Master Thesis Report

Optimization of Valve Damping

by

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Abstract

Öhlins CES Technologies in Jönköping have in the last 30 years been developing control valves for semi active suspension systems used in the car industry. The system, marketed by Öhlins under the brand name CES (Continuously controlled Electronic Suspension), enables a wide working range and ability to adapt to the current road conditions. By controlling the valve in different ways there are also possibilities to decide on a specific damper characteristic such as sport or comfort.

The CES valve is working as a pilot controlled pressure regulator and is continuously controlled with help of an electro magnet. The CES valve is mounted in a uniflow damper which in turn guarantees the flow through the valve to go in only one direction independently of damper stroke direction.

The first part of the thesis investigates the damping characteristics in the latest model of the CES valve (i.e the CES8700). A simulation model is made to approximate the damping in the solenoid plunger. Questions that are answered are: How is damping defined, what creates damping in the valve, how large is the damping, what parameters affect the damping. The second part of the thesis investigates new and already prototyped damping concepts with help of simulation. This has been done in order to optimize the valve damping and in turn the damper performance. The simulation results show that the valve dynamics can be improved but often at the expense of a slower valve.
Acknowledgements

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Chapter 1

Introduction

1.1 Background

Öhlins CES Technologies in Jönköping Sweden have since the 1980s been developing control valves for semi active suspension systems used in the car industry. The system, marketed by Öhlins under the brand name CES, enables a wide working range and ability to adapt to the current road conditions. There are also possibilities to decide on a specific damper characteristic such as sport or comfort by controlling the valve in different ways. Throughout the last years the development of the valve has resulted in changes in the design. The changes have been made to improve the performance of the valve but also to eliminate some of the problems that have been identified during lab or field testing. Problems with noise and other unfortunate behaviours have been detected in earlier designs. The way of solving some of the problems have been to introduce valve damping in different ways. The problems have been of different kind and therefore damping has been introduced in different ways and in different parts of the valve. However, when evaluating a new design, the damping has never been measured due to lack of measurement methods. This means that there is not much knowledge about the actual damping in the valve and how the damping has changed between the valve versions. To be able to know how a new concept affects the damping it is crucial to perform measurements to be able to evaluate the design. It is also important to find new ways of introducing damping to improve the valve performance.

1.2 Purpose

The thesis project aims to investigate the damping in the latest design of the CES valve and to compare it to old and new damping concepts. The work is focused around the following main questions:

- What is the damping in the valve and can it be measured?
Chapter 1. Introduction

1.3 Limitations

The following valves have been chosen for testing and analysing: CES8700N5541K, CES4600-G020K, CES4343A and CES2000A. The valves will not be individually modeled in detail. The damping concepts will be introduced in the already existing simulation model of the CES8700. The working points that are used in the simulations are chosen to cover low, medium and high currents and flows. No validation will be made of the simulated concepts. When analysing simulation results from models that are not validated trends will be analysed instead of exact values. Comparisons between simulation and empirical tests will be done as far as possible. A new test rig will not be built. Instead data from the existing test methods will be analysed together with simulations.

1.4 Method

A theoretical study on damping has been made as well as a mathematical description of the CES valve. The four different valve types have been tested and analysed. New and old damping concepts have been modelled in the simulation program LMS AMEsim. Literature and older thesis works performed at Öhlins Racing have been studied.
Chapter 2

System description

2.1 Damping of a chassis

To have a pleasant ride in a car, the vertical motion of the chassis should be kept as small as possible. In addition the tires should always have contact with the road independently of the road conditions to guarantee the best possible tire grip. This means that the relative motion that is created between tire and chassis needs to be eliminated or reduced. To handle this motion and to absorb the force that wants to accelerate the chassis, a spring is used on each wheel. The downside of the spring is that it has a natural tendency to oscillate when the absorbed energy is released. To handle the oscillations and forces, a damper is introduced with the purpose to convert the spring energy to heat.

