AALBORG UNIVERSITY
STUDENT REPORT

Feasibility Study of a Novel ON/OFF Valve Solution for Velocity Servo Systems

Master Thesis

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Department of Energy Technology

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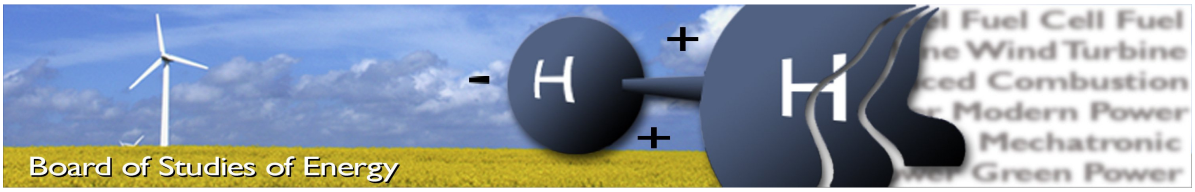

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SYNOPSIS:

The purpose of this report is to analyse the performance, challenges and benefits of using ON/OFF valves as a replacement for a conventional proportional valve in a hydraulic velocity servo system.
Two models are made: one with 4 ON/OFF valves and one with a single proportional valve. All valves are commercially available.
3 main challenges are identified in this replacement: a multi-variable switching strategy should be made, the fluid hammer effect will add pressure transients every time a valve is turned ON or OFF and the switching time for each valve will restrict the switching frequency.
It is shown that the commercial valves are too slow and they should be replaced. A sliding mode controller is designed for each system. It is shown that this control strategy, along with faster valves, has a better performance than the traditional proportional valve.

Copies: 3
Pages, total: 83
Appendix: 1
Supplements: CD

By signing this document, each member of the group confirms that all group members have participated in the project work, and thereby all members are collectively liable for the contents of the report. Furthermore, all group members confirm that the report does not include plagiarism.

Preface

This master thesis is written by Anette Brorsen, group MCE4-1021 on 10th semester at Department of Energy Technology at Aalborg University. To model the system, Matlab and Simulink are used. Some knowledge into these programs is assumed in order to get the full benefit of this report.

Acknowledgement

My gratitude goes to my husband Martin Brorsen for his support and patience.

Reading Guide

Throughout this report there will be references to sources, where the references are placed in the Bibliography. The used method for referring to sources follows Harvard referencing, where it refers with [Name, Year]. If the reference is included in a sentence before the dot, the reference covers the sentence. If the reference is after a dot, it covers the paragraph or section. If there are more than one source with the same name and year the source gets a letter after the year. A reference leads to the Bibliography where the source is given by the author, title, edition, publisher, hyper link and date.

Figures, tables, equations and calculations have numbers that indicates which chapter they belong to. The first figure in Chapter 3 has the number 3.1 and the next figure has 3.2. Appendixes are indicated with a letter.

All data, program files, the report in PDF and PDF-copies of all used websites can be found on the attached CD.

Summary

This report is a Master Thesis written at Mechatronic Control Engineering at Aalborg University. The project proposal is the outcome of a cooperation between Aalborg University and Hydac. The driving ambition is to use their existing components in a new way. The focus area is one of the most common hydraulic servo systems: the velocity servo. The purpose of this report is to analyse the performance, challenges and potential of using ON/OFF valves as a replacement for a conventional proportional valve in a hydraulic velocity servo system.

An ON/OFF valve has two states; ON and OFF. The physical meaning of these two states is open for flow or closed for flow, respectively. The proportional valve may reach intermediate states, i.e. 20% open, 70% open etc. The larger number of valve settings makes the proportional valve the common choice in servo systems since it is easier to control the position and movement of the hydraulic cylinder. However, the ON/OFF valve has the benefit of being about 20 times cheaper than the proportional valve which is why it is of interest to analyse the possibility to utilize this type of valve for the servo system despite the reduced functionality.

Chapter 1 introduces the reader to the servo system and the challenges that arises from using ON/OFF valves instead of a proportional valve. 3 main challenges are identified in this replacement:

- A multi-variable switching strategy should be made since the control consists of controlling 4 ON/OFF valves instead of 1 proportional valve.
- The fluid hammer effect will add pressure transients every time a valve is turned ON or OFF.
- The switching time for each valve will restrict the switching frequency.

The thesis statement, delimitations and the project strategy are given in chapter 2. The project strategy consists of tree parts; the first is to design the servo system and to analyse the 3 challenges, the second is to analyse how the valves should be controlled and the third is to compare the performance of the ON/OFF valves velocity servo system with a standard proportional valve servo system.

Chapter 3 presents and models the hydraulic cylinder. A set of valves, 4 ON/OFF valves and 1 proportional valve are chosen from the Hydac product portfolio and their dynamics are added to the model. Chapter 4 will investigate and analyse the effect of the three identified challenges. The switching frequency is restricted to 1.5 Hz, the fluid hammer effect is added to the model and it is shown that the valves should be paired between the two cylinder chambers.

Next the system behaviour is analysed, as seen in chapter 5. A dead volume is added to each chamber to improve the damping ratio and undamped natural frequency of the cylinder. Chapter 6 concludes that it will not be possible to get a satisfactory performance with a switching frequency of 1.5 Hz, thus the valves need to be upgraded to a faster version with a switching time of 6ms. Finally, a sliding mode control algorithm is developed for the velocity servo.

The performance of the valves: the new 6ms-, the original- and the proportional valves are investigated in chapter 7. It is shown that the original valves are too slow as predicted, and the new valves can reach a better performance than the proportional valve. The simulations show a fluid hammer effect which does not affect the velocity, but this should be confirmed in the laboratory.

The overall conclusion can be summed up to: If a set of new valves of the specified standard can be obtained then ON/OFF valves can be implemented successfully as a replacement to a proportional valve. A set of recommendations to Hydac is given in section 7.3 and the full conclusion can be read in chapter 8.

Table of Contents

Nomenclature

Symbol	Description	Unit
$\square_1$	.... for chamber 1	
$\square_2$	.... for chamber 2	
$\square_v$	.... for valve	
$\dot{\square}$	Time derivative	
$\square_{\square,ref}$	Reference value	
A	Area	m^3
B	Viscous friction Coefficient	$\frac{Ns}{m}$
β	Bulk Modulus	Pa
d	duty cycle	
e	Error	
f	Frequency	Hz
F	Force	N
L	Length	m
M	Mass	kg
ω_n	Natural undamped frequency	$\frac{rad}{s}$
p	Pressure	Pa
p_s	Pump pressure	Pa
p_t	Tank pressure	Pa
ψ	Length ratio	
Q	Flow	$\frac{m^s}{s}$
σ	Switching surface	
T	Time	s
u	Control law	
v	velocity	$\frac{m}{s}$
V	Volume	m^3
$V_{\square,0}$	Dead volume	m^3
x_c	Cylinder position	m
$\dot{x}_c$	Cylinder velocity	$\frac{m}{s}$
$\ddot{x}_c$	Cylinder acceleration	$\frac{m}{s^2}$
x_v	Valve opening	m

Chapter 1

A New Valve Solution for Hydraulic Servo Systems

This report will investigate the performance of a hydraulic cylinder, controlled by ON/OFF valves instead of the conventional proportional- and servo valves. This chapter will showcase the driving ambition behind this replacement and the challenges related to using ON/OFF valves.

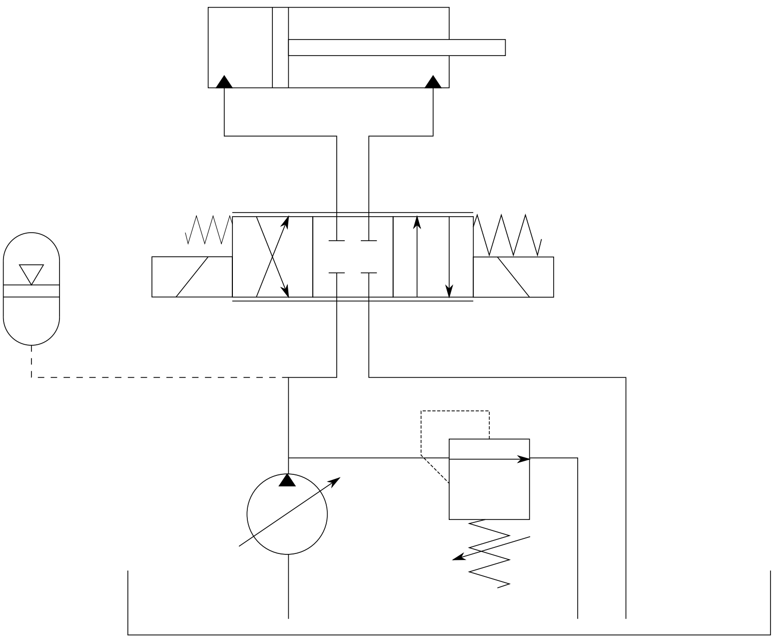


Figure 1.1. A simplified hydraulic servo system with a proportional valve, a pressure relief-valve, a pump, an accumulator and a hydraulic cylinder.

Hydraulic cylinders are essential in numerous applications such as robots, production facilities and construction equipment. Figure 1.1 shows a simplified hydraulic servo system, where the hydraulic cylinder is connected to a pump, two valves and an accumulator. This combination forms the basis of most hydraulic servo systems, along with additional sensors, other types of valves, pumps, cylinder types etc. The functional requirement to the system depends on the application and thus the components may vary accordingly.

The standard applied types of hydraulic servo systems are position servos, velocity servos and torque/force servos. As implied by the names, the desired control reference is a position, a velocity or a force, respectively. Due to the number of application areas of hydraulic servo systems and the different functional requirements, product optimization is of great interest. Focus areas are mainly price, energy consumption, system performance and reliability.

The idea behind the project is straightforward: Can an expensive proportional valve be replaced with cheap ON/OFF valves in a hydraulic servo system? The idea came from Hydac - an important player within the hydraulics and fluid engineering industry. Hydac manufactures and supplies ready-for-use hydraulic control and drive systems. The company has over 7500 employees and a product portfolio which enables them to be a part of all areas of industrial hydraulics. Hydac is a competitive company with an interest in updating their portfolio and utilizing existing products in a new way in order to make a profit.

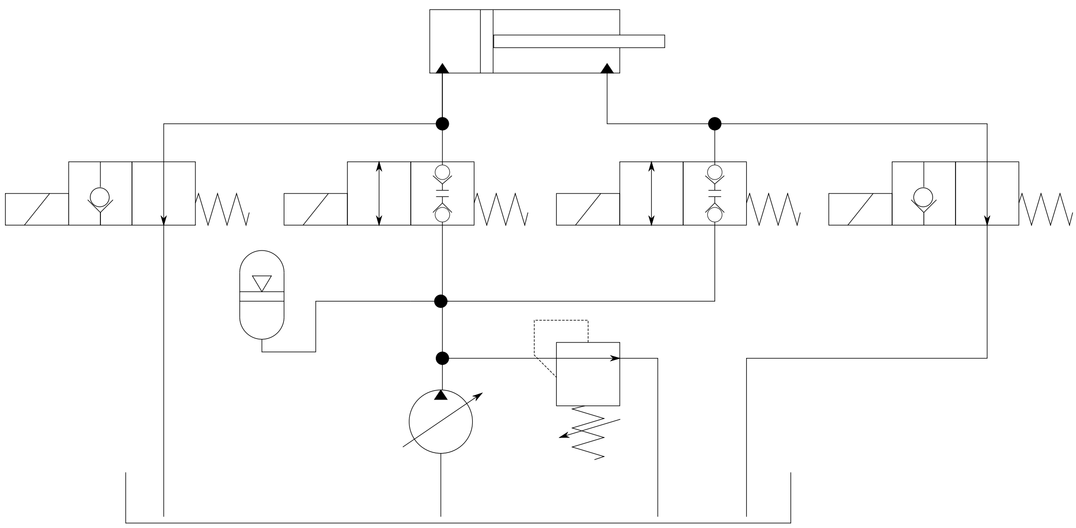


Figure 1.2. Hydac's concept drawing of the hydraulic servo system with 4 ON/OFF valves.

Figure 1.2 shows a concept design from Hydac with 4 ON/OFF valves implemented instead of the proportional valve shown in figure 1.1. The illustrated design is a concept drawing, which showcase one of the problems connected to changing the valve. It shows that if a proportional valve should be replaced then it will need to be replaced with more than one ON/OFF valve.

At the sketch from Hydac, the proportional valve is replaced with 4 ON/OFF valves enabling the pump flow to be redirected to each of the chambers and from each chamber to the tank. The basic functionality of the proportional valve is thus kept as shown in figure 1.3. It will later be discussed if this setup is the optimal configuration. It is seen that the control of the servo system will be multivariable due to the increased number of valves, which implies that it is important to understand if and how the valves influence each other and the importance of designing a sound control strategy.

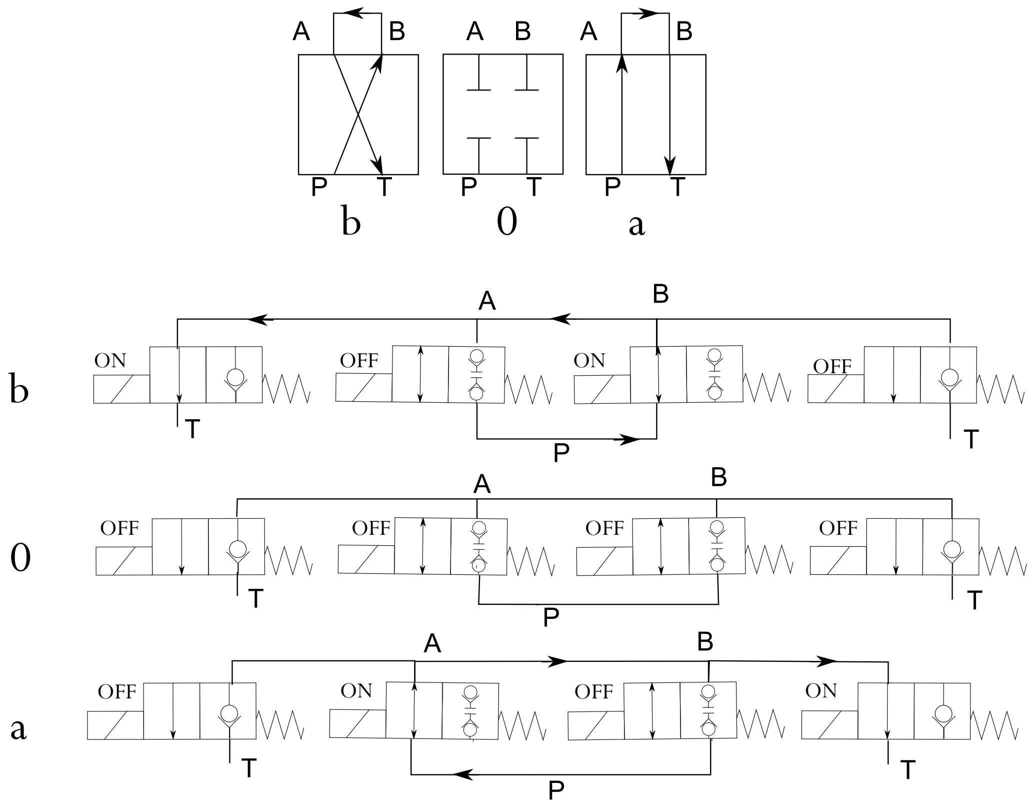


Figure 1.3. The 3 basic states ’b’, ’0’ and ’a’ which can be reached by a standard 4/3-proportional valve and how 4 ON/OFF valves should be turned on and off to reach the same functionality.

The idea of replacing the main proportional valve with multiple valves has been a research area for the past two decades, see [Aardema, 1999], [Liu et al., 2009], [Nielsen, 2005] and [Opdenbosch et al., 2011]. Most of the researchers have implemented 2/2 proportional valves or 4/3 proportional valves in order to control the meter-in and meter-out edges, separately i.e. control the flow in and out of each cylinder chamber independently.

This analysis differs by using ON/OFF valves instead. Within the pneumatic field, some studies have already been made with ON/OFF valves, see [Hejrati and Najafi, 2013] and [Ahn and Yokota, 2015]. [Ahn and Yokota, 2015] show promising results with a tracking error as low as 0.2mm, but it is from a pneumatic system with low pressure and almost no applied force. Therefore the results are not directly transferable.

1.1 Proportional Valve vs. ON/OFF Valve

An ON/OFF valve is a 2/2 valve, which can operate in two states: ON - sometimes referred to as energised or OFF - sometimes referred to as de-energised. Depending on the configuration of the valve the ON state may refer to fully open or fully closed. The proportional valve may reach intermediate states e.g. 20% open, 70% open etc. The larger number of valve settings makes the proportional valve the common choice in servo systems since it is easier to control the position and movement of the hydraulic cylinder.

For example, if the desired flow through the valve is half of the valve's rated flow then the flow through the proportional valve will be controlled with a single voltage setting. With an ON/OFF valve, there are different possibilities instead of the voltage set point: one could be to send a pulse-width modulated control signal (pwm) with a duty cycle of 50% which would give an average flow at the desired level, otherwise it is necessary to change the supply flow or change the pressures before and/or after the valve in order to minimize the flow.

The proportional valve's increased functionality comes with a higher price tag compared to an ON/OFF valve. A low performance proportional valve costs around 4000 DKK whereas the ON/OFF valves typically has a price around 200 DKK[Meeting, 2014]. Even with an additional manifold for the ON/OFF valves then the total price of the ON/OFF system should be lower compared to the proportional valve system.

A proportional valve will open for flow to both chambers of the hydraulic cylinder simultaneously, thereby connecting one chamber to the pump and the other to the tank. By using ON/OFF valves the flow in and out of each chamber can be controlled separately. The throttle losses can hereby be minimized if it is sufficient to open for flow to one instead of two or by selecting a different size of valve for the outlet flow compared to the one for the inlet.[Liu and Yao, 2008]

1.2 The Challenging Aspects of Using ON/OFF Valves

One of the main challenges with ON/OFF valves is the switching time, meaning the time it takes the valve from being fully closed to fully open or the other way around. Figure 1.4 shows the difference between the control signal and the actual placement of the valve spool. The valve opening x_v is either 1 corresponding to energised state ON or 0 corresponding to de-energised state OFF of the valve. It is assumed that the transition from open to closed and back again can be approximated as having constant rate.

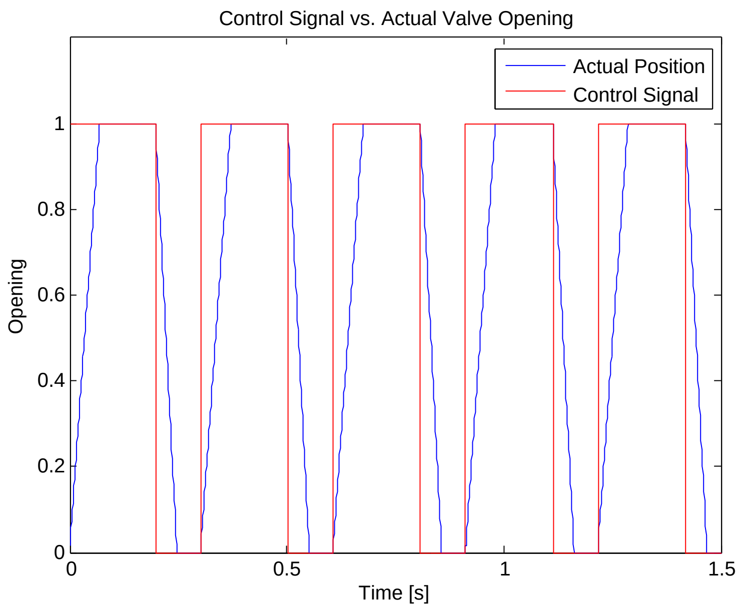
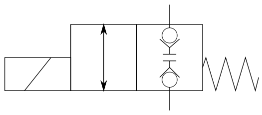


Figure 1.4. Simulated model response showing the difference between the control signal (red) and the position of the valve (blue) due to the switching time.

As seen in table 1.1, which shows a small selection of Hydac's product portfolio, the time varies between 30 and 70 ms depending on the size of the valve and position of the spool. There exists other valves on the market with lower switching time, but for now the analysis will be based on standard low cost ON/OFF valves. The valves will chosen after the system has been dimensioned (section 3.3). The switching time will restrict how fast each valve can be switched and problems may arise if the switching time is too high.



2/2 Poppet Valve Normally Closed Bidirectional			
Name	**Q_{max} [L/min]**	**Opening [ms]**	**Closing) [ms]**
WS08W-01	19	35	50
WSM06020W	19	35	50
WSM06020W-01	25	30	40
WS10W-01	40	50	50
WSM16520W	100	70	50
WSM12120W	110	30	70

Table 1.1. A selection of the product range for 2/2 poppet valve normally closed bidirectional for different flow sizes. Between 30 and 70 ms are used before the valves reach open or closing position. Section 3.3 will show the selected valves for this project after an analysis of the required size and configuration are made.

Another important dynamic aspect of using ON/OFF valves is the fluid hammer effect. The fluid hammer effect covers pressure and flow transients which are generated when the flow conditions change rapidly such as opening or closing for flow. There are rapid changes in the velocity of the fluid when an ON/OFF valve opens or closes, which will send pressure waves through the system. When the valve closes then the upstream flow will be stopped resulting in a momentary increase in pressure, while the downstream flow will decrease in pressure. These changes are reflected at the end of the pipelines, which sends pressure waves back to the valve, which again are reflected.

Depending on the system configurations, pressures, flows and valves, the fluid hammer effect can influence the performance. Yang and Kuo [2001] describes this effect in a monopropellant hydrazine reaction control subsystem for a satellite. 4 thruster valves control the flow to a propulsion system. Each valve is controlled with a pwm signal and has a switching time of 170 ms, which makes it comparable, to some extent, to the hydraulic cylinder system found in this report.

The article documents experimentally that in their hydraulic system with a supply pressure of 24 bar and a switching time of 170 ms, then the fluctuations will have an amplitude of up to 1 bar and converge toward a steady state within 0.5s. The results from the article is shown in figure 1.5. The flucturations affect the propulsion performance parameters, such as thrust and specific impulse due to the high pressure change sensitivity.

