1.78mm Shore 70 and round stopper bolted to align seal and the rod

The top part is the pilot cover (2 in Figure 18) and is bolted to the main block with oil seal in between. It also consists of the protruding rod to be used for timing measurement of CV. The primary block or main stage, holds the CV and inlet/outlet channel fixed on to the main flow of the system.

2.1.7 Throttle

To create load pressure in the system, a variable throttle (8a in Figure 12) is attached at the low-pressure side to increase the system pressure. The use of variable throttle is to be able to increase pressure load in steps in increasing order, depending on the operating conditions defined in case setup. Basically, throttling is done by adjusting the sleeve of the valve to constrict the area for the flow, which is formed between the valve body and sleeve. The area can be varied steplessly.

Figure 19. Throttle valve Rexroth MG20G (left) and Parker Colorflow N2000 [26]

Valve's full flow capacity is 200 l/min with a pressure drop of 5.5 bar, as listed in the technical data sheet measure with HLP46 fluid. While, the second throttle, labeled 8b in Figure 12, was added after initial test runs to create a higher-pressure difference, with similar performance rating and flow rate up to 250 l/min. The principle equation of throttle valid for steady incompressible flow is,

$$q_v = C_d A_o \sqrt{\frac{2\Delta p}{\rho}} \tag{3}$$

Where q_v is the volume flow rate, which is proportional to the square root of pressure drop Δp, C_d denotes the discharge coefficient, A_o is the cross-sectional area, which is variable in the mentioned throttle, and ρ is the density of the fluid.

2.1.8 Pilot unloading valve

The pilot unloading valve, denoted with PUV and shown in Figure 20, is essentially an unloading valve that prevents the system pressure from rising higher than is the preset load pilot pressure. The PUV pilot pressure can be adjusted by varying the spring load. The flow path from P to T is normally blocked, and the spool is moved when the pilot pressure attains the set level, which allows the unloading path to the tank. The valve is installed in parallel to the CV to bypass the flow when the pressure rises above the safety level. The maximum flow is limited to 200 l/min and the operating pressure up to 210 bar.

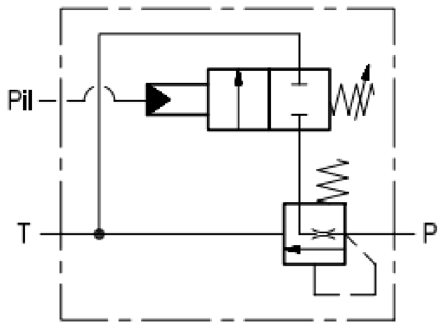


Figure 20. Pilot unloading valve Rexroth BM-N [27]

2.1.9 Check valve

The check valve is also installed in parallel to the CV to bypass the flow when the CV is closed and to create a pressure difference of 8 bar for initial testing of the system. The foremost reason to use it as a bypass valve is that, it rapidly opens up for the flow without delay when the CV closes to prevent higher pressure peaks back to the pump and closes when CV is opening, that has the least resistance path. The check valve consists of a spring and ball, which allow the flow in one direction and block it in opposite, and is basically governed by the pressure differential across the valve.

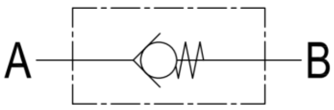


Figure 21. Check valve Rexroth 04.31.17.00/05/01 schematic

$$p_{crack} = p_A - p_B \tag{4}$$

Where p_A is the inlet port pressure, p_B is outlet port pressure and p_{crack} is the cracking pressure. The ball is lifted off its seat when the pressure difference is equal to or greater than the cracking pressure and the area is further opened until maximum orifice area and corresponds to the maximum pressure differential across the valve.

2.1.10 Accumulator

The hydraulic accumulator is used for temporarily storing the hydraulic fluid and it is analogous to the capacitor in the electrical system. The accumulator comes in different internal structure, in terms of separating the preloaded gas and/or spring from the hydraulic fluid, such as piston, bladder or diaphragm. Based on the requirement and availability, diaphragm type accumulators have been installed on the high-pressure side, SB330H-3.5E1/663U-330AB and low-pressure side, SB0210-1.4E1/663U-210AB, shown in Figure 22.

Figure 22. Low-pressure side accumulator (left) and High-pressure side accumulator (right)

The piston type accumulator has the highest fluid storing capacity and high operating pressure in excess of 700 bar and it can be used for higher flow rate and energy density requirements. The diaphragm accumulators are less susceptible to contamination, are compact in size, but have limited pressure ratio which needs to be taken care of, to prevent diaphragm puncture due to fluid over pressure in the system.

Accumulators are useful in many dynamic situations in a hydraulic system such as transient peak power supply, damping the pressure shocks in the system and fluid buffer as a low-pressure reservoir for later use. The diaphragm type accumulators used in the experiment have been pre-charged with nitrogen gas to a certain level based on pressure ratio and operating pressure of the experiment.

Basically, the accumulator's pre-charged pressure is lower than the system pressure, and the accumulator is connected in parallel to the system as shown in Figure 12. The accumulator is filled with fluid until the pressure rises equivalent to system's pressure and volume during operation changes as shown in Figure 23.

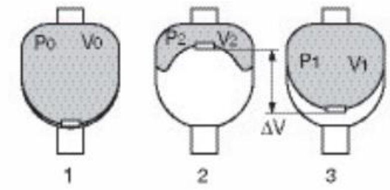


Figure 23. Diaphragm accumulator - 1) Pre-charge pressure, 2) Maximum working pressure, and 3) Minimum working system pressure [28]

Sizing of accumulator can be done based on the Boyle's law and incorporating maximum and minimum system working pressure

$$V_{o,g} = \frac{\Delta V}{\left(\frac{p_o}{p_1}\right)^{1/n} - \left(\frac{p_0}{p_2}\right)^{1/n}} \quad (5)$$

$$\Delta V = V_{1,g} - V_{2,g} \quad (6)$$

Where the ΔV is the change in volume, and $V_{o,g}$ is the effective total gas volume, $V_{2,g}$ is the volume of gas at maximum pressure p_2, $V_{1,g}$ is volume at minimum system pressure p_1 and p_o is the pre-charge gas pressure. In this case, polytropic exponent 'n' is taken as 1.4 for isothermal process, mainly due to rapid change in volume and pressure due to rapid valve switching.

In this case, the accumulators are merely used for dampening sharp pressure transients, preventing cavitation and fluctuations in the system and smoother displacement readings of CV displacement rather than oscillations, as shown in switching time figures in the paper [17]. The accumulators used, Low-pressure accumulator (LPA) has a nominal volume of 1.4 l, the pressure ratio of 8:1 and operating pressure up to 210 bar, whereas the High-pressure accumulator (HPA) has corresponding values of 3.5 l, 4:1 and 330 bar rating, both are rechargeables. The pre-charge gas pressure was changed for HPA from 12 bar to 23 bar for experiments with operating pressures above 40 bar.

2.1.11 Hoses, tubes, and joints

The conductors are an important part of hydraulic systems and need to be sized according to the requirements, especially in the case of a high-performance system where operating pressure, flow rate, and pressure drop are important factors. Since fittings and the conductors connect the sub-systems with each other, the selection should be made on the basis of the safety factor, minimum bends to prevent pressure losses, system's size and adjustment tolerance and the flow capacity.

In this setup, the selection that is based on the flow rate, high operating pressure, least pressure drop and the length of the conduit, plays an important aspect in the system. Mostly, flexible hoses have been used, then steel tube in the low-pressure side, steel union elbows, union tees for the fittings, dimensions and bending schematic and placement given in Appendix 3.

Figure 24. Flexible hoses in the setup

Due to high flow operation, heavy duty flexible hoses have been used for easier installation of the components in the system and to reduce pressure oscillations in the flow. For conduits selection, the material, length, inside diameter and the fitting type needs to be defined. Excessive length could cause significant pressure loss per unit length due to friction, which is covered in a latter section to account for supply pressure rise and losses in the system.

And also, in the case of these measurements, the longer length of flexible hoses may cause pressure waves traversing back and forth for a short period of time. All of the fittings have either threaded connections or the socket type for the main inlet and outlet from the valve block. In addition, all of the conduits are capable of operating pressure at 350 bar or greater. Secondly, the cross-sectional area of the lines should be large enough to prevent generating excessive average fluid velocity, that is defined by

$$v_{avg} = \frac{q}{A} \tag{7}$$

It is understood, that the thumb rule is to keep the flow velocity approximately at 1.2 m/s at the suction or tank lines to prevent cavitation and 6.1 m/s at the pressure lines to prevent head losses, turbulent flow, and excessive temperature in the lines [9]. Using Equation (7) with quoted values, the minimum internal diameter is found to be 1´´ (25 mm) at 200 l/min flow. However, in the system 1.5 inch and 1 inch lines are being used for the main stage flow and 3/8´´ lines for pilot flow, where the flow rate is tended to be lower.

2.2 Sensors, data acquisition and electrical setup

The data acquisition setup consists of National Instrument PCI card NI 6259 with analogue to digital conversion (ADC) resolution of 16 bits. It has 16 differential or 32 single-ended analog input channels as well as digital in and digital out channels. The card is adequate enough for the sampling cycle set at 2 kHz and for the number of sensors installed. The card is installed in a target computer (also called slave) and is connected to the master computer through an ethernet cable.

The program Matlab/Simulink 2015a version is used to interface data signals from sensors and to control the valve. The Simulink real-time function is utilized to monitor the real-time changes on the screen, for setting the operational parameters and to store the collected data from the sensors. The collected data is raw values, that is in voltage form, which can be converted by applying a proportional multiplier based on the sensor's range during post-processing or in the results analysis phase. Model and script for references are presented in Appendix 6 and Appendix 7.

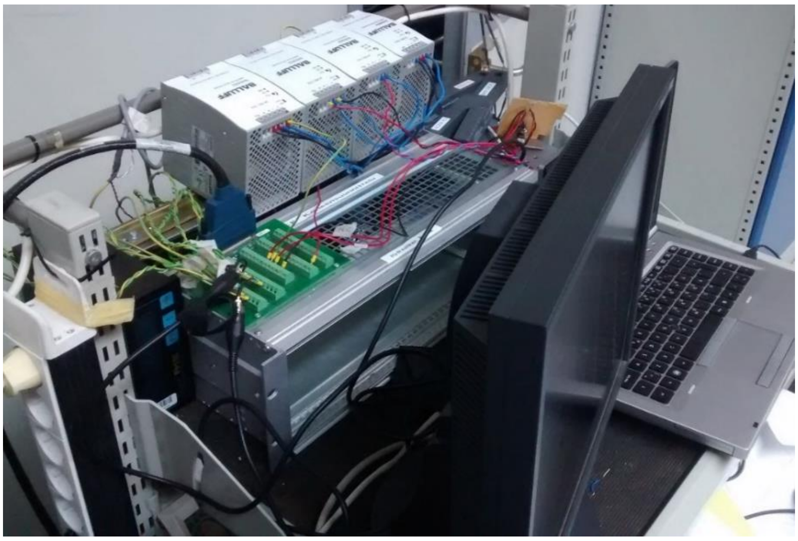


Figure 25. Power supply unit (PSU) and data acquisition (DAQ) system

In the system, four power supplies are used for different components, that is 48 VDC for solenoid valves controller, 24 VDC for flow counter and laser displacement sensor, 12 VDC for pressure transmitters and 5 VDC from NI card for the temperature sensor. The wire shielding and PSUs except 48 VDC supply have been grounded into the NI card, while PSU of valves is isolated. The TTL signals to control valves are also isolated from valve power controller using optocoupler circuit (see Appendix 2) to prevent electrical fluctuations into the sensor signals during valve activation. The valves TTL signals are supplied from two different digital output (DO) channels, flow counter and laser sensor signals are fed into data acquisition card through differential analog input (AI) channels and remaining ones are single ended. For electrical schematic, see Appendix 2.

2.2.1 Valve controller box

The two on/off solenoid valves are powered by the boosted voltage for a fraction of second at the recommended 48 VDC [23, p. 2] for first 5−6 ms and then dropped down to the holding voltage of 12 VDC. For closing, the supply of current is switched off causing a slight back-EMF in the coil as shown in left Figure 26. The valve control box, provided by the Tampere University of Technology, takes in the TTL signal from the NI card and the 48 VDC from the main power source.

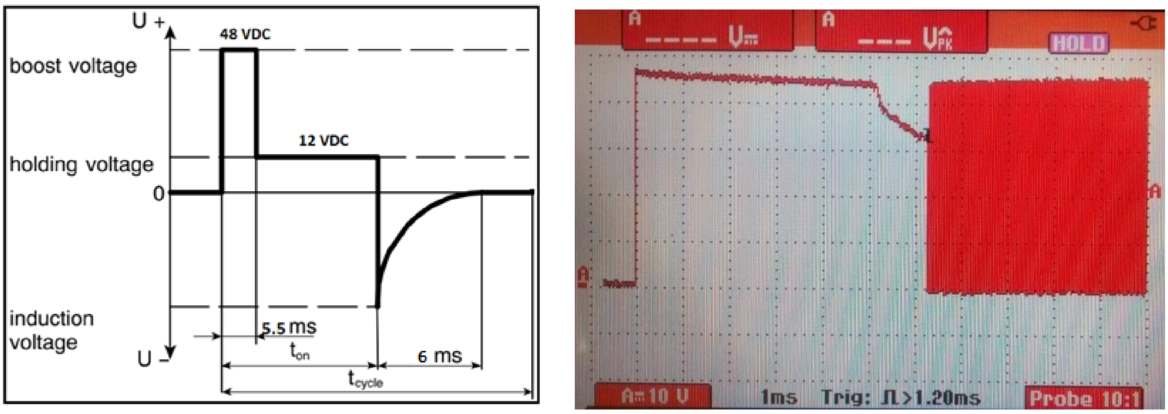


Figure 26. Boosted voltage supply scheme (left) and oscilloscope voltage profile supplied by valve controller (right) – horizontal scale 1 box = 1 ms, vertical scale 1 box = 10 VDC

The measured (from oscilloscope) voltage profile shows 48 VDC for first 6 ms and then reduced to 13 VDC at 24 kHz as holding voltage. The holding voltage can be externally adjusted to set the pulse width modulation (PWM) output. The additional 1 V is to compensate for the voltage drop across total 6 m length of wires from the controller to the valve and back.

2.2.2 Pressure transmitters

In the experimental setup, total five pressure transmitters are being used to record the pressures in the system. One for the pilot chamber pressure of CV and two at the inlet of CV or main stage flow line and remaining two at the outlet, as depicted in Figure 12. The two transmitters, at inlet and outlet, are individually placed one near the power pack and second one installed in the valve block and similarly for the outlet. The two sensors at inlet and outlet each, are placed apart, to account for the pressure loss in hose due to friction and cross check the reading data for any error.

Figure 27. Pressure transmitter Hydac HDA 4700 [29]

The pressure transmitters used are of 100 bar range, single-ended relative pressure measurement devices and give out an analog output (AO) signal from 0−10 V and accuracy to DIN 16086 to be ≤ ± 0.25% of full scale and rise time ≤ 1 ms, which makes them suitable for the setup.

2.2.3 Temperature sensor

One temperature sensor has been fitted at the outlet port of the system to monitor the temperature in order to conduct the experiment at the same viscosity of the fluid while varying the operating parameters. Its operating range is from -50 to 150 ºC with a full-scale error of 1%.

Figure 28.Temperature sensor model IQAN-ST [30]

2.2.4 Laser displacement sensor

The laser displacement sensor is used in the experiment to measure the displacement of the poppet motion of the cartridge valve over switching duration. The sensor can do the sampling at minimum of 2.55 µs, good enough to capture the poppet displacement down to fraction of milliseconds. For that purpose, a timing rod is fixed into poppet's pilot chamber with spring as mentioned in the previous section, and the laser spot is set on the flat surface of the rod protruding out of the valve block cover.

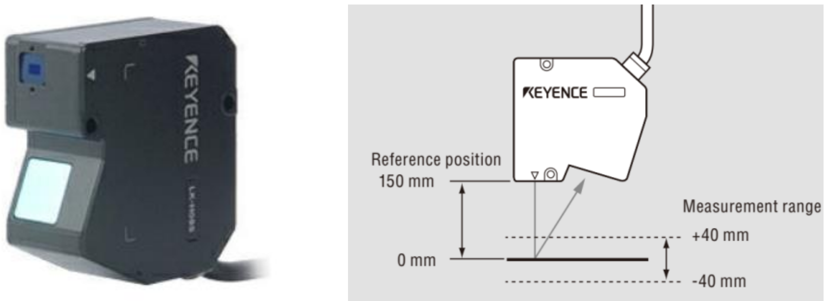


Figure 29. Laser displacement sensor Keyence LK-H150 [31]

The laser sensor head, shown in Figure 29, works on the triangulation method and is suitable for opaque material when positioning it perpendicular to the reference surface. The laser is applied to the target and the light is reflected back on to the receiving surface. The measurement range is ±40 mm, shown in the figure scheme, and the generated signals through laser head are scaled via laser control box that gives out AO of ±10 V. Because the stroke of the poppet is limited to 8.5 mm, the AO conversion is set to 1 mm = 1 V to have good accuracy of the result. The sensor is set at an approximate distance of 150 mm from the head when the poppet is seated and the measuring accuracy set to 0.01 mm.

As the measurements are primarily transient and sampling cycle is set at 0.5 ms for all the sensors, even though the laser sensor is super responsive down to microseconds, there is an initial delay since from the laser head to AO signal output. This delay has to be offset negatively in the experimental data to plot correct displacement over time. The delay is dependent on the controller settings, provided by the manufacturer in Equation (8),

$$5\ \mu s\ (Sampling\ cycle) * 74 + 5\ \mu s * 4\ (reset\ time) + 50\ \mu s = 440\ \mu s \quad (8)$$

2.2.5 Flowmeter

In the setup one flow rate sensor is used to monitor the flow rate passing through the main line from the pump to the tank. It is a gear type flow meter by Kracht, where a nominal volume between each tooth generates a pulse when driven by the flow. The pulse counter device, model FM 16, converts the pulses into flow rate reading and proportionally gives an AO signal over ±10 V in forward or reverse direction of flow.

Figure 30. Flow meter Kracht VC 5F1

The sensor model used is Kracht VC 5, which covers the range up to 250 l/min and operating pressure up to 315 bar, with the minimum flow counter resolution of 5.22 ml and measuring accuracy of ±3%. One limitation of this flow counter is the response time, due to which the sudden change in flow rate with respect to change in poppet displacement cannot be captured. When the mechanism rotates by one tooth pitch, it emits one pulse equal to 5.22 ml or 191.50 pulses per liter, so for 200 l/min,

$$\left(\frac{200 * 191.5}{60 * 1000}\right)^{-1} = 1.567\,\frac{ms}{pulse} \quad (9)$$

If data is acquired using pulses, the least possible response time at the mentioned flow rate would be 1.567 ms. which is a lower than the other sensors sampling frequency, that is about 2 kHz. Moreover, flow signals are being taken from the log device that samples from two pulse channels and then converts the pulses into AO signal, which also adds up undefined delay at DAQ.

2.2.6 Current Clamps

The current clamps have been used to measure the electric power consumption and approximating the position of pilot valve’s poppet based on current profile, for both NO and NC valve. The current profile can be used to estimate the delay in poppet movement from the point of activation command to full stroke, where the maximum peak starts to fall. The current clamps give out proportional voltage 100 mV/A, which is fed into the AI pins of the DAQ card. Image of current clamp used in the setup shown in Figure 31.

Figure 31. Current clamp for AC/DC current [32]

2.3 Pressure mapping

In the actual system, the flow elements such as, conduit, unions, throttles, and any contractions causes drop in pressure with respect to flow rate. To account for pressure losses in the assembled system, the pressure map in Figure 32 is presented. The maximum pressure difference across each component is mentioned at continuous flow of 200 l/min for mainline components and conduits. For the pilot flow line, the flow rate is less than 30 l/min and corresponding pressure drop Δp is quoted.

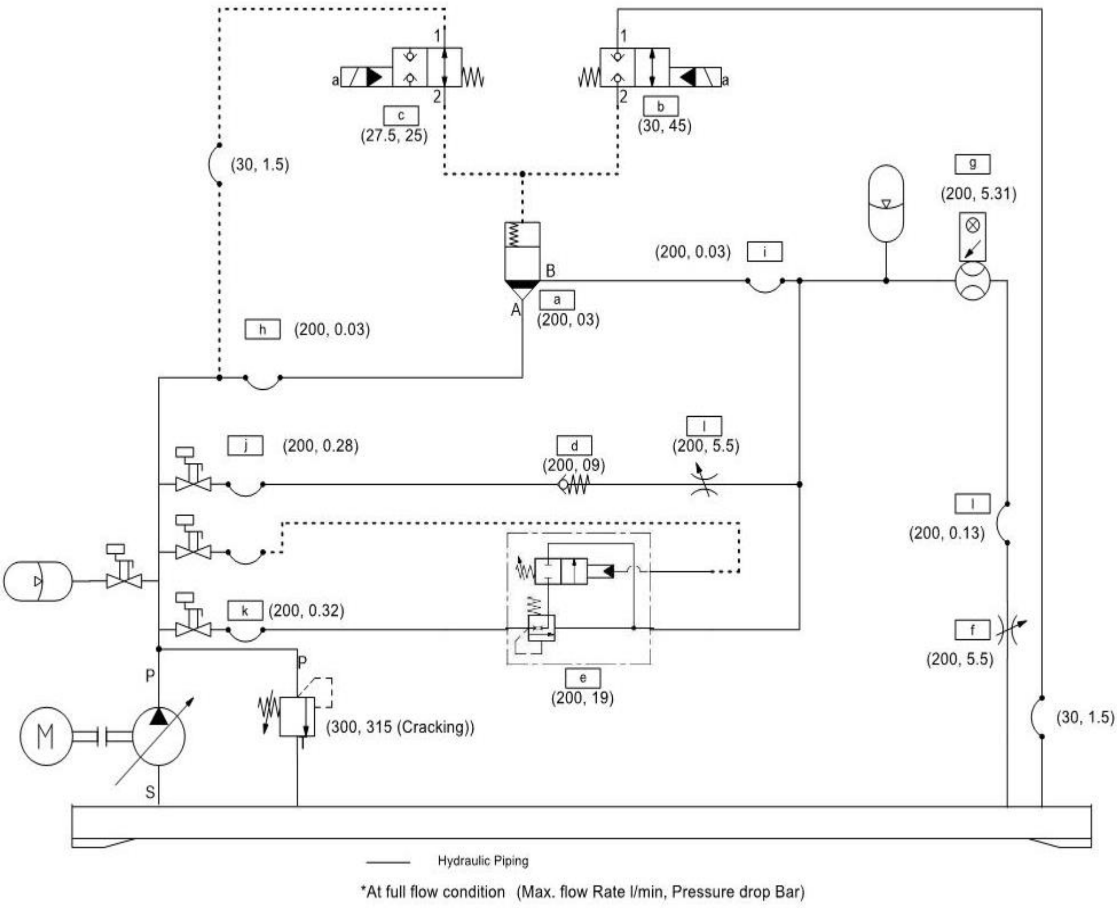


Figure 32. Pressure map of the schematic at maximum flow 200 l/min in the system

The pressure drop at rated flow for each component is taken from the technical data sheet. However, for the hoses it has been referenced and converted into standard units from one of the flexible hoses manufacturer's provided pressure chart corresponding to the internal diameter and flow rate given in the catalog [33].

For the full range of flow, the pressure drop variance and governing principles have been covered in simulation chapter to have refined simulation model and a benchmark for experimental results.