2.1.1 The suspension mass spring system

In Figure 2.1 the velocity of the mass (car) and the road will both contribute to the total velocity of the damper. They are therefore split up in $\dot{z}$ (vertical velocity of the car) and $\dot{u}$ (vertical velocity of the road, e.g a road bump) to be studied separately. From Figure 2.1 the differential equation of motion becomes:

$$m_c \ddot{z}_c + B_p \dot{x}_d + k_s x_d = 0 \quad (2.1)$$

Damper position is calculated as:

$$x_d = z_c - u_r \quad (2.2)$$

Damper speed is calculated as:

$$\dot{x}_d = \dot{z}_c - \dot{u}_r \quad (2.3)$$
Figure 2.1: Sketch of the suspension mass spring system

If equation (2.2) and (2.3) is put into (2.1) the following is obtained when simplified [8].

\[
\ddot{z}_c + \frac{B_p}{m_c} \dot{z}_c + \frac{k_s}{m_c} z_c = \frac{1}{m_c} (B_p \dot{u}_r + k_s u_r)
\] (2.4)

Since equation (2.4) is on standard form, the constant in front of \( z_c \) can be identified according to equation:

\[
\frac{k_s}{m_c} = \omega_n^2 \Rightarrow \omega_n = \sqrt{\frac{k_s}{m_c}}
\] (2.5)

where \( \omega_n \) is the natural angular frequency.

The constant in front of \( \dot{z}_c \) is identified as:

\[
\frac{B_p}{m_c} = 2\zeta \omega_n \Rightarrow \zeta = \frac{B_p}{2\omega_n m_c} \Rightarrow \zeta = \frac{B_p}{2\sqrt{m_c k_s}}
\] (2.6)

where \( \zeta \) is the relative damping ratio.
2.2 The damper

A sketch of the damper is seen in Figure 2.3. The damper consists of an outer cylindrical tube with two cylindrical tubes inside. The two tubes on the inside form a ring tube for the oil to flow in. In the piston as well as in the base, there is a check valve that makes sure that the flow will go in only one direction through the CES valve. There is also a blow off valve in the piston and in the base that prevents a too high pressure build up. A simplified hydraulic scheme of the damper can be seen in Figure 2.2. The damper is not the main focus of this study but will be described to understand the complete system.

![Figure 2.2: Hydraulic scheme of the damper](image)

2.2.1 Compression

The damper is defined to be in compression when the piston is moving down in Figure 2.3. During compression of the damper, the check valve in the piston opens and forces the oil to flow according to the arrows in Figure 2.3 while the check valve in the base is closed. The blow off valve in the base prevents the pressure from building to high in the compression chamber. The flow is created due to the different size of volume in the compression and rebound chamber. The oil displacement will be equal to the piston rod that is forced inside the rebound chamber. The oil flows through the ring tube, through the CES valve and in the gas chamber. When more oil enters the gas chamber the gas pressure increases.

2.2.2 Rebound

Rebound is defined as the piston moving up in Figure 2.3. During rebound of the damper the oil no longer flows through the piston. The check valve in the piston is now closed
and prevents the oil to pass. Instead the check valve in the base opens and the blow off valve in the piston prevents the pressure from building to high in the rebound chamber. As for compression the oil flows through the inner ring tube and on through the CES valve according to the arrows in Figure 2.3. In this case the displaced oil volume will be calculated as \((A_{\text{piston}} - A_{\text{piston,rod}}) x_{\text{damper,position}}\). By using this damper design, the oil will flow in one direction through the CES valve independently of stroke direction. This is a main advantage when it comes to designing the valve.
2.3 The CES valve

CES stands for Continuously controlled Electronic Suspension and enables a certain amount of control over the damper characteristics. In a conventional damper, used in most cars, oil is forced through a fixed orifice or spring loaded check valve which creates damping. In the CES system on the other hand, that orifice (i.e the CES valve) is electronically controlled by the car control unit. The principals of this system will be described in detail in this section. The system consists of a uniflow damper and a CES valve. When the damper is compressed or in rebound the oil is forced to flow through the CES valve. The valve is working as a controllable restriction which creates a pressure drop between the two damper chambers. It can be seen as a more complex pressure regulator which can be continuously controlled.