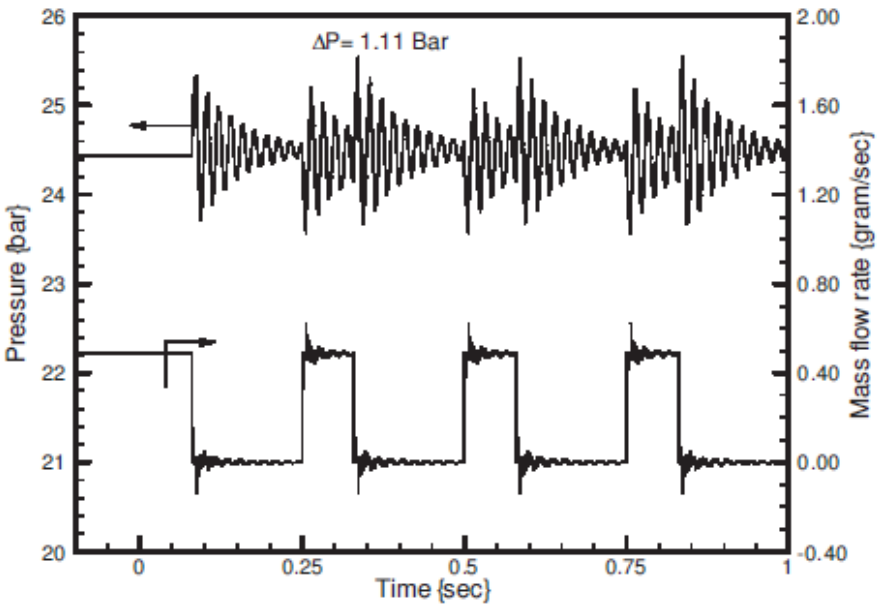


Figure 1.5. The documented fluid hammer effect from opening and closing an on/off valve in a system. The top response goes with the left axis and the bottom with the right axis. It is seen that fluid hammer influences both the system pressure and the mass flow rate. Image courtesy of Yang and Kuo [2001,page 298].

If the same oscillations are present in the hydraulic servo system for this thesis, then it will make the position of the piston in the hydraulic cylinder inaccurate each time a valve switches. Based on the results shown, then the fluid hammer effect should be taken into consideration in the model. An approximation of the size of the fluid-hammer forces and spectra will be made after the system is dimensioned.

1.3 Summary: The 3 Identified Problems

As mentioned earlier in this chapter using ON/OFF valves in a traditional servo system will add the following challenges compared to a traditional proportional valve servo system:

- **Multivariable Switching Strategy:** It would be necessary to replace a proportional valve with several ON/OFF valves. The control strategy will therefore be multivariable, which expands the problem to a series of different control scenarios and configurations. The use of a hydraulic cylinder in different servo systems present the possibility that not all valves may need to be turned on and off and that the control strategy would vary with the objective.

- **Switching time:** A standard hydraulic ON/OFF valve uses between 30 and 70 ms to open or close. This time restricts the switching frequency of each valve. If the switching restriction causes the control strategy to loose performance, then it should be investigated how fast each valve needs to switch in order to meet the performance demands.

- **Fluid hammer effect:** Switching between on and off is not without costs. Pressure and flow transients are generated each time an ON/OFF valve switches state.

> Depending on the magnitude and duration of the transients then the system can be subject to undesired disturbance and cylinder movement. This effect may also restrict the performance of the system.

Another concern could be the energy consumption of using several ON/OFF valves instead of a single proportional valve. However, a proportional valve continuously uses power to maintain the position setpoint due to a fluctuating external force caused by the flow. Conversely an ON/OFF valve is either fully open or fully closed most of the time, which significantly reduces the necessary power to maintain the setpoint. For this reason, power consumption is not considered an issue relevant for this thesis, but it can be investigated by others if the conclusion speaks in favor of using ON/OFF valves.

These three main differences are going to determine how the ON/OFF valves will perform compared to the more expensive proportional valve.

Chapter 2

Thesis Statement

The previous chapter identified the main goal and the main problems involved in achieving this goal. There exists many different servo systems, where the three most common systems are the position-, velocity- and the force servo. It will not be possible to investigate all three, or other possible servo systems, therefore only one is chosen as the objective for this report. The velocity servo is chosen for this thesis due to it having a wide application range.

The thesis statement is defined as

The purpose of this thesis is to analyse the feasibility of using ON/OFF valves instead of the conventional proportional valves for controlling a hydraulic cylinder. A performance analysis of the ON/OFF valve velocity servo system is compared to a conventional proportional valve velocity servo system. As a conclusion, the consequences, limitations and benefits of using ON/OFF valves instead of proportional valves for a velocity servo system are stated.

2.1 Delimitation

The project focus will be strictly on controlling the valves and their performance in regard to moving the hydraulic cylinder. Thus the pump will not be a part of the analysis and will be seen as delivering a constant pressure and flow. Adding a variable pump gives a new dimension to the control strategy as a weighting between how to control the pump and how the valves should be made. As a result, each ON/OFF valve will be controlled by a modulation signal instead of changing the surrounding pressures. Having 4 independent valves gives the possibility to The analysis is based on cheap low performance valves in order to show the potential and problems with this technology if it was implemented today as a competitor to a low performance proportional valve. There has not been time for building a test setup, so the results are only verified by simulations.

2.2 Project Strategy

In order to fulfill the thesis statement, the project will be divided into 3 parts: System understanding, system control and evaluation. The purpose of each part are shown below, and a set of goals for each part will be presented at the beginning of the part.

1. **System understanding**
   The purpose of this part is to design the servo system and to analyse the problems found in section 1.3. It is necessary to design and size the basic servo system which would be the foundation of the analysis. Then the three identified problems can be analysed. A reference system with a proportional valve is also needed in order to compare results.

2. **System control**
   The purpose of this part is to analyse how the valves should be controlled. A dynamic analysis will be made in order to show the limitations and possibilities in regard to controlling the system. The size of different key parameters will have an influence on the final perfomance and should be optimized accordingly. Controllers will be designed for both the proportional and ON/OFF valve setup.

3. **Evaluation**
   The purpose of this part is to analyse if the thesis statement is met. The performance of the ON/OFF valve system with the chosen control will be compared to the performance of a standard proportional valve. By presenting the challenges and results found, a set of statements can be made which show if and when ON/OFF valves can be used instead of proportional valves and how the performance is expected to be outside these areas.

Part I

System Understanding

Chapter 3

System Modelling and Choice of Components

This chapter describes how the different servo system components will be chosen and sized. The dynamic behaviour of a proportional valve and a digital valve will be investigated and a model of each system will be made.

3.1 Basic System

Figure 3.1 shows the cylinder with notation. The cylinder will not change between the proportional valve setup and digital valve setup and the notation will likewise be the same.

Table 3.1 shows the size requirements given by Hydac.

If the cylinder is completely extended then it will contain 7.85 L in chamber 1 and contracted 2.82 L in chamber 2. The contribution from the dead volumes in each end are not taken into account in these numbers. If chamber 1 contains a pressure of 150 bar and chamber 2 is open towards the tank, then the cylinder will be able to counterbalance a force of 118 kN or the same as 12 tonnes.

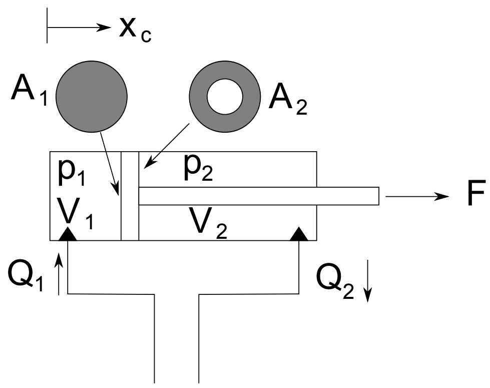


Figure 3.1. The cylinder with notation. The cylinder does not change depending on the valves.

Name	Notation	size	unit
Diameter of piston	D_{piston}	100	mm
Diameter of rod	D_{rod}	80	mm
Length of cylinder	L	1	m
Maximum cylinder velocity	$v_{\text{c max}}$	150	mm/s
Pump pressure	p_s	150	Bar

Table 3.1. Hydac's requested sizing of the cylinder and system.

From table 3.1 the maximum flow to chamber 1 and 2, denoted Q_1 and Q_2 can be calculated:

$$Q_1 = v_{\text{c max}} \cdot A_1 = 70.68 \qquad \left[\frac{L}{min}\right] \tag{3.1}$$

$$Q_2 = v_{\text{c max}} \cdot A_2 = 25.45 \qquad \left[\frac{L}{min}\right] \tag{3.2}$$

The requirement to the flow will be used when the valves in the two concepts are sized.

3.1.1 Modelling the Cylinder

The following section is based on Andersen and Hansen [2003] and Andersen and Hansen [2007]. The effective stiffness (Bulk modulus) β and the density of the hydraulic oil ρ are assumed constant. The continuity equation defines the flow in and out of the cylinder.

$$Q_{in} - Q_{out} = \dot{V} + \frac{V}{\beta} \cdot \dot{p} \qquad \left[\frac{m^3}{s}\right] \tag{3.3}$$

where Q_{in} is the flow into the chamber, Q_{out} is the flow out of the chamber, V is the volume, $\dot{V}$ is the change in volume, β is the bulk modulus of the fluid and p is the pressure.

A force equilibrium can be made for the cylinder. The sign of the system follows the sketch of the system given in figure 3.1. It is assumed that there is no leakage between the two chambers.

$$F_{res} = F_{p_1} - F_{p_2} - F_D + F \qquad [N] \tag{3.4}$$

where F_{res} is the resulting force, F is an external load force, F_{p_1} is the force on the piston from the pressure in chamber 1, F_{p_2} is the force on the piston from the pressure in chamber 2 and F_D is damping of the system.

The governing equations for the cylinder are:

$$Q_1 = \dot{V}_1 + \frac{V_1}{\beta} \cdot \dot{p}_1 \qquad \left[\frac{m^3}{s}\right] \tag{3.5}$$

with $V_1 = x_c \cdot A_1 + V_{1,0}$.

$$0 - Q_2 = \dot{V_2} + \frac{V_2}{\beta} \cdot \dot{p_2} \qquad (3.6) \qquad \left[\frac{m^3}{s}\right]$$

with $V_2 = (L - x_c) \cdot A_2 + V_{2,0}$.

$V_{1,0}$ and $V_{2,0}$ are the dead volumes at each end of the cylinder. It is assumed that there is no leakage flow between the two chambers.

After substituting the expression for the volumes, the pressure change rate can be found for each side:

$$Q_1 = \dot{x_c} \cdot A_1 + \frac{x_c \cdot A_1 + V_{1,0}}{\beta} \cdot \dot{p_1} \qquad (3.7) \qquad \left[\frac{m^3}{s}\right]$$

$$\Updownarrow$$

$$\dot{p_1} = \frac{\beta \cdot (Q_1 - \dot{x_c} \cdot A_1)}{x_c \cdot A_1 + V_{1,0}} \qquad (3.8) \qquad \left[\frac{Pa}{s}\right]$$

and

$$0 - Q_2 = -\dot{x_c} \cdot A_2 + \frac{(L - x_c) \cdot A_2 + V_{2,0}}{\beta} \cdot \dot{p_2} \qquad (3.9) \qquad \left[\frac{m^3}{s}\right]$$

$$\Updownarrow$$

$$\dot{p_2} = \frac{\beta \cdot (-Q_2 + \dot{x_c} \cdot A_2)}{(L - x_c) \cdot A_2 + V_{2,0}} \qquad (3.10) \qquad \left[\frac{Pa}{s}\right]$$

3.1.2 Eigenfrequency and the Mass

The external load force is not defined in the requirements from Hydac, and will be left undetermined. Additionally, it is assumed that the system includes a mass which must be moved, designated by M. This includes fluid mass, weigth of cylinder piston, external weigth etc. An eigenfrequency analysis will be carried out in order to show the influence of the mass.

The forces in equation (3.4) can be rewritten into the following expression, but with the .

$$F_{res} = F_{p_1} - F_{p_2} - F_D + F \qquad (3.11) \qquad [N]$$

$$\updownarrow$$

$$M \cdot \ddot{x_c} = A_1 \cdot p_1 - A_2 \cdot p_2 - B \cdot \dot{x_c} + F \qquad (3.12) \qquad [N]$$

A Laplace transformation gives the following

$$s^2 \cdot M \cdot x_c(s) = A_1 \cdot p_1(s) - A_2 \cdot p_2(s) - s \cdot B \cdot x_c(s) + F(s) \qquad (3.13) \qquad [N]$$

Both Q_1 and Q_2 are set equal to 0 in order to find the lowest eigenfrequency. The two equations are then Laplace transformed

(3.14) $$Q_1 = \dot{x_c} \cdot A_1 + \frac{V_1}{\beta} \cdot \dot{p_1} \qquad \left[\frac{m^3}{s}\right]$$

$$\downarrow$$

(3.15) $$0 = s \cdot A_1 \cdot x_c(s) + s \cdot \frac{V_1}{\beta} p_1(s) \qquad \left[\frac{m^3}{s}\right]$$

(3.16) $$-Q_2 = -\dot{x_c} \cdot A_2 + \frac{V_2}{\beta} \cdot \dot{p_2} \qquad \left[\frac{m^3}{s}\right]$$

$$\downarrow$$

(3.17) $$0 = s \cdot A_2 \cdot x_c(s) - s \cdot \frac{V_2}{\beta} p_2(s) \qquad \left[\frac{m^3}{s}\right]$$

(3.18)

Combining these expressions with the Laplace transformed force equilibrium gives

(3.19) $$\frac{x_c(s)}{F(s)} = \frac{1}{s^2 \cdot M + s \cdot B + \beta \cdot \left(\frac{A_1^2}{V_1} + \frac{A_2^2}{V_2}\right)}$$

the cylinder eigenfrequency ω_n can then be found as

(3.20) $$\omega_n = \sqrt{\frac{\beta}{M}\left(\frac{A_1^2}{V_1} + \frac{A_2^2}{V_2}\right)} \qquad [Hz]$$

The eigenfrequency as a function of the mass M can now be found. First, the length of the cylinder can be divided into two sublenghts L_1 and L_2, where L= L_1 + L_2. Then $\psi = L_1/L$ thus the eigenfrequency can be defined as

(3.21) $$\omega_n = \sqrt{\frac{\beta}{M}\left(\frac{A_1}{L_1} + \frac{A_2}{L_2}\right)} \qquad [Hz]$$

$$\updownarrow$$

(3.22) $$\omega_n = \sqrt{\frac{\beta}{M \cdot L}\left(\frac{A_1}{\psi} + \frac{A_2}{1-\psi}\right)} \qquad [Hz]$$

for $0 < \psi < 1$. The hydraulic stiffness k_h will be defined as $\omega_n = \sqrt{k_h/m}$. Thus the minimum eigenfrequency can be found when the hydraulic stiffness reaches its minimum.

(3.23) $$\frac{dk_h}{d\psi} = \frac{\beta}{L}\left(\frac{A_2}{(1-\psi)^2} - \frac{A_1}{\psi^2}\right) = 0$$

$$\downarrow$$

(3.24) $$\frac{A_1}{\psi^2} = \frac{A_2}{(1-\psi)^2}$$

$$\updownarrow$$

(3.25) $$\psi = \frac{\sqrt{A_1}}{\sqrt{A_1} + \sqrt{A_2}}$$

For the areas given then $\psi = 0.625$. The hydraulic stiffness gradient is negative for $\psi < 0.625$ and positive for $\psi > 0.625$ thus a minimum is found.

$$k_{\text{h min}} = \frac{\beta}{L}\left(\sqrt{A_1} + \sqrt{A_2}\right)^2 \tag{3.26}$$

$$\Downarrow$$

$$\omega_{nmin} = \sqrt{\frac{k_{\text{h min}}}{M}} \quad [Hz] \tag{3.27}$$

Figure 3.2 shows how the eigenfrequency changes with the mass. The system can balancethe equivalent of 12 tonnes, thus the plots are shown for up to 10 tonnes.

As seen from figure 3.2, the mass will change the eigenfrequency of the system. Since a pwm signal will be used to control each valve, then the switching frequency can make the system unstable if the cylinder frequency are in the same area. Thus it is important to know how the mass influences the eigenfrequency and choosing it accordingly.

The basic system is now defined and sized. The next section will choose and model a proportional valve such that the ON/OFF valves can be compared.

3.2 Proportional Valve Setup

There are two different 4/3- proportional solenoid valves to choose from in Hydac's standard product portfolio. P4WE06 is rated to 40 L/min and P4WE10 is rated to 90 L/min. The maximum required flow was 70.68 L/min as shown in equation 3.1, thus in order to meet this requirement P4WE10 is chosen.

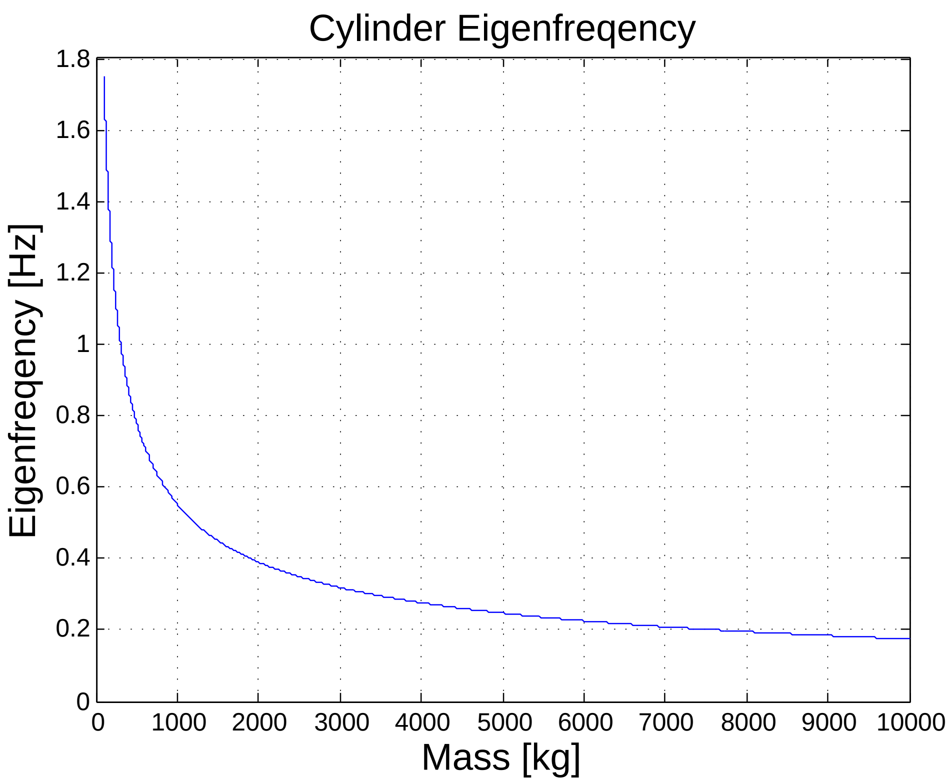


Figure 3.2. Cylinder eigenfrequency as a function of the mass.

The valve can operate in three states as shown in figure 1.3. In state 'a', the valve allows flow to pass from the pump into chamber 1 of the cylinder and allows flow to pass from chamber 2 into tank. In state '0', no flow is allowed through the valve. Finally in operation state 'b', the valve allows flow to pass from the pump into chamber 2 and flow from chamber 1 into the tank.

The valve is controlled by a voltage input of ± 15 [V]. The flow characteristic is shown in figure 3.3. At a reference signal of -15 [V], the opening area of the servo valve will be maximum in operation mode B. At a reference signal of +15 [V], the opening area of the servo valve will be in operation mode A. Operation mode 0 can be obtained by a reference voltage of -1.5 to 1.5 [V].

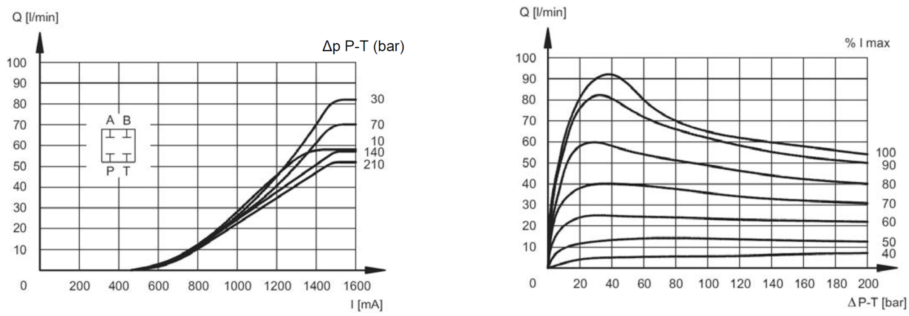


Figure 3.3. The flow through the proportional valve at different pressure drops and voltage settings. [Hydac International, 2013,p.825].

The flow characteristic shown in figure 3.3 is implemented in the model as a look-up table.

The dynamic response has not been investigated for the specific chosen valve, but Hydac states that the characteristics for the z version of the valve is representative [Meeting, 2014]. The response is shown in figure 3.4.