2.4 Case setup and operation

The experimental setup has been arranged to perform the valve actuation and to acquire the measurements by systematically increasing the operating parameters in steps. Such as system pressure, pressure difference over CV, and flow rate while keeping the system's intensive property, temperature, constant. Initially, one-time switching was performed and then the continuous switching, ballistic and independent pilot control switching method. In all test cases, before the start of the DAQ and valve switching, the system is run to adjust for desired operating conditions. The valve activation time is followed as shown in the Figure 33.

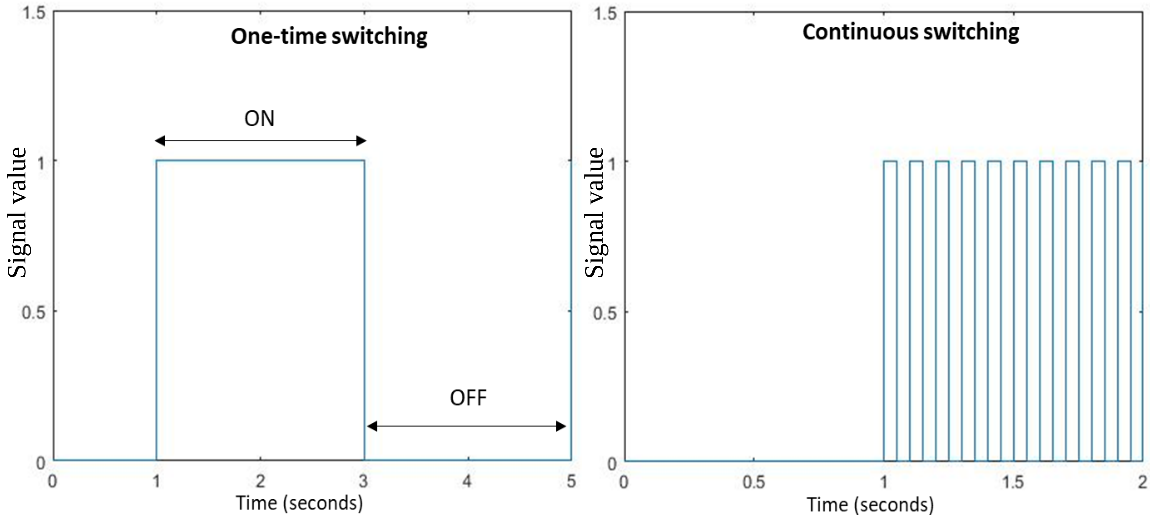


Figure 33. Valve switching condition

The one-time switching was done at full flow range in steps and at different pressure conditions to measure the fastest switching time and to monitor the flow directional control, peak pressures or probable cavitation and valve motion stability. The reason for extended opening or closing duration, i.e. two seconds is to let the flow stabilize and account for any anomalies and greater number of data points. As for continuous switching at different frequency and duty cycles, this could serve a purpose in digital valves application where flow and pressure can be averaged out by controlling the valve switching frequency and duty cycle through pulse width modulation.

2.4.1 Operating parameters and cases

The conducted experiment cases have been defined in Table 4 and the measurements outcome in the Chapter 4.

Table 4. Experimental operating parameters case setup at system temperature 29.5 ºC

Switching type	**Switching frequency (Hz) and duty cycle (%)**	**Pressure difference over CV (bar)**	**Pump pressure (bar)**	**Flow Rate (l/min)**
Fully open (static test)	-	Proportional to flow rate	Proportional to PD (P to T)	0–200–0
One time	0.25, 50%	8 8 20 40 60 80	20 40 40 60 80 100	200
One time (pilot valve switching control)	- CV stroke control - Ballistic mode	13	26	150
Continuous	5, 75% 5, 25% 10, 50% 20, 30%	08	40	150
Pilot Leakages test	a) No switching	08	40	0−150
	b) 2, 50%, 9s duration (18 cycles)	08	20	150
		08	40	100
		08	40	150
		80	100	200
	c) Zero pressure, CV fully open, 10s duration	08	40	150

The operation of valve actuation is briefed in the simulation chapter and also various method of controlling in later headings. The first case, static test, is done to test the whole system workability, for potential leakages from the connections and also to measure the pressure drop at varying flow rate. While the rest are dynamic switching operations to measure the performance characteristics at different operating conditions and capability. The oil used for the experiments, is ISO VG 46, brand Neste, with a density of 858 kg/m^3 at 15 ºC.

3 Modeling and Simulation Setup

This chapter includes the simulation model built in virtually on the platform Simulink/Simscape environment within Matlab software. First, a low-fidelity model is built before the lab experiments to confirm the flow directional control functionality using the internal piloting valve scheme and to have a general idea of the dynamic analysis to be conducted. Further on, the model is fine-tuned based on the components selected for the physical setup to replicate similar results and to analyze the factors, that possibly limit the operation of the setup.

The Simscape platform is an obvious choice for the simulation modeling due to its integration within the Matlab, a routine software for scripting, graphs plotting, and data processing. Another benefit is to be able to use Simulink library objects to formulate the equations for processing the input/outputs coupled with the object-oriented block models. It consists of a physical component library available to choose from, mechanical, electrical systems, and fluid power components. This is a convenient option for developing the tier one level physical representation of the system and subsequent sub-systems.

The physical blocks can be parametrized on a detailed level, for example, using data extracted from the technical data sheet of the stock part or empirical data from the research. Moreover, the variables or expressions can be defined as input parameters in the Matlab script, to be able to sweep simulation runs and control the model operating parameters and simulation runs without ramifications in the original model.

3.1 Simulation model for uni-flow setup

The simulation model is made similar to the physical connection shown in Figure 12, with the addition of sensors and linking blocks for signal conversions, between Simulink blocks and Simscape. Moreover, additional parametric objects have been connected with the corresponding components to imitate the dynamic behavior of the real objects.

In the model, the cartridge valve and pilot valves need to be explicitly parameterized to function as the real objects. The hydraulic layout and piping have also been included to account for pressure losses and velocity change in the fluid due to contraction or expansion. The inertia of the fluid and compressibility in the hoses and system is not accounted for the reason, primarily, that the topic is focused on making a correlation with the valve dynamics and experimental switching results rather than the flow distribution system such as conduits and joints.

Figure 34 shows the simulation model built to simulate the switching of on/off valve to allow flow to pass in one way based on the pilot control signal. To account for pressure losses similar to the experiment, the flow elements such as channels and joints are also included using the built-in library components. For throttle valves and check valve orifice's coefficient and empirical data is sourced from the given data sheets of the components. The flow vs. pressure drop performance charts are provided in the combined graph in Appendix 8.

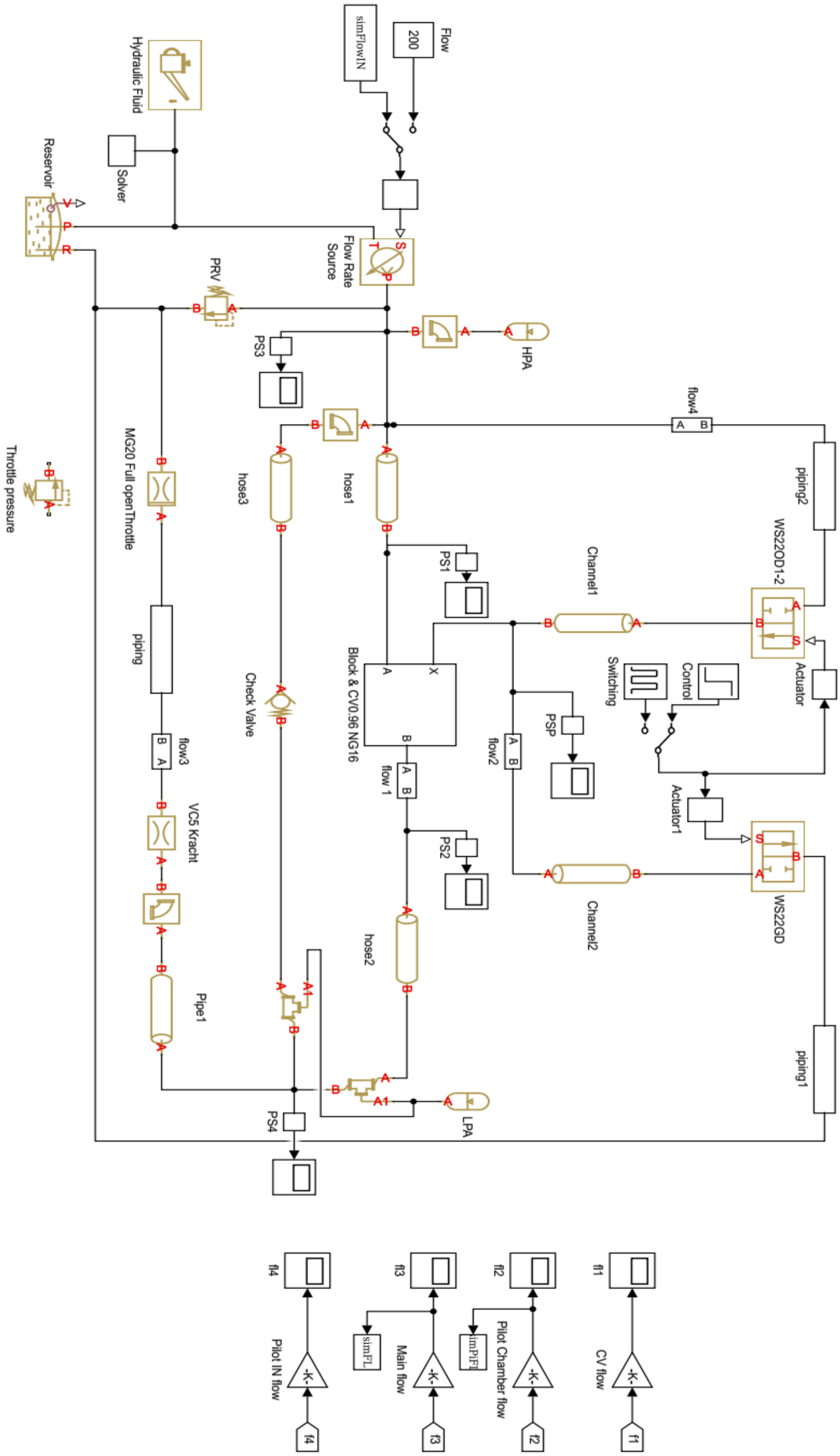


Figure 34. Uni-flow simulation model for static run

Initially, the simulation is run for the static flow condition, varying the flow rate from 0–200–0 l/min over the time period while keeping the CV valve open. The pressure difference is measured between HPA and LPA as the flow passes through the main line and is plotted against the simulation's result in Figure 35, to validate the model.

The difference at some points in the measured results could be due to difference in discharge coefficient in actual and the model's, earlier closing in simulation, turbulent flow in the system, further analysis is covered in the results section.

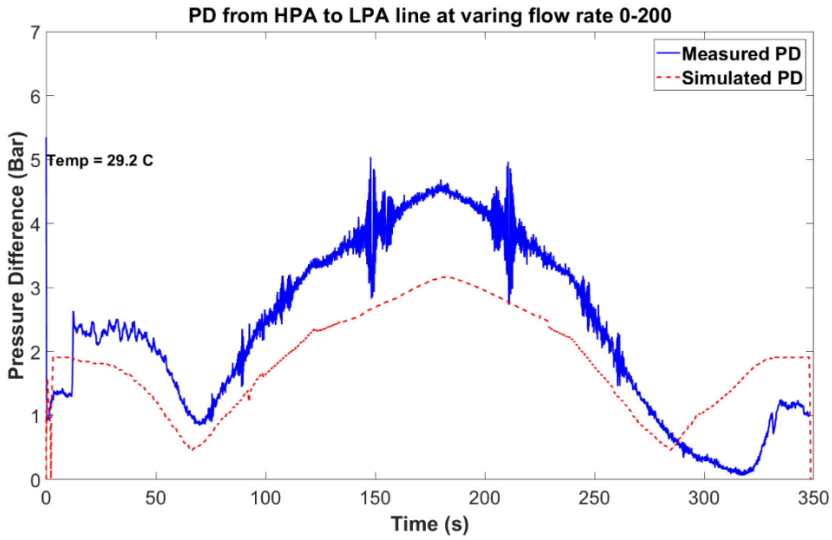


Figure 35. PD at varying flow rate from HPA to LPA line in measured and simulation

Further, to imitate the physical system in simulation, the internal piloting, accumulator, check valve to create pressure difference across the CV are modeled. As for the throttle, a simple empirical model of the orifice is used for fully open condition (MG20 throttle in Figure 34). For the fluid power source, an ideal flow source object is utilized for the variable flow rate instead of a real pump model, as the subject of study is limited to the valve block.

To account for the pressure loss due to wall friction, in pipping, at a particular flow rate is modeled and defined with the diameter, pipe shape, surface roughness and fluid density using the Darcy-Weisbach equation,

$$\Delta p = f_D l_c \frac{\rho}{2} \frac{v^2}{D} \tag{10}$$

The Equation (10) is in general form for both laminar and turbulent flow, and it accounts for pressure loss Δp over the length l_c of a cylindrical pipe with fluid density ρ, mean flow velocity v, hydraulic diameter D and the Darcy friction factor f_D, which is different for laminar and turbulent flow, and whose value is based on the Reynold number.

Each subcomponent and model are defined with parameters in detail in the next section. Some of the basic principle equations are already covered in the previous chapter for the primary components, while object values and construction are detailed in next Chapter to have an accurate representation of physical objects based on the measured empirical data and assumed values.

The final model for dynamic analysis used for benchmarking is without components such as the accumulator, return lines, and safety valves to keep the model simple without varying multiple parameters. Apparently, the two-stage valve is destined for application where flow behavior may be different than the one conducted in the experiment. Nevertheless, the irregularities such as pressure oscillations and peaks are analyzed in the succeeding sections, which are directly related to the valve motion or caused by the rapid switching.

Another point to note regarding the type of switching based on piloting are: **Passive switching** and **Active switching.**

In Passive switching, the pilot pressure is kept constant and supplied from external source without loss of flow from the main line hydraulic fluid, also can be termed as External piloting. In Active switching, the pilot pressure is referenced from the higher-pressure side of the main valve port which makes it highly dependent on the main valve dynamics and pressure levels in the system, termed as internal piloting. In the rest of the measurements, internal pilot switching is performed except where specifically mentioned.

3.1.1 Modeling components and parameters

Two-way on/off valve
The schematic arrangement of the pilot two-way on/off valve is similar to the actual system. The internal piloting valve model is created from the library consisting of an ideal actuator and switching valve, whose models are based on the Bucher Valve WS22GD/OD size 5.

The flow discharge coefficient is computed from the technical data sheet, using performance curves with near approximation. Stroke length was measured using the laser displacement sensor and nozzle area equal to the total area of outlet holes. Both pilot valves, NO and NC, are controlled with a single signal generator and are arranged as shown in Figure 36, included is the signal delay and the conversion from Simulink signal source to physical signal to the actuator pin (acting as solenoid actuator) extending and retracting the valve to the alternative positions.

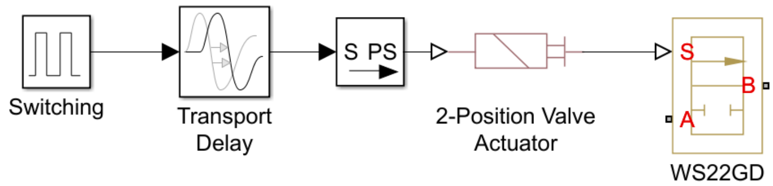


Figure 36. Pilot valves simulation model

As the initial delay in start of motion is not provided in the technical data sheet, it is approximated from the current profile and CV poppet displacement curves under pressure (wet measurement). From Figure 37 and Figure 38, it is determined to be 4.5 ms and used as a constant in the Transport delay block in Figure 36. Whereas, actuator pin switching time on and off is 3.5 ms and 4.5 ms, respectively for both NC and NO valves. Detailed block parameters are provided in Appendix 9.

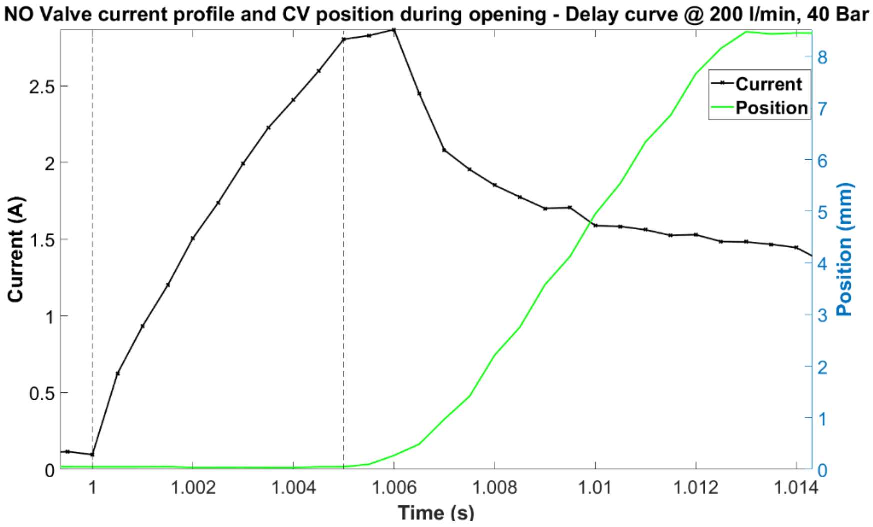


Figure 37.Pilot valve current and CV displacement profile during opening under load to measure delay

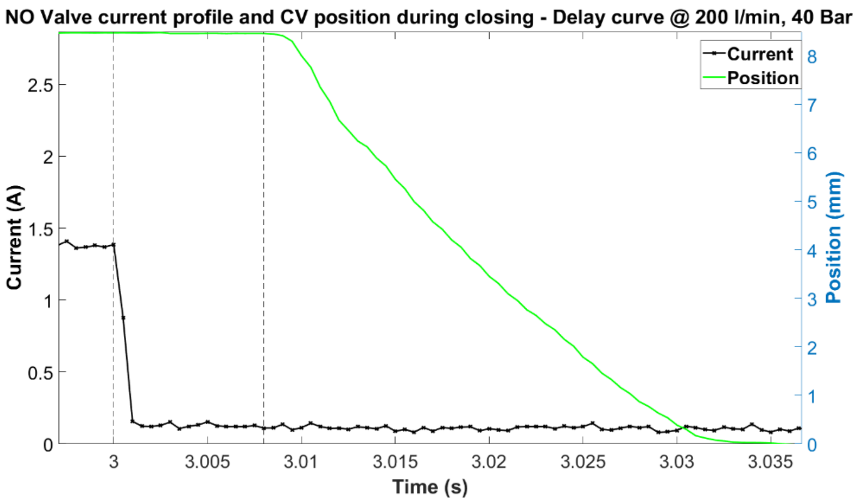


Figure 38. Pilot valve current and CV displacement profile during closing under load to measure delay

The delay due to hydraulic forces under pressure and leakages are not computed, but the cross-flow lost to tank does occur during overlapping strokes of the valves, and has been measured and presented in results Chapter.

Cartridge Valve

The cartridge valve is modeled extensively using several components consisting of separate blocks such as a cylinder, spring and valve orifice along with the modeling of flow forces, Figure 39. The poppet in the CV is basically a differential piston with three effective surface areas, two cylinder components have been used to model it. The pilot side area has been divided into two halves, whereas for the bottom side, one is port A and second one is port B area based on the ratio of A_c : 1.04 A_A / 25 A_B and stroke of 8.5 mm. Similarly, the mass is also defined and the linear spring with a perforce of cracking pressure, equal to around 1.9 bar, is attached in parallel to the poppet.

The difference between the default cartridge seat valve library component and the self-built is the flexibility in parametrizing the objects and the use of the mathematical model to account for the forces that are neglected in the ready-made object.

The 'Pilot to high pressure' object is defined with the flow discharge coefficient, valve stem diameter and the seat angle. The detailed parameters are given in Appendix 10. The basic output of concern is the poppet displacement over time during the cycle and is saved and converted into an output variable.

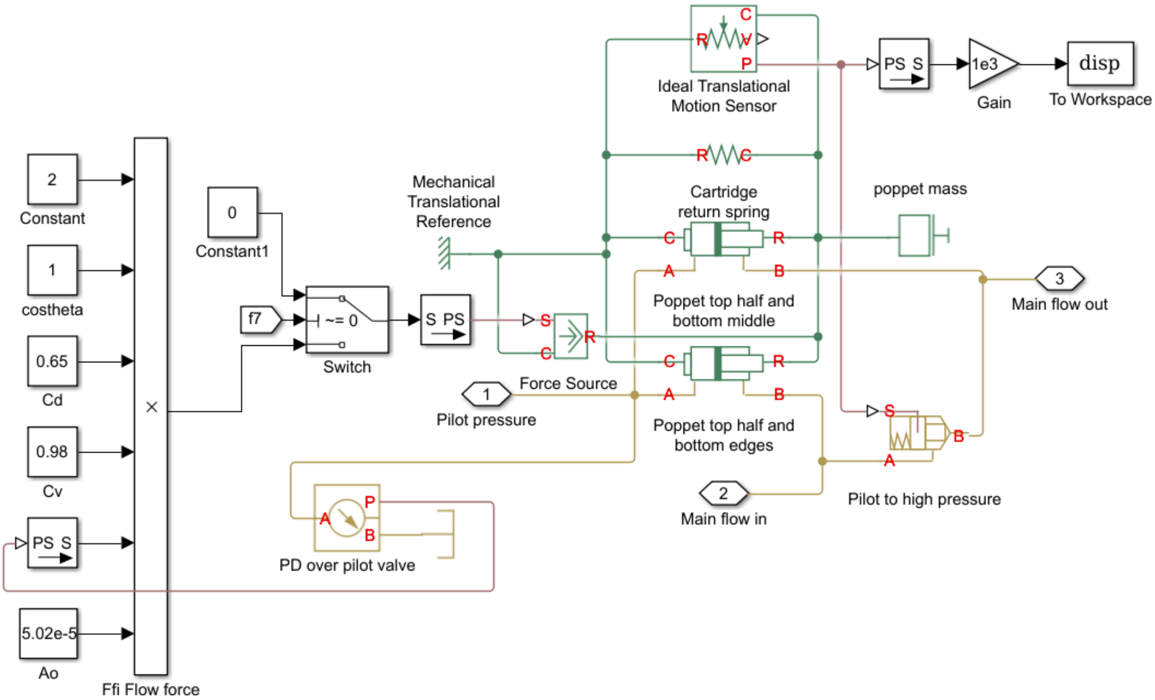


Figure 39. Cartridge valve simulation model

The flow forces, which tend to close the valve from pilot chamber (port X in Figure 14), when the pilot volume is being filled causing the poppet to work against the acting pressure difference on A and B ports, can be evaluated by the equation taken from [1],

$$F_{fi} = 2\, C_d C_v A_o cos(\emptyset)\Delta p \qquad (11)$$

C_d is discharge coefficient of orifice into the pilot chamber taken to be 0.65. Velocity coefficient C_v taken to be the 0.98 [34] for fairly uncontracted orifice, while its defined as the actual velocity of the jet divided by theoretical velocity. Orifice area A_o, the diameter is 8 mm of orifice in the he cap of the pilot chamber, angle Ø of flow jet axially to the orifice, that is zero, and pressure difference Δp across pilot chamber. The flow force is switched to zero during the valve opening to simulate the flow leaving the pilot volume.