2.3.1 Principles of the 2-stage controlled pressure regulator

Figure 2.5: Pressure regulators
Figure 2.5a shows a sketch of a simple pressure regulator. When the pressure \( P_1 \) (acting on the poppet area) overcomes the spring force, the poppet will open. This means that the opening pressure is determined by the force needed to compress the spring. If instead designing the pressure regulator as in Figure 2.5b a pressure is created inside the main poppet due to the jd restriction. As long as the pilot poppet is closed there will be no flow through the jd or pd restriction and therefore the pressure inside the main poppet (B) will be \( P_1 \). The closing force will now be the spring force plus the pressure inside the main poppet acting on the inner main poppet area. This will, with the same dimensions as in Figure 2.5b, create a larger closing force. For the main poppet to open, the volume inside the poppet (B) needs to be evacuated. When the force created by \( P_2 \) (C) together with the pilot poppet spring and the force \( F \) is smaller than the force created by \( P_1 \) (on the pilot poppet) the pilot poppet will open. When this occurs the pressure inside the main poppet (B) decreases, the volume inside can be evacuated through the pilot orifice and the main poppet can open. The largest portion of the flow will then flow through the main poppet orifice. A smaller portion will flow through the jd restriction, on through the pd restriction and through the pilot orifice according to the arrows in Figure 2.5b. By using this concept the dimensions can be made smaller since the pressures that is controlled, is amplifying the pressure regulator force. It also means that the closing force of the pressure regulator will be dependent on the pressures \( P_1 \) and \( P_2 \). This is the principle of the CES8700 valve. By controlling the force \( F \) by an electro magnet, the valve becomes continuously controlled.

### 2.3.2 Already prototyped valve models

The principle function of the valve has always been the same, meaning that a solenoid is acting on the pilot poppet and in turn controlling the pressure needed to open the main poppet. Some parts have changed between the valve generations and some of the changes are interesting from a damping perspective. When the valve is described it is often divided into three parts: main stage, pilot stage and solenoid. Throughout the years damping has been introduced in all three of these parts in different ways. In the model CES2000, shown in Figure 2.6a, damping was introduced on the main poppet by using a damping volume. This concept is described more deeply in section 5.3.1. This method was not always favourable since the damping of the main poppet introduced sudden pressure peaks when shocking the valve with a sudden flow. On the other hand it was pleasurable to use when driving on a constantly ”bad” road, since the main stage never had time to fully close. This gave a soft and comfortable feeling. The design was dropped in the next version of the valve since the sudden pressure peaks could not be allowed due to an uncomfortable driving experience.

In the model CES4300, shown in Figure 2.6b the problem of ”singing” was discovered, which is a high frequency noise occurring when the damper has been exposed to heavy use, rested for 24 hours and then used again. The reason for this was believed to be gas bubbles
mixed into the oil that could not be evacuated which in turn changed the bulk modulus of the oil. This is described in [4]. No specific damping design existed in this model. In the following model CES4600 the problem of singing disappeared when adding a damping cylinder to the pilot poppet seen in Figure 2.6c. The disadvantage of this was lowered bandwidth. In the latest model, i.e CES8700, this damping cylinder is taken away and the problem with singing is solved by letting the oil flow around and pressurize the solenoid. By doing this the gas can be evacuated.
Chapter 3

Theory

3.1 Damping

3.1.1 Step response measurement

To be able to evaluate a step response fully, different measurements can be used. Different types of step responses do sometimes demand different types of measurement methods. In this section four methods are presented. Often these methods needs to be combined to get an estimation of the damping. Depending on the demands on the step response, one method can be of greater significance than the other.

3.1.1.1 Damping ratio

The damping ratio is a measurement of the logarithmic decay. Figure 3.1 shows a plot of some different damping ratios. ζ < 1 is defined as under damped. ζ = 1 is defined as critically damped, meaning the lowest amount of damping possible without getting any overshoot. Damping ratios where ζ > 1 is defined as over damped. Damping ratio is an indicator of how oscillative a system is.

To compare theoretical damping ratios to empirical test results the following formula can be used:

\[
ζ = \frac{\ln\left(\frac{y_1}{y_2}\right)}{\sqrt{(2\pi)^2 + \ln\left(\frac{y_1}{y_2}\right)^2}}
\]

(3.1)

where \(y_1\) and \(y_2\) are two subsequent peaks in the graph. The formula is only valid if the system is under damped and at least two peaks can be observed [4].

If the system only gives one overshoot with no following oscillations the following formula
can be used [16] for damping ratios between 0.5-0.8:

$$\zeta = \sqrt{\frac{\ln\left(\frac{M}{100}\right)^2}{\pi^2 + \ln\left(\frac{M}{100}\right)^2}}$$ (3.2)

Where $M$ is the percentage overshoot calculated by taking the maximum overshoot value minus the step value, divide by the step value.