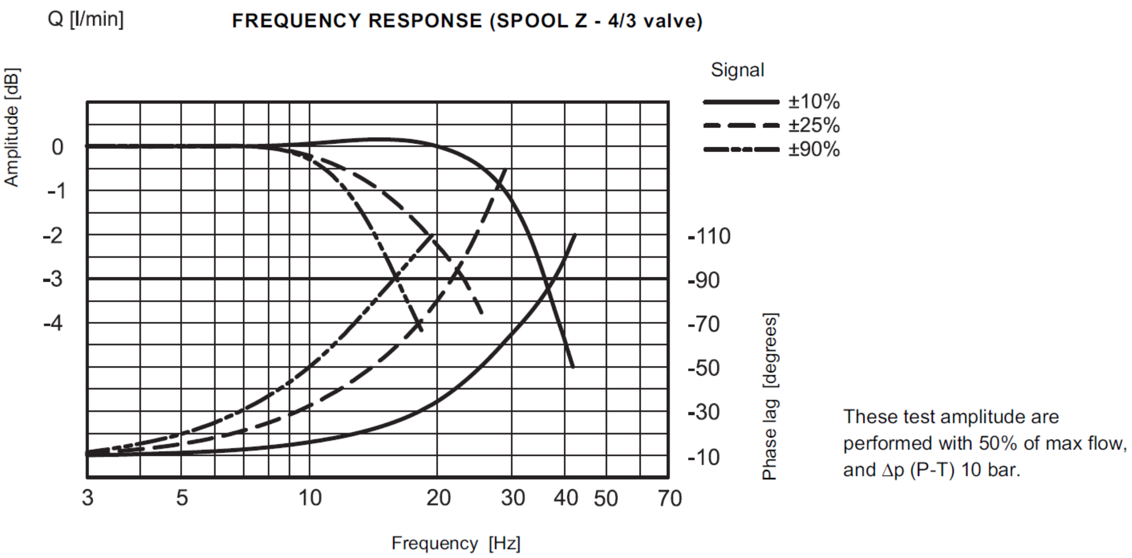


Figure 3.4. The bode plot for the proportional valve. It is for another valve than the chosen, but it should be representative. Image courtesy of Hydac.

From the appearance of the dynamic behaviour, it is reasonable to approximate it as a second order transfer function, as shown below.

$$G_{valve} = \frac{x_v}{x_{v,ref}} = \frac{\omega_v^2}{s^2 + 2 \cdot \zeta \cdot \omega_v \cdot s + \omega_v^2} \tag{3.28}$$

The valve was chosen to be larger than needed, thus an approximation are made after the $\pm 25\%$ line in figure 3.4. ω_v is found to be 200 and $\theta = 0.8$ in order to resemble the one given in figure 3.4. The bode plot for this approximation is shown in figure 3.5.

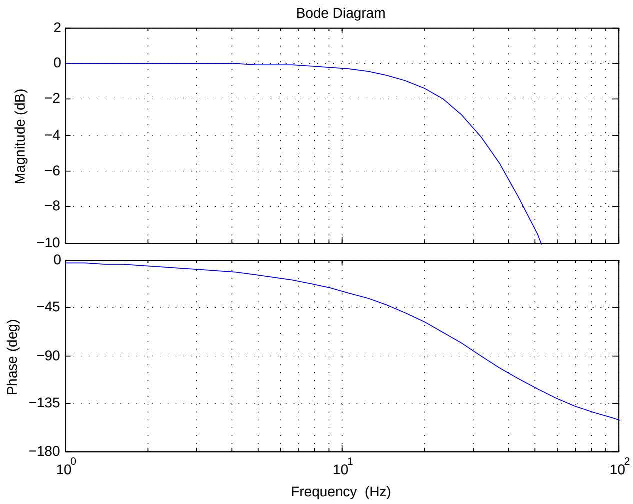


Figure 3.5. The resulting bode plot found by adjusting the parameters to fit figure 3.4 given by Hydac.

The proportional valve has now been modelled and is ready to be used for comparison.

3.3 ON/OFF Valve Setup

Before the ON/OFF valves are chosen, it is necessary to review the concept drawing and see if anything should be altered. As stated earlier, the idea of replacing the proportional valve with several valves is not new. The literature contains different solutions and arguments about how many valves are needed and their configuration in the hydraulic circuit. Most of the suggestions resemble the Hydac draft (figure 1.2) with either the 4 ON/OFF valves or 2 3/2 valves [Opdenbosch et al., 2011], [Liu and Yao, 2008], [Hejrati and Najafi, 2013].

Ahn and Yokota [2015] suggests using two ON/OFF valves at each inlet - one rated with 1/3 of the flow and the second with 2/3 of the flow. A concept inspired by this idea is shown in figure 3.6. This configuration allows a greater variety of directly available flows. Instead of having 0 % and 100 % available then 2 more possibilities 33% and 66% are added as shown on 3.7.

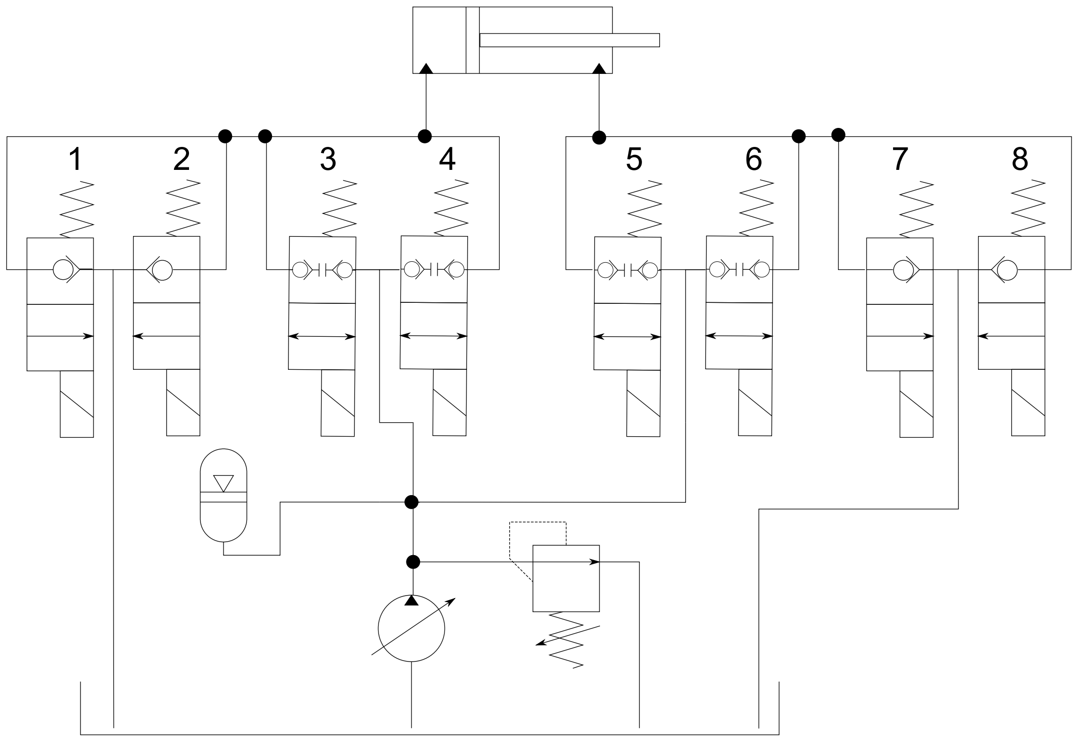


Figure 3.6. A concept inspired by the pneumatic work of [Ahn and Yokota, 2015].

More valves could be added after same principle until the functionality of a proportional valve is reached. Since price optimization is a driving ambition, then the solution with 4 valves will be analysed first. The next chapter will analyze the 3 problems identified in Chapter 1, if the results are deemed inferior, then an analysis based on 8 valves will be used instead.

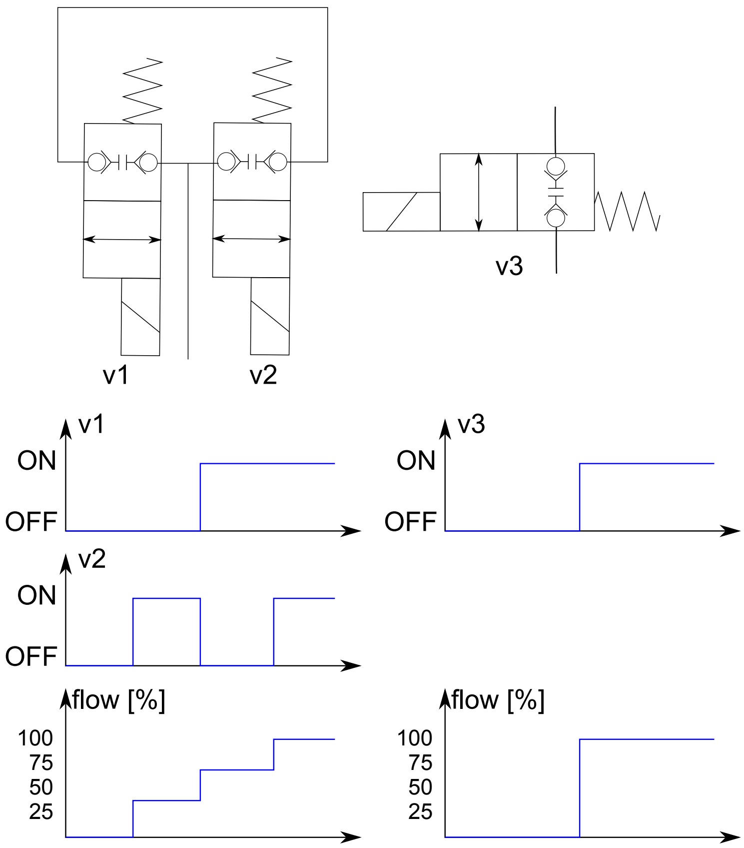


Figure 3.7. The difference in flow by having two ON/OFF valves instead of one.

The two normally open valves in Hydac's concept are changed to two normally closed valves. If something breaks the power turns off which closes every valve thus preventing further damage. Thereby ON is defined as open, and OFF is defined as closed. Figure 3.8 shows the system chosen for the analysis.

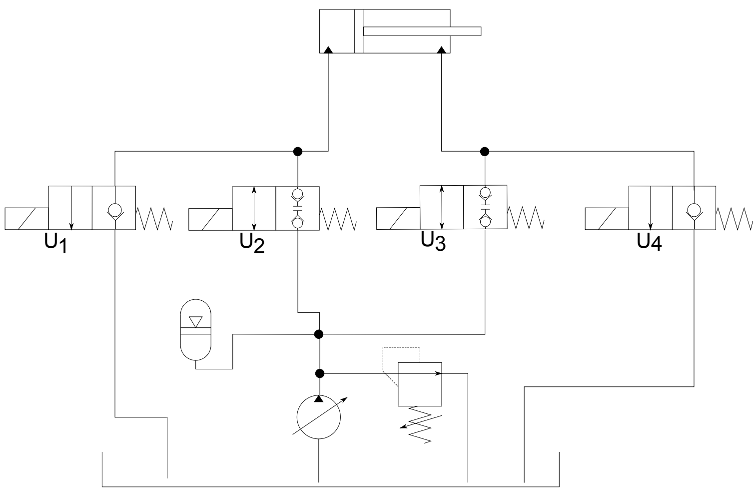


Figure 3.8. The chosen concept with valve numbers. The two valves to the tank is changed in order to ensure break-down safety.

4 valves are chosen from the product portfolio. There is not a perfect match for each of the rated flows, therefore valve 1,2 and 4 are chosen larger than needed, and valve 3 is 0.45L/min smaller than the required, which are judged to be OK. The specifications can be seen in table 3.2.

The chosen valves			
Name	$\mathbf{Q_{max}}$ **[L/min]**	$T_{ON,0}$ **opening time [ms]**	$T_{OFF,0}$ **closing time [ms]**
Valve 1: WS10Z	75	30	60
Valve 2: WSM16520W	100	70	50
Valve 3: WSM06020W-61	25	30	40
Valve 4: WS08Z-01	38	35	50

Table 3.2. The 4 chosen valves. Each valve is rated to 350 Bar and is solenoid actuated. [Hydac International, 2013,p.299,301,377,381]

The flow characteristics from the data sheets are shown in figure 3.9. The implementation in the model consists of four look-up tables - one for each valve.

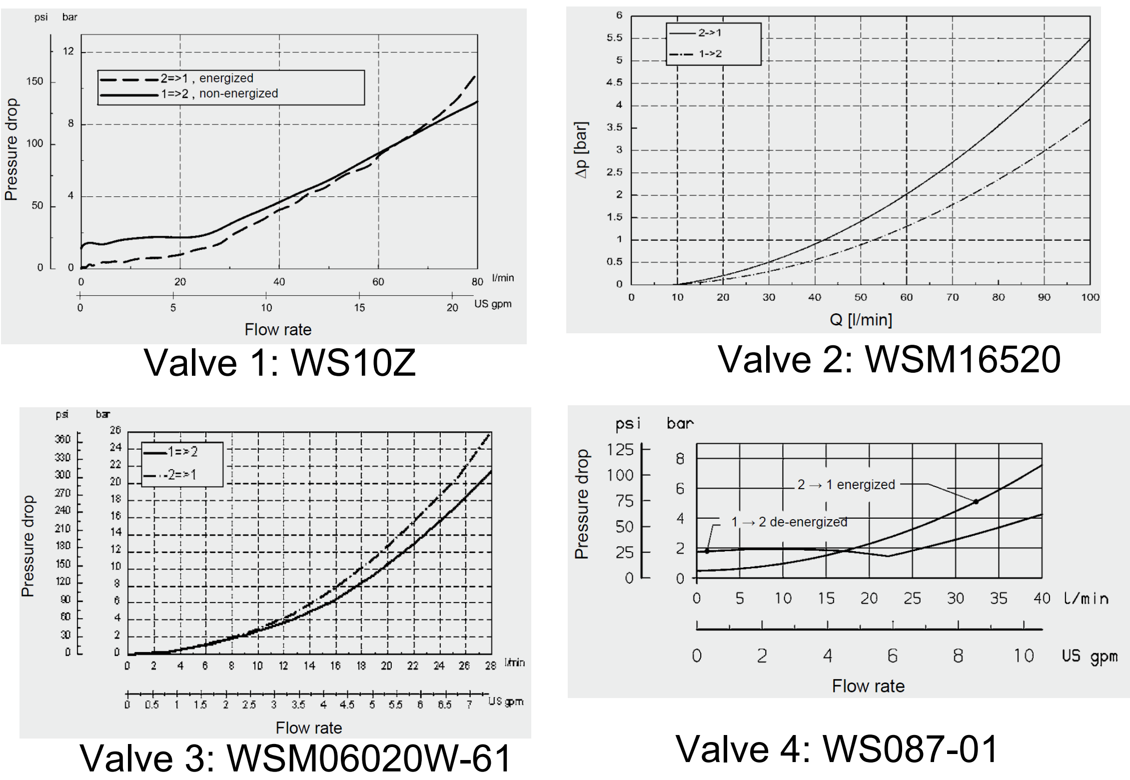


Figure 3.9. The flow characteristics for each of the 4 chosen valves. State '1' is closed, while state '2' is fully open.

The datasheets do not provide any information on the position of the valve spool during the switching time. Based on [Taghizadeh et al., 2009] the transition can be assumed to be constant rate, and a rate limiter is implemented in simulink for simulation purposes.

The block has two parameters: the rising rate and the falling rate, where the values from table 3.2 are implemented. The response to a pwm signal with 10 % duty cycle at 1 Hz for the 4 valves are shown in figure 3.10.

Lei et al. [2010] shows that the transition should be viewed as having constant rate. For analysis purposes a linear system is desired, thus it is followingly investigated whether a first order system has a sufficiently similar response. A standard first order system is shown below where τ is the time constant.

$$G(s) = \frac{1}{\tau s + 1} \tag{3.29}$$

If a step is given to a first order system then at 4τ 98.2% of the gain value is reached [Philips and Parr, 2011,p.136]. By approximating the system response as a first order system, the difference in the opening and closing time will be lost. The average switching time for the 4 valves will then be: 45ms, 60ms, 35ms and 42.5ms respectively. The time constant for each of the systems can therefore be found as

$$\tau = \frac{T_{avg}}{4} \tag{3.30}$$

The 4 time constants will then be 0.0113, 0.0150, 0.0088 and 0.0106, respectively. Figure 3.11 shows the responses for the valves to a step.

A simulation can be performed which shows that the two different models does not change the simulation results significantly. Figure 3.12 shows the position and velocity of the piston if valve 2 and 4 are controlled by a duty cycle of 70% at 1.5 Hz. The mass is set to 1000 kg, an initial pressure of 140 bar at p_1 and 140 bar at p_2 and a force of -70kN.

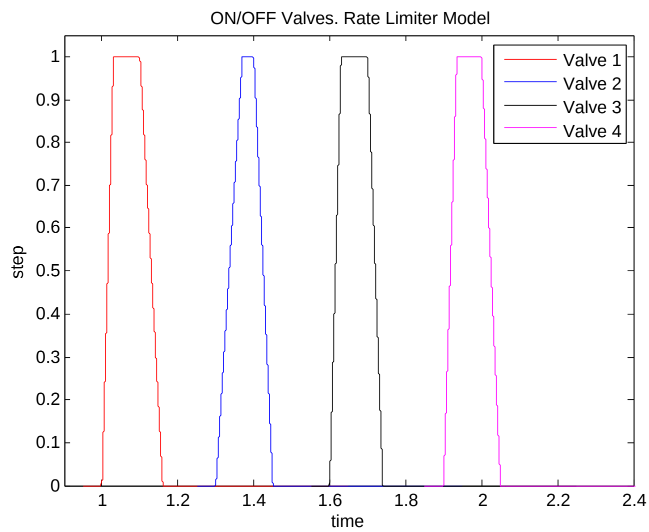


Figure 3.10. The implementation of the switching time in the model by a rate limiter block. The response to a single pwm for the valves are shown.

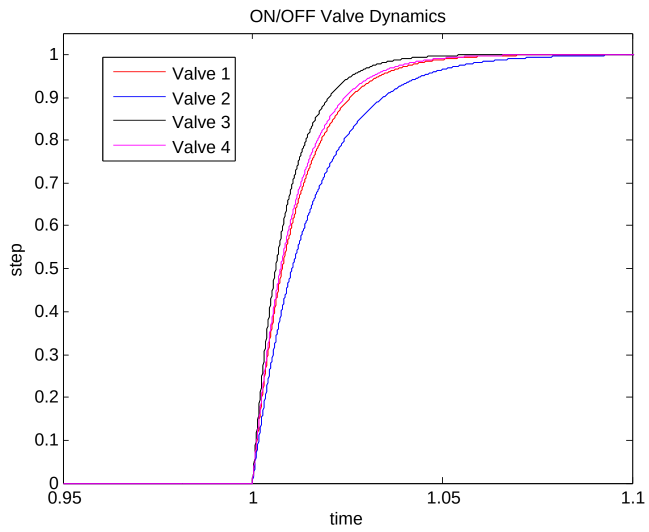


Figure 3.11. The step response for the 4 ON/OFF valves when they are modelled as a first order system.

The differences between implementing the valve dynamics as a rate limiter and a first order system are within 2 mm, and 2.7mm/s which are assumed to be acceptable. The true characteristic for the valves are not known, so both systems are assumed to be representative. The first order system approach are necessary for controller design, but most of the papers with experimental results show behaviours as given by the rate limiter. Therefore both modelling approaches are kept.

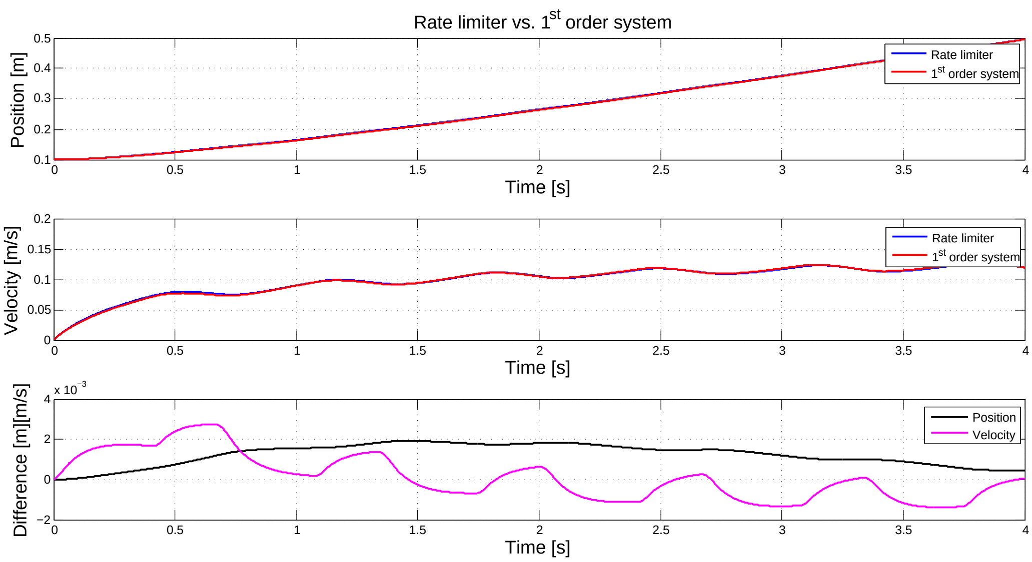


Figure 3.12. The difference in position and velocity if the valves are modelled as 1^{st}' order systems compared to a rate limiter.

It is noted that the velocity oscillate, a deeper analysis of this are made in chapter 5 and 6

3.4 Final Models

The two systems have been modelled, and figure 3.13 shows an overview of how the different model parts are connected. The red and blue lines show how the calculation of different boxes are connected to variables other places in the model i.e. flows depend on the pressures and the volumes depend on the position of the piston.