Throttle and bypass valve

Check valve is used as a bypass valve to create the pressure difference across the CV and to block the pressure peaks traversing to the high-pressure line during switching. The PRV is used to create pressure load in the system and various range of cracking pressures are set and simulated to map the dynamic operation and the timing of valve opening and closing.

The valve dynamics such as opening delay of bypassing and load valves are not considered, and the check valve and the PRV are assumed to allow the fluid to pass rapidly.

Flow source and fluid

In the model, the ideal flow source is used to provide a constant flow rate, defined by an input variable to the main line and internal piloting. The hydraulic fluid specified in the model is similar to the fluid used in the experiment, that is ISO 46 VG and density of 873 kg/m^3 at system temperature 29.5 ºC. The fluid is returned to the reservoir.

3.2 Basic model for valve's dynamic measurements

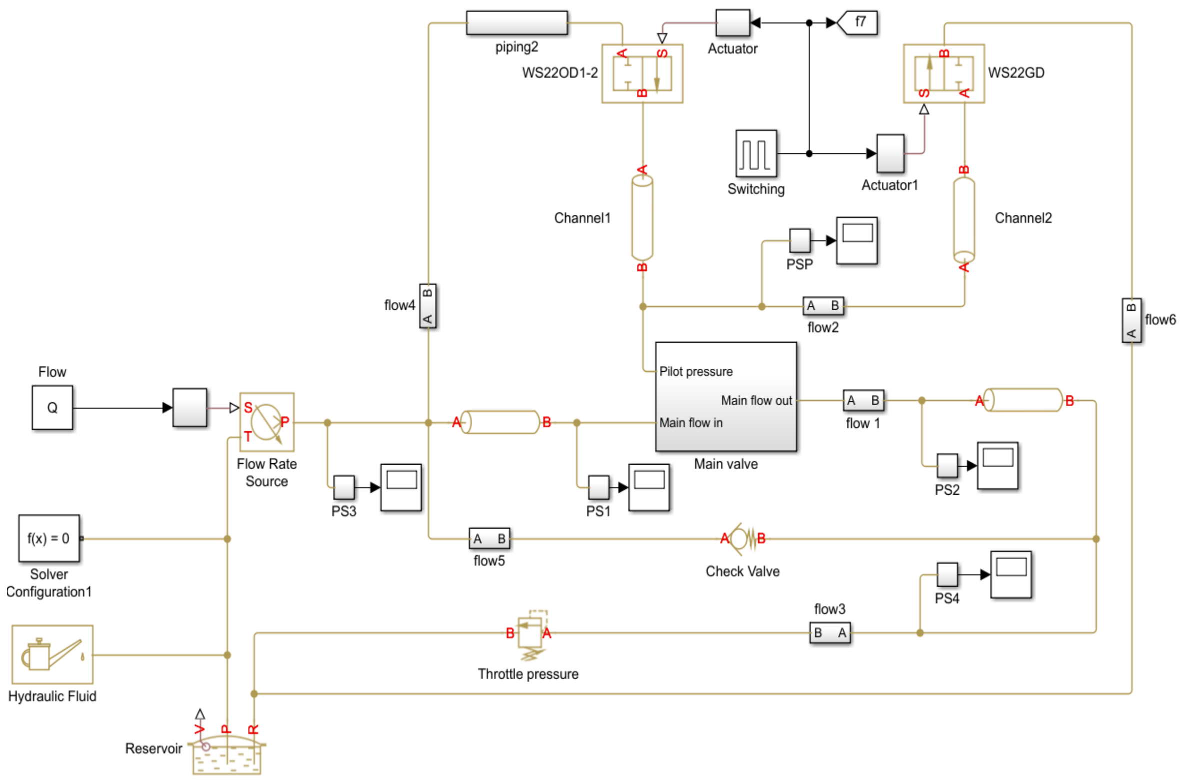


Figure 40. Uni-flow valve setup simulation model

The model in Figure 40 is utilized to create a reference map for different operating conditions and could be used for replicating the intermittent, continuous or discrete switching cases. The additional piping on the low-pressure sides have been removed from the model, since they are the only constraints of the experimental test system causing pressure drop.

4 Measurement Results and Analysis of Uni-Directional flow Actuation

In this chapter, measurements acquired from the experiment are presented for uni-flow directional control valve. Along with that, simulations results are compared with measured results, and the model is tuned so that it can be used as a reliable model for two-stage valve for further analysis.

This Chapter is divided into four sections, where the first is static measurements at full open condition to test the functionality of the system and operate at varying flow rate. In the second section, the leakages are measured in both, static and dynamic conditions at pilot line, to deduce loss of flow and energy while in operation.

In the third section, under the dynamic measurements, different switching modes are performed and measurements are plotted as transient data. The valve actuation type includes one-time switching, in which valve is kept open and closed for extended period of time, continuous switching, where valve is actuated at different frequencies and duty cycle ranges to observe any progressive delays or consistency in operation. Further two are ballistic switching and independent pilot valve controlled switching.

In the final section of this chapter, the analysis is drawn as a conclusion on performance and operating requirements such as energy consumption, valve dynamics at certain operating conditions, optimization measures which can be taken to reduce switching time, and the anomalies in result due to current experiment setup.

It is to be noted for the rest of the measured results, where continuous fluctuation is high have been smoothened (or filtered) mildly without the loss of significant data. The fluctuations may well be because of the induced voltage, due to noisy power supplies or main motors. As the sensor with AO in voltage signals are susceptible to the electrical noise and in this case offset the readings within ±0.2 V range, but were found to be within acceptable tolerance due to use of high-resolution sensors.

4.1.1 Fully open condition

In the first experiment, the steady state flow characteristics of the valve was recorded, where the flow rate is varied over the time from 0–200–0 l/min. The pilot valves are activated to cut-off the pilot pressure and to let CV open. The pressure difference is also plotted across the CV, for both in measurements and simulation to show the pressure loss with respect to the flow rate.

In Figure 41, it can be noticed in the first 60 secs, that the pressure difference rises to 2 bar at a lower flow rate, which is the pre-load pressure of the CV. Afterwards, the poppet starts to open proportionally to the flow rate, as seen in Figure 42, until it reaches the maximum stroke of 8.5 mm.

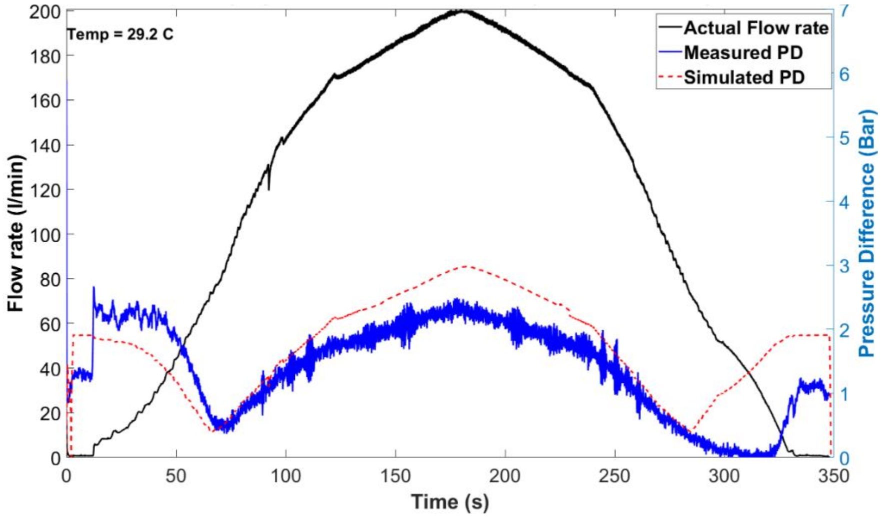


Figure 41. Flow rate from 0−200−0 l/min varied continuously at zero pilot pressure and pressure drop plotted across the CV

The measured pressure drop curve almost follows the performance curve provided in the datasheet, see Appendix 12, and simulation output is also well within ±0.5 bar range. The slight oscillations can be observed in the symmetric curve of the pressure difference in measured data, when transitioning into higher flow rates from 130 l/min and above and similarly when transitioning into lower flow rates in second half. This could be narrowed down due to the flow entering into the transition zone, between laminar and turbulence. For this transition zone the Reynolds number, N_R, which can be calculated with Equation (12), is between 2000 and 4000 for the pipe flow.

$$N_R = \frac{v_{avg} D}{\mathrm{v}} \tag{12}$$

Where v_{avg} is the velocity of the flow, and can be calculated from the Equation (7), D is the hydraulic diameter that is the flow passage diameter in the block, in this case 17.5 mm, and v is the kinematic viscosity taken at 29.5 ⁰C to be 76 cSt.

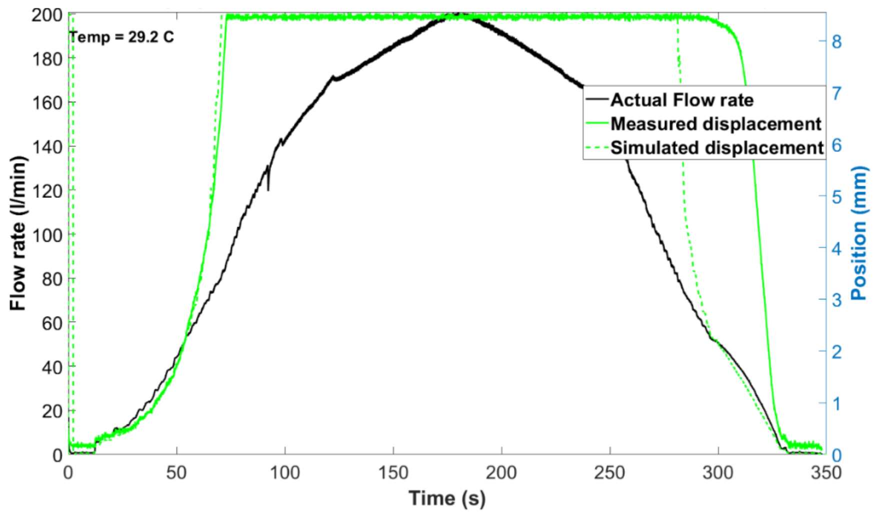


Figure 42. The CV poppet displacement over the varying flow rate at static condition

The movement of CV poppet starts when the pressure difference over the poppet becomes equal to the preload cracking pressure of the valve, and the poppet starts to close down at around 280 secs (Figure 42), and closes fully when the flow rate diminishes to zero and then the pressure difference rises back to the cracking pressure. Pressure drop from two sets of pressure sensors at inlet and outlet, and the position of poppet as function of flow rate is also presented in Appendix 12.

4.2 Pilot leakages

In the experiment, multiple scenarios were tested to measure any potential leakages and possible loss of pressure and flow due to it, particularly in the pilot lines. Total three cases were run to measure the leaked flow volume. The simple method approach was that the return line is unscrewed from the reservoir and drained to a clear plastic bottle to measure the average flow loss over different test cases over time. Specified number of cycles were run for each operating condition to measure the amount of leakages.

The weight the drained hydraulic oil i.e. ISO VG 46 was measured and the lost oil volume was calculated with the density Equation (13).

$$V = \frac{m}{\rho} \tag{13}$$

Where ρ is taken to be 873.5 kg/m^3 at 29.5 ºC, m is the measured fluid mass, and the V to be the total volume lost at pilot line actuation.

No switching
Two types of leakages were checked, first from solenoid valve under pressure and second keeping the CV fully open to check for leakages from the main stage into the the pilot chamber.

In first case, CV is kept closed while the flow rate is varied for a stipulated time, from 0–150 l/min at 40 bar pressure supply in the main line. The internal piloting valves were kept at normal condition (non-active) to check for the leakages through the solenoid valves. Few drops were accumulated and assumed to be the 5 drops or 0.2 ml/min quoted in the on/off solenoid valve section.

In the second case, possible fluid seepage from the main line into the pilot chamber within CV was checked by shutting off the pilot line with ball valve and keeping the CV fully open at 150 l/min and 40 bar pressure for a brief period of time. In this case the quantity of leakage was found to be less than 0.5 ml.

Leakage per cycle
In the third case, the cyclic switching is performed to measure the average fluid lost per cycle, inclusive of the volume lost through cross-flow during overlapping of pilot valves in the opening and closing period and also the poppet’s pilot volume that is equal to 2 ml.

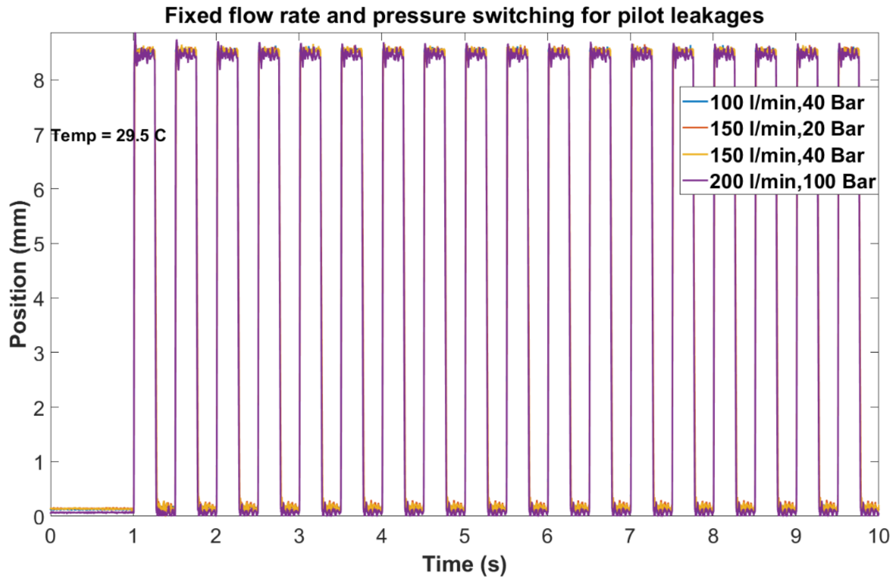


Figure 43. The switching cycles at different operating conditions to measure leakages

As shown in Figure 43, the full stroke poppet movement and continuous switching is performed at the frequency of 2 Hz and 50% duty cycle at different operating conditions. The flow lost (includes cross flow and CV pilot volume) to the measuring bottle over 10 seconds of operation is divided by the 18 valve actuation cycles. In Table 5, the fluid loss is listed and appears to be the average volume of 2.58 ml at the full cycle and stands similar to the value computed in the simulation, at 2.5 ml, see Figure 44 for reference.

Table 5. The fluid loss per switching cycle at different operating conditions

Case	**Crossflow loss per cycle in ml (gm) + 2 ml of pilot volume**
100 l/min, 40 bar	0.51 (0.445)
150 l/min, 20 bar	0.67 (0.583)
150 l/min, 40 bar	0.48 (0.417)
200 l/min, 100 bar	0.67 (0.583)

From Figure 44 it can be observed, keeping in view the delay and switching time (4.5 ms + 3.5 ms) until 1.08 ms, initially it is the cross-flow and then the pilot volume. The optimization and analysis are further discussed in the last section.

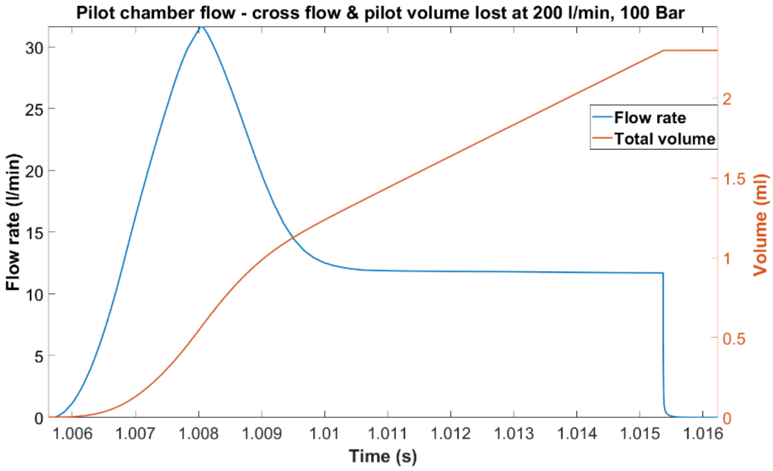


Figure 44. The flow rate (or loss of fluid at opening) out to the return line of the pilot valve during switching

4.3 Dynamic measurements

This section covers the dynamic switching of the on/off valve at various operating conditions, such as supply pressure, pressure difference across CV and the flow rate. In addition to that, different switching operations are performed, which are one-time, continuous and controlled switching.

4.3.1 One-time switching

First of all, the model created for switching measurements, Figure 40, was used to do a sweep parameter simulation varying the pressure difference across the valve, from 0 to 200 bar in steps of 20 bar and for each step of 25 l/min flow rate up to 250 l/min. A 3D surface graph is plotted in Figure 45, to map for both, the opening and closing duration of the valve under different conditions. To note that, the switching time duration is measured from the point of activation command given until the full stroke of the CV's poppet, which is also inclusive of the solenoid valve delay, otherwise explicitly mentioned.

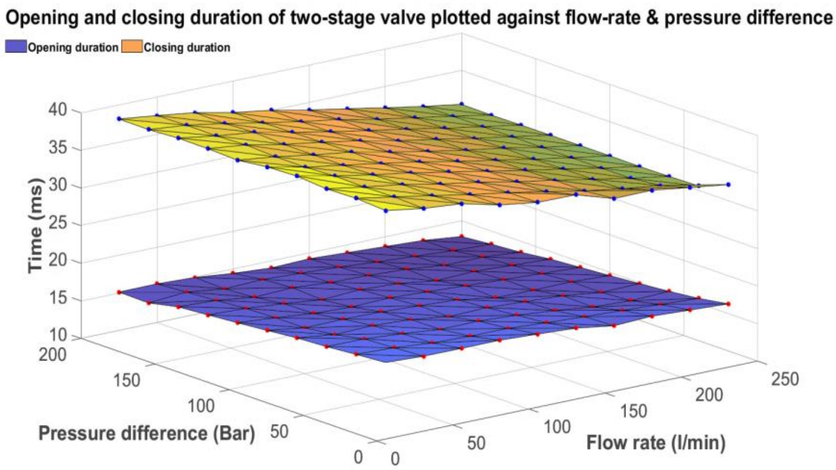


Figure 45. Simulation sweep of fixed step flow rate and PD plotted with respect to switching times

It is observed that at high pressure operating conditions (200 bar), the duration reduces to as low as 13 ms and 30 ms, respectively, keeping pilot valves switching duration constant in simulation. The increase in flow rate negligibly effects the switching duration. While, the system operating conditions, influencing the delay before the start of movement of poppet is covered in the measured results. It is deduced from the results that as the net force, based on Equation (1), acting on the CV's poppet is in positive direction, it starts to lift upwards, as the pressure difference gets smaller at the specified flow rate, the velocity gradient starts to decrease due to reductions in the pressure forces, see velocity-position figure in Appendix 13. The poppet motion can be defined based on the flow rate passing through the valve and the pressure difference,

$$q_{vm} = C_d A_{(X)} \sqrt{\frac{2\Delta p}{\rho}} \tag{14}$$

Where q_{vm} is the flow rate passing through in main line, $A_{(X)}$ area as a function of displacement of the poppet. The equation for $A_{(x)}$ is defined [1] as,

$$A_{(X)} = \pi d_s X sin\alpha \ (1 - \frac{X}{2.d_s}.sin2\alpha) \tag{15}$$

It defines the immediate orifice area at the specified distance of poppet from the seat, where d_s is the stem diameter, 18 mm in this case, X is the distance from seat, α is the poppet angle which in this case is 120 degrees.

The change in gradient of the motion during switching can be observed in the measurements and could be an important aspect in reducing the switching duration. This is further discussed in the analysis section. The dominating criteria for closing are the flow rate, pressure difference across CV's pilot port, stiffer spring stiffness, the flow forces and the piloting valves.

The measurements were acquired in several operating conditions and presented here are two different extreme operating conditions, the lower (100 l/min and 40 bar) and the higher operating point side (200 l/min and 100 bar).

The difference in poppet motion over actuation period at both conditions is paramount and it is imperative to highlight effect on system operation such as, pressure peaks, oscillations, retardation or increase in speed of valve actuation, as these would potentially become a good set of reference for a refined simulation model besides the potential use in the listed application.

The measurement system was equipped with high-pressure accumulator (HPA) and low-pressure accumulator (LPA) to dampen pressure peaks and sudden back pressures. Initially, before switching measurements, the system was run for considerable time to set the operating conditions as desired. This included setting the load pressure by varying the throttle, adjusting the required flow rate and pre-charge pressures of HPA and LPA which were set to 12 bar and 10 bar, respectively.

The first sets of data in graph illustrations, Figure 46, shows the operated condition at 100 l/min flow rate and 8 bar pressure difference across CV with 40 bar supply pressure. The switching mode is one-cycle of actuation for the period of 4 seconds, kept open for 2 seconds and closed in the remaining.

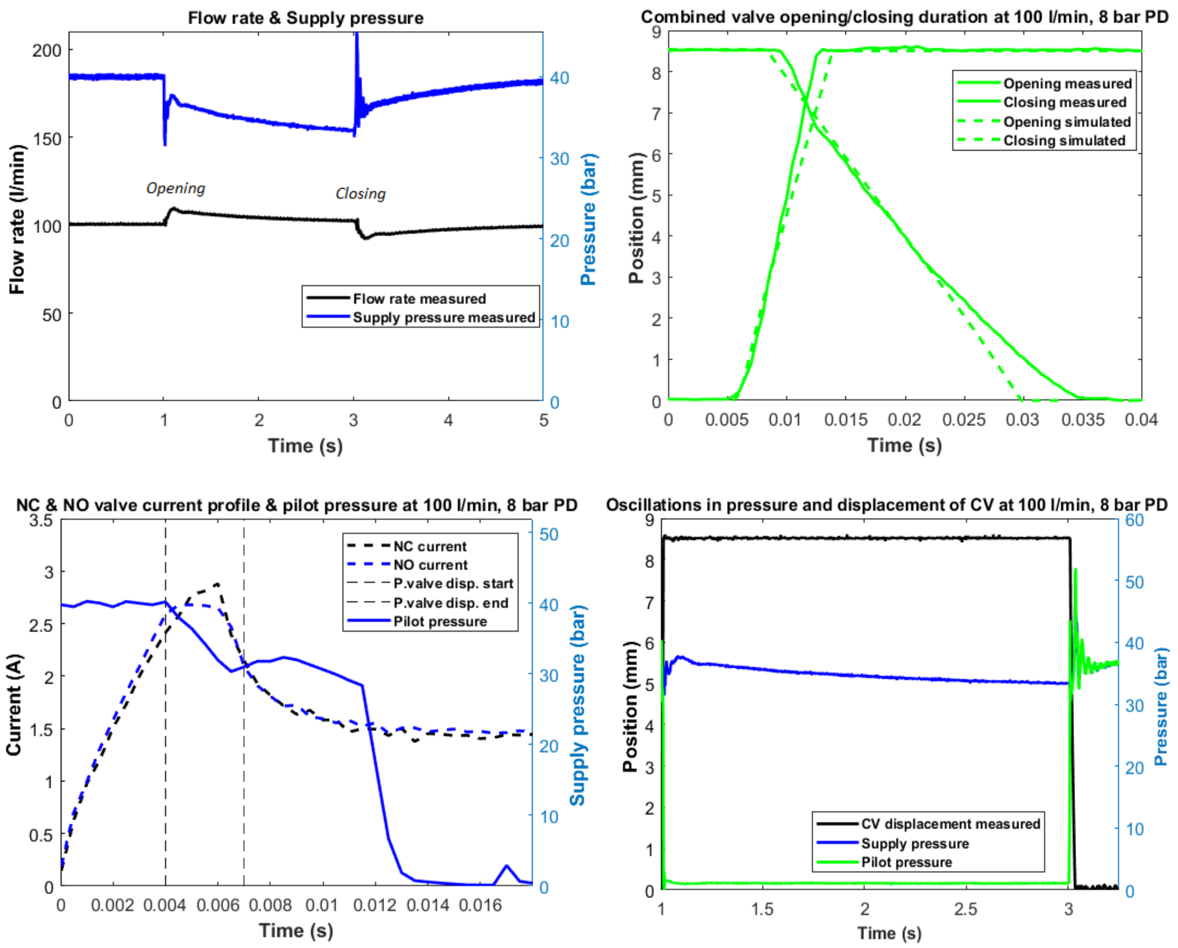


Figure 46. One-time switching performance and characteristics at 100 l/min and 8 bar PD

On top-left graph is the flow rate and pressure supplied graph of 100 l/min and 40 bar supply. When the CV is switched open, the supply pressure drops to a level equal to the load + pressure drop across CV and restores back to cracking pressure of check valve + load pressure upon switching off of the main valve. Flow rate rises slightly at lower supply pressure region due to excess flow rate supplied by the charged HPA.