### 3.1.1.2 Percentage overshoot

Percentage overshoot is a measurement of how much the system exceeds the target value before stabilizing. It is given in percentage of the step value and is calculated as $M = (x_1 - x_0)/x_0$. In Figure 3.2 the overshoot is shown.

### 3.1.1.3 Rise time

Rise time is defined as the time required for the response to rise from $x$ to $y$ of its final value. In Figure 3.2 an example of how rise time is measured is shown. The common way to observe rise time is to measure the time it takes for the response to rise from 10% to 90% of its final value. The rise time is then calculated as $t_3 - t_1$ [3]. However, it does not handle the initial lag that might occur between 0-10%. To capture the lag when measuring, the definition for time constant can be used instead [3]. By measuring the time it takes for the
step response to rise from 0% to 63%, the initial lag is captured. Rise time is an indicator of how quickly the system responds to a change.

![Step response diagram with labels](image)

Figure 3.2: Example of step response measurement

### 3.1.1.4 Settling time

In Figure 3.2 it can be seen how settling time is measured. Settling time can be measured in time or in number of oscillation periods. Settling time is an indicator of how quickly the step response reaches a steady level.

### 3.1.2 Friction

All kinds of friction will contribute to damping of a system since the friction force always counteracts the direction of motion and in turn transforms energy into heat [12]. For a moving object this means that the motion eventually will stop, due to the opposing forces, if no new energy is put into the object. The total energy that is dissipated from the system is shown in equation (3.3).

\[
| \int B_p \dot{x} dx | + | \int \mu N dx | = W_{damp} \tag{3.3}
\]

Friction occurs in mainly two forms in a hydraulic system and can be modelled in a large number of ways [9] [11]. By combining different friction theories, approximations can be made with the help of simulation.
3.1.2.1 Coulomb friction

The most basic and commonly used friction model is the coulomb friction model. It is often called the dry friction model but is used in dry as well as boundary and mixed lubricated contacts.

\[
F = \begin{cases} 
F_c & \text{if } v > 0 \\
F_{\text{app}} & \text{if } v = 0 \text{ and } F_{\text{app}} < F_c 
\end{cases} 
\]  

(3.4)

where \( F_c \) is the coulomb friction force defined by

\[
F_c = \begin{cases} 
\mu_k N & \text{if } v > 0 \\
\mu_s N & \text{if } v = 0 
\end{cases} 
\]  

(3.5)

where \( \mu_s \) is the static friction coefficient, \( \mu_k \) is the kinematic friction coefficient and \( N \) is the normal force. In Figure 3.3a the damping due to coulomb friction can be seen. Coulomb friction is known to give hysteresis and static errors and does not contribute to stabilizing a system according to [14].

3.1.2.2 Viscous friction

Viscous friction occurs in the fluid when the fluid is given a velocity and is created due to shear of the fluid layers. The force created by the viscous friction can be calculated according to equation (3.6). The damping due to viscous friction can be seen in Figure 3.3b.

\[
F = B_v \ddot{x} 
\]  

(3.6)
where $B_p$ is the viscous damping coefficient measured in $Ns/m$ and $\dot{x}$ is the velocity of the fluid.

From the equation (3.6) it’s clear that an increased speed will increase the viscous dependent friction force.

### 3.1.2.3 Striebeck friction

A commonly used friction model is the Striebeck friction model [11] [9]. With this model the total friction force can be approximated taking in consideration the static coulomb friction, the velocity dependent viscous friction and the break away friction (stiction) as seen in equation (3.7).

Figure 3.4 shows the four regimes that characterize the Striebeck friction model [10]. Regime I represents the break away friction or steady state friction, regime II represents the boundary lubrication regime, regime III represents the partial fluid lubrication and the last regime IV represents full fluid lubrication. Regime II and III are the most complex since the friction force is dependent on material properties of the sliding surfaces (Youngs modulus, poisson’s ratio, surface roughness etc.) as well as fluid properties of the lubricant (Viscosity, temperature etc).