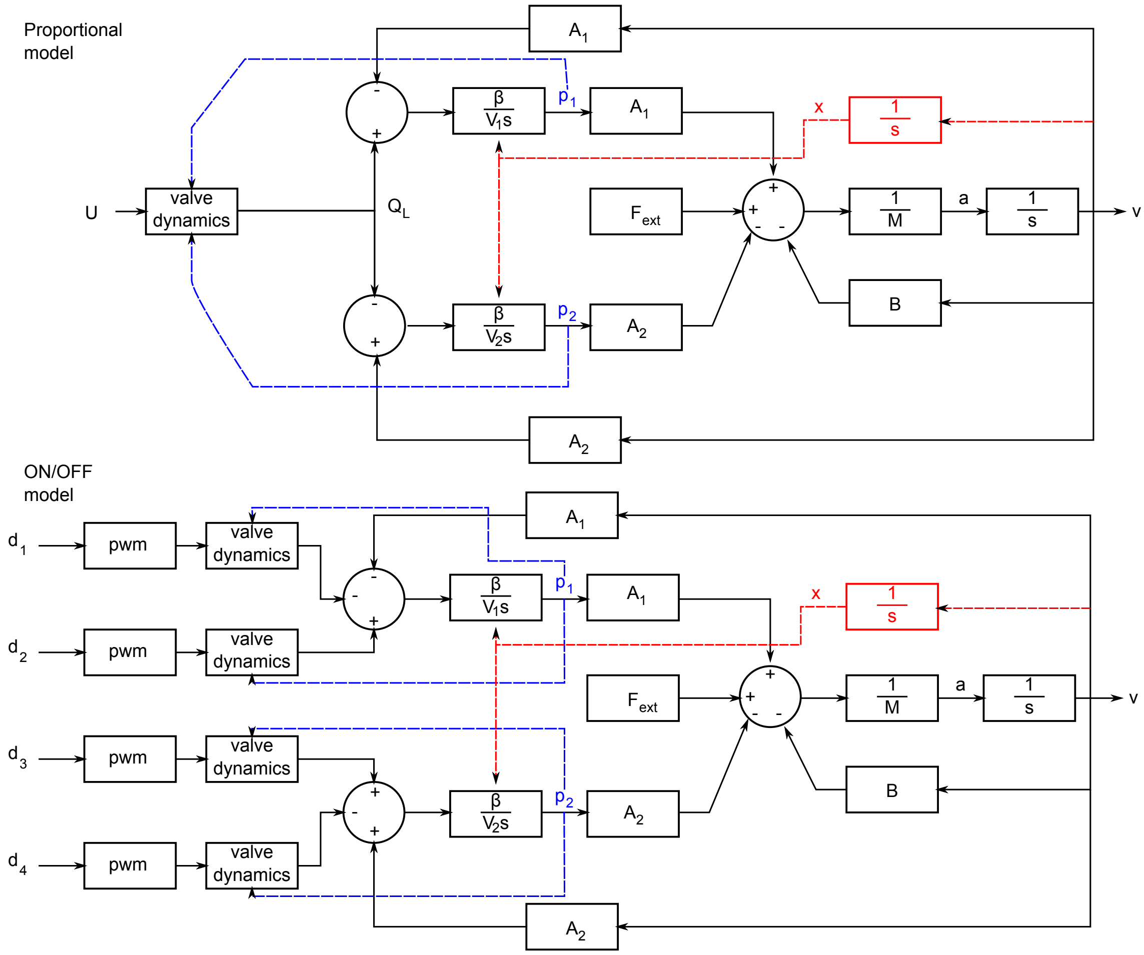


Figure 3.13. The proportional model at the top vs. the ON/OFF model at the bottom. The blue and red lines show how the calculation of different boxes is connected to variables other places in the model i.e. flows depend on the pressures and the volumes depend on the position of the piston.

It is seen, that the standard proportional valve will be a SISO system (single input single output), whereas the ON/OFF system shall be controlled as a MISO system (multivariable input single output). The main problem with control of the hydraulic cylinder are the nonlinear behaviour of the flow through the valves, which can cause problems both for control of a SISO system but also for the MISO system.

The next chamber will analyse the effect of the 3 identified problems for the ON/OFF valve setup and then the control analysis will be made.

Chapter 4

The Three Challenges

This chapter will investigate and analyse the effect of the three challenges introduced in Chapter 1. First the restriction from switching time will be analysed, then an approximation of the size of the fluid-hammer forces and spectra will be made and finally the effect of different switching algorithms will be presented. The next chapter will analyse which switching strategy should be used at different servo systems.

4.1 Switching Time

The switching time is the time it takes an ON/OFF valve to turn on or off, as presented in section 3.3. The purpose of this section is to show how much it restricts the switching of each valve. Since none of the valves has the same switching time then an average max switching frequency can be found for each valve.

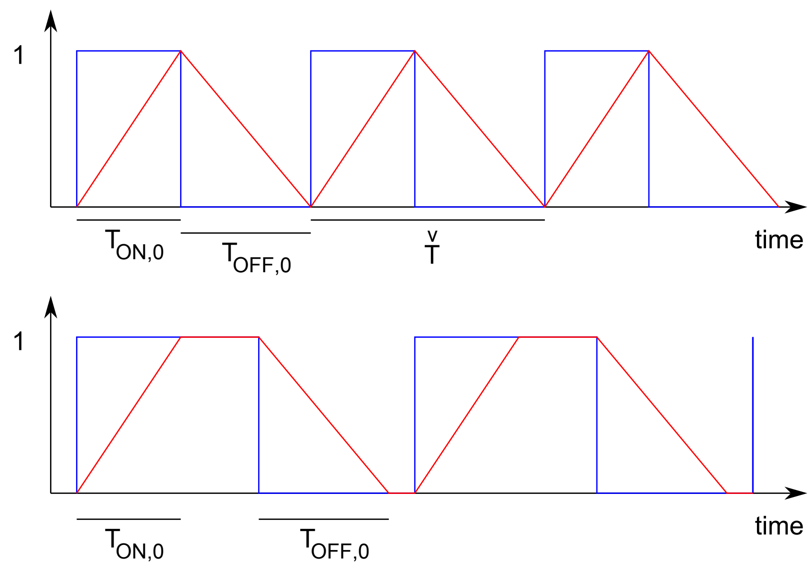


Figure 4.1. The minimum required period can be found as shown at the top picture. In order to change the duty cycle then the period needs to be longer as shown at the bottom picture.

The minimum period $\check{T}$ can be found as the time it takes the valve to switch on added to the time it takes the valve to turn off, as shown below:

$$\check{T} = T_{ON,0} + T_{OFF,0} \qquad [s] \tag{4.1}$$

and thus the maximum switching frequency $\hat{f}$ is found as

$$\hat{f} = \frac{1}{\check{T}} \qquad [Hz] \tag{4.2}$$

This restriction is visualised in figure 4.1. The top picture shows the minimum period.

It is therefore necessary to choose a lower switching freqency in order to allow the system to be on and off in longer time than it takes the system to switch, as shown at the bottom picture in figure 4.1.

Table 4.1 shows the restriction on switching frequency for each of the chosen 4 valves.

Maximum Switching frequencies	
Valve 1	11.11 Hz
Valve 2	8.33 Hz
Valve 3	14.29 Hz
Valve 4	11.76 Hz

Table 4.1. The maximum swithching frequencies. Based on Table 3.2and Equation 4.1.

If the valves were operated at the maximum frequency then the duty cycle will be 0.33, 0.58, 0.43 and 0.41 for valve 1-4 respectively. As the switching frequency is lowered then a range of duty cycles become available as shown in figure 4.2. Valve 2 is chosen to show the possibilities since it has the lowest maximum switching frequency, and thus the restrictions would be most significant for this valve.

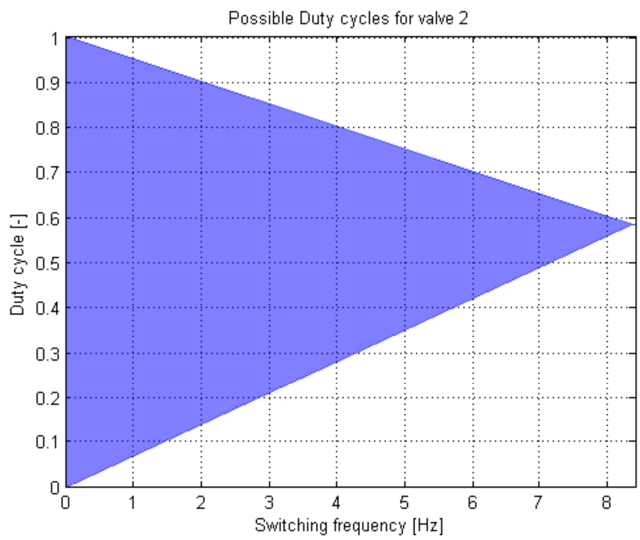


Figure 4.2. The duty cycle range is restricted depending on the switching frequency and -time.

As seen in figure 4.2 the switching frequency should be lowered to at least 1.47 Hz if the valve should be required to give between 10% and 90% of the flow. Thus choosing a switching frequency is restricted by the switching times of the valves and if faster switching is required, then the valves should be replaced.

4.2 Fluid Hammer Effect

As mentioned earlier, an instantaneous change in the fluid flow conditions will cause pressure rises and falls which gives pressure transients in the system, meaning waves will propagate and be reflected in the system. By opening or closing an ON/OFF valve, the momentum of the fluid will change. The rate of change in the momentum of a system is equal to sum of the forces exerted on the system, as given by Newton's second law of motion.

There are different ways to model the fluid hammer effect based on how the phenomenon is considered. In the distributed-system approach, the transients travels as waves whereas the lumped-system approach assumes that any change in fluid conditions will travel instantaneously throughout the system [Chaudhry, 2014]. Based on the experimental results from Yang and Kuo [2001] it is fair to assume that a distributed-system approach should be chosen. Chaudhry [2014] supports this choice and notes that the lumped-system approach is best suited for simulations of surge tanks and other slow oscillating systems.

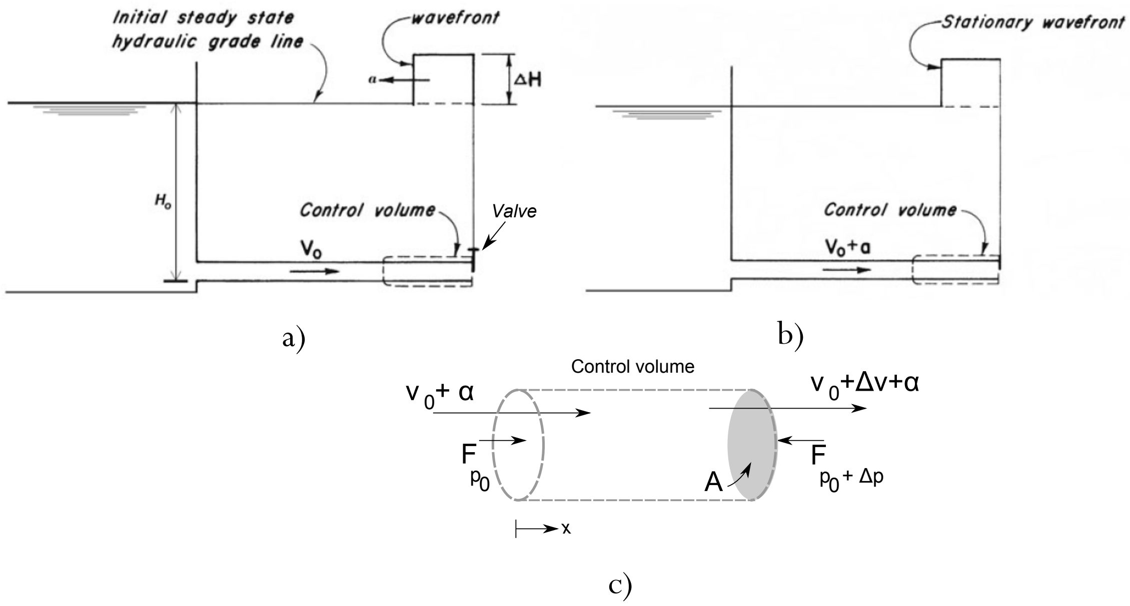


Figure 4.3. The valve is closing at the end of pipeline which has an initial pressure head of H_0 and velocity of V_0. When the valve closes situation a) will happen: An unsteady flow will rise in the pipe, leading to a pressure change at the end with the size of ΔH resulting in a moving wave front with velocity a. By superimposing the velocity a, situation b) is given. c) shows the forces and flow through the control volume. [Chaudhry, 2014] has image courtesy for a) and b).

Figure 4.3 shows a sketch of the system. A pipeline with a valve at the end in which a fluid is flowing with a velocity v_0. The initial pressure is shown by the initial pressure head H_0 created by the reservoir. At time t_0 the valve will close and situation a) will happen: An unsteady flow will rise in the pipe, leading to a pressure change at the end with the size of ΔH resulting in a moving wave front with velocity α. The velocity will change to $v_0 + \Delta v$ and the pressure head to $H_0 + \Delta H$.

By assuming the pipe is rigid, the wave velocity can be found as

$$\alpha = \sqrt{\frac{\beta}{\rho_0}} \qquad \left[\frac{m}{s}\right] \tag{4.3}$$

where β is the bulk modulus of the fluid and ρ_0 is the fluid density. For oil with a bulk modulus of 1.5 GPa and a density of 900 kg/m^3 the wave speed will be 1291 m/s.

By superimposing the wave velocity α, the initial velocity is $v_0+\alpha$. Then the moving wave front will appear stationary. The outflow velocity of the control volume will be $v_0+\Delta v+\alpha$ as given in situation b) in figure 4.3. Then the change in momentum through the control volume (as defined at c)) can be estimated as [Chaudhry, 2014,p.8]

$$\frac{dM_{sys}}{dt} = \rho_0 \cdot A \cdot (v_0 + \alpha)\left[(v_0 + \Delta v + \alpha) - (v_0 + \alpha)\right] \tag{4.4}$$

$$= \rho_0 \cdot A \cdot (v_0 + \alpha) \cdot \Delta v \tag{4.5}$$

A is the area of the control volume. The change in momentum is equal to the sum of the forces. If the friction force is neglected then the force would be:

$$F = p_0 - (p_0 + \Delta p) = \rho_0 \cdot g \cdot H_0 \cdot A - \rho_0 \cdot g \cdot (H_0 + \Delta H) \cdot A \qquad [N] \tag{4.6}$$

$$= -\rho_0 \cdot g \cdot \Delta H \cdot A = -\Delta P \tag{4.7}$$

Hence, it follows from equation 4.4, 4.6 and Newton's second law of motion ($\dot{M_s}ys = \sum F$) that

$$\Delta p = -\rho_0 \cdot (v_0 + \alpha) \cdot \Delta v \tag{4.8}$$

Since $\alpha \gg v_0$ (1291 m/s $\gg$ 0.15 m/s) then the first term can be neglected. The expression is then reduced to:

$$\Delta p = -\rho_0 \cdot \alpha \cdot \Delta v \tag{4.9}$$

The analysis can be done again for the other side of the valve, where the velocity will decrease after closing of the valve. It can be shown that the change in pressure is equal to:

$$\Delta p = \rho_0 \cdot \alpha \cdot \Delta v \tag{4.10}$$

4.2.1 Influence on the system

The pressure wave will be completely reflected at the ends, thus changing sign and move back to the source. The time between each wave will be the time it takes for the wave to reach one end, be reflected and return. At time t_0 a pressure wave will be sent, it reaches the other end at $t_1 = L/\alpha$, changes sign and moves back. At $t_2 = 2 \cdot L/\alpha$ the negative pressure wave will hit the valve, be reflected and move again. This is shown in figure 4.4, where the friction in the pipe is neglected thus the wave will continue to move up and down with the same pressure change.

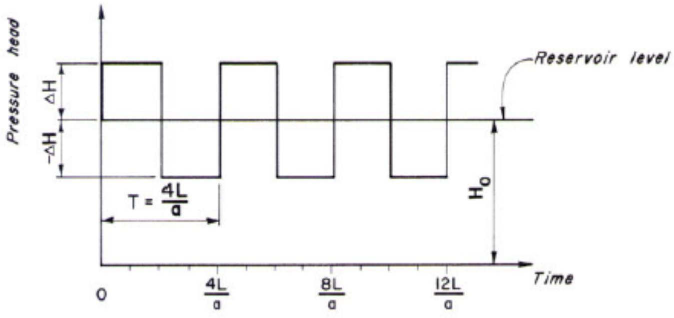


Figure 4.4. The transient pressure wave if the pipe is considered frictionless.[Chaudhry, 2014]

The effect at the maximum cylinder velocity can be seen in figure 4.5. The flow through valve 2 is 75 l/min, the initial pressure is 150 Bar and the pipeline has a length of 1 m. The friction loss is taken into consideration followingly.

$$\text{Friction Loss} = \frac{\pi \cdot d \cdot f \cdot \alpha \cdot |\alpha| \cdot \rho \cdot \Delta x}{4} \tag{4.11}$$

where d is the diameter of the pipeline, f is the Darcy Weisbach friction factor and Δx is the travelled distance. The Darcy Weisbach is found from a Moody's diagram to be 0.0025 assuming a Reynolds Number of 16866 and a relative roughness of 3.33e-5.

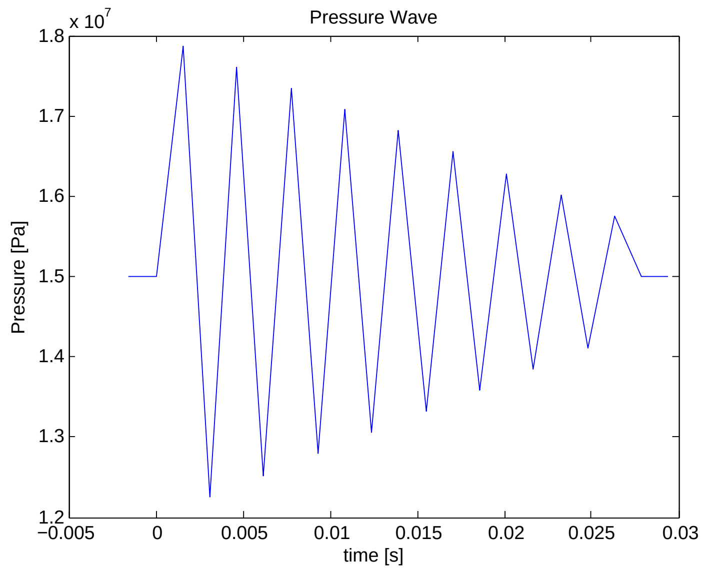


Figure 4.5. The transient pressure wave in the system with friction included. Initial velocity is set to 0.15 m/s, initial pressure 150 Bar and a pipeline length of 1 m.

From figure 4.5 it can be seen that the overshoot is 25 % and a rise time of 2 ms, thus a second order system can be approximated for the response with $\zeta = 0.38$ and $\omega_n = 1800$. The step response of the modelled fluid hammer effect is shown in figure fig:stepresponse.

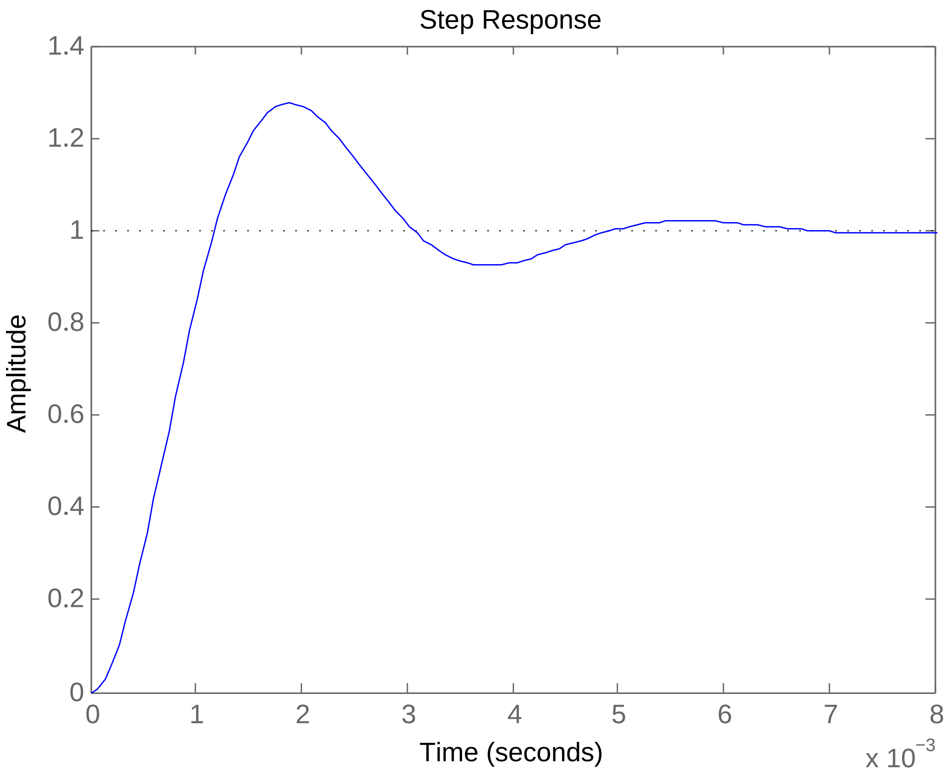


Figure 4.6. The applied second order system to the model to fit the transients of having a fluid hammer effect.

If the valves could be synchronised so the pressure transients will hit the piston wall at the same time from both sides, then the effect will be reduced. There would be a different instantaneous force at the same velocity drop due to fact that the area of the piston wall in chamber 2 is 36% of the one found in chamber 1. The different switching times and the position of the piston will have an influence on when a valve should be opened or closed in order to meet the effect. The next section will investigate the effect of different switching combinations, and analyse the possibility to match the switching of the valves for the two chambers.

4.3 Multi Variable Switching

The basic system consists of 4 ON/OFF valves. Each valve has two states: ON and OFF. This section will present what flow there will be present if a valve is turned on and rank the different ON/OFF combinations for the valves.

Chamber 1 is connected to the tank by valve U_1 and to the pump by valve U_2. Thus 4 different flow combinations can be found as seen below. 0 represents the valve is OFF (closed), and 1 means ON (open), the notation is as found in figure 3.1.

$$Q_1 = 0 \qquad for\ U_1 = 0,\ U_2 = 0 \tag{4.12}$$
$$Q_1 = Q(P_1, P_S) \qquad for\ U_1 = 0,\ U_2 = 1 \tag{4.13}$$
$$Q_1 = -Q(P_1, P_T) \qquad for\ U_1 = 1,\ U_2 = 0 \tag{4.14}$$
$$Q_1 = Q(P_1, P_S) - Q(P_1, P_T) \qquad for\ U_1 = 0,\ U_2 = 1 \tag{4.15}$$

And the same goes for chamber 2 with valve U_3 to the pump and U_4 to the tank.

$$Q_2 = 0 \qquad for\ U_3 = 0,\ U_4 = 0 \tag{4.16}$$
$$Q_2 = -Q(P_S, P_2) \qquad for\ U_3 = 1,\ U_4 = 0 \tag{4.17}$$
$$Q_2 = Q(P_2, P_T) \qquad for\ U_3 = 0,\ U_4 = 1 \tag{4.18}$$
$$Q_2 = Q(P_2, P_T) - Q(P_S, P_2) \qquad for\ U_3 = 1,\ U_4 = 1 \tag{4.19}$$

These equations show how the flows will be separately, but not how to combine them with opening and closing of the other valves. There are 4 valves which each can reach 2 states giving a total of 16 different operation modes. Table 4.2 shows the 16 options. Some of the operation modes will in practice never be desired. If the two valves at the chamber side are matched, i.e. the valve to the tank and the valve from the pump allow the same flow, then turning both valves ON will only give flow to the tank which means lowering the energy efficiency of the system. Therefore operation mode IV, VIII, XII, XIII, XIV, XV and XVI can be considered unsuitable and will be neglected.