In the second graph, top-right, the measured and simulated CV displacement is plotted over the time and duration recorded for measurement is about 13 ms, from the point of actuation to the full stroke of 8.5 mm. Closing is at about 35 ms, however, in simulation the valve closes earlier due to slightly reduced pressure drop (in the CV virtual model, but follows good approximation in the current and later results.

In the third figure, bottom-left, the current profiles of pilot valves (NC and NO) are plotted with respect to the pilot supply pressure. The pilot supply pressure affixed in the pilot cover block can be one way to deduce the pilot valve opening and complete closing against the amperage readings by monitoring the fall and rise of pressure in pilot volume chamber. It can be observed, that the delay in solenoid switching, dwelling under pressure forces combined with the solenoid resistance, is about 4 ms from the activation signal. It can be ascertained from the graph, that the pilot valve motion starts from 4 ms until 7.5 ms, based on the higher current consumption and drop in pilot pressure. From 7.5ms to 12 ms, the pilot pressure reading remains stable as the pilot volume is still passing through the pilot valves and to the tank. The closing plot current profile is almost similar to one in Figure 38, shown with the delay and start of CV's poppet movement.

The fourth figure, on bottom-right, shows the full displacement curve combined with pressure at the pilot and high-pressure lines to show the pressure transient effects due to switching. It can be noticed, that at both maximum opening and closing, a small degree of oscillations (pressure-displacement curves in Figure 46) is visible that could be due to fluid compressibility or spring effect. Whereas higher amplitude of oscillations can be observed for both pilot and supply pressure readings following the same trend and frequency at the closing point, explanations covered in the next measurement. The slight lifting of poppet after full closing takes place that is under +0.1 mm within the first 50 ms and also pressure oscillations gradually subsides by 150 ms after closing.

These above described measurements preceding the next case of 200 l/min, 100 bar supply pressure, and 80 bar pressure difference, set a good reference for evaluation of the valve characteristics at higher rated flow and operating pressure for high-performance system and the homogeneous pattern data is presented in Figure 47.

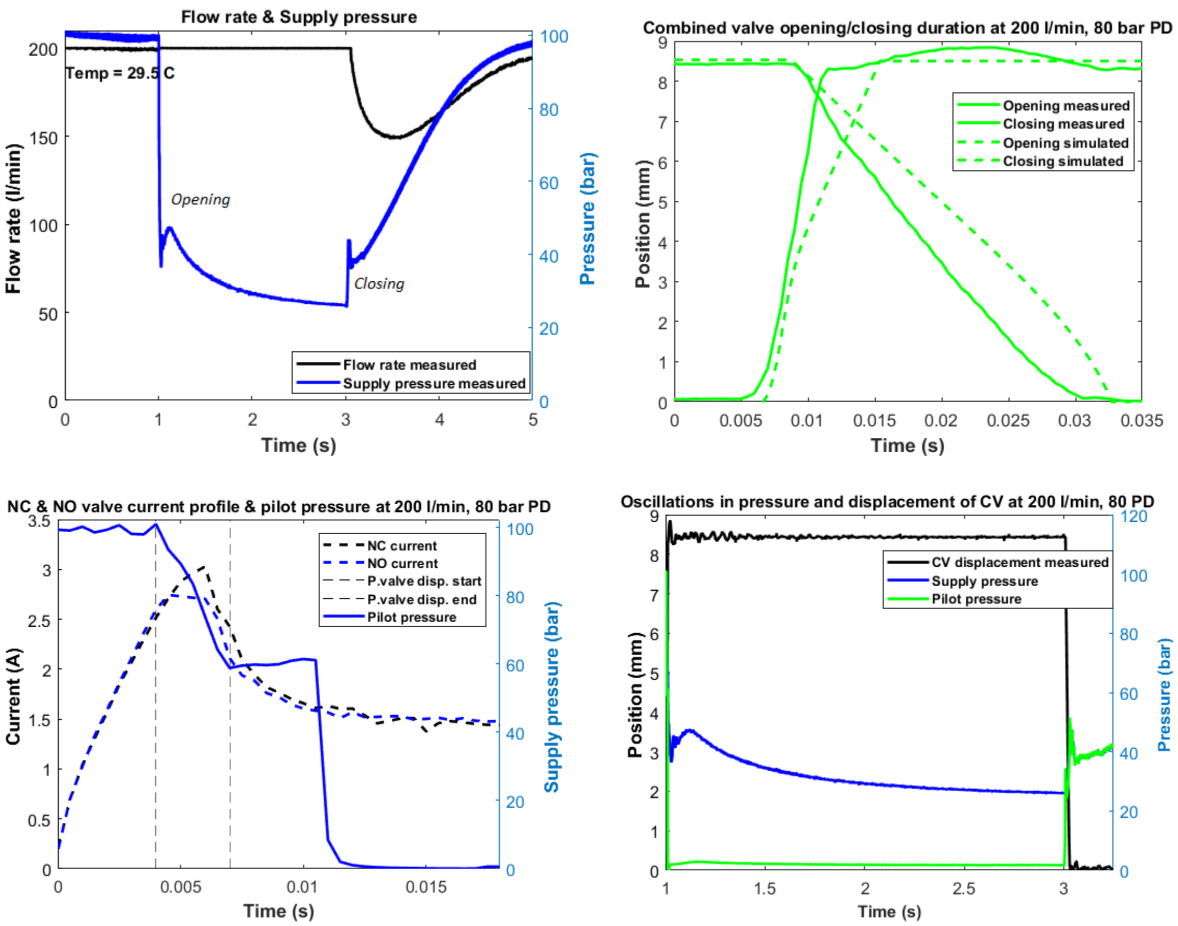


Figure 47.One-time switching performance and characteristics at 200 l/min and 80 bar PD

The analysis of the results at the above-mentioned operating conditions presented in Figure 47, are congruent to Figure 46 results but with the perceptible difference in the outcome. In this case, the flow rate is set to 200 l/min and 100 bar supply pressure. Even though the flow rate from pump was set to 200 l/min and also the corresponding flow sensor analog output to 10 V, the excess flow rate due to accumulator has not been captured compared to operation at 100 l/min, in Figure 46.

Furthermore, the poppet opening time is reduced to 12 ms at higher pressure, comparatively to 40 bar, just a minuscule reduction of 1 ms, however, there is a higher gradient in the first half of displacement. For closing, it was reduced to 30 ms, and again with a little bump in reopening of around +0.1 mm. The delay in the pilot valve is increased to 4.5 ms (from 2.5 ms in dry measurement and 0.5 ms from 40 bar supply pressure case in Figure 46) and full opening may well cross 8 ms or more. As the result of the extended opening duration of the pilot valve can cause pressure to act on the pilot chamber longer than the lower pressure level case, slowing down the CV opening.

In the current case, the oscillations in pressure and displacement curves, at opening and closing tend to be larger than the previous case. This can be pinpointed to be due to pressure waves traversing through the long and slender flexible hoses in the high-pressure side between HPA and the valve ports A and X. The fourth illustration from Figure 47, is presented in enlarged view in Appendix 12. In line with the undesired oscillations, in the intended applications the long lines are largely avoided when building a compact system and digital valving system is essentially compact and within the rigid block of the manifold. In one article [17] the integrated accumulators nearby the main valve are used in the test system to dampen the oscillations in the displacement. The cause of oscillations discussed above are further explored in an article [1] through simulation at three different lengths, is provided in Figure 48.

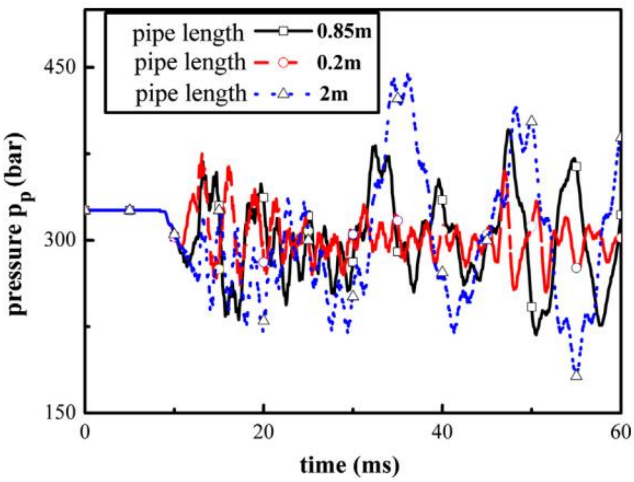


Figure 48. Pressure fluctuations between the accumulator and valve at different lengths [1]

The amplitude and the period of fluctuation vary with the hose length and operating pressure upon sudden stagnation of the flow, in hydraulics also known as water hammer effect. In this physical system, the hoses installed are ranging from 0.8 m to 1.75 m. This analysis can be utilized in future experiments to reduce the conduit sizes in aimed at valve analysis to reduce uncontrolled oscillations.

Further, the CV poppet's displacement duration is mapped for both opening and closing in Figure 49 and Figure 50.

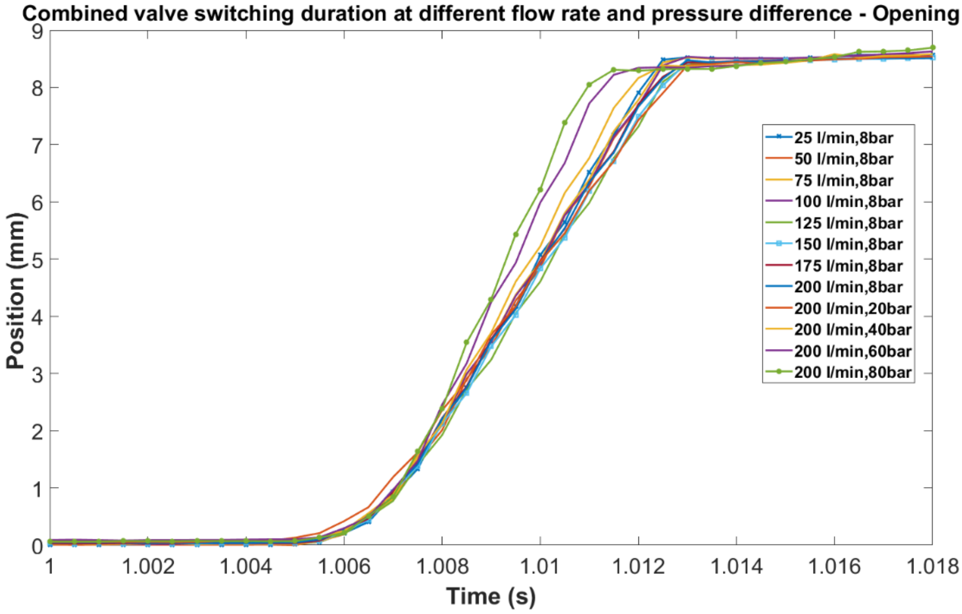


Figure 49. Combined valve switching duration at a varied flow rate and PD - opening

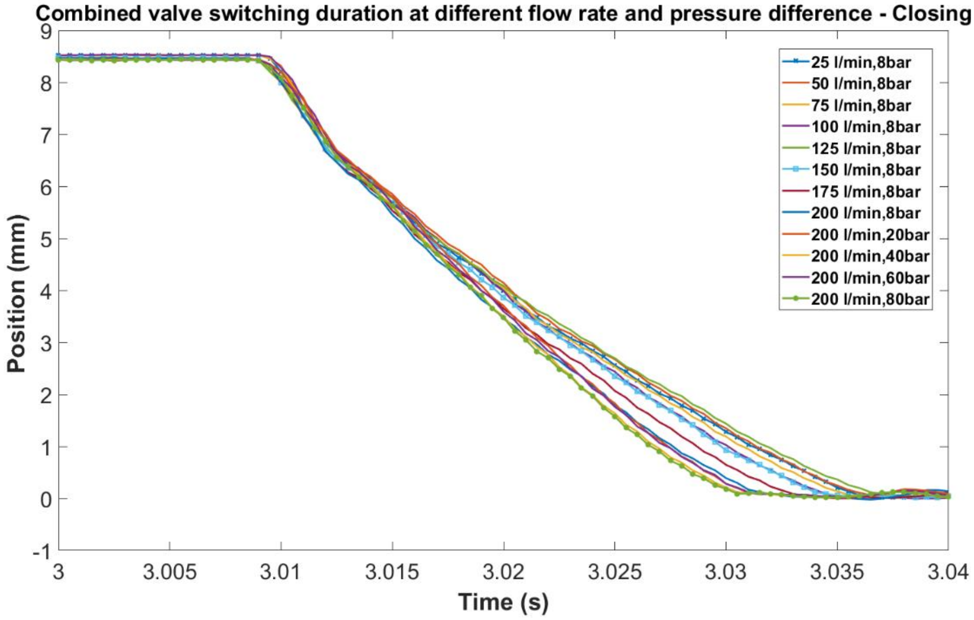


Figure 50. Combined valve switching duration at a varied flow rate and PD – closing

The high-pressure conditions are the far-left ones to be the fastest at 200 l/min and 20 bar PD and above and the far-right curves at 8 bar PD. However, the switching delays (until 5 ms) and second half of the motion (displacement over 5 mm) takes up most of the time during the opening. This is discussed in the last section of this Chapter dealing results analysis.

4.3.2 Continuous switching

The second type of switching method tested, in the thesis, is the continuous switching, where the two-stage valve is actuated with a defined frequency and duty cycle for a brief period of time. Similar to digital electronics where modulated signals can be used to control the voltage level, this same principle can be applied in the digital valve scheme, however, new digital valve technologies are not supposed to switch continuously, but supply flow in a discrete manner in conjunction with the parallel valves of same sizes. The continuous switching based on PWM signal has been studied in one article [35], where it is explained the error due to unpredictable valve opening, valve delays may affect the flow control and also in another article [36], the inappropriate installation may induce vibrations, pressure peaks and excessive noise in the system.

One of the reasons to do continuous switching was to measure the repeatability of the displacement and any lag in switching time after consecutive switching in a definite interval of time. In total, four different frequency and duty cycles were selected, enough to perform full opening and closing stroke of CV consistently. But remained unsuccessful, a reference to Figure 51, due to pulsating of flow, charging and discharging of accumulator and pressure traversing at high flow rate caused non-uniform cycles, however, the system operated well even at 20 Hz switching and high flow rate without physical fallout in the system.

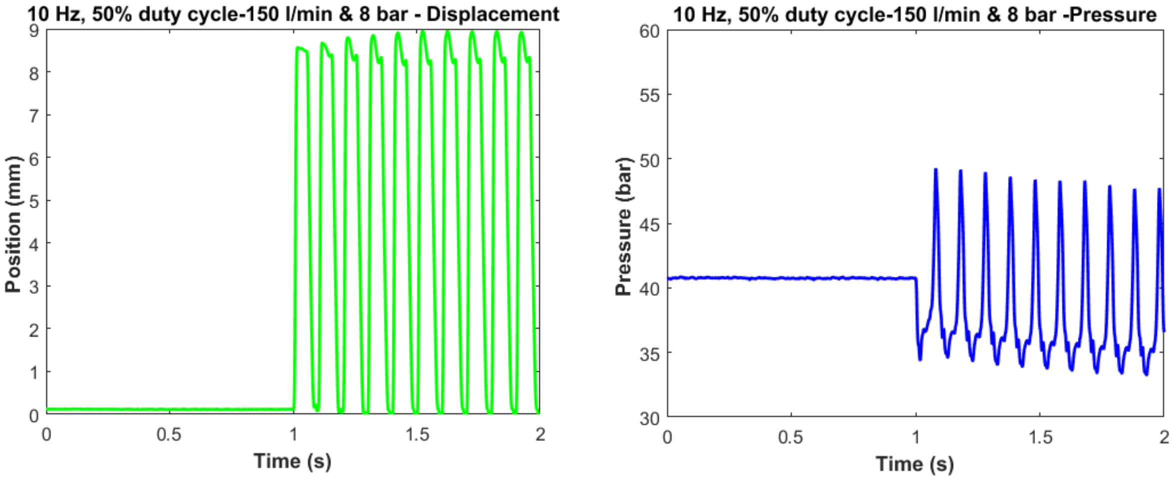


Figure 51. Continuous switching of the valve at 10 Hz and 50% duty cycle - Displacement and supply pressure graph

In one dissertation [15], the continuous switching is discussed by utilizing the PWM signals to control the switching. This method results into the pressure being averaged out between the high pressure and low-pressure line based on the pulse width control. With the availability of high-pressure source, this method allows adjusting the system pressure to be suitable for the required load pressure. Along with that, the proportional flow rate can be supplied through the same control by varying the signal control at high frequency.

The system was operated at an even higher frequency of 20 Hz, where it showed the pattern of the valve position hovering over the seat at about 1.5 mm (see Appendix 12). This could enable the throttling of the flow and overall reduced flow rate for minor position control may be achieved. But it might not be feasible enough in comparison with the direct acting solenoid digital valve because of two-stage control and pilot leakages resulting in flow losses. Also, the Bucher valve's operating frequency is limited to a maximum 60 Hz and 30% duty cycle, escalating the values further results in excess coil temperature rise.

4.3.3 Pilot valve-controlled switching

Controlling the pilot valve signal opens more ways to control the CV. This can be achieved by modulating the switching signals of the two pilot valves, either individually or combined. Two types of control have been defined, which could be useful for following purpose, limiting flow rate, main poppet switching duration control, and flow release in smaller steps for the slightest movement of the actuator.

CV stroke control

In CV stroke control the pilot valves, both NC and NO, are independently supplied the switching signal opposed to one source signal in previous cases. If the flow is to pass from the main stage for duration of x seconds, the NO valve is activated and cuts off the pressure port, in parallel pulse width to NC valve is supplied for short period of time in the fraction of x. By this method, the pilot volume returning to tank line is stop short before the poppet completes its full stroke and locking the fluid in the pilot chamber and in turn proportionally limiting the stroke of the CV.

The schematic in Figure 52 shows the valve at the deactivated condition, keeping the CV closed due to pressure from NO. When NC is activated for a short period of time it lets the pilot flow to pass to tank and enables lifting of the poppet to a desired position. This method would be suitable in limiting the stroke of the CV poppet, as well as creating the higher-pressure difference can be created across CV metering at the corresponding flow rate due to reduce flow area.

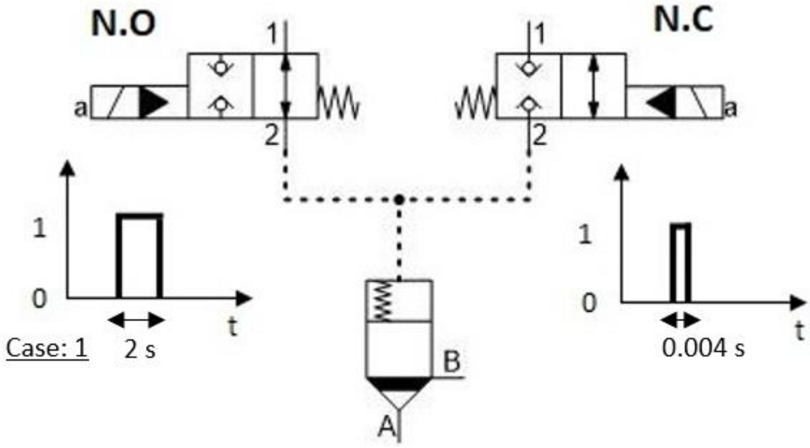


Figure 52. CV poppet's stroke control through the pilot valve's independent signal pulse control

Two sets of cases were experimented, listed in Table 6, where the first experiment is run for 5 sec, and the signal starts at time 2 sec. The CV is kept open for first 2 sec and closed for the remaining time. NC valve remains active for 4 ms.

Table 6. CV stroke control case setup

Case	Flow rate (l/min) – Pressure (bar)	NO valve pulse	NC valve pulse	CV dis-placement
1	150 - 26	4 s period – 50% duty cycle	4 s period – 0.1% duty cycle	5.06 mm
2	150 - 26	2 s period – 50% duty cycle	2 s period – 0.25% duty cycle	6.8 mm

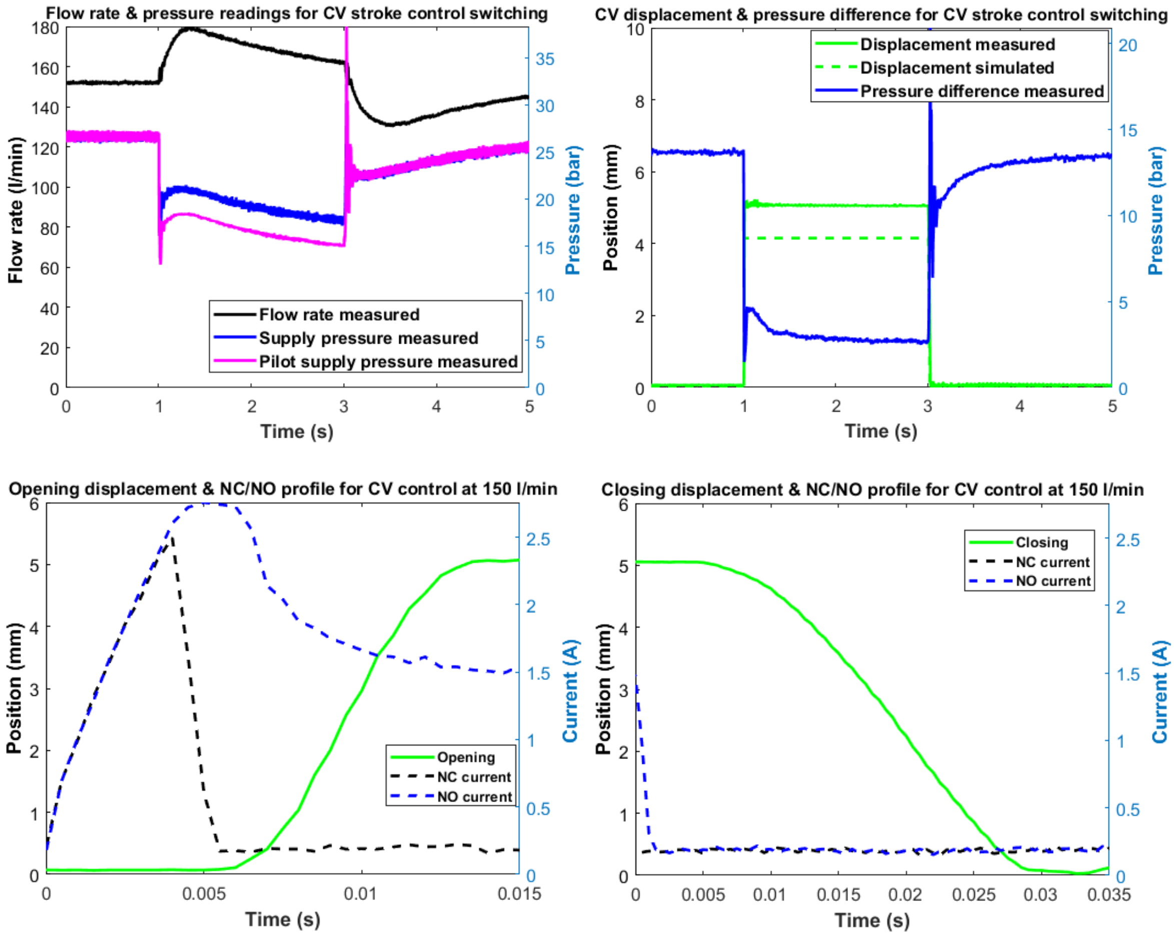


Figure 53. CV control switching graphs for case 1

In Figure 53, the respective graphs for the case 1 are plotted, where on top-left are the total flow rate passing through flow sensor, supply pressure, and pilot supply pressure. It should be noted that in previous switching cases, pilot pressure remained at 0 bar when CV was at open position while here a pressure lesser than the high-pressure line (supply pressure) is exerted in the closed chamber. In top-right graph simulated and measured displacement is shown at level of about 5.06 mm at the listed pulse signal of NC. The pressure drop remains at 3 bar, same as for fully open condition.