To capture this behaviours a number of other more complex friction models have been developed, e.g Dahl and Karnopp, this is documented in [9], [10] and [11].

$$F = (F_c + (F_{brk} - F_c)^{-\frac{1}{n}})\text{sign}(v) + B_pv$$  \hspace{1cm} (3.7)
3.1.3 Valve damping

3.1.3.1 The turbulent flow equation

When the flow is turbulent it is proportional to the square root of the pressure difference over the restriction according to equation (3.8). \( \rho \) is the density of the fluid, \( A \) is the area of the restriction, \( \Delta p \) is the pressure difference and \( C_q \) is the turbulent flow coefficient that varies with restriction geometry.

\[
q = C_q A \sqrt{\frac{2}{\rho} |\Delta p| \text{sign}(\Delta p)} \tag{3.8}
\]

3.1.3.2 The laminar flow equation

When the flow is laminar the flow is directly proportional to the pressure difference over the restriction according to equation (3.9). \( K_{\text{laminar}} \) is the laminar flow coefficient and \( \Delta p \) is the pressure difference over the restriction.

\[
q = K_{\text{laminar}} \Delta p \tag{3.9}
\]

3.1.3.3 The continuity equation

This equation describes the dynamics of the flows going in and out of a volume, taking in consideration the volume change and the hydraulic capacitance. The equation is shown on its standard form in equation (3.10).

\[
\sum q_{\text{in}} = \frac{\partial V}{\partial t} + \frac{V}{\beta_e} \frac{\partial p}{\partial t} \tag{3.10}
\]

In the case where a linear actuator or a poppet is studied, the volume change is often rewritten as equation (3.11). The pressure gradient is also rewritten in a more simple way.

\[
\sum q_{\text{in}} = A \dot{x} + \frac{V}{\beta_e} \dot{p} \tag{3.11}
\]
3.1.3.4 The damping orifice

This section aims to show why an orifice gives rise to a velocity dependent force that acts opposite to the direction of motion. This is what is usually called viscous damping and is caused by the viscous friction created when the fluid is forced through a restriction. In Figure 3.5 a simple pressure regulator is shown.

\[ q = A_p \dot{x} - \frac{V_1 - A_p x}{\beta_e} \dot{p}_1 \]  

where \( \dot{x} \) is the velocity of the poppet.

The flow equation becomes:

\[ q = C_q A_o \sqrt{\frac{2}{\rho}} (p_1 - p_T) = K_t A_o n/\sqrt{p_1 - p_T} \]  

where \( K_t = C_q \sqrt{\frac{2}{\rho}} \), \( C_q \) is the turbulent flow coefficient, \( \rho \) is the density of the fluid and \( n \) is a number that states the degree of turbulence \([1 < n < 2]\) where \( n = 2 \) denotes fully developed turbulent flow.
The differential equation of motion becomes:

\[ m \ddot{x} + kx = p_2 A_{poppet} - p_1 A_{poppet} - F_c \text{sign}(\dot{x}) \]  

(3.14)

where \( m \) is the mass of the poppet, \( k \) is the spring stiffness and \( F_c \) is the coulomb friction force.

**Laminar flow**

The following derivation will assume laminar flow and therefore \( n = 1 \).

The laminar flow equation becomes:

\[ q = K_{lam} A_{orifice} (p_1 - p_T) \]  

(3.15)

where \( K_{lam} \) is the laminar flow coefficient.

Since equation (3.12) is equal to (3.15) we get:

\[ p_1 = \frac{A_p \dot{x} - \frac{V_1 - A_p x}{\beta_e} \dot{p}_1}{K_{lam} A_o} + p_T \]  

(3.16)

If equation (3.16) is put into (3.14) the expression becomes:

\[ m \ddot{x} + \left( \frac{A_{pl}^2}{K_{lam} A_o} \right) \ddot{x} + \left( k + \frac{A_p^2}{\beta_e K_{lam} A_o} \dot{p}_1 \right) x = \]  

\[ = p_2 A_p + \frac{A_p V_1}{\beta_e K_{lam} A_o} \dot{p}_1 - p_T A_p - F_c \text{sign}(\dot{x}) \]  

(3.17)

The coefficient in front of \( \dot{x} \) is the viscous damping coefficient by definition. One can see that a decreased \( A_o \) will increase the damping coefficient and in turn the damping force.

**Turbulent flow**

The following derivation will assume fully developed turbulent flow and therefore \( n = 2 \).