Operation Modes				
Operation mode number	**Valve 1**	**Valve 2**	**Valve 3**	**Valve 4**
I	OFF	OFF	OFF	OFF
II	OFF	OFF	OFF	ON
III	OFF	OFF	ON	OFF
IV	OFF	OFF	ON	ON
V	OFF	ON	OFF	OFF
VI	OFF	ON	OFF	ON
VII	OFF	ON	ON	OFF
VIII	OFF	ON	ON	ON
IX	ON	OFF	OFF	OFF
X	ON	OFF	OFF	ON
XI	ON	OFF	ON	OFF
XII	ON	OFF	ON	ON
XIII	ON	ON	OFF	OFF
XIV	ON	ON	OFF	ON
XV	ON	ON	ON	OFF
XVI	ON	ON	ON	ON

Table 4.2. There are 16 different operation modes to choose from when controlling the 4 valves. ON means an open valve, while OFF means a closed valve.

The remaining 9 operation modes can be summed up to 3 different control strategies:

- **Not controlling:** In operation mode I no valves are opened. If the load changes then the system will react as a spring-damper system which will oscillate until a steady state position of the piston is found. This state does not provide any influence of controlling the piston except by changing the load.

- **Controlling a single side:** For operations mode II, III, V and IX, only a single valve is controlled. The operation mode is thus solely dependent on either pumping flow into one chamber to build up pressure, or to remove some by letting flow out. This operation mode will be viewed as an intermediate mode since there is a physical limit to how much the fluid in the chamber not connected to the tank or the pump can be compressed or expanded before it will behave as a vacuum. Furthermore, the fastest response will be found by pumping flow into one chamber and allowing flow to the tank from the other chamber. Thus the benefit of this mode would be to open one valve earlier in order to minimise the impact from the fluid hammer effect as described in section 4.2.

- **Controlling both sides:** Operation mode VI and XI are the classical control strategies given by a proportional valve. There is flow from the pump into one chamber, and the other chamber is open to the tank. These two operation modes are expected to be used at most times due to their faster response compared to only controlling a single valve. Operation mode VI and X where both valves open from the pump or to the cylinder are not a desired mode to be used. If they are used then it would mean that there is a need for either building up pressures in both chambers or to decrease the pressures in the chambers.

Operation mode VI and XI are found to be the best options for controlling the cylinder, but as described in section 4.2, using state II and III a few milliseconds before reaching state VI and XI can reduce some of the fluid hammer effects.

4.4 Summary

Figure 4.7 shows an overview of the system. At this stage all the black boxes have been modelled and the dynamic responses added. As seen by the red signal flow, a pwm signal will be made based on a duty cycle reference, then the effect of the switching times will be added, followed by the fluid hammer effects. This input to the model gives the velocity of the piston. The next chapter will look at the control strategy in order to decide how the different duty cycles should be calculated for the 4 valves.

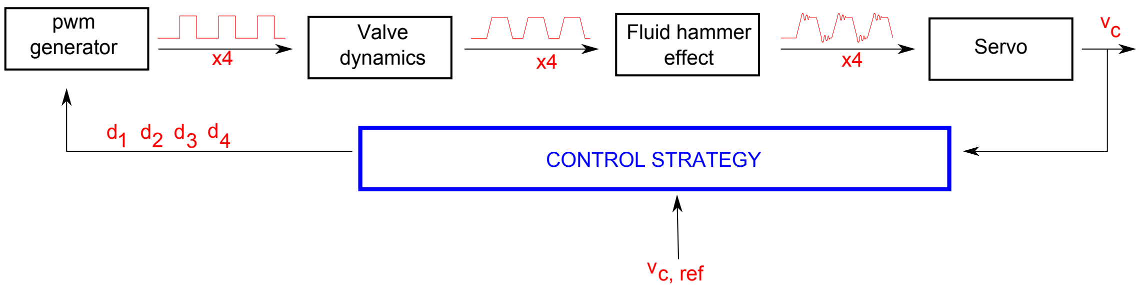


Figure 4.7. The different parts of the system. The black boxes represent the subjects which have been modelled, the blue box represents the control strategy, which will be analysed in the next part, and the red parts show how the signals change between the boxes.

The control strategy should be based on matching valves 2 and 4 and matching valves 1 and 3.

Part II

System Control

Chapter 5

System Behaviour

The main task before defining a control strategy is to analyse how the system behaves, and if whether there are some minimum or maximum demands to the variable parameters such as the dead volumes and the mass. It is important to analyse the dynamic frequency response, since the switching frequency should not be in the area of the system frequency in order to maintain control with the system. The switching frequency for the valves should be faster than the rest of the system since any desired flow under 100% will be reached by opening and closing the valves.

There are 4 valves in the system: 2 to the tank and 1 for each chamber. The simplest control would be to control only one of the valves. This can be done for two different setups: either by controlling the flow into the chamber or the flow out of it. As described earlier, the two cases are known as meter-in control and meter-out control, respectively. The two cases can be seen in figure 5.1.

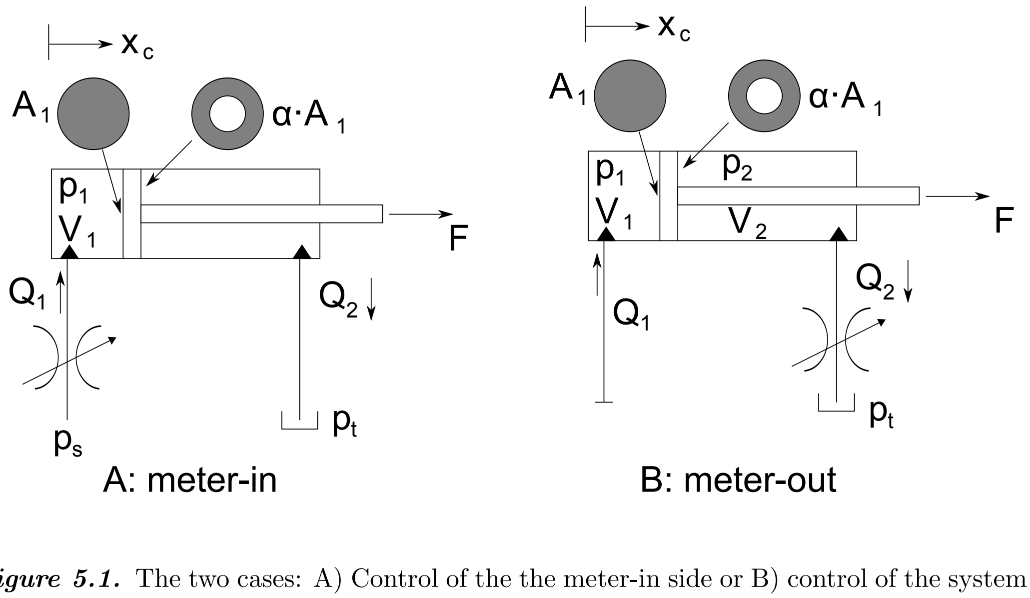


Figure 5.1. The two cases: A) Control of the the meter-in side or B) control of the system using the meter-out side. The orifice controls the flow and the other chamber is connected to the tank.

5.1 Meter-in Control Response

First, the effect of situation A will be analysed. The ON/OFF valve is replaced by an orifice with a variable opening area. This is done in order to analyse the connection between the velocity of the cylinder and flow through the valve. There is open towards the tank for chamber 2, where the flow Q_2 is equal to the volume change. Variable notation is continued from section 3.1.

The flow into chamber 1 can be found as

$$Q_1 = \dot{V}_1 + \frac{V_1}{\beta}\dot{P}_1 \tag{5.1}$$

$\updownarrow$

$$\dot{P}_1 = \frac{\beta}{V_1}\left(Q_1 - \dot{V}_1\right) \tag{5.2}$$

The flow out of chamber 2 can be found as:

$$-Q_2 = \dot{V}_2 + \frac{V_2}{\beta}\dot{P}_2 \tag{5.3}$$

(5.4)

The valve is open for flow into the tank, and it is assumed that it will change with the movement of the piston, thus:

$$-A_2 \cdot v = -A_2 \cdot v + \frac{V_2}{\beta}\dot{P}_2 \tag{5.5}$$

$\downarrow$

$$\dot{P}_2 = \frac{\beta}{V_2}\left(A_2 \cdot v - A_2 \cdot v\right) = 0 \tag{5.6}$$

The flow through the orifice at Q_1 can be modelled as

$$Q_1 = C_d \cdot A_{v1}(x_{v1}) \cdot \sqrt{\frac{2}{\rho}\left(p_s - p_1\right)} \tag{5.7}$$

where C_d is the discharge coefficient, $A_{v1}(x_{v1})$ is the opening area as a function of the spool position, ρ is the fluid density. The discharge coefficient is assumed to be 0.6 since the flow is assumed to be turbulent around a sharp edge spool orifice. [Andersen and Hansen, 2003,p.14]

The opening area can be modelled as

$$A_{v1}(x_{v1}) = A_{1,0} \cdot x_{v1} \tag{5.8}$$

where $A_{1,0}$ is the maximum opening and x_{v1} is the opening in percent. The area is estimated based on an expected flow of 40 L/min at a density of 870 kg/m^3 and a pressure difference of 10 bar. Thus $A_{1,0}$ is $0.000023m^3$

Linearising the flow through the orifice can be done with a 1^{st} order Taylor expansion as seen below. The difference in the flow is found to eliminate $Q_1(x_{v1,0}, p_{1,0})$.

$$Q_1 = Q_1(x_{v1,0}, p_{1,0})\frac{\delta Q_1}{\delta x_{v1}}|_{x_{v1,0},p_{1,0}}\Delta x_{v1} + \frac{\delta Q_1}{\delta p_1}|_{x_{v1,0},p_{1,0}}\Delta p_1 \tag{5.9}$$

$\downarrow$

$$\Delta Q_1 = \frac{\delta Q_1}{\delta x_{v1}}|_{x_{v1,0},p_{1,0}}\Delta x_{v1} + \frac{\delta Q_1}{\delta p_1}|_{x_{v1,0},p_{1,0}}\Delta p_1 \tag{5.10}$$

The two linearisation points can now be found as:

$$K_{q1} = \frac{\delta Q_1}{\delta x_{v1}}|_{x_{v1,0},p_{1,0}} = C_d \cdot \pi \cdot d_1 \cdot \sqrt{\frac{2}{\rho}\left(p_s - p_{1,0}\right)} \tag{5.11}$$

$$K_{qp1} = \frac{\delta Q_1}{\delta p_1}|_{x_{v1,0},p_{1,0}} = \frac{-\sqrt{2} \cdot C_d \cdot \pi \cdot d_1 \cdot x_{v1,0}}{2 \cdot \rho\sqrt{\frac{p_s - p_{1,0}}{\rho}}} \tag{5.12}$$

K_{qp1} is negative. The sign of the equation are changed in order to keep the coefficients positive, therefore

$$\Delta Q_1 = \frac{\delta Q_1}{\delta x_{v1}}|_{x_{v1,0},p_{1,0}}\Delta x_{v1} - \frac{\delta Q_1}{\delta p_1}|_{x_{v1,0},p_{1,0}}\Delta p_1 \tag{5.13}$$

Depending on the chosen linearisation points then the parameters K_{qp1} and K_{q1} will change. The linearisation point for p_1 depends on the rest of the system i.e. position of the cylinder, p_2 and the external force applied. Figure 5.2 shows the two basic cases: with and without a gravitational load. There exists more options than shown since the cylinder can be placed at different angles, but the minimum and maximum case are shown.

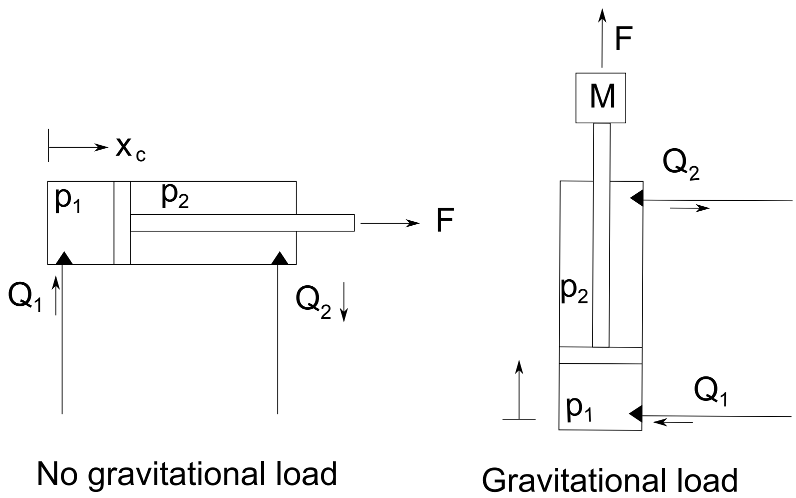


Figure 5.2. The two basic cases: no gravitational load - only contributions are from the two pressures and the external force, and a gravitational load in addition to the pressures and the external load.

If a static steady state position should be reached i.e. the piston does not move, and the pressure from the pump cannot produce more than 150 bar then the external force can be calculated as a function of the two pressures

$$\begin{aligned} 0 &= A_1 \cdot p_1 - A_2 \cdot p_2 + F(-M \cdot g) \\ F &= A_2 \cdot p_2 - A_1 \cdot p_1(+M \cdot g) \end{aligned} \quad \downarrow \tag{5.14}$$

Figure 5.3 shows how the different pressures change the need for external force in order to keep a static position. The cylinder is assymmetrical, so if the external load is low then the two pressures should be matched accordingly, which is why the intervals are shown from the pump pressure of 150 bar down to 50 bar. It is seen from the left figure that if no external loads are applied then $p_1 = 54bar$ and $p_2 = 150bar$. It is fair to say, that the system has a hard time holding a static position if the force is pulling the piston outwards since the required force is negative for nearly all values of p_1 and p_2. If a gravitational load is added then the static position can be reach for both pushing and pulling external forces.

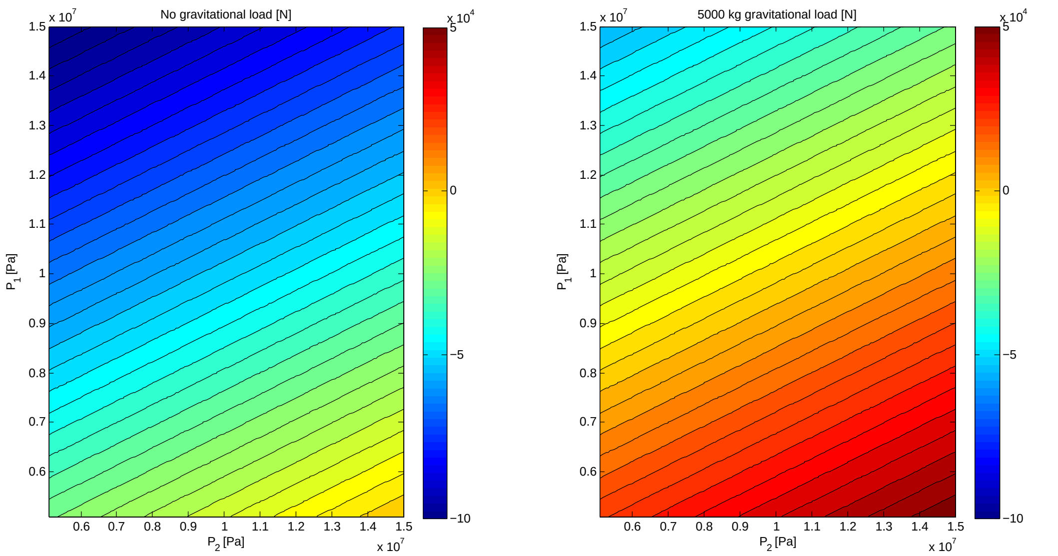


Figure 5.3. The external force [N] as a function of the pressures p_1 and p_2 for two different cases: no gravitational load and 5000 kg gravitational load.

It is assumed that by choosing a relatively large cylinder with a matching pump pressure that the cylinder should be used near its maximum operating conditions. An operating point of 140 bar for both chamber 1 and 2 are chosen in order to ensure that there is an operating window for pressure build up.

This operating point influences the linearisation constants K_{qp1} and K_{q1} .

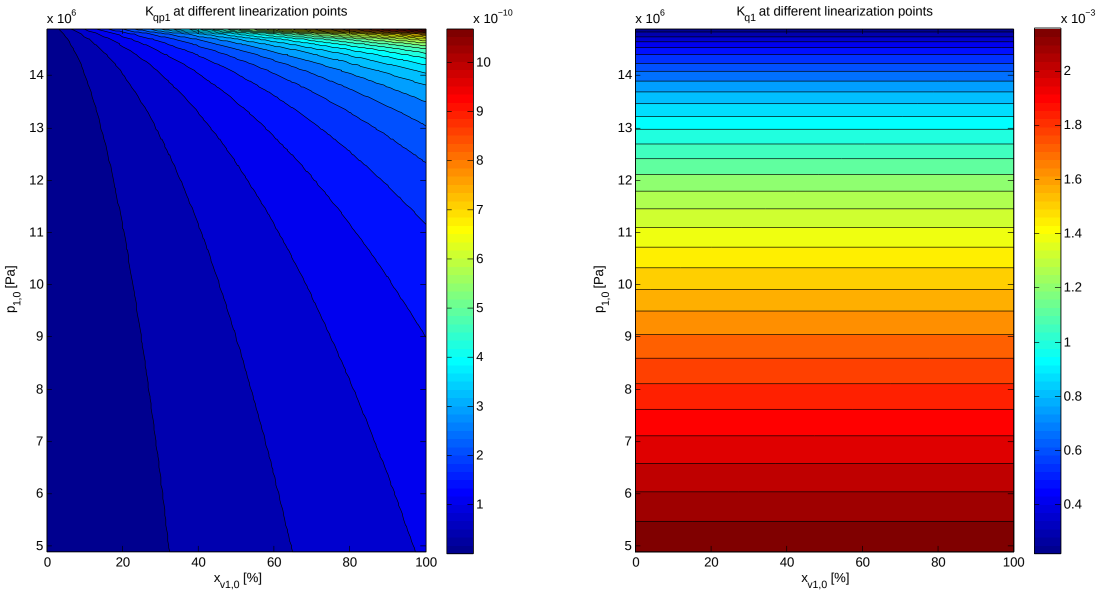


Figure 5.4. The linearisation coefficients for the valve to chamber 1.

With $p_1 = 140bar$ then $K_{q1} = 6.9879e-04$. At this point K_{qp1} varies linearly between 0 and 3.46e-10. A linearisation point of 1.0 are chosen since the valve will operate as open or closed. The consequences of this choice will be investigated later.

To summarize, the flow Q_1 is equal to

$$\Delta Q_1 = K_{q1} \cdot \Delta x_{v1} - K_{qp1} \cdot \Delta p_1 \tag{5.15}$$

Substituting this into equation 5.2 and finding the difference in the pressures and velocity gives

$$\Delta \dot{p_1} = \frac{\beta}{V_1} \left(K_{q1} \cdot \Delta x_{v1} - K_{qp1} \cdot \Delta p_1 - A_1 \cdot \Delta v \right) \tag{5.16}$$

The equation can then be Laplace transformed in order to find an expression for Δp_1

$$\Delta p_1 \cdot s = \frac{\beta}{V_1} \left(K_{q1} \cdot \Delta x_{v1} - K_{qp1} \cdot \Delta p_1 - A_1 \cdot \Delta v \right) \tag{5.17}$$

$\Updownarrow$

$$\Delta p_1 \cdot \left(s + \frac{\beta \cdot K_{qp1}}{V_1} \right) = \frac{\beta \cdot \left(K_{q1} \cdot \Delta x_{v1} - A_1 \cdot v \right)}{V_1} \tag{5.18}$$

$\Updownarrow$

$$\Delta p_1 = \frac{\beta \cdot K_q \cdot \Delta x_{v1} - A_1 \cdot \beta \cdot \Delta v}{V_1 \cdot s + \beta \cdot K_{qp1}} \tag{5.19}$$

The force balance for the system seen in figure 5.1 can be found as seen in equation (5.20). The equation can then be Laplace transformed and changed to find the difference in variable parameters. It is assumed that the external force F is constant, thus when finding the difference in the parameters then F disappears from the equation.

$$M \cdot \dot{v} = A_1 \cdot p_1 - B \cdot v + F \qquad \downarrow$$

$$M \cdot s \cdot \Delta v = A_1 \left(\frac{\beta \cdot K_{q1} \cdot \Delta x_{v1} - A_1 \cdot \beta \cdot \Delta v}{V_1 \cdot s + \beta \cdot K_{qp1}} \right) - B \cdot \Delta v \tag{5.20}$$

A transfer function for this system can now be found as:

$$\Delta v \cdot \left(s \cdot M + B + \frac{A_1^2 \cdot \beta}{V_1 \cdot s + \beta \cdot K_{qp1}} \right) = \Delta x_{v1} \cdot \frac{A_1 \cdot \beta \cdot K_{q1}}{V_1 \cdot s + \beta \cdot K_{qp1}} \tag{5.21}$$

$$\frac{\Delta v}{\Delta x_{v1}} = \frac{A_1 \cdot \beta \cdot K_{q1}}{V_1 \cdot M \cdot s^2 + s \cdot (B \cdot V_1 + \beta \cdot M \cdot k_{qp1}) + \beta \cdot \left(A_1^2 + B \cdot k_{qp1}\right)} \tag{5.22}$$

$$\frac{\Delta v}{\Delta x_{v1}} = \frac{\frac{A_1 \cdot \beta \cdot K_{q1}}{V_1 \cdot M}}{s^2 + s \cdot \frac{B \cdot V_1 + \beta \cdot M \cdot k_{qp1}}{V_1 \cdot M} + \frac{\beta \cdot \left(A_1^2 + B \cdot k_{qp1}\right)}{V_1 \cdot M}} \tag{5.23}$$

Then the eigenfrequency of the system can be found as

$$\omega_n = \sqrt{\frac{\beta \cdot \left(A_1^2 + B \cdot k_{qp1}\right)}{V_1 \cdot M}} \tag{5.24}$$

and the damping

$$\zeta = \frac{\frac{B \cdot V_1 + \beta \cdot M \cdot k_{qp1}}{V_1 \cdot M}}{2 \cdot \sqrt{\frac{\beta \cdot \left(A_1^2 + B \cdot k_{qp1}\right)}{V_1 \cdot M}}} \tag{5.25}$$

There are three variables here, which have not been finally determined. The viscous friction coefficient B, the volume for chamber 1 V_1 and the system mass M. From the cylinder analysis in section 3.1.2, it is known that the cylinder has the lowest frequency if the piston is placed at $\psi = 0.625$, equivalent to x=0.625 m, and increasing the mass will converge towards zero logarithmically. Since this thesis is not based on an existing system, then the influence of choosing the cylinder should be investigated. Three different coefficients are tested: one for a high performance cylinder [Bosch Rexroth AG, 2013], and two standard cylinders with different friction [Hvoldal and Olesen, 2011] and [Bosch Rexroth AG, 2013], giving B = 80 Ns/m, 530 Ns/m and 4000 Ns/m, respectively. With a volume of 6.9L (position at x=0.625 and contribution from surrounding pipe as modelled in fluidhammer) and a mass of 5000 kg, the difference in dynamic responses can be seen in figure 5.5.