Bottom graphs show the illustration of position with respect to the current profiles, wherein the left image the NC current starts to drop to zero after 4 ms. However, the opening time even up to 5.06 mm, contrary to 8.5 mm of a full stroke, remains the same at 13 ms, while, a slight improvement in closing duration at around 30 ms, 5 ms less at similar pressure operating condition. The reason for the similar opening time duration can be concluded, based on pressure-force equation on poppet, due to pressure force also acting on the pilot chamber.

Ballistic valve control

The ballistic valve control, a term originally adopted from the article [37] and also mentioned in the dissertation [15], is used to achieve fine positioning of the actuator in the application by passing enough volume of flow to displace the actuator at a desired position by switching the valve for a short period of time.

The ballistic control is achieved by exciting the valve for a short interval in a combined manner, which opens to let the flow pass in minute quantity, as shown in Figure 54.

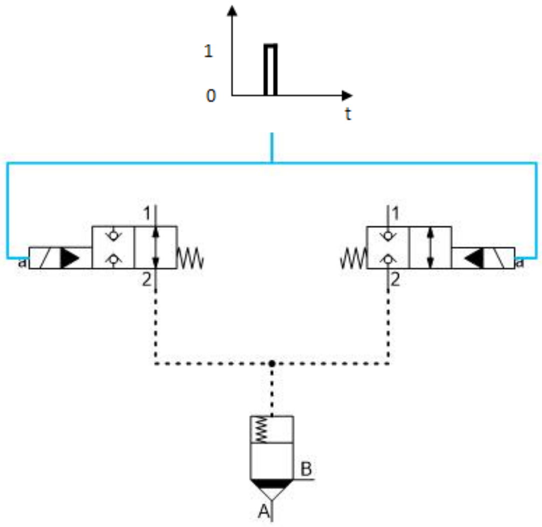


Figure 54. Ballistic control of CV opening via short pulse switching of pilot valves

Only one case was tested at conditions listed in Table 7, for about 4 ms at which the CV opens up to 3.5 mm for short period time. The relevant flow rate and supply pressure are presented in Figure 55. The total flow could not be measured as the flow rate sensor is installed at the end of the line after the bypass valve, else, it would be possible to measure the pulse per tooth of the flow sensor and measure the total volume passed.

Table 7. Ballistic switching control case setup

Case	Flow rate (l/min) – Pressure (bar)	NO/NC valve Pulse	CV displacement
1	150 - 26	1 s period – 0.4% duty cycle	3.5 mm

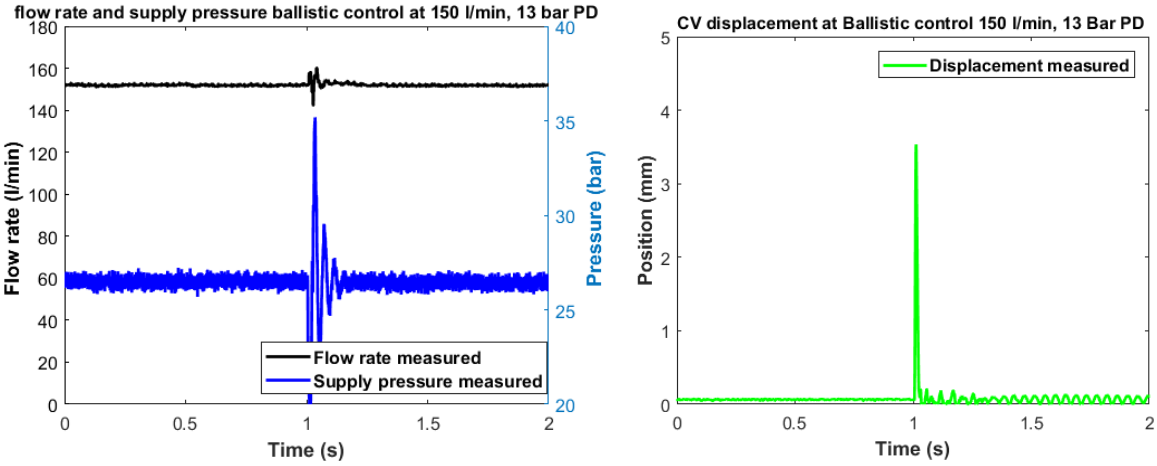


Figure 55. Ballistic control graph for case 1

Figure 56 shows the simulated results of ballistic control at varying pulse widths from 0.1% to 1% (in time 1 ms to 10 ms) in steps of 0.05% (0.5 ms). The passed total volume (in black markers) is plotted against the displacement (in green markers). This analysis could be useful in programming the algorithm for slight actuation in the application for high performance rated two-stage valves discussed in this thesis.

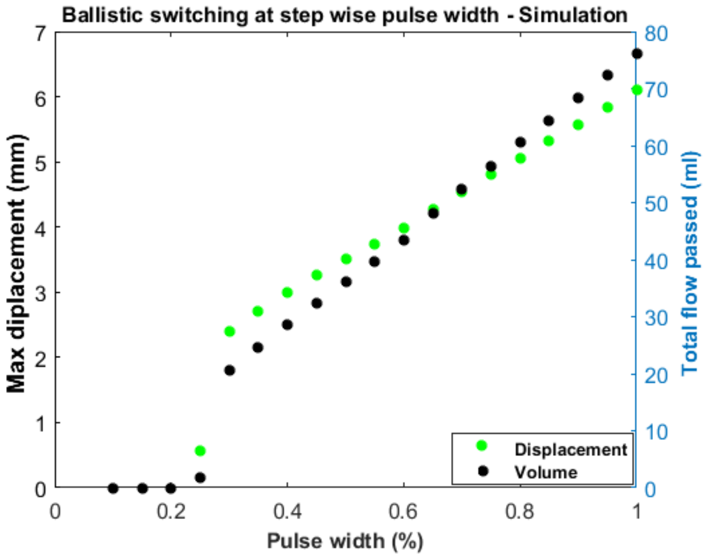


Figure 56. Ballistic switching of the valve at a fixed flow rate of 150 l/min, pressure supply 26 bar, with varying pulse width at the 1 sec period to pilot signal

4.4 Results analysis

In this section, the measurements are further discussed and analyzed on possible optimization in reducing the switching duration, recommendations and the energy lost as conclusion of the Chapter. Reference to Figure 57, the two different flow rates 100 l/min and 200 l/min at 8 bar and 80 bar pressure differences on the metering edge of the poppet have been plotted.

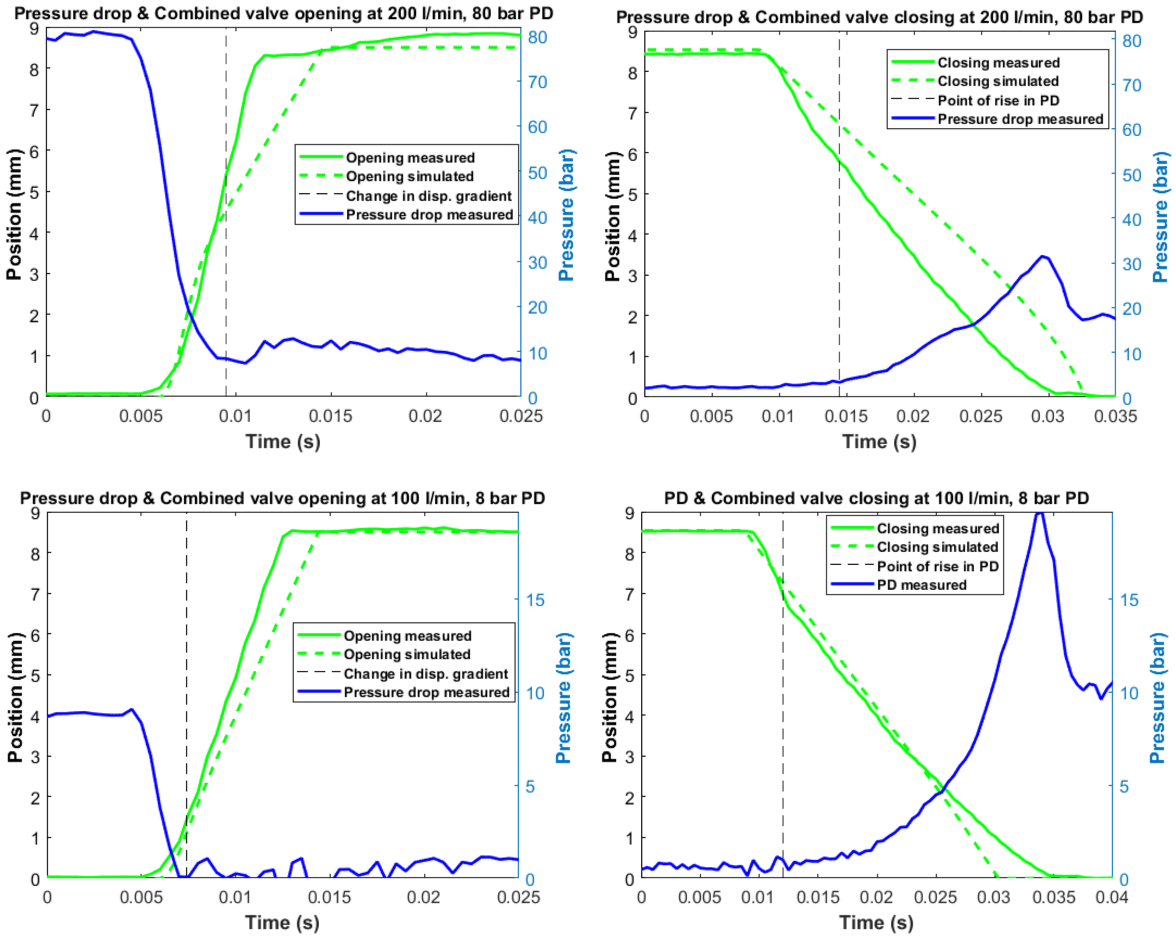


Figure 57. Pressure difference stabilization and poppet displacement relationship at 100 l/min, 8 bar PD and 200 l/min, 80 bar PD

It is observed that the pressure difference drops to the steady state value across CV at the specified flow rate, just at quarter of stroke at 100 l/min and half of stroke at 200 l/min during opening. For this reason, if the switching time is computed for full stroke, the latter half travel of the stroke takes another 4−5 ms of the time. Similarly, closing duration is also higher because of pilot volume to be filled proportionally to the stroke length,

$$V_p = \pi \frac{d_s^2}{4} s_{CV} \tag{16}$$

The pilot volume V_p can be calculated by multiplying the area of the poppet at pilot chamber with stroke length s_{CV}. In similar studies, the correlation with poppet diameter and stroke length is researched, to reduce pilot volume capacity and switching time by employing smaller diameter with the use of multi-poppet valve in studies [4], [5].

In Figure 58, only the CV's poppet stroke duration is plotted with limited stroke for two samples of operating conditions (100 l/min and 200 l/min), from the point where the poppet lifts off till reaching the stable pressure difference across CV. Only one parametric change in simulation model, Figure 40, was done is the changing of stroke length value, while all pilot dynamics and delays are inclusive.

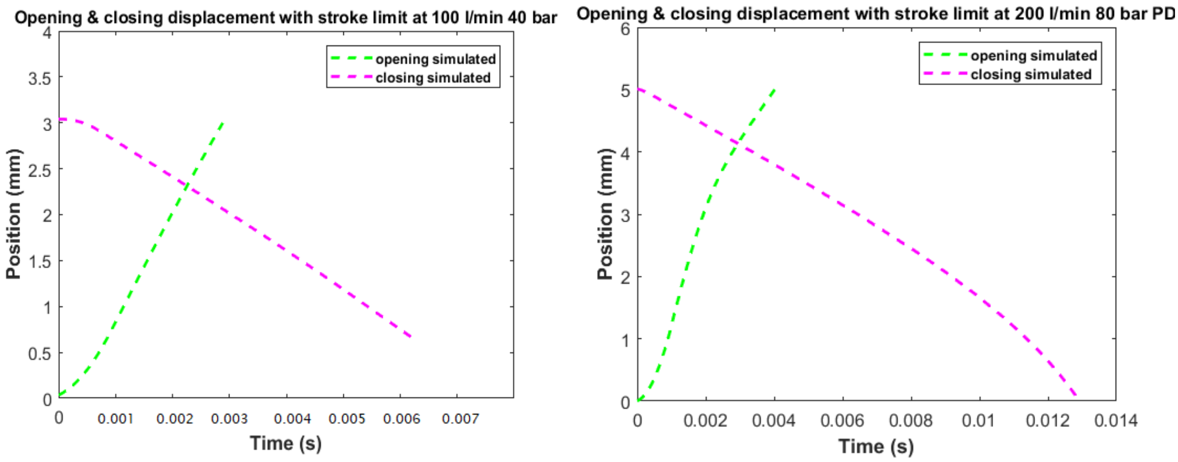


Figure 58. Opening and closing curves at limited stroke of CV at 100 l/min and 200 l/min – Simulation

The limited stroke values extracted from Figure 57, at 7 ms and 9 ms to be 2 mm and 5 mm for two different cases are used to apply the hard stop to the poppet in the simulation. These are the displacement values where flow rate saturates to its normal flow rate value without significant increase in pressure drop than the one at fully open condition. The 2 mm stroke at 100 l/min was later on changed to 3 mm due to twice the pressure drop at 1.92 bar than approximate 1 bar at full open condition. The on/off state time for opening is almost halved for both opening and closing at 100 l/min it is reduced from 27 ms to 6.5 ms for fully closed condition, and correspondingly for 200 l/min, from 25 ms to 13 ms.

Based on the application, required flow capacity and operating pressure, the stroke length can be restricted by fixing the required length of spacer between the cap and poppet. Another option explored in measurements, was the independent pilot control switching to limit the stroke, but its only benefit is slightly reduced closing duration, as during opening the poppet velocity is reduced to exerting pilot pressure based on Equation (1). Secondly, the combined energy consumption of the pilot valves is halved due to switching off the NC valve current.

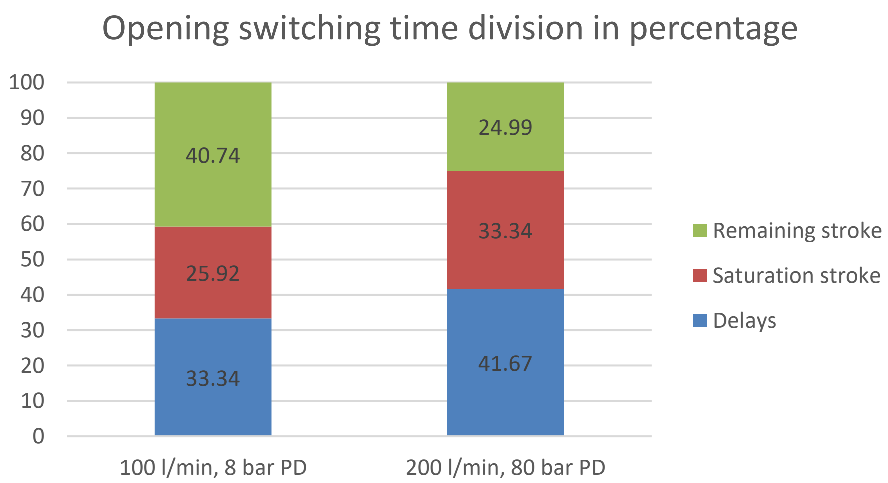


Figure 59. Opening switching time division in percentage - Solenoid and pressure force delay (blue), saturation stroke (red) and remaining partial stroke (green)

It is implied from Figure 59, limiting the stroke length would reduce the on/off switching time but also reduces the marginal gap between opening and closing duration. The reducing of gap can be complemented with using stiffer springs and independent pilot controlling, which would increase the opening duration but bring closing time closer. Secondly, the opening duration can also be retarded by employing extra orifices to slow down the movement.

Other major part is the delay, which is largely due to the impedance of the solenoid and slight delay in opening and further delayed movement due to moving against high pressure. Another recommendation is the use of miniature digital valve which has claimed switching time of about 1 ms, mentioned before in literature, but has a limited flow capacity of 9 l/min. This may limit the flow rate leaving the pilot chamber during opening and also rate of filling in during closing. By installing multiple parallel pilot valves, to bring the flow capacity around and equal to Bucher's 27–30 l/min and also limiting the stroke, may potentially shave off the switching duration.

4.4.1 Energy consumption

The consideration of energy efficiency is an integral part in implementation in a new system and so it is imperative to compute the energy consumption during the complete switching cycle. This could set a base reference to compare the conventional directional flow valves with the on/off valves assembled in set.

Initially, the energy consumed in one-time switching is calculated from the activation signal to full stroke and back to fully closed at 200 l/min, with 100 bar pressure supply and at full CV stroke of 8.5 mm. Equation (17) is used to calculate the total energy consumption to operate valve switching.

$$E_{Tloss} = \int_{t_{o1}}^{to_2} \left(V_{pvs}\, I_{NC} + V_{pvs}\, I_{NO} + \Delta p_p q_p \right) dt + \int_{t_{c1}}^{tc_2} \left(\Delta p_1 q_p \right) dt \tag{17}$$

Where E_{Tloss} is energy loss in switching, V_{pvs} is the voltage supplied for pilot valve, I_{NC} and I_{NO} current flow at both NC and NO, respectively. Δp_p is the pressure drop across pilot valves, q_p is the crossflow + pilot volume flow leaving the system. to_1 and to_2 is the time window between activation signal to full opening of the CV. tc_1 is the signal cut off point, and tc_2 is the full closing of CV, see Figure 60.

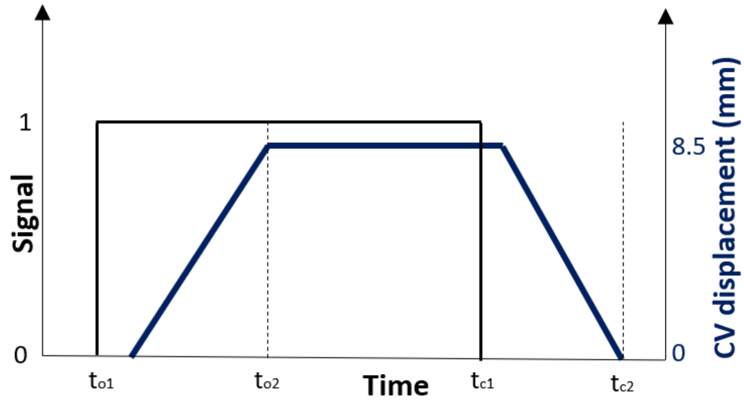


Figure 60. Signal-Valve displacement over time

During closing, only the crossflow is the loss energy, while no current is supplied to pilot valves and pilot volume remains in the system. From Equation (17) the electric energy consumption and the flow losses in terms energy, for full cycle valve actuation, has been computed from the measured data in Equation (18)

$$E_{Tloss} = 0.694\,J + 0.683\,J + 24.3\,J = 25.67\,J \tag{18}$$

Further, the power loss can be calculated using Equation (19), when the CV valve is opened to pass the main line flow, Q_m, and the pressure drop across the CV is Δp_{CV} at operating flow rate and the average given power consumption at 12 VDC for one Bucher valve is 15 W which can be taken as constant during open condition.

$$P_{loss} = \Delta p_{CV}\, q_m + V_{pvs}\, I_{NC} + V_{pvs}\, I_{NO} \tag{19}$$

The energy loss value of full switching cycle can be utilized to consider the energy required to operate the one two-stage valve in the system at given operating conditions.

This section concludes the portion of two-stage internal piloting valve for uni-flow direction control that includes experimental and simulation measurements and numerical calculation included in results and analysis. The chapter provides an important information on the operational capability of the two-stage pilot operated valve, the flow rate capacity, effects of pressure on performance, time required to switch the main valve from 0 to 1 position and induced effects on system due to internal pilot controlled and rapid actuation of valve. The data could further pave the way for building the bi-directional simulation and experimental setup for the application, to actuate the actuators such as cylinder, requiring higher flow and immediate responsive actuation.

5 Bi-Directional Valve Simulation Setup and Analysis

In this chapter, Bi-directional valve functionality is studied with a model containing an added actuator and an external load. The simulation performed to present its operation based on flow direction from either side of the CV ports (A and B). The two types of piloting, internal and external, to control two-stage bi-directional valve scheme is included along with its advantages and drawbacks related to the switching and controlling of the valve. Further, the potential applicability of bi-directional on/off valve scheme is discussed, that is Multi-pressure actuator system, and the generic changes which may require have been proposed.

5.1 Simulation model and changes

The simulation model was built with similar parameters as in the previous case of uni-flow directional control. However, the CV's poppet geometric dimensions have been modified, as shown in Figure 61. The available poppet design with the effective area ratio at metering edges $A_A = 0.6A_X$ and $A_B = 0.4A_X$ is used in simulation, from herein referred as CV06. The lesser area ratio at the inlet side causes slightly higher switching duration at the opening and lower time during closing, based on the net force poppet Equation (1). The velocity-position graph at both area ratios and same operating condition is plotted in Appendix 13.

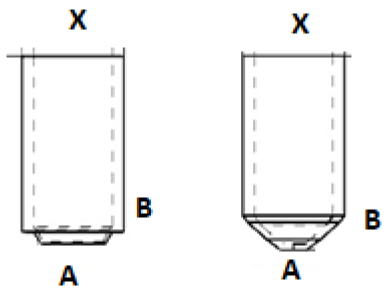


Figure 61. CV poppet design. Area ratio of $A_A = 0.96A_X$ (left) and $A_A = 0.6A_X$ (right)

Secondly, the simulation model is slightly modified to operate in both directions of flow from port A to B and inversely from the actuator under the load. In the thesis study, initially the simple operating conditions are defined and the model is simulated to highlight the requirements and controlling method required to operate a bi-directional on/off valve, without introducing the throttling losses in the flow. To test the bi-directional functionality, the actuator with an external load is included to control the position of the cylinder.