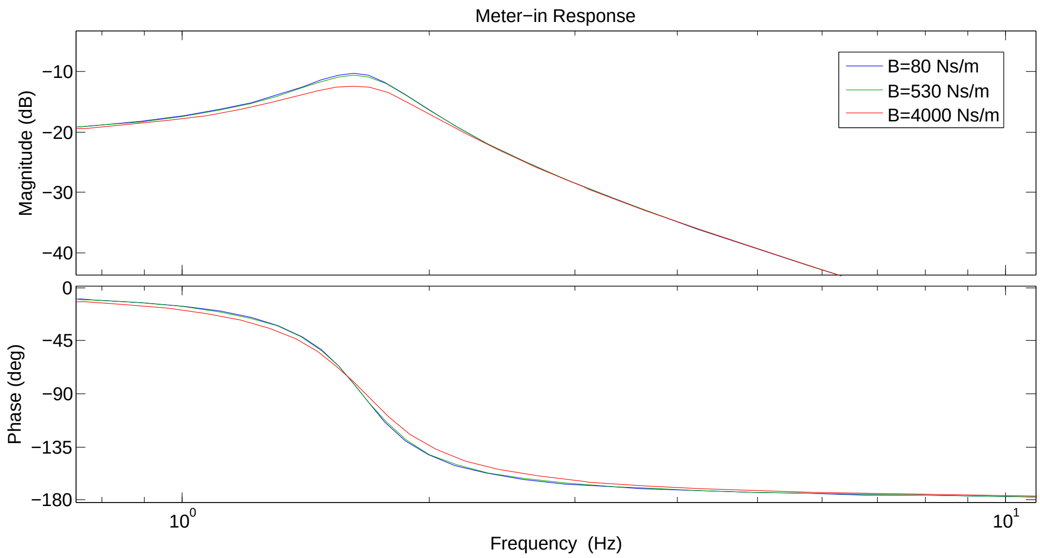


Figure 5.5. System bode plot with varying viscous friction coefficient. A higher viscous friction coefficient dampens the system more.

It is seen from the figure bode plots that the different cylinders do not affect the dynamic response significantly. The standard cylinder with a higher viscous friction coefficient reduces some of the resonance peak compared to the high-performance cylinder. A pressure feedback should be implemented to remove the resonance peak, but considering the scope of this project, it is noted that it would be a waste of money to buy an expensive high-performance cylinder. It would only add more complications to the control. With 4000 Ns/m, the friction loss is a little under 1% of the total energy used to move the cylinder, which seems reasonable. Thus the work is continued with a viscous friction coefficient of 4000 Ns/m.

Having modelled the cylinder, the influence of the volumes and mass will then be investigated.

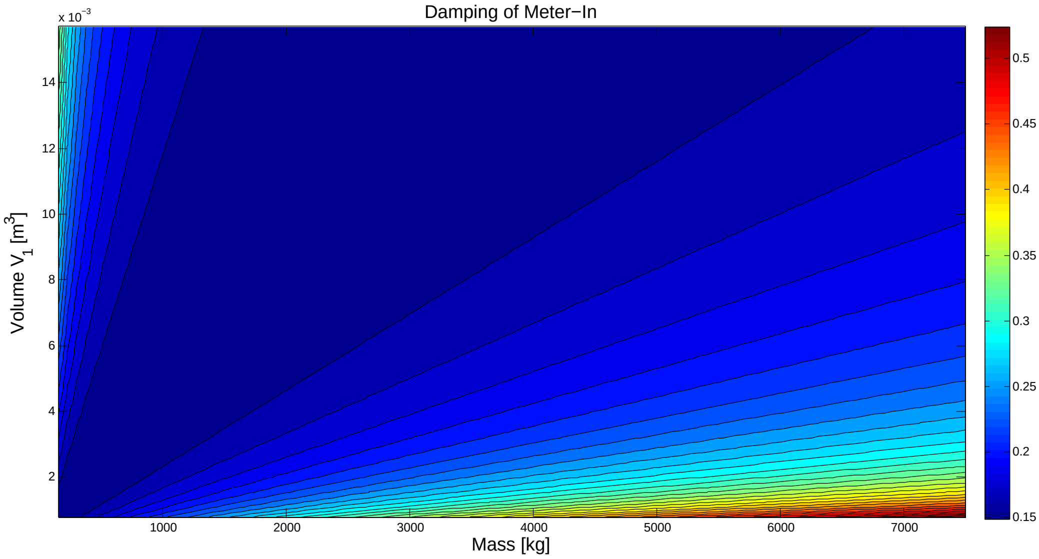


Figure 5.6. The damping ratio as a function of the mass and volume.

Figure 5.7 and 5.6 show how the frequency and damping of the system will be at different volumes and masses. The damping ratio shows that the system will be underdamped at all combinations of the mass and volume. Normally, an underdamped damping ratio of 0.707 is desired [Franklin et al., 2010] but as seen from 5.6 then this cannot be achieved at any point. It is noted that the mass should either be low with a large volume or the system should have a large mass and a smaller volume in order to get a better performance. But the system specifications cannot be based solely on the damping raio.

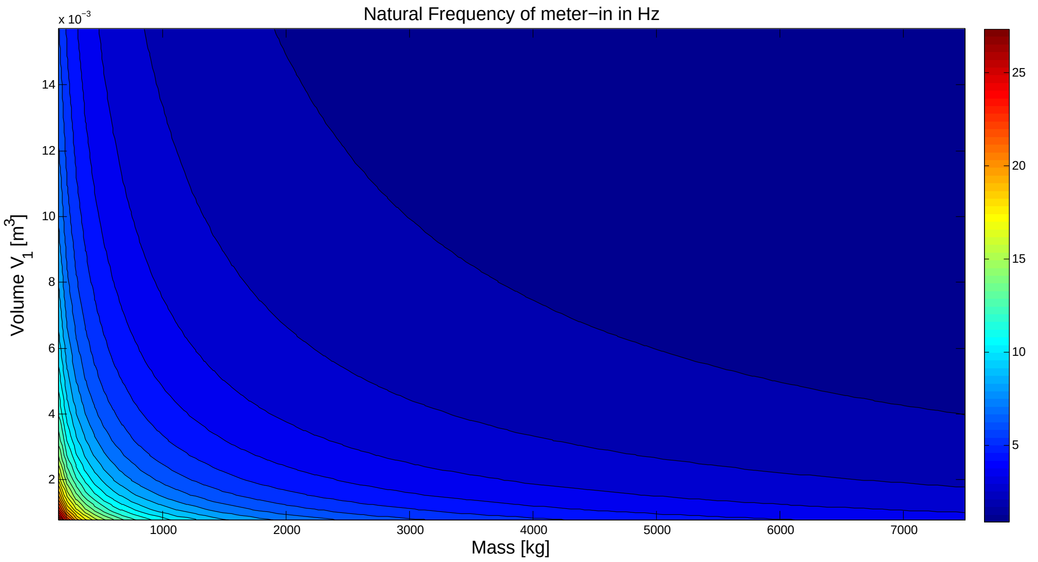


Figure 5.7. The system frequency as a function of volume and mass.

In order to increase the controllability of the system, the switching frequency of the valves should be higher than the system frequency. The chosen valves have a maximum switching frequency of 10 Hz but in order to switch within 10 to 90% duty cycle then a switching frequency of maximum 1.5 Hz was found reasonable. Figure 5.8 shows the area where this condition is met. It is noted that this area is restricted to the 1.5 Hz limit with any value above being marked by dark red. Note that the remaining area is still close to 1.5 Hz which set high requirements to the valves and control.

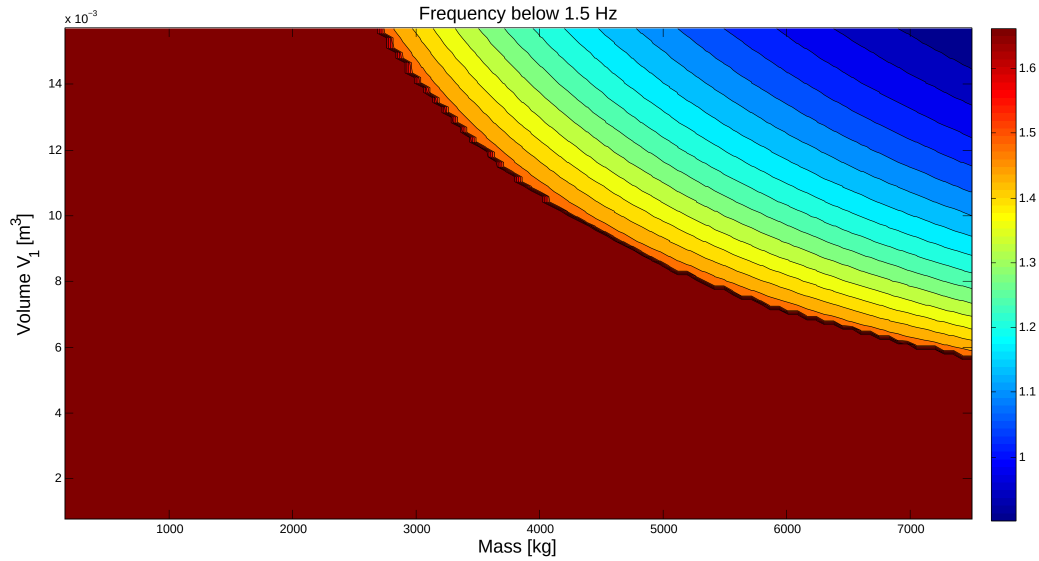


Figure 5.8. The working space if the system should be slower than the valves. The dark red area has a frequency faster than 1.5Hz, which should be avoided.

If the piston should be able to be in all positions from contracted to extended without leaving this zone, then it would need a dead volume of 7.9L and a mass of 5421kg. This dead volume is 0.1L larger than a filled cylinder which is possible, but inefficient. Since the cylinder and pump can handle more than that, 7500 kg is chosen as dead weight. This reduces the dead volume requirement to 6.1L, however a large mass also leaves little room for additional, external forces. Furthermore, the cylinder can not handle such a large mass if it is placed below the cylinder thereby exerting a gravitational pull. The damping ratio for the two points are 0.19 and 0.22 which are still so low that it is a concern whether the system can be controlled.

As seen from the previous figures, the volume will change the damping ratio and natural frequency, which will influence the response depending on the position of the cylinder. Figure 5.9 shows how much the dynamic response will change by the position of the piston. The mass is 7500 kg, the viscous friction coefficient is 4000 and there is a dead volume of 6.1L.

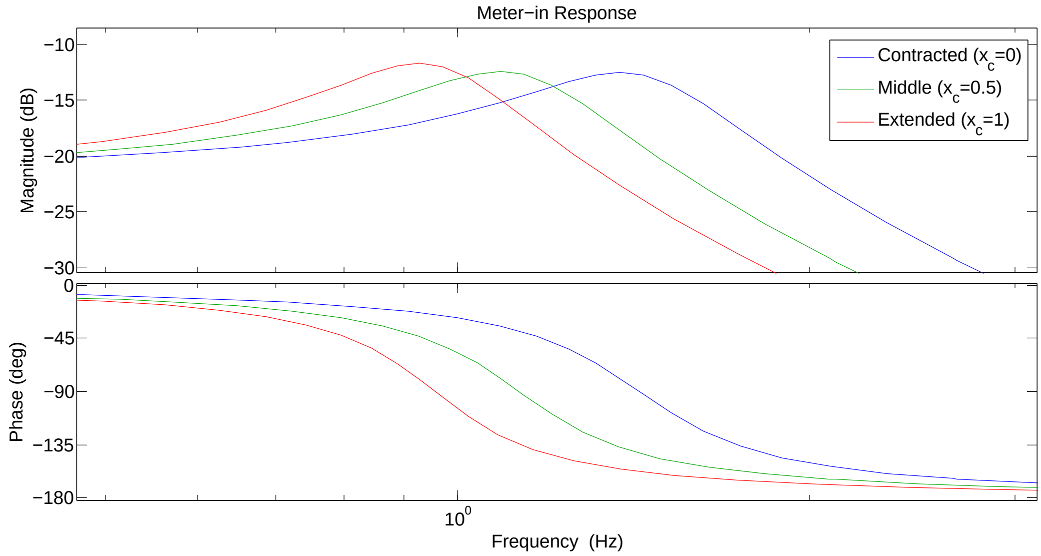


Figure 5.9. The dynamic response at three different positions of the piston. The mass is 7500 kg, the viscous friction coefficient is 4000 and the dead volume is 6.1L.

There is a significant difference in the dynamic response depending on the position of the piston, which should be taken into consideration when the control strategy is made. This is illustrated by the bode plots in figure 5.9.

5.2 Meter-out Control Response

If the analysis is redone with the case where the meter-out edge is analysed, as seen in figure 5.1, then a demand to the dead volume of chamber 2 can be found. The cylinder is still moving in the same direction, but instead of pumping flow into chamber 1, then it is drained from chamber 2 thus applying a meter-out control strategy. Chamber 1 is open towards the tank so Q_1 will follow the volume change, thus

$$Q_1 = \dot{V}_1 + \frac{V_1}{\beta}\dot{p}_1 \tag{5.26}$$

$$\updownarrow$$

$$\dot{p}_1 = (v \cdot A_1 - v \cdot A_1) \cdot \frac{\beta}{V_1} = 0 \tag{5.27}$$

The flow out of chamber 2 can be modelled as

$$-Q_2 = \dot{V}_2 + \frac{V_2}{\beta}\dot{p}_2 \tag{5.28}$$

$$-Q_2 = -v \cdot A_2 + \frac{V_2}{\beta}\dot{p}_2 \tag{5.29}$$

$$\dot{p}_2 = \frac{\beta}{V_2}\left(-Q_2 + v \cdot A_2\right) \tag{5.30}$$

Again, the valve is assumed to be an orifice, thus the flow out of the valve can be defined as:

$$-Q_2 = -C_d \cdot A_{v2}(x_{v2}) \cdot \sqrt{\frac{2}{\rho}(p_2 - p_t)} \tag{5.31}$$

$$A_{v2}(x_{v2}) = A_{v2,0} \cdot x_{v2} \tag{5.32}$$

where $A_{v2,0}$ is the maximum opening and x_{v2} is the opening area in percent. At 25 l/min and a pressure drop of 150 bar then $A_{v2,0} = 0.000004m^3$.

With the same procedure as before the flow can be linearised giving:

$$-\Delta Q_2 = -\frac{\delta Q_2}{\delta x_{v2}}|_{x_{v2,0},p_{2,0}} \Delta x_{v2} - \frac{\delta Q_2}{\delta p_2}|_{x_{v2,0},p_{2,0}} \Delta p_2 \tag{5.33}$$

the two linearising constants are found as:

$$K_{q2} = \frac{\delta Q_2}{\delta x_{v2}}|_{x_{v2,0},p_{2,0}} = C_d \cdot \pi \cdot d_2 \cdot \sqrt{\frac{2}{\rho}(p_{2,0} - p_t)} \tag{5.34}$$

$$K_{qp2} = \frac{\delta Q_2}{\delta p_2}|_{x_{v2,0},p_{2,0}} = \frac{\sqrt{2} \cdot C_d \cdot \pi \cdot d_2 \cdot x_{v2,0}}{2 \cdot \rho \sqrt{\frac{p_{2,0} - p_t}{\rho}}} \tag{5.35}$$

Again, the linearisation can be seen in figure 5.10. The linearisation point of 140 bar was chosen earlier thus $K_{q2} = 0.002604$ and $K_{qp2} = 1.624e - 11$

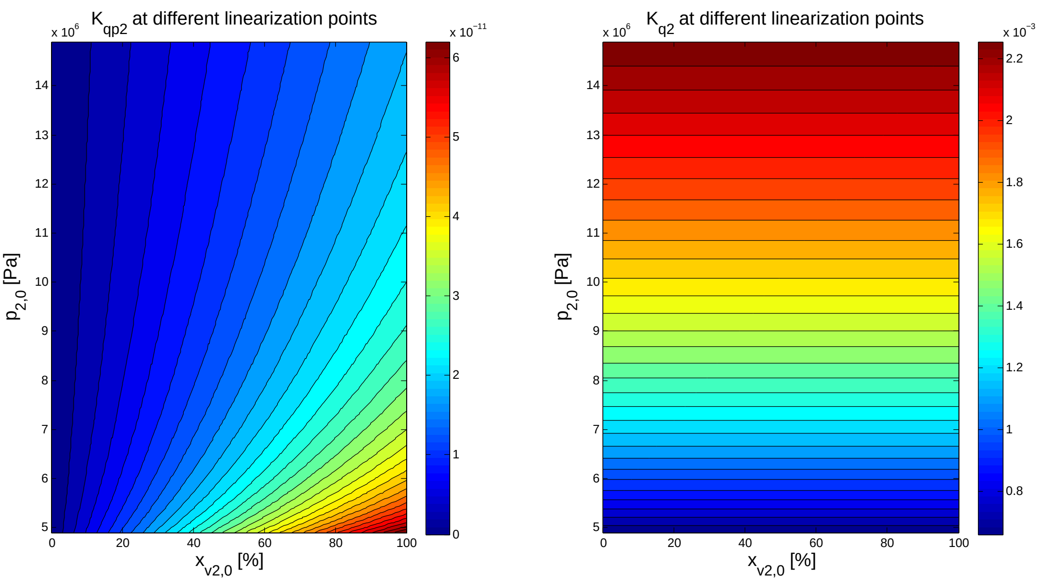


Figure 5.10. The linearisation coefficients for the valve to chamber 2.

$$\Delta p_2 = \frac{\Delta v \cdot A_2 \cdot \beta - \Delta x_{v2} \cdot K_{q2} \cdot \beta}{s \cdot V_2 + K_{qp2}\beta} \tag{5.36}$$

$$M \cdot s \cdot v = -A_2 \cdot p_2 - B \cdot v + F \tag{5.37}$$

$\downarrow$

$$M \cdot s \cdot \Delta v = -A_2 \cdot \left(\frac{\Delta v \cdot A_2 \cdot \beta - \Delta x_{v2} \cdot K_{q2} \cdot \beta}{V_2 \cdot s + K_{qp2} \cdot \beta} \right) - B \cdot \Delta v \tag{5.38}$$

$$\frac{\Delta v}{\Delta x_{v2}} = \frac{A_2 \cdot \beta \cdot K_{q2}}{M \cdot V_2 \cdot s^2 + s \cdot (B \cdot V_2 + \beta \cdot M \cdot K_{kp2}) + \beta \cdot (A_2^2 + B \cdot K_{qp2})} \tag{5.39}$$

Then the natural frequency of the system can be found as

$$\omega_n = \sqrt{\frac{\beta \cdot \left(A_2^2 + B \cdot k_{qp2}\right)}{V_2 \cdot M}} \tag{5.40}$$

and the damping

$$\zeta = \frac{\frac{B \cdot V_2 + \beta \cdot M \cdot k_{qp2}}{V_2 \cdot M}}{2 \cdot \sqrt{\frac{\beta \cdot \left(A_2^2 + B \cdot k_{qp2}\right)}{V_2 \cdot M}}} \tag{5.41}$$

The viscous friction coefficient was chosen in the previous section to B=4000 Ns/m. Thus the natural frequency and damping ratio can be investigated, as illustrated in figures 5.11 and 5.12.

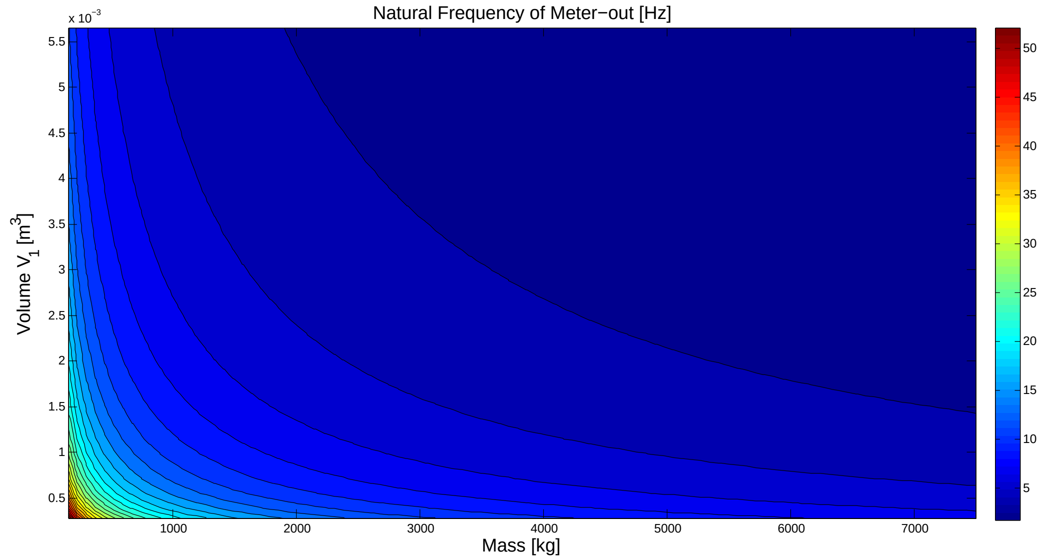


Figure 5.11. The natural system frequency at meter-out. The desired frequency is max. 1.5 Hz, which is not found in the figure.