For the actuator modeling, consisting of load and hydraulic cylinder, dimensions are referenced from a catalog [38] used in medium sized construction machinery and geometrically larger than the one used in multi-pressure actuator articles [3], [13] to have power rating at the actuator greater than 10 kW. The model is simulated to move the piston at constant velocity condition, that is 0.27 m/s velocity (or flow rate at 200 l/min) and output power to be 25 kW set as design benchmark value. The geometric dimensions of the cylinder are given in Appendix 13 and lifting external load of 10,000 N is applied.

The simulation model is given in Figure 62, to generate results related to switching time and control of actuator and piloting type is internal.

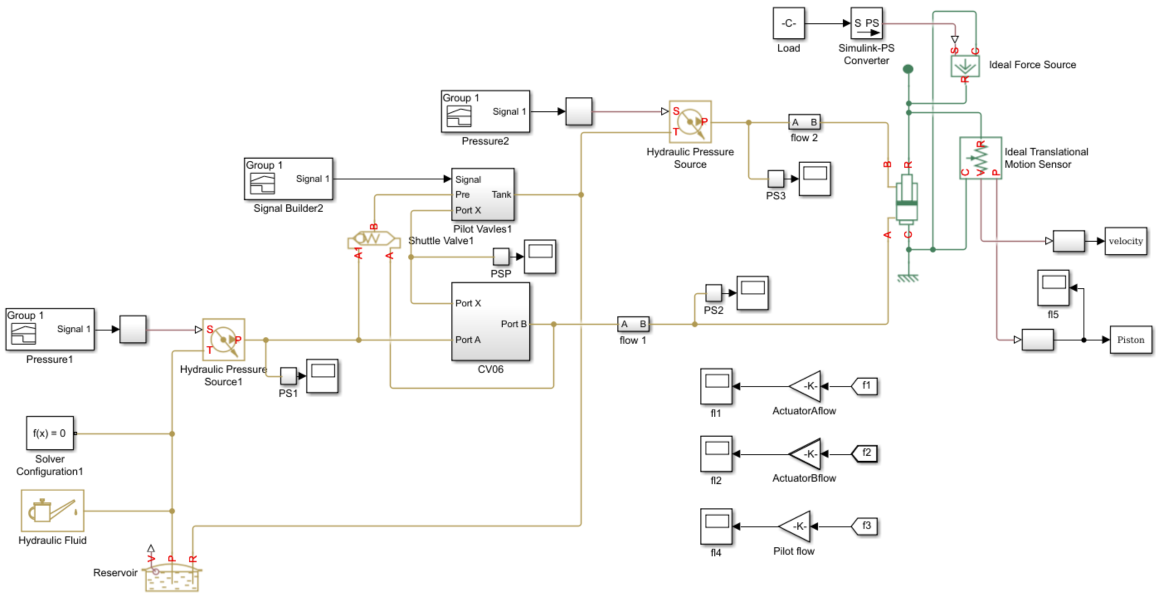


Figure 62. Bi-directional flow simulation setup – active switching

Another change with respect to the uni-flow simulation model, is the addition of shuttle valve to supply highest pressure value from either side of metering edge of CV06. The pilot valve scheme remains the same and is masked into sub-system for clarity, double-acting hydraulic cylinder is added with cap side diameter of 125 mm and rod is of 56 mm. Furthermore, position sensor and external load is added and the pressure source at cylinder rod side to replicate the pressure-force system. Some information about the system used as a reference, to simulate this model, is give in the next section.

Multi-pressure actuator system

A discrete fluid power system studied in the articles [3], [13], simplified schematic given in Appendix 13, has number of solenoid on/off valves in set that are used to control the hydraulic cylinder through the available multiple pressure levels to control the force and velocity of the load on the actuator. The is the pressure-force control system that optimally selects pressure range to be applied on both side of piston, cap and rod, to lift or lower it without the use of flow control throttle. Moreover, the load has to have inertia and system is controlled by applying appropriate forces to control the load.

However, here in the simplified simulation model, Figure 62, only one valve is used to pass the flow at the cap side with the use of pressure source and at piston rod side direct pressure source is defined. Both of the pressure sources are varied at predefined time maintaining constant pressure difference across the CV, almost equal to pressure drop at 200 l/min to keep the constant velocity. This is to create sort of open loop control to test effects of valve delay in the position control of the actuator. The simulation was run to position the actuator over time, as shown in Figure 63.

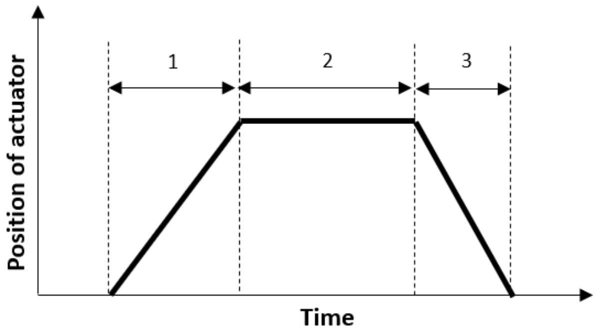


Figure 63. Hydraulic cylinder position control in bi-directional valve scheme. 1) From port A to port B flow direction, 2) Holding load 3) Port B to port A flow bringing back to original position

5.2 Outcome and Results

The simulation is run to operate the actuator for five seconds, and the cylinder is set at 0.1 m above its seat initially, which is the point of start of the lifting motion. Simulating the system as open loop circuit and the distinct pressure sources has been applied, enough to maintain the constant velocity of 0.27 m/s during lifting and lowering. As in Multi-pressure system, where multiple pressure sources can be accessed within milliseconds, but with a finite volume of fluid, here it is considered consistent with two pressure ranges.

Figure 64 shows the velocity and position of actuator curves side by side with the switching duration at both opening and closing during lowering motion. The overrunning of the piston position can be observed at 4 s due to higher switching duration (at closing during lowering) than the opening (during lifting). Also, the pressure value is higher during the lifting phase set to be 231 bar and 78 bar in third phase at cap side. While, 184 bar at piston rod side and zero gauge pressure during lowering, so to simulate bringing piston to original position by its own load. The switching delay and slightly higher-pressure difference assimilates into 10 mm of overrunning of the piston.

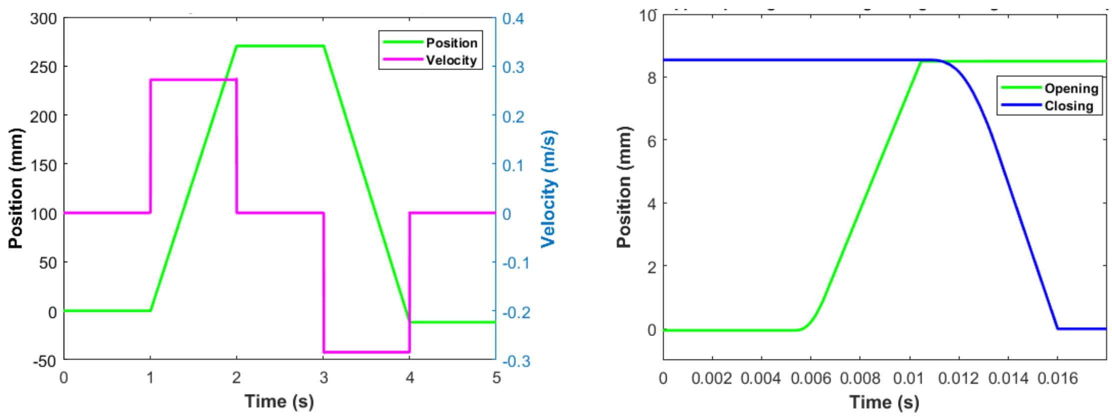


Figure 64. Bi-directional flow curves with active (internal) switching. Velocity and actuator position (left) and CV switching duration at lowering (right)

It can be ascertained that for pressure-force control of actuator, the timing of valve actuation is crucial with the two-stage valve scheme when considering the fine positioning and velocity control of the actuator. As well as maintaining constant pressure difference across the valve to keep the losses under minimum level (< 5 bar) and ideally maintaining the speed of the actuator. The lower closing time observed, compared to the uni-flow model, is due to reduced area ratio of main valve's poppet and also simulation conducted at higher operating pressure.

In passive switching, the pressure source to the pilot line is externally supplied and can be set higher than the maximum system working pressure. This will ensure faster closing duration and make the pilot flow independent of the main flow line, shown in Figure 65.

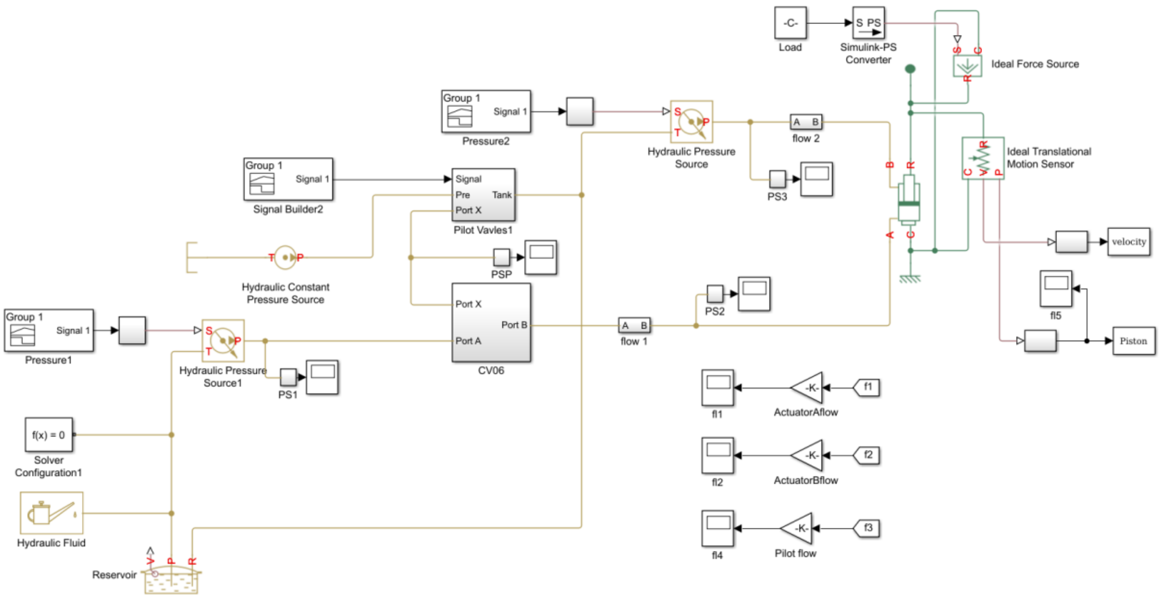


Figure 65.Bi-directional flow simulation setup – passive (external) switching

The simulation is run at the same operating condition with pilot pressure supply at 250 bar, the only difference in simulation from active pilot control is the slightly higher opening duration and lower in closing. The perceptible difference in simulation results with the passive controlling, is the marginal difference between opening and closing duration can be reduced for accurate positioning of the actuator, Figure 66.

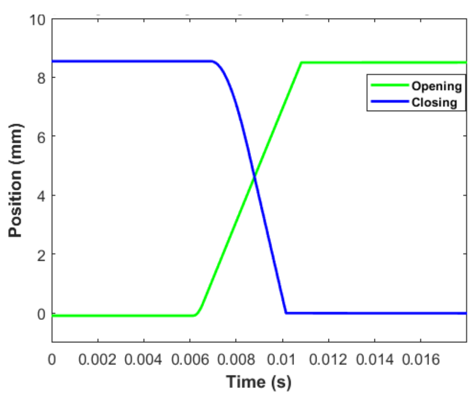


Figure 66. Bi-directional flow curves at passive switching. CV switching duration at lowering

In conclusion of this chapter, the two-stage bi-directional valve can potentially be utilized in Multi-pressure actuator system as target application to scale for higher performance rating. Considering the measurements from uni-flow directional valve measurements and functionality test using simulation of bi-directional valve, it is concurred that pilot operated valve can perform and deliver similar performance as direct operated solenoid on/off valve, in addition of higher flow rate operating capacity. However, the switching time duration is higher which is rationally in line with the increase in volumetric size. Moreover, the piloting structure (internal or external) and control of valve actuation time is an important factor which affects the dynamics of the operation, that controls the fine positioning of the actuator, timing gap between the opening and closing of the valve, pilot flow into and out of the system. Additionally, there is still need to investigate more by building physical experimental system with the actuator and load and consequential effects of controlling the position with bi-directional flow control valve with either of the piloting structure type.

6 Conclusions

This thesis work has been performed in anticipation of utilizing the on/off directional flow control valves in the hydraulic power applications, where high flow rate is required and able to perform switching within shortest possible time. The two-stage pilot operated valve performed well under various operating conditions in the experiment, and it was operated at maximum flow rate of up to 200 l/min and 100 bar pressure supply and can be operated at even higher values up to 325 l/min. The switching time was found to be lowest at higher operating pressure for both closing and opening of the valve. However, there was marginal difference between opening and closing duration at maximum operating pressure with a difference of 18 ms. This could be reduced, based on simulation results of bi-directional valve, by the use of poppet with lower area ratio $0.5A_x$, with addition of limiting the stroke of the poppet as per desired flow rate requirement and orifices.

It was observed that piloting of the main valve is susceptible to unstable controlling, if it is internally piloted (supplied pressure from main line), specially at high operating pressure and remains one of the cause of oscillations during opening and closing. One recommended solution is the use of external supply of pressure level, greater than the working system pressure, based on the application. It is imperative to highlight that two-stage valve scheme was also able to perform various switching operations such as, continuous, ballistic and can even proportionally control the poppet movement through independent pilot valves controlling. Though, further investigation between controlling of the pilot valve and throttling of the flow at main valve needs to be performed for potential controlling of the hydraulic actuator with fine positioning.

Similar analysis is also applicable for the bi-directional flow control valve, which is limited to the simulation in this study. It is investigated that the precise time control between the activation signal for valve actuation and the actuator movement is an important factor to prevent overrunning of hydraulic cylinder in either direction of flow. To conclude, the study to utilize the catalog components with slight modification for large scale discrete fluid power system with the requirement of higher performance rating and measurements for its dynamic characteristics has been met.

6.1 Future work

The proposed valve scheme does fulfill most of the objectives laid out in the first Chapter, and recommendations have been given to achieve the slower switching duration. Inclusive is the leakages in the pilot line except in the main flow channel, CV, when it is closed that can be further studied in the future experiments.

It is also proposed, the use of miniature digital valves, developed by Tapio Lantela, to control the piloting of the hydraulic operated cartridge valve. This could significantly shave-off the actuation time and reduce the overall size of the valve block. As the miniature valves has lower flow rate capability than the ones, Bucher size NG 5, used in this study, it can be compensated by utilizing two valves in parallel at both high-pressure side and low-pressure side. Furthermore, in discrete fluid power systems, both types of piloting internal and external can be applied as presented in Figure 67 and Figure 68. In external piloting, the external pressure supply, $P_{X,}$ can be attained using miniature piston pump or branch-off supply in the discrete fluid power system.

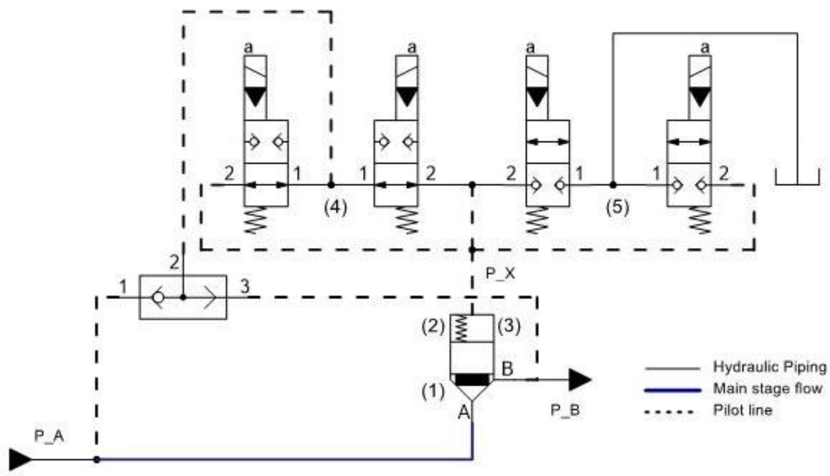


Figure 67. Internal piloting with miniature valve

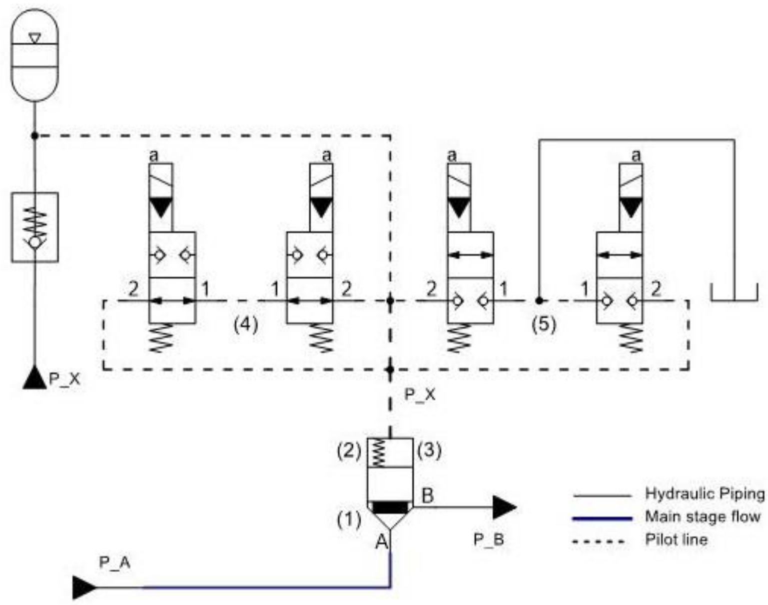


Figure 68. External piloting with miniature valve

Bibliography

[1] B. Xu, R. Ding, J. Zhang, and Q. Su, “Modeling and dynamic characteristics analysis on a three-stage fast-response and large-flow directional valve,” *Energy Convers. Manag.*, vol. 79, pp. 187–199, Mar. 2014.

[2] E. Lisowski, W. Czyżycki, and J. Rajda, “Multifunctional four-port directional control valve constructed from logic valves,” *Energy Convers. Manag.*, vol. 87, pp. 905–913, Nov. 2014.

[3] M. Huova, A. Aalto, M. Linjama, and K. Huhtala, “Study of Energy Losses in Digital Hydraulic Multi-Pressure Actuator,” 2017, pp. 214–223.

[4] A. H. Hansen, H. C. Pedersen, and T. O. Andersen, “Design of Bidirectional Check Valve for Discrete Fluid Power Force System for Wave Energy Converters,” p. V001T01A009, Oct. 2013.

[5] B. Winkler, A. Ploeckinger, and R. Scheidl, “A Novel Piloted Fast Switching Multi Poppet Valve,” *Int. J. Fluid Power*, vol. 11, no. 3, pp. 7–14, Jan. 2010.

[6] B. Zardin, M. Borghi, G. Cillo, C. Alberto Rinaldini, and E. Mattarelli, “Design Of Two-Stage On/Off Cartridge Valves For Mobile Applications,” *Energy Procedia*, vol. 126, pp. 1123–1130, Sep. 2017.

[7] M. Karvonen, M. Juhola, V. Ahola, L. Söderlund, and M. Linjama, “A MINIATURE NEEDLE VALVE,” p. 19.

[8] T. Lantela, J. Kajaste, J. Kostamo, and M. Pietola, “Pilot operated miniature valve with fast response and high flow capacity,” *Int. J. Fluid Power*, vol. 15, no. 1, pp. 11–18, Jan. 2014.

[9] Esposito, *Fluid Power with Applications*, 7 edition. PE, 2013.

[10] M. Vilenius, A. Laamanen, and M. Linjama, “Is It Time for Digital Hydraulics?,” p. 20.

[11] K.-E. Rydberg, “Hydraulic Servo Systems : Dynamic Properties and Control,” *Hydraul. Servo Syst.*, p. 111.

[12] M. Jelali and A. Kroll, *Hydraulic Servo-systems: Modelling, Identification and Control*. Springer Science & Business Media, 2012.

[13] M. Linjama and M. Huova, “Model-based force and position tracking control of a multi-pressure hydraulic cylinder,” *Proc. Inst. Mech. Eng. Part J. Syst. Control Eng.*, vol. 232, no. 3, pp. 324–335, Mar. 2018.

[14] M. Huova, A. Laamanen, and M. Linjama, “Energy Efficiency of Three-Chamber Cylinder with Digital Valve System,” *Int. J. Fluid Power*, vol. 11, no. 3, pp. 15–22, Jan. 2010.

[15] “Miniature Digital Hydraulic Valve System - Pilot operated design with fast response and high flow capacity.” [Online]. Available: https://aaltodoc.aalto.fi/handle/123456789/31298. [Accessed: 13-Jun-2018].

[16] “2-Way Slip-In Cartridge Valve - Series CE, CP | Parker NA.” [Online]. Available: http://ph.parker.com/us/en/ce-cp-2-way-slip-in-cartridge-valves. [Accessed: 19-Jun-2018].

[17] B. Winkler and R. Scheidl, “Development of a fast seat type switching valve for big flow rates,” *Tenth Scand. Int. Conf. Fluid Power SICFP07*, May 2007.

[18] K. Dasgupta and J. Watton, “Dynamic analysis of proportional solenoid controlled piloted relief valve by bondgraph,” *Simul. Model. Pract. Theory*, vol. 13, no. 1, pp. 21–38, Jan. 2005.

[19] B. Rexroth, “Proportional pressure relief valve, pilot operated RA 29160/04.05.” .

[20] "What's the Difference Between Poppet and Piston Spool Valves?," *Machine Design*, 23-Feb-2017. [Online]. Available: http://www.machinedesign.com/whats-difference-between/what-s-difference-between-poppet-and-piston-spool-valves. [Accessed: 02-Jul-2018].
[21] "CHAPTER 11: Slip-in Cartridge Valves (Logic Valves)," *Hydraulics & Pneumatics*, 19-Feb-2007. [Online]. Available: http://www.hydraulicspneumatics.com/other-technologies/chapter-11-slip-cartridge-valves-logic-valves. [Accessed: 02-Jul-2018].
[22] "2-way Slip-in Cartridge Valves Directional and Pressure Function," p. 56, 2013.
[23] "2/2 Cartridge Seat Valve, Size 5 Series WS22GD…/ WS22OD..." Bucher Hydraulics.
[24] J. Peurala, "Model-Based Design, Modelling and Simulation of Digital Hydraulic Gas Admission Valve," May 2014.
[25] M Balakrishnan, "Detection of Plunger Movement in DC Solenoids." Texas instruments.
[26] "Throttle and throttle check valve RE 27219/01.09." Bosch Rexroth AG.
[27] "Pilot operated unloading BM-N, RE 18310-11/04.10." Rexroth Bosch.
[28] "Sizing Accumulators - Hydac." Hydac.
[29] "Hydac Electronic Pressure transmitter," 25-Jun-2018. [Online]. Available: https://www.hydac.com/de-en/products/sensors/pressure-sensors/pressure-transmitters/hda-4700.html. [Accessed: 08-Jul-2018].
[30] "IQAN-ST Sensors Electronic Control Systems temperature sensor." Parker.
[31] "High-speed, High-accuracy Laser Displacement Sensor LK-G5000 series catalogue." KEYENCE.
[32] "Fluke 80i-110s AC/DC Current Clamp." [Online]. Available: http://en-us.fluke.com/products/all-accessories/fluke-80i-110s.html. [Accessed: 12-Jul-2018].
[33] "Technical SPEC it, STAMP it catalog 4400US for Hoses." Parker.
[34] J. . Lienhard V, "Velocity coefficients for free jets from sharp-edged orifices," *J. Fluids Eng.*, vol. Vol. 106, 1984.
[35] M. LINJAMA, "Digital Hydraulics – Towards Perfect Valve Technology," *Digit. Hidravl.*, vol. Ventil 14 /2008/ 2.
[36] "PWM — the unconventional way to regulate pressure," *Hydraulics & Pneumatics*, 01-Jun-2004. [Online]. Available: https://www.hydraulicspneumatics.com/other-technologies/pwm-unconventional-way-regulate-pressure. [Accessed: 04-Sep-2018].
[37] M. Flor, "Digital hydraulics at Bosch rexroth - A trend evolves to real applications," vol. The Fifth Workshop on Digital Fluid Power, October 24-25, 2012, Tampere, Finland, pp. 5–13.
[38] "Volvo excavator EX210B LC Monoblock/2-piece boom." VolvoCE.