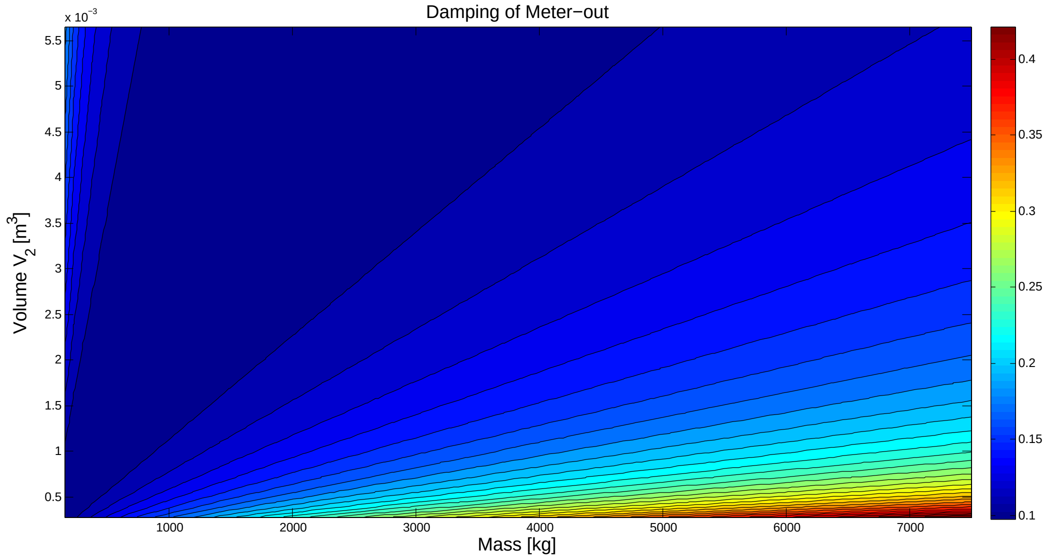


Figure 5.12. The damping ratio at meter-out control.

As seen from figures 5.11 and 5.12 the damping ratio still needs to be improved for meter-out, and not at any point is it possible to get the system to be slower than 1.5 Hz. The demands need to be reviewed or the valves should be changed if meter-out control should be possible. Figure 5.13 shows the bode plot at three different volumes. A weight of 7500 kg is applied and the dead volume is assumed to be 1 L for chamber 2.

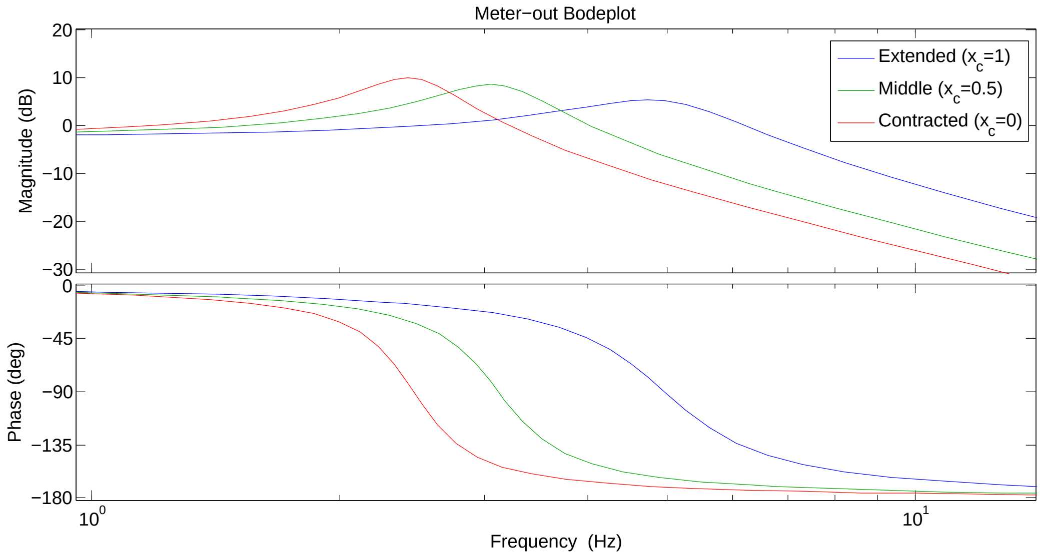


Figure 5.13. The difference in the dynamic response to different volumes.

Chapter 6

Control Strategy for a Velocity Servo System

The previous chapters have analysed and modelled the behaviour of the valves and the hydraulic servo system. This chapter will deal with controlling the two servo systems. First, a few comments on the system response from the previous chapter, then design of a controller for the proportional system, followed by a control strategy for the ON/OFF servo system.

The main problem with controlling a hydraulic servo system is the combination of large parameter ranges, nonlinearities in the flow characteristics and a low natural damping [Andersen et al., 2005]. The hydraulic servo system in this report has these problems: the natural low damping can be seen in figures 5.6 and 5.12, the nonlinearities in figure 3.3 and 3.9 and the parameter ranges that are investigated in chapters 3 and 5.

It has been shown that linear controllers will have difficulties in achieving a satisfactory performance in the entire workspace due to the linear approximations and the limitation of the linearisation points [Andersen et al., 2005]. Nonlinear control theory can be applied instead in order to achieve satisfactory performance in the entire workspace. The nonlinear control theory gives different choices when it comes to designing a nonlinear controller. Slotine and Li [1991] covers feedback linearisation, adaptive control and sliding mode control. The sliding mode control is chosen as the nonlinear control structure for this report due to its strength within maintaining stability and consistent performance [Slotine and Li, 1991,p.277].

6.1 Controller for the Proportional Valve

This section will design a controller for the proportional valve. The different challenges described in chapter 4 does not apply to the proportional valve. The system is SISO, no fluid hammer effect or switching time. The only problem is to design a controller which can handle the nonlinearities and track the reference successfully.

6.1.1 Sliding Mode Control

Variable notation in this section continues from those defined in chapter 5. In order to design a sliding mode controller, the servo system is put into a standard form: [Slotine and Li, 1991]

$$x^{(n)} = f(\mathbf{x}) + b(\mathbf{x}) \cdot u \tag{6.1}$$

where u is the control input, $\mathbf{x}$ is the state vector.

The control signal u is identified to be the signal to the valve as seen in figure 3.13. The flows Q_1 and Q_2 are implemented by a table look up but can be redesigned to the following form where u should be multiplied:

$$Q_1 = \begin{cases} u \cdot Q(P_s, P_1) \text{ for u} > 0 \\ u \cdot Q(P_1, P_t) \text{ for u} < 0 \end{cases} \tag{6.2}$$

$$Q_2 = \begin{cases} u \cdot Q(P_2, P_t) \text{ for u} > 0 \\ u \cdot Q(P_S, P_2) \text{ for u} < 0 \end{cases} \tag{6.3}$$

This is abbreviated to $Q(p_s, p_1, p_t)$ and $Q(p_s, p_2, p_t)$, where a modified look up table can be combined with sign(u) in order to determine the flow.

Then the rest of the equations for the cylinder are put on the standard form

$$\dot{p_1} = \frac{\beta}{V_1} \left(u \cdot Q(p_s, p_1, p_t) - \dot{x} \cdot A_1 \right) \tag{6.4}$$

$$\dot{p_2} = \frac{\beta}{V_2} \left(-u \cdot Q(p_s, p_2, p_t) + \dot{x} \cdot A_2 \right) \tag{6.5}$$

$$M \cdot \ddot{x}_c = A_1 \cdot p_1 - A_2 \cdot p_2 - B \cdot \dot{x}_c + F \tag{6.6}$$

In order to combine equation 6.4 and 6.5 with 6.6 then equation 6.6 should be differentiated.

$$M \cdot \dddot{x}_c + \dot{M} \cdot \ddot{x}_c = \dot{A}_1 \cdot p_1 + A_1 \cdot \dot{p_1} - \dot{A}_2 \cdot p_2 - A_2 \cdot \dot{p_2} - \dot{B} \cdot \dot{x}_c - B \cdot \ddot{x}_c + \dot{F} \tag{6.7}$$

The mass, areas, viscous friction coefficient and external force are assumed constant, and as a result their time derivative will be 0. Equation 6.4 and 6.5 can be substituted into equation 6.7.

(6.8)

$$M \cdot \dddot{x}_c = A_1 \cdot \frac{\beta}{V_1} \left(u \cdot Q(p_s, p_1, p_t) - \dot{x}_c \cdot A_1 \right) - A_2 \cdot \frac{\beta}{V_2} \left(-u \cdot Q(p_s, p_2, p_t) + \dot{x}_c \cdot A_2 \right) - B \cdot \ddot{x}_c$$

$$\updownarrow$$

$$\ddot{x}_c = \frac{-M \cdot \dddot{x}_c - \dot{x}_c \cdot \left(\frac{\beta \cdot A_1^2}{V_1} + \frac{\beta \cdot A_2^2}{V_2} \right)}{B} + u \cdot \frac{Q(p_s, p_1, p_t) \cdot \frac{\beta \cdot A_1}{V_1} + Q(p_s, p_2, p_t) \cdot \frac{\beta \cdot A_2}{V_2}}{B} \tag{6.9}$$

By comparing 6.8 and eq:0 then $f(\mathbf{x})$ and $b(\mathbf{x})$ can be identified as:

$$f(\mathbf{x}) = \frac{-M \cdot \dddot{x} - \dot{x} \cdot \left(\frac{\beta \cdot A_1^2}{V_1} + \frac{\beta \cdot A_2^2}{V_2} \right)}{B} \tag{6.10}$$

$$b(\mathbf{x}) = \frac{Q(p_s, p_1, p_t) \cdot \frac{\beta \cdot A_1}{V_1} + Q(p_s, p_2, p_t) \cdot \frac{\beta \cdot A_2}{V_2}}{B} \tag{6.11}$$

The objective of the velocity servo is to track a velocity, thus the switching surface becomes:

$$\sigma \left(\dot{x}_c - \dot{x}_{c,ref} \right) = \sigma(e) = e = 0 \tag{6.12}$$

where e is the tracking error. In order to determine a stable control law, that keeps the system on the switching surface then Lyapunov theory will be used. A positive definite candidate for the Lyapunov function is proposed:

$$V(e) = \frac{\sigma^2(e)}{2} \tag{6.13}$$

Next, a control law should be chosen that makes the the time derivative of the Lyapunov candidate negative definite:

$$\dot{V(e)} = \sigma(e) \cdot \sigma(\dot{e}) \tag{6.14}$$
$$= e \cdot \dot{e} \tag{6.15}$$
$$= e \cdot (\ddot{x_c} - \ddot{x}_{c,ref}) \tag{6.16}$$
$$= e \cdot (f(\mathbf{x}) + b(\mathbf{x}) \cdot u - \ddot{x}_{c,ref}) \tag{6.17}$$

If equation 6.17 should be negative definite then the control law is chosen to be:

$$u = \frac{1}{b(\mathbf{x})} (\ddot{x}_{c,ref} - f(\mathbf{x}) - sign(e)) \tag{6.18}$$

Then

$$\dot{V(e)} = e(-sign(e)) \tag{6.19}$$
$$= -\sigma(e) \cdot sign(\sigma(e)) \tag{6.20}$$

$\dot{V(e)}$ is negative definite, $V(e)$ is positive definite and radially unbounded ($V(e) \to \infty$ as $e \to \infty$), thus the control law is asymptotically stable, which means the error will tend to zero as given by Theorem 3.3 Global Stability in Slotine and Li [1991,p.65].

The final control law is as seen below

$$u = \frac{B \cdot (\ddot{x}_{c,ref} - sign(e)) + M \cdot \dddot{x_c} + \dot{x}_c \cdot \left(\frac{\beta \cdot A_1^2}{V_1} + \frac{\beta \cdot A_2^2}{V_2}\right)}{Q(p_s, p_1, p_t) \cdot \frac{\beta \cdot A_1}{V_1} + Q(p_s, p_2, p_t) \cdot \frac{\beta \cdot A_2}{V_2}} \tag{6.21}$$

This control law contains the risk of division by 0. It is small but existing, which is why a restriction is made:

- The possible flow should not be zero for both chambers i.e. the two following scenarios are not allowed: $p_2 = p_t$ when $p_1 = p_s$ or $p_1 = p_t$ when $p_2 = p_s$.

The physical meaning of reaching the limitation is pressures that equalize at the supply and tank pressure, respectively, thus the piston has hit the end wall without a adjustment of the valve. The control law states it cannot control it at this point, which makes sense.

Finally, a small gain is tuned to the signal u in order to scale it for the valve input of -1.5 to 1.5A. 0.133 is found to be sufficient.

The design of a control strategy for the proportional valve is completed. The next chapter will compare its performance to the one found with the ON/OFF valves.

6.2 Control Strategy for ON/OFF Servo

The ON/OFF control sets higher demands to the control strategy due to the identified challenges and the bode plots shown in figures 5.9 and 5.13.

The bode plots from chapter 5 show the undamped natural frequency of the servo system is in the vicinity of the found switching frequency for the valves. In figure 5.8 it was possible to identify an area where the valves were a bit faster than the servo system for the meter-in case. This was not possible for the meter-out case. The identified working space is, however, still in the vicinity of the switching frequency so the switching may still cause problems.

Since the valves only have two states to be in, every switching would be equal to a step input. Figure 6.1 shows how the switching frequency will influence the response from the system. The figure is made by giving a pwm signal of 50% with a frequency of 1.5 Hz, 3 Hz and 15 Hz respectively to the transfer function for the meter-in case at $x_c = 0.5$, M=7500kg and a dead volume of 6.1L.

It is clear that switching at system frequency is not an option. If a design with 8 valves were made as suggested at 3.6, then by switching between the two valves in each set, then the system could behave something like a 3 Hz system. However, this is still not fast enough as seen from fig 6.1.

From data sampling theory it is known that the sampling frequency should be 10 times as fast as the system frequency in order to get accurate measurements. Figure 6.1 supports that around 10 times as fast would be a satisfactory rule of thumb for controlling the system. Switching at the system frequency or slower will give an oscillating response and any control implemented for that frequency is not expected to work well.

There are two obvious solutions to this: either change the servo system or change the

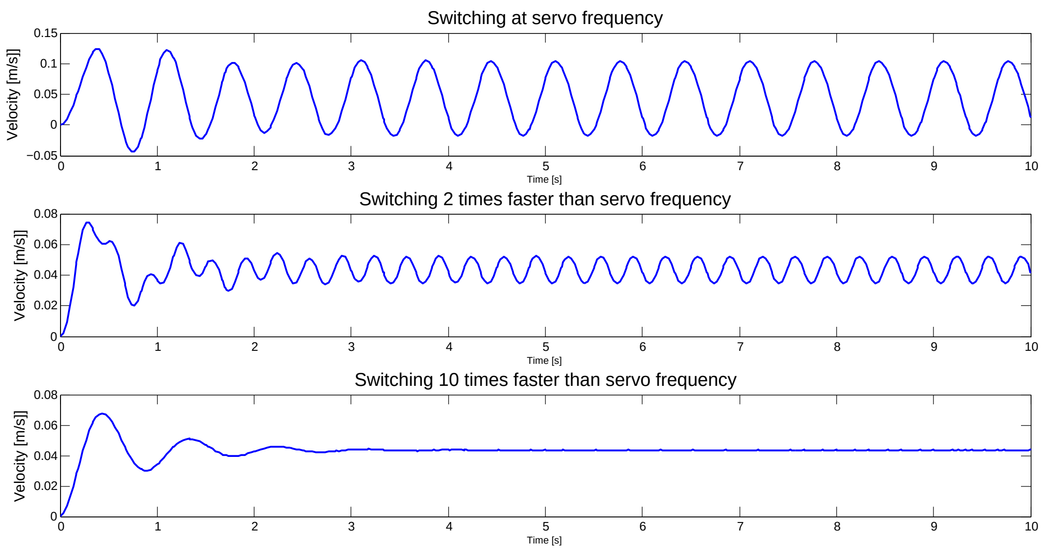


Figure 6.1. The effect of switching at system frequency 1.5 Hz, 2 times faster 3 Hz and 10 times faster than the servo system frequency 15 Hz. It is clear that it is not desirable to switch at system frequency or twice as fast.

valves. There has already been made a parameter sensitivity study, where the dead volume was allowed to be equal to the volume which could be in the cylinder. The mass was varied up to a limit where the cylinder would still be able to move it. Allowing these parameters to be even bigger is not plausible if the system should be build and used. The used areas, fluid properties and viscous damping coefficients are also within reason, so instead the demands to the valves should be changed.

If the valves should be able to switch with 15 Hz then the maximum switching frequency should be around 85 Hz, based on the used ON/OFF valves were having a maximum frequency of 8.33 Hz. At 85 Hz then $T_{ON} + T_{OFF} = 0.0118s$ as seen from equation 4.2 thus the T_{ON} and T_{OFF} should be 5.89 ms. However, if the solution should be used for smaller loads and with a small dead volume the valves need to be even faster, if the [10:90] % duty cycle interval is necessary. A new set of valves are designed with a switching time of 6 ms. The next chapter will compare the performance for the new valves with the original Hydac valves and the proportional valve.

Changing the switching frequency to 10 times as fast is not without consequences. The fluid hammer effect will send pressure transients through the cylinder 10 times as often. With a natural frequency at 1800 rad/s= 286 Hz then the switching is not fast enough to remove this added dynamic. The effect of the fluid hammer on the overall performance will be discussed in the evaluation part, because it is not reasonable to get valves which can switch with 0.025 ms.

6.2.1 ON/OFF servo: The switching strategy

As found in section 4.3, the valves should be paired, so valve 2 and 4 turn ON and OFF simultaneously and valve 1 and 3 do the same. Instead of having 4 different input duty cycles the problem is reduced to control and design two duty cycles. Furthermore, there is no sense in having all 4 controllers ON at the same time, since it will only lead to an energy loss.

This argues for changing between two control laws: the first where a duty cycle is sent to valve 1 for valve 3 and a constant off to valve 2 and 4, and another where the duty cycle instead is sent to valve 2 and 4 and a constant off to 1 and 3. The flows through the valves at different ON/OFF signals are given by equation 4.13, 4.14, 4.17 and 4.18 - thus the strategy can be formulated as a function of the error:

- If $\dot{x}_c - \dot{x}_{c_ref} < 0$ then a duty cycle should be sent to valve 2 and 4. Thereby letting flow from the pump into chamber 1 and from chamber 2 towards the tank.
- If $\dot{x}_c - \dot{x}_{c_ref} > 0$ then a duty cycle should be sent to valve 1 and 3. Thereby letting flow from the pump into chamber 2 and from chamber 1 towards the tank.

6.2.2 ON/OFF servo: Controller

Then the two sliding mode controllers can be designed. The design approach from section 6.1 are almost the same since the cylinders are the same for the two systems, but with a few adjustments:

at $\dot{x}_{c,ref} > 0$ then the flows for valve 2 and 4 are

$$Q_1 = u \cdot Q(p_s, p_1) \tag{6.22}$$

$$Q_2 = u \cdot Q(p_2, p_t) \tag{6.23}$$

where u is the duty cycle d.

Thus the control law will be

$$u = \frac{B \cdot (\ddot{x_{c,ref}} - sign(e)) + M \cdot \dddot{x} + \dot{x} \cdot \left(\frac{\beta \cdot A_1^2}{V_1} + \frac{\beta \cdot A_2^2}{V_2}\right)}{Q(p_s, p_1) \cdot \frac{\beta \cdot A_1}{V_1} + Q(p_2, p_t) \cdot \frac{\beta \cdot A_2}{V_2}} \tag{6.24}$$

Again, it is noted that there is a slight risk of division by 0, but physically it will not be reached. A gain is tuned for u in order to fit the duty cycle range of 0 to 1. 0.05 is found to be sufficient.

For $\dot{x_{c,ref}} < 0$ then the flows for valve 1 and 3 are

$$Q_1 = u \cdot Q(p_1, p_t) \tag{6.25}$$

$$Q_2 = u \cdot Q(p_s, p_2) \tag{6.26}$$

Since the duty cycle cannot be negative, then u=(1-d).

$$u = \frac{B \cdot (\ddot{x_{c,ref}} - sign(e)) + M \cdot \dddot{x} + \dot{x} \cdot \left(\frac{\beta \cdot A_1^2}{V_1} + \frac{\beta \cdot A_2^2}{V_2}\right)}{Q(p_1, p_t) \cdot \frac{\beta \cdot A_1}{V_1} + Q(p_s, p_2) \cdot \frac{\beta \cdot A_2}{V_2}} \tag{6.27}$$

As previously, it is noted that there is a slight risk of division by 0, but physically it will not be reached. A gain is tuned for u in order to fit the duty cycle range of 0 to 1. -0.05 is found to be sufficient.

6.2.3 Summary

It has been shown that the ON/OFF valves are too slow to control the system. As a consequence all 4 ON/OFF valves have been replaced with 4 valves which have a switching time of 6ms. The control strategy for the ON/OFF valves are pairing valve 2 and 4, which get the same duty signal for a positive velocity reference. For a negative velocity reference then valve 1 and 3 are paired instead to the same duty cycle. Finally 3 sliding mode controllers have been designed: 1 for the proportional valve and 2 for the ON/OFF valves. The next chapter will compare and evaluate the performance.

Part III

Evaluation

Chapter 7

Performance Evaluation

This chapter will present and compare the performance of the proportional valve system with the ON/OFF system including new and original valves. The results will be discussed and a set of recommendations will be made to Hydac concerning the future work and prospects of using ON/OFF valves.

7.1 Test Description

There will be performed 4 different tests in order to evaluate the performance of the servo systems with control. The requirement from Hydac was tracking of a velocity reference at max 150mm/s. The 4 tests are a combination of 2 different tracking signals and 2 different weights as shown below:

- **Test 1:** Tracking a combination of a constant reference and a ramp with M=1000 kg
- **Test 2:** Tracking a combination of a constant reference and a ramp with M=7500 kg
- **Test 3:** Tracking sine signal with an amplitude of 0.15, a frequency of 0.5 rad/s and with M=1000 kg
- **Test 4:** Tracking sine signal with an amplitude of 0.15, a frequency of 0.5 rad/s and with M=7500 kg

The combination of velocity references: constant-, ramp- and sine signal should be sufficient to outline the performance of the different valves.

The sine signal has been chosen to have a frequency of 0.5 rad/s =0.08 Hz in order for the original Hydac valves to have about 20 samples to change the duty cycle in. This should be slow enough to allow the valves a chance to control the cylinder, but at the same time without being unrealistically slow. The mass is varied in order to simulate different load cases.

The initial pressure of chamber 1 and 2 are 140 bar for all tests. An external force of -70kN is applied in order to have a stable starting point. The initial position of the cylinder is $x_c = 0.6m$ for test 1 and 2, whereas test 3 and 4 have a starting point at $x_c = 0.1m$.

7.2 Comparison of Results

Figures 7.3, 7.6, 7.1 and 7.2 show the velocity profile and tracking error for the 3 different servo systems.

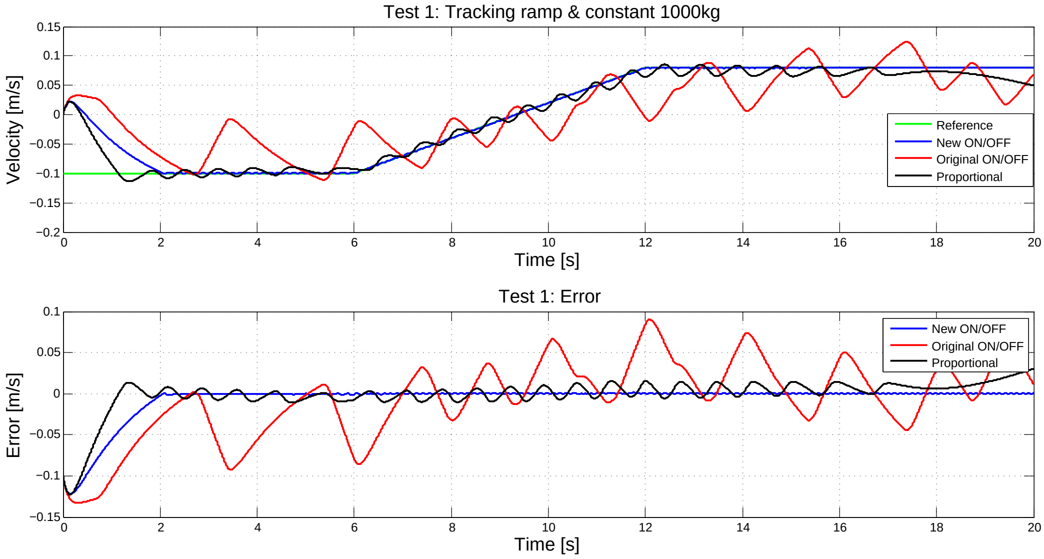


Figure 7.1. Test 1: Tracking a combination of a constant reference and a ramp with M=1000 kg

The proportional valve is a standard model, so it is not dimensioned after the two chambers having different areas. The difference in area leads to different requirements in flow as shown in section 3.1.

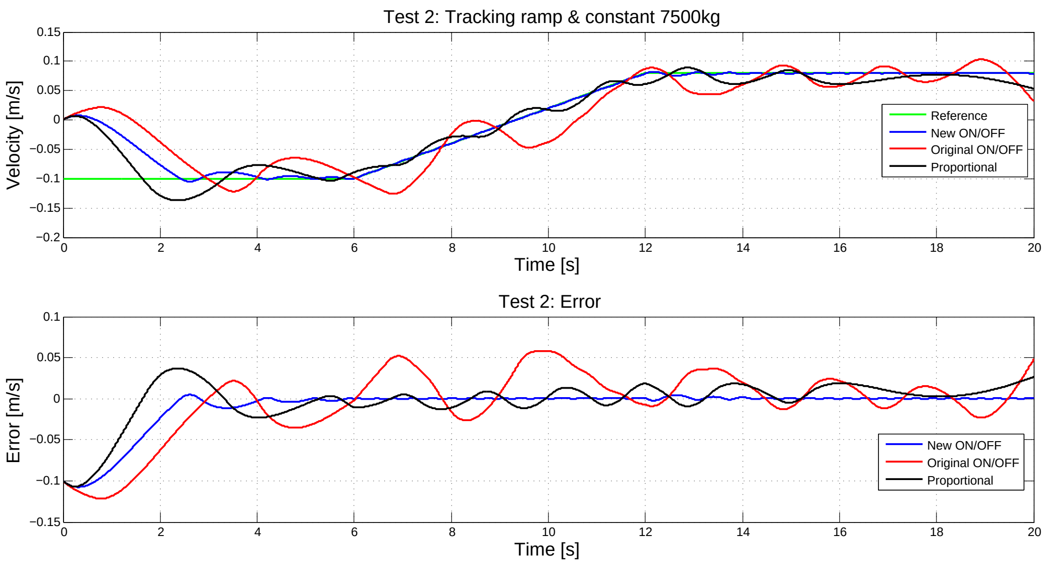


Figure 7.2. Test 2: Tracking a combination of a constant reference and a ramp with M=7500 kg

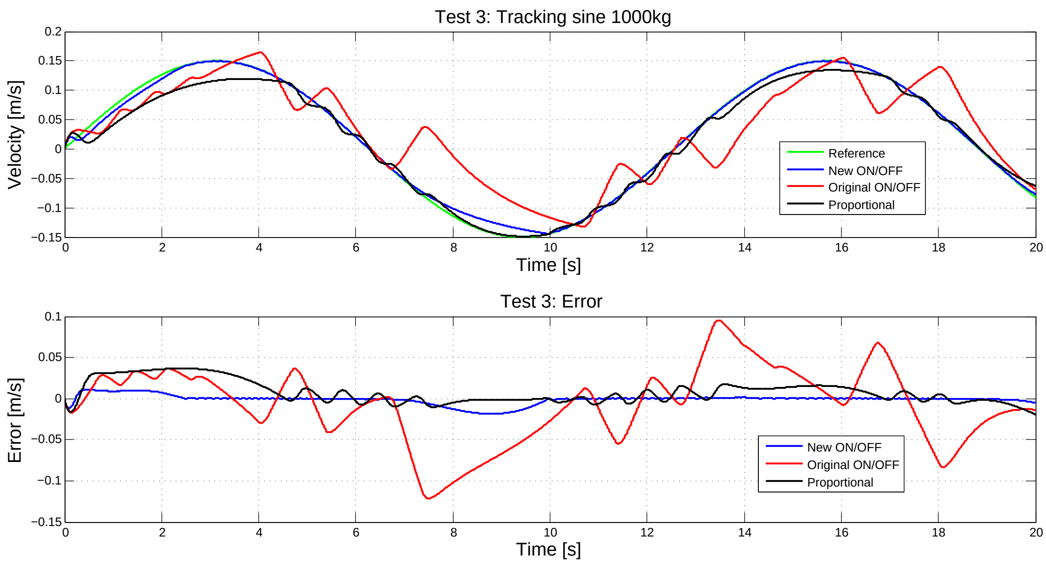


Figure 7.3. Test 3: Tracking a sine signal with an amplitude of 0.15, a frequency of 0.5 rad/s and a mass of 1000 kg.

Thus, the throttle loss is large and the tracking has an average error of 5.9 % and a standard deviation of 4.7%. The first 2s of test 3 and 4 are disregarded in these numbers. This is a satisfactory performance considering it cannot control the flow into the chambers independently.

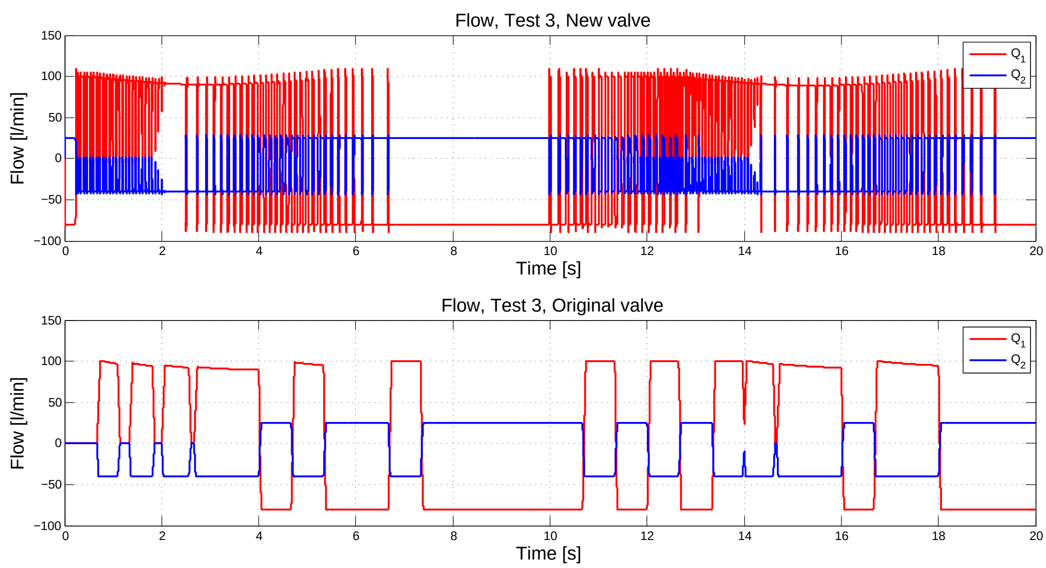


Figure 7.4. The flows for test 3. It can be seen that the new valves have an area where they are open for a longer time. When compared to figure 7.3 it can be seen that valve 3 has been chosen too small. The proportional valve allows more flow through to the tank than the two ON/OFF setups, which gives higher throttle losses as expected.

It is clear, that the new valves track the system with high accuracy even for a low mass. The 4 new valves only show some small problems in figure 7.3. Figure 7.4 shows the flows Q_1 and Q_2 for this test. The new valve 1 and 3 are fully open at the given problem area, which shows that the selection of valves has not been good enough - one or both are too small to meet the demands. Valve 3 was the only valve which was chosen to be smaller than the demand (0.45L/min), therefore it should be replaced with a larger model if the system was to be build. However, the average error is 0.88 % with a standard deviation of 1.21%, where only the first 2s of test 3 and 4 are disregarded. With a larger valve, even lower average errors should be obtainable. This performance is considered successful and satisfactory for most hydraulic control purposes.

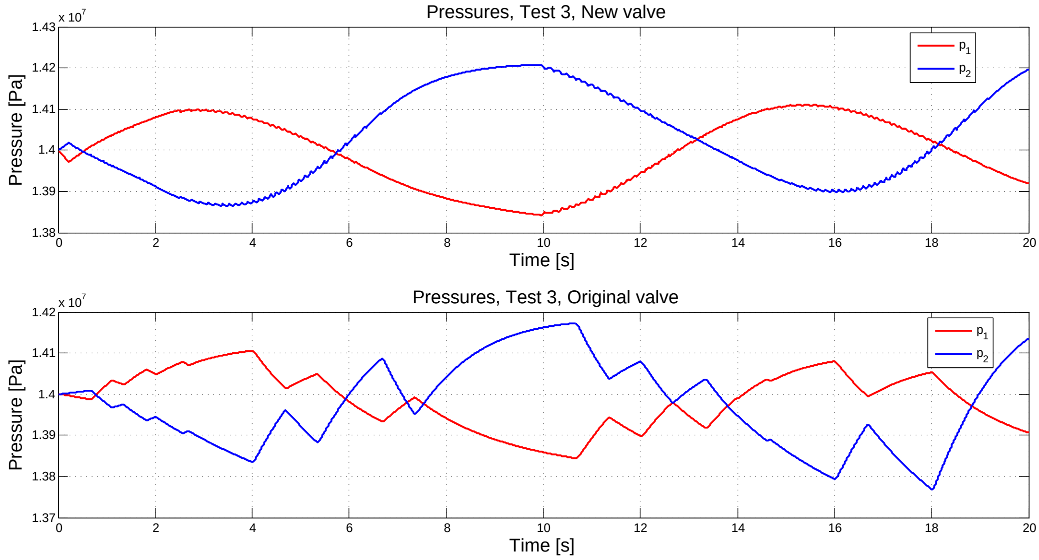


Figure 7.5. The pressures for test 3.

The original Hydac valves do not perform better than the proportional valve. The mean error for the Hydac valves is 19.4%, and a standard deviation of 14.9%. The first 2s of test 3 and 4 are disregarded in these numbers. The results are considered to be so poor that it is not an alternative, and the expectations from section 6.2 are met.

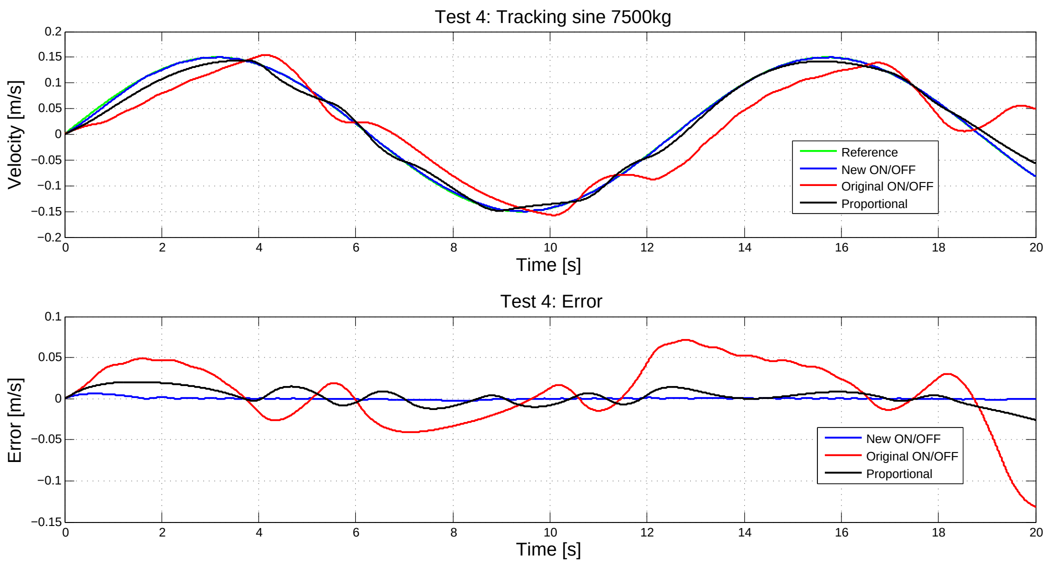


Figure 7.6. Test 4: Tracking a sine signal with an amplitude of 0.15, a frequency of 0.5 rad/s and a mass of 7500 kg.

In all, the results are as expected, but one question remains: why is the fluid hammer effect not more distinct? It can be seen in the flows for the new valves in figure 7.7, but the effect on the pressures are small, and it is therefore not visible in the velocity. In the simulation the transients will hit the piston wall at the same time with opposite sign, but since the area of chamber 2 is 36% of chamber 1, then some effect should be seen. One reason could be that the volumes are larger than in the analysis which would reduce some of the magnitude of the transients.

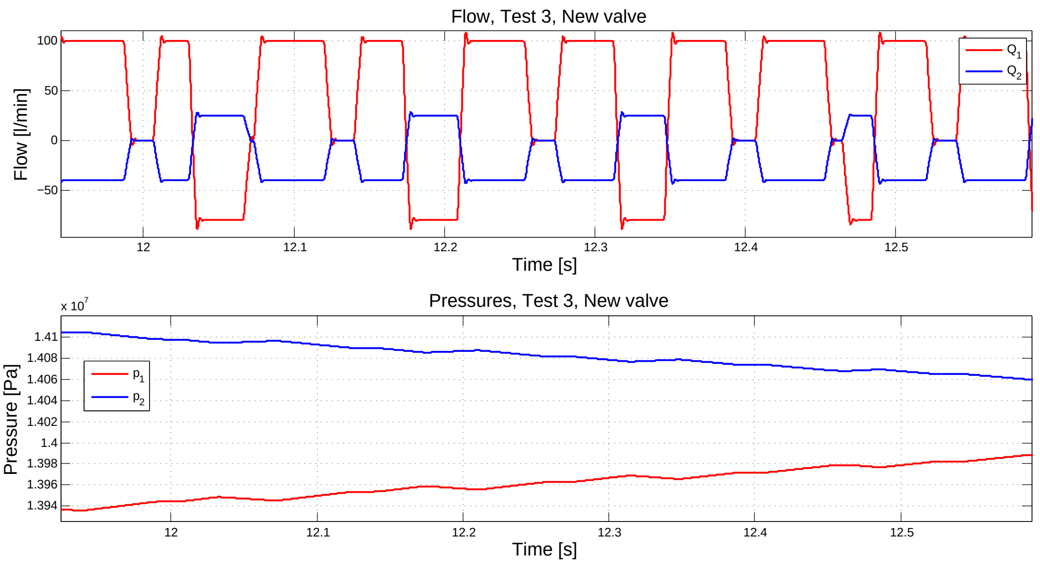


Figure 7.7. The fluid hammer effect is present at the flows, but it does not change nearly as much in the pressures as expected.

The servo system contains a damping, a large mass and external forces where the combination of these could ensure that the system does not show any response to the transients, i.e. the contribution from the fluid hammer effect is small compared to the other forces which position the valves. Yang and Kuo [2001] showed it for a system with small volumes and a pressure of 22 bar and no external forces. This system has 140 bar at each chamber and an external force of -70kN. However, it is also possible that the chosen implementation in the model (figure 4.7) is insufficient and the simulation dynamics wrongfully removes this effect. Without any laboratory test to document the response, it is not possible to judge which of the two theories is correct. This work is left to be continued by others.

7.3 Recommendations for Hydac

This last section will give a set of recommendations to Hydac about the prospect of this technology and areas that still need some investigations.

- The current valves are not suitable for this scenario - unless there is no problem in having a mean error of 19.4% with a standard deviation of 14.9%.
- It is unclear whether the missing presence of fluid hammer effect in the results are due to the larger volumes, stiffness of the cylinder and timing of collision with the piston wall or if it is just implemented in an inexpedient way in the model. A laboratory test should be performed to further investigate the fluid hammer effect.
- If the fluid hammer effect is implemented correctly, then use valves which can switch 10 times faster than the undamped natural frequency of the system. For the analysed system, it has been shown that if the valves have a switching time of 6 ms, then it is possible to track a velocity reference with an average error of 0.87% with a standard deviation of 1.21%.
- Valve 3 should be upgraded to a larger size if the system is to be build.
- A study of the saved throttle loss and energy consumption should also be performed. Currently, different researchers have presented ON/OFF valves with switching times of 10-15 ms [Taghizadeh et al., 2009] [Hansen et al., 2013], so it is assumed that the current price for a valve with a switching time of 6 ms is high, or that it has yet to be designed. As a consequence it might take a few years before this solution becomes strictly price competitive with a low performance proportional valve, but if the savings and tracking are a part of the business case it can be made commercial earlier.
- Noise, wear and tear of the valves have not been investigated and should also be investigated.

Chapter 8

Conclusion

The main objective of this project has been to analyse the feasibility of using ON/OFF valves instead of the conventional proportional valves for controlling a hydraulic cylinder. The velocity servo is chosen to be the analysis basis. A concept was proposed by Hydac with an asymmetrical cylinder where the proportional valve was replaced with 4 ON/OFF valves. The process of analysing the feasibility of this design consisted of 3 steps: System understanding, system control and evaluation.

First, the dimensioning of different components is made, as shown in chapter 3. Two models are made: an ON/OFF valve model and a proportional valve model for comparison. Both models include the cylinder and their respective valve dynamics. The pump is assumed to deliver a constant pressure and flow, and the fluid is assumed to have a constant bulk modulus. 3 challenges are identified for the ON/OFF model: Switching time, fluid hammer effect and multi-variable switching. The switching time restricts the switching frequency to 1.5 Hz (section 4.1), the fluid hammer shows that pressure transients will be present every time a valve switches (section 4.2) and the multi-variable system becomes MISO instead of SISO (section 4.3).

Second, the response to controlling the cylinder is analysed, as shown in chapter 5. The servo system has an natural frequency of 1-4 Hz and a low damping ratio which makes the control difficult. An area is identified where the parameter values are slightly better, and a dead volume size is determined and added to each chamber. A switching algorithm is designed for the ON/OFF valves, which utilizes control of valve 2 and 4, and 1 and 3 synchronously (section 6.2). The ON/OFF valves are estimated to be too slow, and 4 new are designed which can operate at 15 Hz. Finally, 3 sliding modes controllers are designed: one for the original valves, one for the new and one for the proportional valve (section 6.1.1 and 6.2.2).

Third, the performance of the valves are investigated as seen in chapter 7. 4 tests are made with 2 different velocity references and at 2 different masses. The original ON/OFF valves demonstrate that they are too slow as predicted: a mean error of 19.4% with a standard deviation of 14.9%. The proportional valve has a satisfactory performance with an average error of 5.9 % and a standard deviation of 4.7%. The new valves show an average error of 0.87% with a standard deviation of 1.21%, which is considered a satisfactory performance.

The next step would be to build a testing stand in order to verify the results. The simulations show a fluid hammer effect which does not affect the velocity. This result are conflicted with the expectations from section 4.2, and should be investigated.

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Appendix A

Contents on CD

This is the contents of the attached CD. The numbering is the same on the CD.

1. A pdf copy of the report
2. Pdf copies of the used references
3. Hydac Datasheets
4. Proportional Valve Model
5. Original Hydac Valve Model
6. New Valve Model
7. Different Matlab Scripts