Appendices

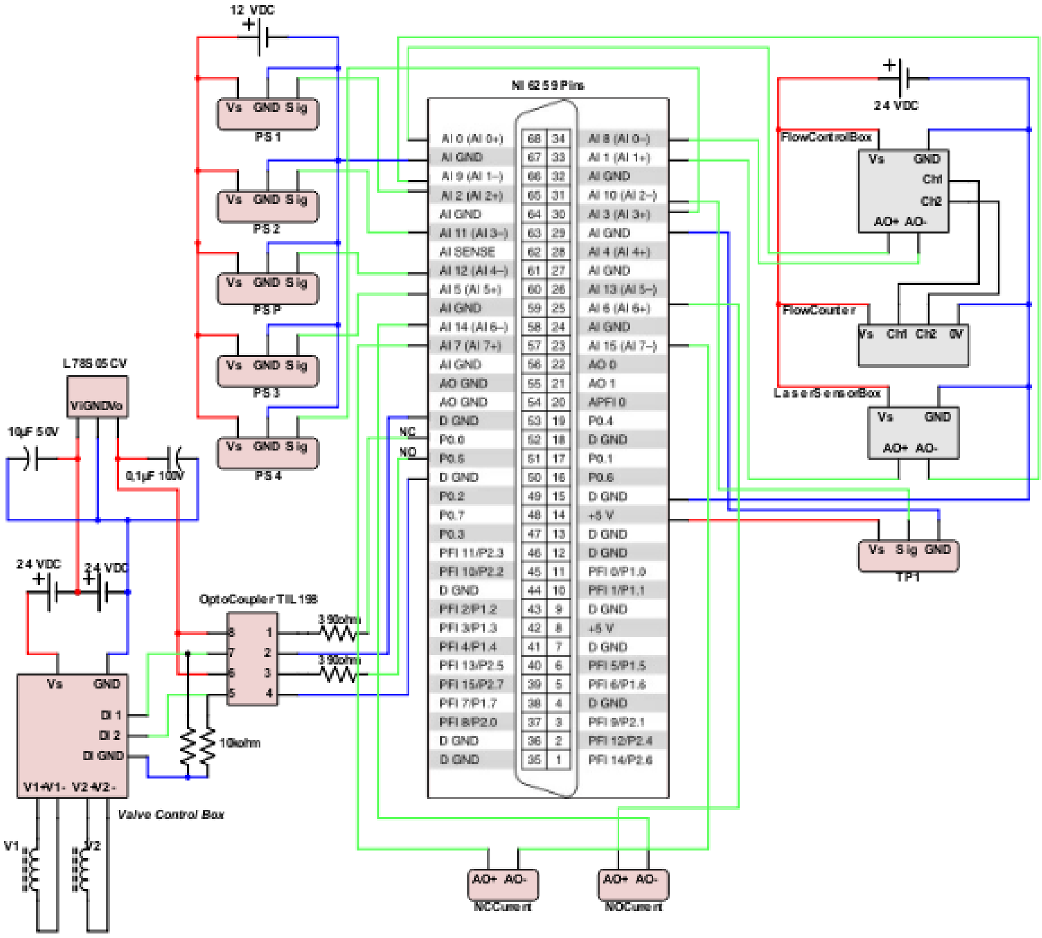
Appendix 1. Comparison table of commercially available on/off solenoid valves

	Bucher WS22GD Size 5	Hydac WS08W-01	Bucher WS22GDA-10	Bosch Rexroth SEC 6	The Lee Co SDBB3321003A	Bosch Rexroth KKDER1
Type	Poppet 2/2	Poppet 2/2	Poppet 2/2	Poppet 4 x 2/2	Poppet 2/2	Spool 4/2 connected as 2/2
Open/close response time without boosting [ms]	-	<35/<50	?	<70/<45	<30	<80/<50
Response time with boosting [ms]	6–30	5–8	14–21	6	?	10–14
Flow capacity @ Δp = 0.5 MPa [l/min]	10	12	37	7	2.8	43
Maximum flow rate [l/min]	30	35	140	35	?	120
Leakage	?	5 drops/min @ 25 MPa	< 0.0001 l/min @ 20 MPa	?	0.002 l/min @ 21 MPa	0.05 l/min @ 20 MPa
Max. operating pressure [MPa]	35	25	?	42	21	35
Dimensions [mm]	90 x ø26	99 x ø36	140 x ø47*	212 x 90 x 49	48 x ø25.4	175 x ø37
Flow density [1/min]	209	119	152	15	115	229
Weight [kg]	0.2	0.33	?	2.14	0.068	0.6
Electrical power consumption [W]	15	18	17	30	8	22
Max. ISO 4406 fluid cleanliness class	20/18/15	21/19/16	?	20/18/15	?	20/18/15

* Diameter estimated from image

The table lists the commercially available on/off type solenoid operated valves, comparing the properties. These valves can essentially be boosted to have increased response time, while, question mark signifies unavailable figure or not provided in the data sheet by the manufacturer. This chart has been referenced in the dissertation [15].

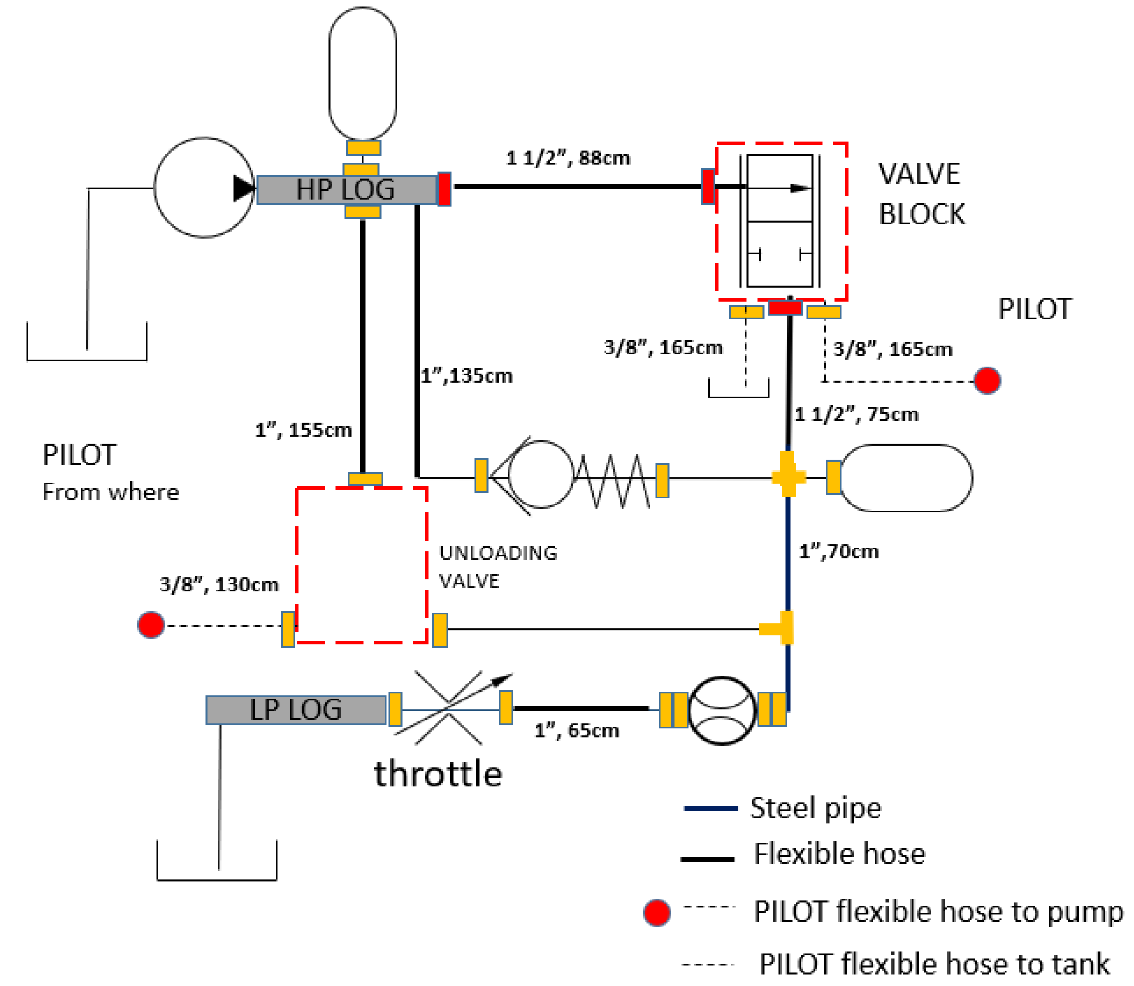
Appendix 2. Electrical connections of DAQ with sensors and valve control



The electrical connection figure is of the setup being used in the experiment, utilizing National Instrument data acquisition card through sensors and controlling of the valve

.

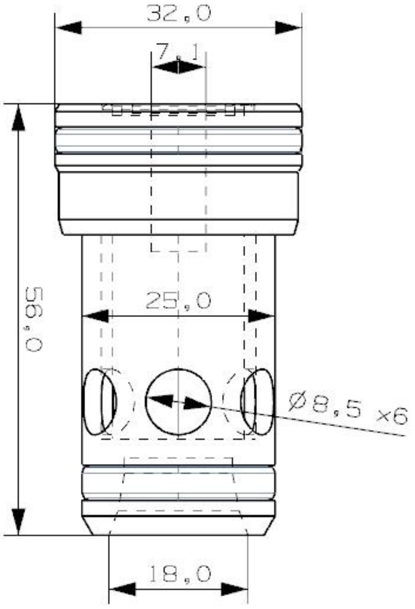
Appendix 3. Uni-direction flow schematic with fittings and hoses detail



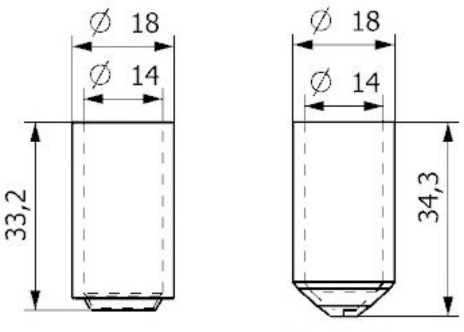
The figure details the log fittings and unions, flexible hoses and steel pipes used in setting up the experimental setup between the components. All of the conduits have pressure rating of 350 bar and above. (Courtesy: Advisor Jyrki Kajaste)

- Diameter defined is internal, in inches.

Appendix 4. Cartridge valve dimensions



Cartridge valve CE16 dimensions measured and included for modeling and geometric dimension purpose.

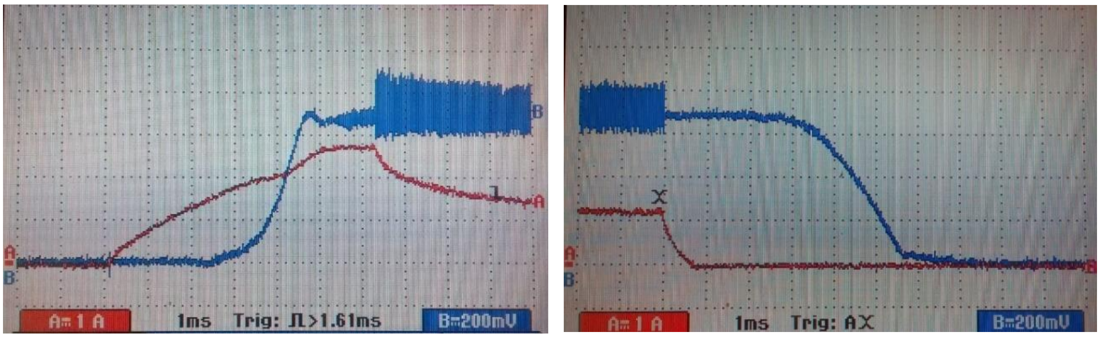


Two available poppets with geometric dimensions, and different area ratios for simulation data.

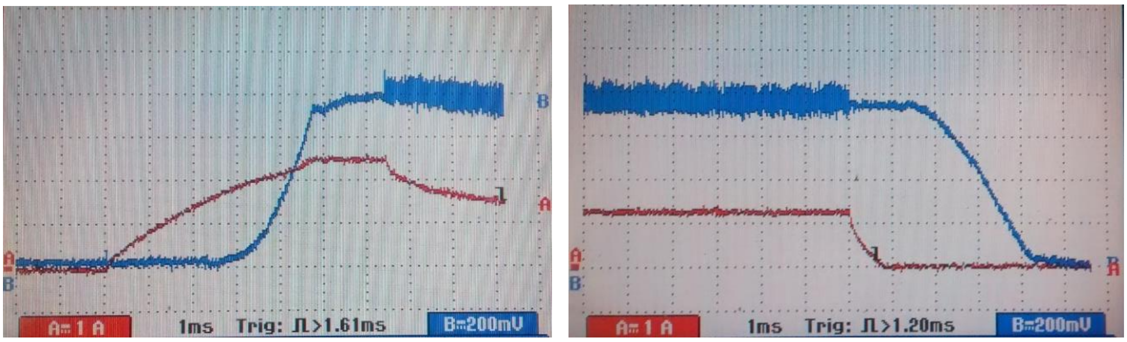
- Preload total spring length = 49.5 mm

**All dimensions in mm for CE16 Parker*

Appendix 5. Bucher valves current profiles w.r.t displacement



NC valve current profile (Red) w.r.t voltage (Blue) during opening (left) and closing (right) with offset positively by 0.44 ms.

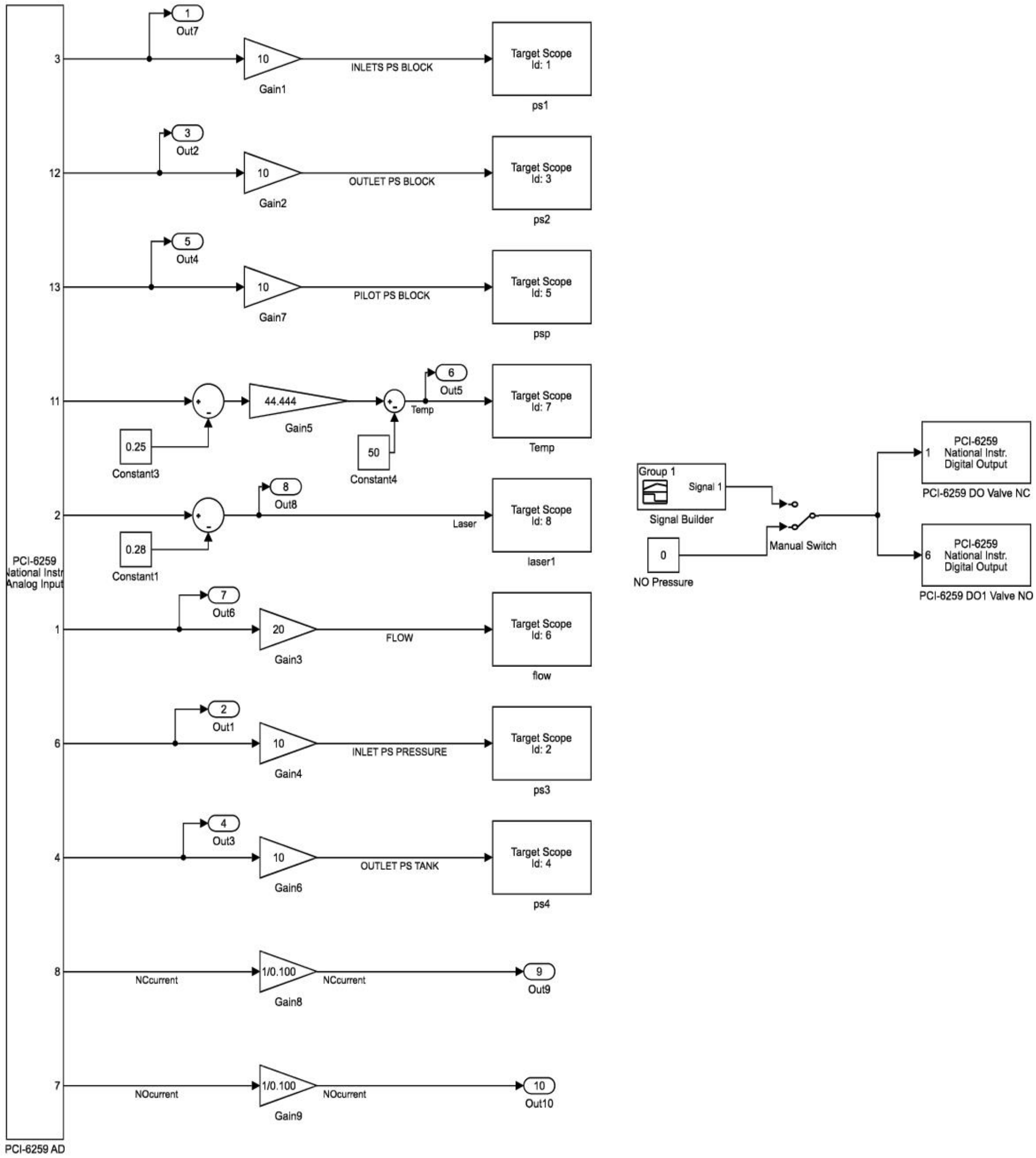


NO valve current profile (Red) w.r.t voltage (Blue) during opening (left) and closing (right) with offset positively by 0.44 ms.

1 V = 1 mm (displacement)

The scale in all figures is 1 A block in vertical and 0.2 V in horizontal axis. The maximum current touches at 2.6 A at no external load condition and holding current to be 1.48 A.

Appendix 6. Simulink real-time model for DAQ



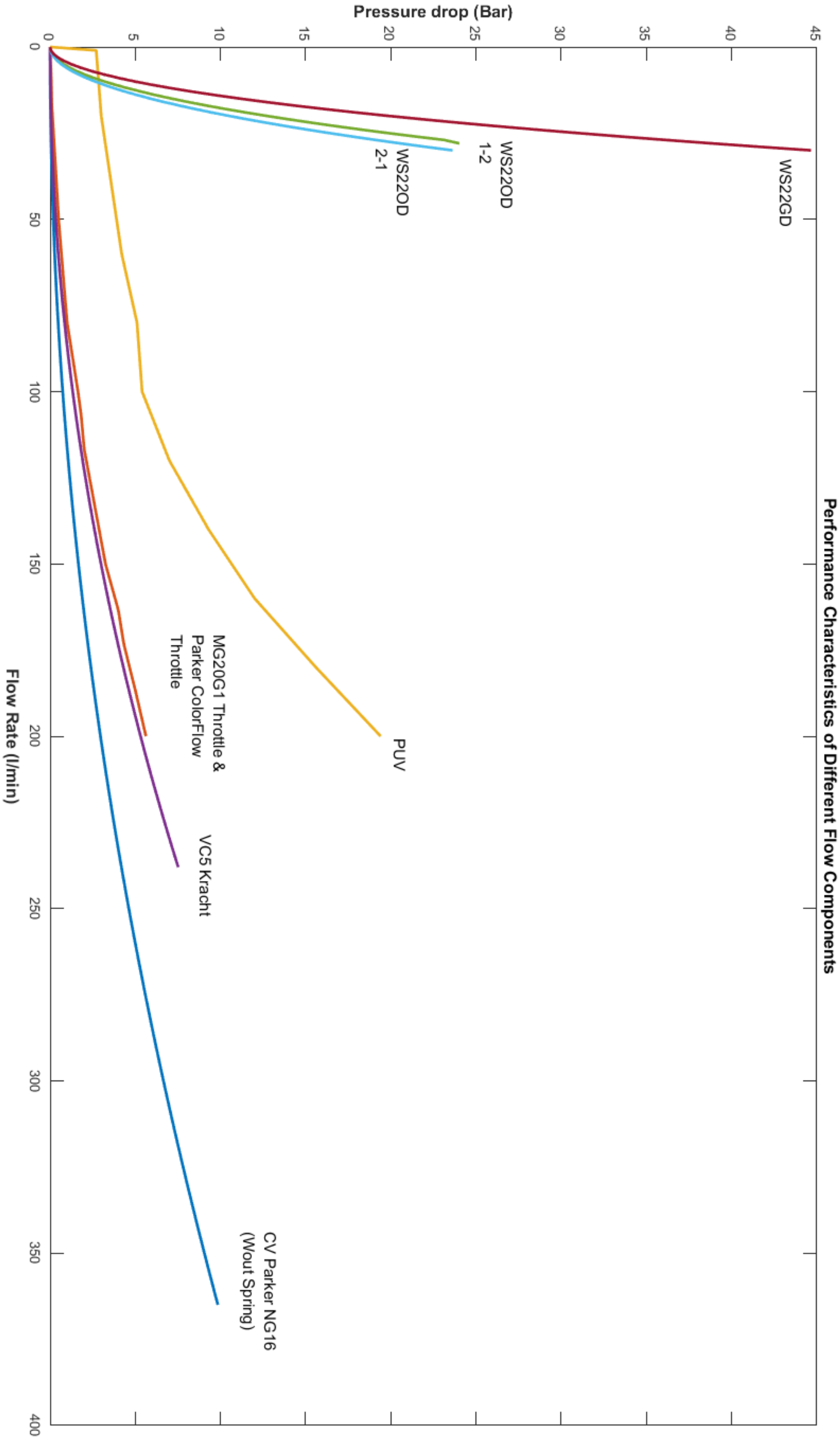
Simulink model for real-time data acquisition. For valve actuation, the manual switch is changed to signal builder for predefined switching frequency and duty cycle

Appendix 7. Matlab script for data acquisition

For auto Target PC connection and data acquisition.

```
%%For Live monitoring and real time control of valve

[r pingtest]=size(slrtpingtarget); % for ethernet connection test and
Ping to target PC
while pingtest==6
    [r pingtest]=size(slrtpingtarget);
end
set_param('SensorInterface','SimulationMode','external','StopTime',
'1000'); %simulation setting
rtwbuild('SensorInterface'); %code build
set_param('SensorInterface','SimulationCommand','connect');
set_param('SensorInterface','SimulationCommand', 'start'); %simulation
start

%%For measurements acquisition at set time and valve control
%manual switch to signal command and live simulation needs to be
stopped from simulink first
set_param('SensorInterface','SimulationMode','external','StopTime',
'5'); %simulation setting and run time
rtwbuild('SensorInterface');
set_param('SensorInterface','SimulationCommand','connect');
set_param('SensorInterface','SimulationCommand', 'start');
while tg.Status=='running'
    pause(10);
end
%for auto data transfer from memory to host pc
eval([input('data variable name=','s'), '=tg.OutputLog']); %output raw
data log
eval([input('time variable name=','s'), '=tg.TimeLog']); %Output time
log
```

Appendix 8. Catalog performance curves of the components



Combined performance curves, extracted from Technical data sheet for each installed component in the assembly, for comparison with simulation results.

Appendix 9. Pilot valve model parameters

2-Way Directional Valve

This block models a 2-way directional valve in a hydraulic network. To parameterize the block, 3 options are available: (1) by maximum area and control member stroke, (2) by the table of valve area vs. control member displacement, and (3) by the pressure-flow rate characteristics.

Ports A and B are hydraulic conserving ports associated with the valve inlet and outlet, respectively. The control member displacement is set by the physical signal input S. A positive displacement opens the connection between ports A and B, permitting liquid flow.

Settings

Parameters

Model parameterization:	By maximum area and opening	
Valve passage maximum area:	3.05e-5	m^2
Valve maximum opening:	0.00069	m
Flow discharge coefficient:	0.2	
Initial opening:	0.00069	m
Leakage area:	1e-12	m^2
Laminar transition specification:	Pressure ratio	
Laminar flow pressure ratio:	0.999	

2-Position Valve Actuator

The block is a data sheet-based model of an actuator that drives 2-position directional discrete valves and assumes 2 positions: extended and retracted. The actuator is activated if the input signal crosses 50% of its nominal value. The actuator can be actuated only by positive signal, similar to the case of AC or DC electromagnets. The push-pin reaches a hard stop after "switching-on" time, and retracts in "switching-off" time after the control signal is removed. The motion can be interrupted. The motion profile does not depend on load. The block has one physical signal input port and one physical signal output port. The push-pin moves in positive or negative direction, depending on the "Actuator orientation" parameter setting.

Source code

Settings

Parameters

Push-pin stroke:	0.00069	m
Switching-on time:	3.5e-3	s
Switching-off time:	4.5e-3	s
Nominal signal value:	1	
Initial position:	Retracted	
Actuator orientation:	Acts in negative direction	

Appendix 10. CV model parameters

Poppet Valve

The block models a poppet valve with orifice created by a cylindrical sharp-edged stem and a conical seat. The flow rate through the valve is proportional to the orifice opening and to the pressure differential across the valve. The model accounts for the laminar and turbulent flow regimes by monitoring the Reynolds number and comparing its value with the critical Reynolds number.

Connections A and B are conserving hydraulic ports associated with the valve inlet and outlet, respectively. Connection S is a physical signal port. The block positive direction is from port A to port B.

Settings

Parameters

Valve stem diameter:	18e-3	m
Seat cone angle:	120	deg
Initial opening:	0	m
Flow discharge coefficient:	0.5	
Leakage area:	1e-12	mm^2
Laminar transition specification:	Pressure ratio	
Laminar flow pressure ratio:	0.999	

Double-Acting Hydraulic Cylinder (Simple)

The block is a model of a double-acting hydraulic cylinder developed for applications in which only the basic cylinder functionality must be reproduced in exchange for better numerical efficiency. For these reasons, factors such as fluid compressibility, friction, and leakages are assumed to be negligible. The hard stops are assumed to be fully inelastic to eliminate any possible oscillations at the end of the stroke. The model is suitable for real time or HIL simulation if such simplifications are acceptable.

Connections R and C are mechanical translational conserving ports corresponding to the cylinder rod and cylinder clamping structure, respectively. Connections A and B are hydraulic conserving ports. Port A is connected to chamber A and port B is connected to chamber B. The block directionality is adjustable with the Cylinder Orientation parameter.

Settings

Parameters

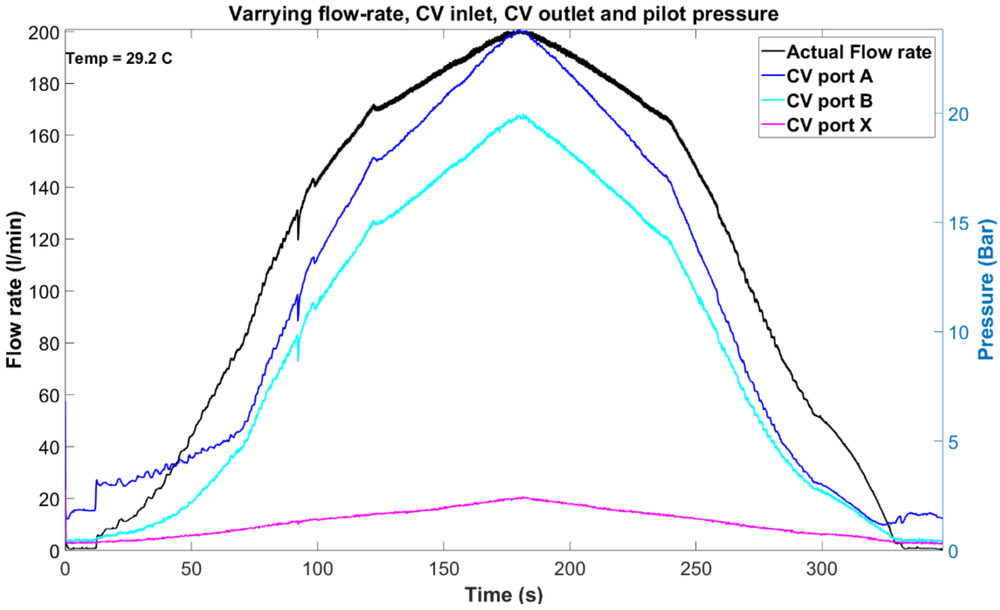
Piston area A:	1.2723e-4	m^2
Piston area B:	2.4429e-4	m^2
Piston stroke:	8.5e-3	m
Piston initial distance from cap A:	8.5e-3	m
Penetration coefficient:	1e12	s*N/m^2
Cylinder orientation:	Acts in positive direction	

Appendix 11. Matlab script for uni-flow direction simulation run

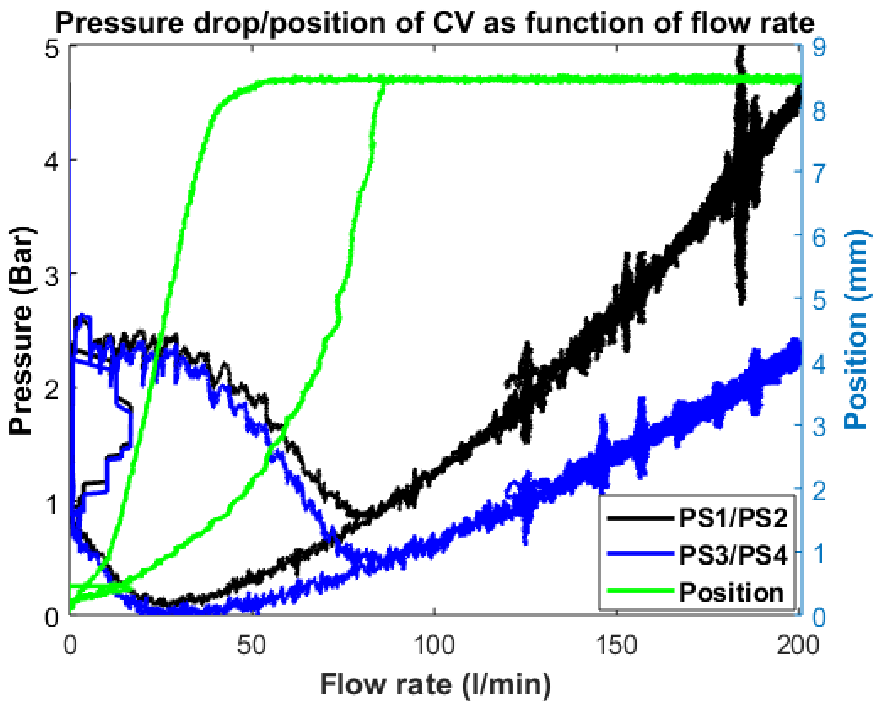
```
%Simulation sweep at varying flow rate, pressure and pressure differ-
ence
N_sim=0;
for i = 1:10
Q=i*25; %main pump flow rate
Load=20e5; %Load at CV port B

for j =1:10
    N_sim=N_sim+1;
    PDValve=j*20e5;
    PDValveMax=PDValve+1e5;
    sim('Logiikkaventtiilit_Onetimeswitch2')

    open_dispindex=find(tout>=1,1); %from switching activation
    fullopen_dispindex=find(disp>=8.075,1); %95percent of poppet total
displacement
    valveopen_t=(tout(fullopen_dispindex)-tout(open_dispin-
dex))*1000;%time duration between command and 95% poppet stroke
    close_dispindex=find(tout>=3,1); %from switching activation
    temp_disp=disp;
    temp_disp(1:close_dispindex)=[];
    fullclose_dispindex=find(temp_disp<=0.42,1)+close_dispindex;
%95percent of poppet total displacement
    valveclose_t=(tout(fullclose_dispindex)-tout(close_dispin-
dex))*1000;%time duration between command and 95% poppet closing
stroke

    open_PDCVindex=find(tout>=2,1);
    Supply_PS1index=find(tout>=0.5,1);
    flowlost_pilotindex=find(tout>=3.5,1);

        switching_onetimes(N_sim).valveopen_t = valveopen_t;
        switching_onetimes(N_sim).valveclose_t = valveclose_t;
        switching_onetimes(N_sim).PDValve = PDValve/1e5;
        switching_onetimes(N_sim).Q =Q;
        switching_onetimes(N_sim).PDCV =PS1(open_PDCVindex)-
PS2(open_PDCVindex);
        switching_onetimes(N_sim).SPS =PS3(Supply_PS1index);
        switching_onetimes(N_sim).pilot_lostflow =pilot_lostflow(flow-
lost_pilotindex);

    end
end

%Simulation sweep for Ballistic measurmenets constant flow rate and PD
with
%varying duty cycle at fixed period of 1s to measure displacement and
flow
%volume

N_sim=0;
PW=0;
Q=150; %main pump flow rate
Load=13e5; %Load at CV port B
PDValve=13e5;
PDValveMax=13.1e5;
    for PW=0.1:0.05:1
    N_sim=N_sim+1;
```

```
Valve_maxopen=max(disp);
flowvol_maxpassed=max(Main_flowvol);
sim('Logiikkaventtiilit_Onetimeswitch2')
Ballistic(N_sim).PW = PW;
Ballistic(N_sim).disp = Valve_maxopen;
Ballistic(N_sim).volume = flowvol_maxpassed;
end
```

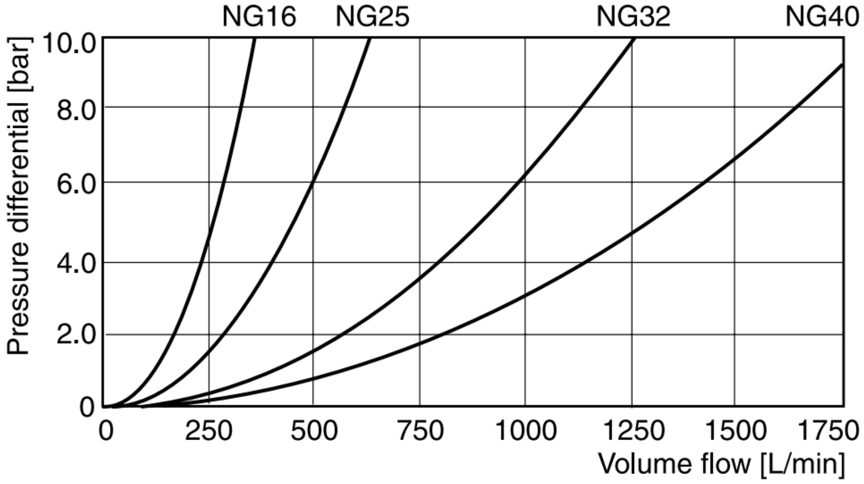
Appendix 12. Experiment operating parameter graphs



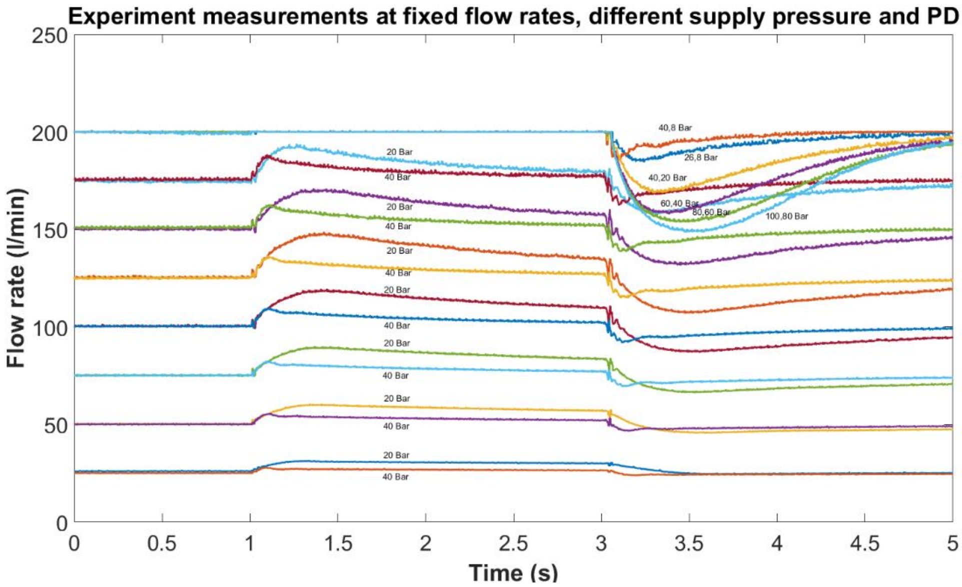
Total supply pressure needed to pass flow rate from 0 – 200 – 0 l/min in the setup. Also, pilot and outlet pressure plotted.



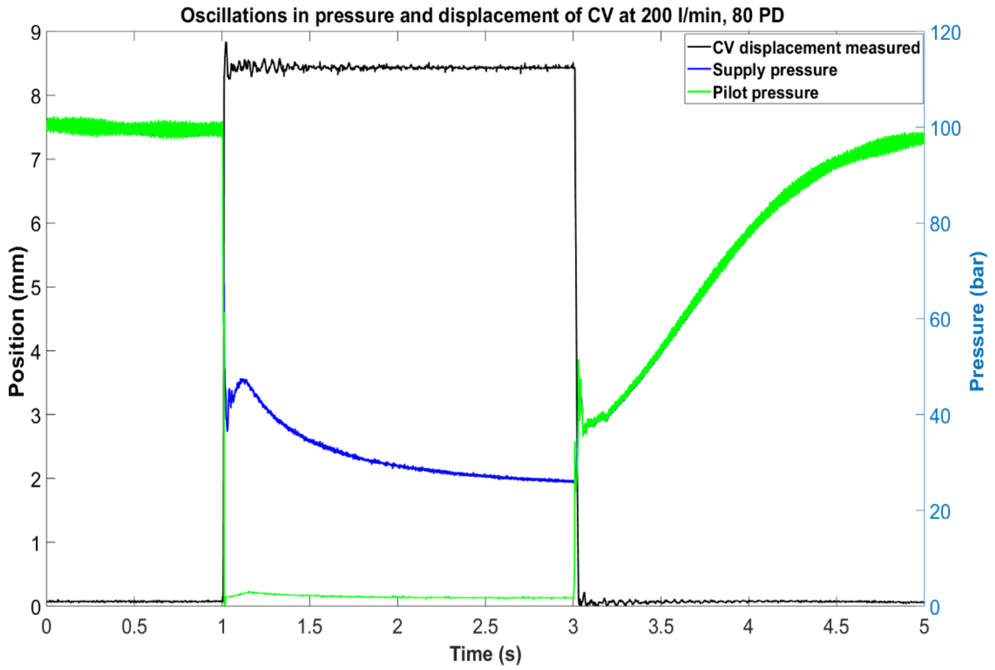
PS1 and PS2 sensors are installed in valve block's inlet and outlet of CV
Second set of pressure sensor PS3 and PS4 are similarly installed on the start of high-pressure line conduit and end of low-pressure conduit.
Pressure drop across CV and position is plotted with respect to varying flow rate from 0 -200 - 0 l/min at zero pilot pressure



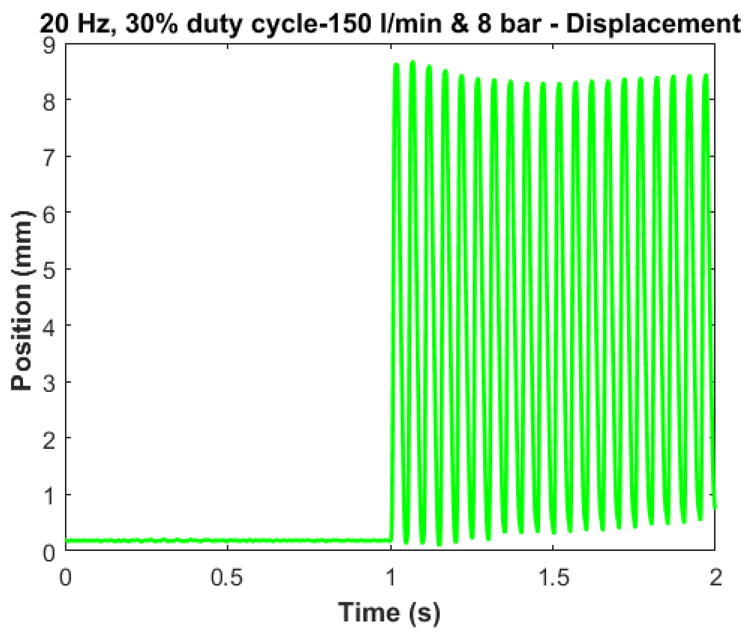
Performance curves of Parker CE 2 way valve [16]



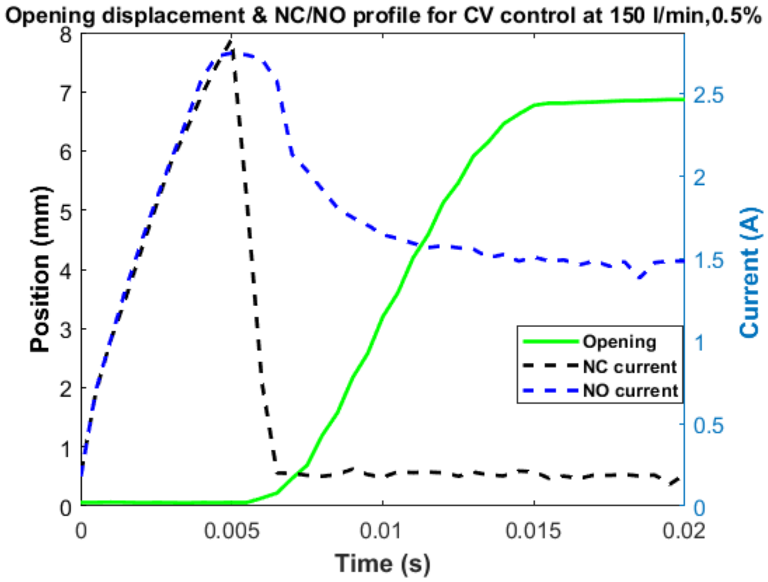
Flow rate transition due to switching in experiment



Oscillations depicted in the figure, at 200 l/min and 80 PD, during one time switching. Oscillations in after full stroke displacement of CV poppet and after closing in both pilot/Supply pressure and lift off of poppet from its seat occurring under 150 ms.

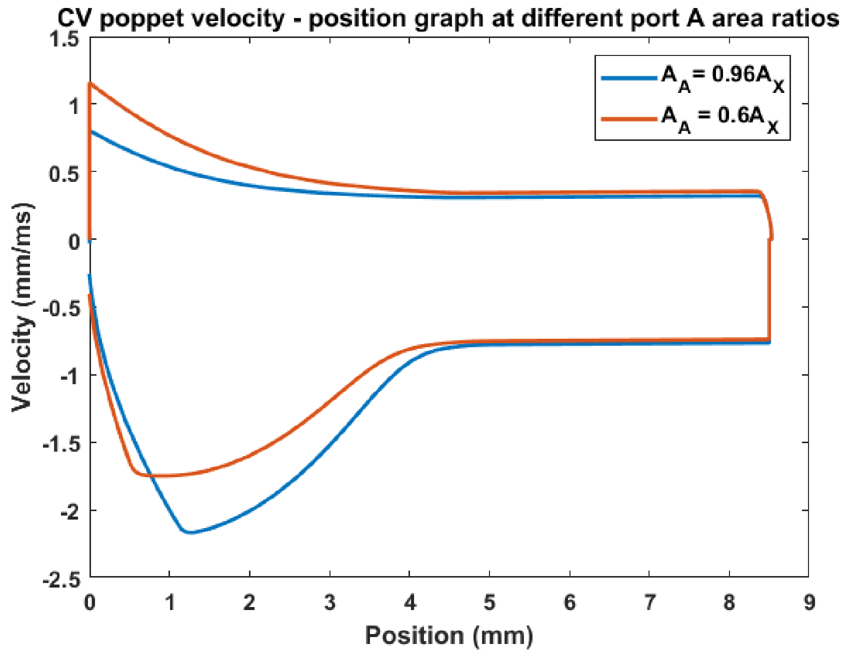


Continuous switching of the valve at 10Hz and 50% duty cycle - Displacement graph

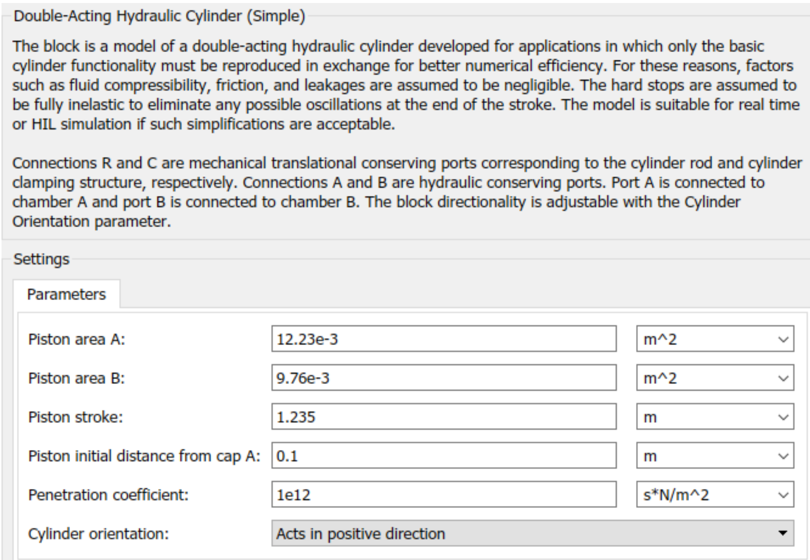


Opening displacement and NC/NO valve current profile for CV control switching – NC valve period = 1 s and 0.5 % duty cycle

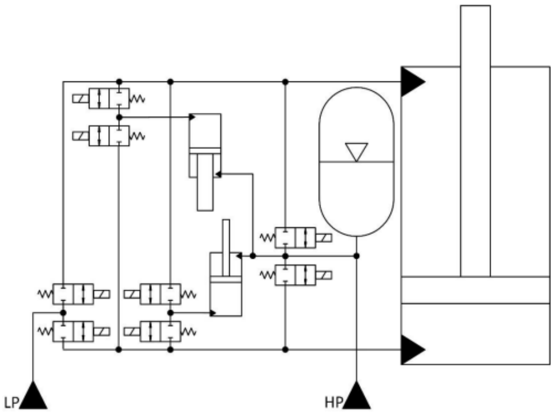
Appendix 13. Bi-Directional flow case



The velocity-position curves of CV poppet at different area ratios of port A at 200 l/min, 80 bar pressure difference



The hydraulic cylinder model’s specification used in simulation



Hydraulic diagram of digital multi-pressure actuator with four pressure levels to generate desired actuator force by controlling pressure levels at cap side and rod side of the cylinder through on/off valves [3]