Since equation (3.12) is equal to (3.13) we get:

\[ p_1 = \frac{1}{(K_i A_o)^2} \left( A_p \dot{x} - \frac{V_1 - A_p x}{\beta_e} \dot{p}_1 \right)^2 + p_T \]  

(3.18)
when expanded, the expression becomes:

$$p_1 = \left(\frac{A_p}{K_tA_o}\right)^2 \dot{x}^2 - \left(\frac{2A_p\dot{p}_1(V_1 - A_p x)}{(K_tA_o)^2 \beta_e}\right) \ddot{x} + \left(\frac{A_p\dot{p}_1}{K_tA_o\beta_e}\right)^2 x^2 -$$

$$- \left(\frac{2V_1 A_p \dot{p}_1^2}{(K_tA_o\beta_e)^2}\right) x + \left(\frac{V_1 \dot{p}_1}{K_tA_o\beta_e}\right)^2 + p_T \tag{3.19}$$

If equation (3.19) is put into (3.14) the expression becomes:

$$m\ddot{x} + \left(\frac{A_p^3}{(K_tA_o)^2}\right) \dot{x}^2 - \left(\frac{2A_p^2 \dot{p}_1(V_1 - A_p x)}{(K_tA_o)^2 \beta_e}\right) \ddot{x} + \left(\frac{A_p^3 \dot{p}_1^2}{(K_tA_o\beta_e)^2}\right) x^2 +$$

$$+ \left(k - \frac{2V_1 A_p \dot{p}_1^2}{(K_tA_o\beta_e)^2}\right) x = p_2 A_{poppet} + \frac{A_p(V_1 \dot{p}_1)^2}{(K_tA_o\beta_e)^2} - p_T A_p - F_c \text{sign}(\dot{x}) \tag{3.20}$$

The classical way of identifying the damping constant would be to simply pick the constant in front of the velocity $\dot{x}$. But from observing equation (3.20) it can be seen that the final equation of motion is on a more complex form than in the standard case. In this case the coefficients in front of the velocity $\dot{x}$ and the velocity in square $\dot{x}^2$ will create the viscous damping coefficients. We also see that a decreased $A_o$ will increase the damping coefficients and in turn the damping force. The complete damping force can be written as equation (3.22) and consists of both a velocity dependent and a constant part. One can also see that there will be two terms containing $\dot{p}_1$ and $x$. These terms, seen in equation (3.21), will act as a spring and will be important in the damping point of view since they will vary with the magnitude of the pressure gradient and the poppet position. The pressure gradient appears in two other terms which makes the complete force equilibrium a complex equation to interpret.

$$F_{spring} = \left(\frac{A_p^3 \dot{p}_1^2}{(K_tA_o\beta_e)^2}\right) x^2 + \left(k - \frac{2V_1 A_p \dot{p}_1^2}{(K_tA_o\beta_e)^2}\right) x \tag{3.21}$$

$$F_{damp} = \left(\frac{A_p^3}{(K_tA_o)^2}\right) \dot{x}^2 - \left(\frac{2A_p^2 \dot{p}_1(V_1 - A_p x)}{(K_tA_o)^2 \beta_e}\right) \ddot{x} + F_c \text{sign}(\dot{x}) = B_p \dot{x}^2 - B_r \ddot{x} + F_c \text{sign}(\dot{x}) \tag{3.22}$$
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In Figures 3.6 and 3.7 the velocity dependent coefficients are investigated further. The variables in the plots have been normalized, where 1 corresponds to the following:

- Orifice area \(m^2\): 1 corresponds the same size as the pd orifice in the CES8700.
- Poppet area \(m^2\): 1 corresponds same size as the main poppet area in the CES8700.
- Poppet position \(m\): 0 means closed and 1 means fully open.
- Pressure gradient \(Pa/s\): 1 is a number calculated with help of simulating a flow step.

If comparing the magnitude of \(B_p\) in Figure 3.7 and \(B_r\) in Figure 3.6a it can be seen that \(B_p\) is a factor 1000 larger than \(B_r\). This implies that \(B_p\) will be the dominant coefficient. However, the forces created by \(B_p\) and \(B_r\) when they are multiplied with velocity will approach each other for small velocities. At the velocity 0.0014 m/s the forces are equal and receives the value 0.006 N. In Figure 3.6b the value of the viscous coefficient can be seen when varying the pilot position and the pressure gradient. It can be seen that the magnitude of change is small compared to 3.7.

![Figure 3.6: Sensitivity of \(B_r\)](image)

(a) Varying orifice and poppet area  
(b) Varying pressure gradient and poppet position
Figure 3.7: Sensitivity of $B_p$ when varying orifice and poppet area
3.2 Calculations on the CES valve

In this section the CES valve is investigated through calculations and derivations.

![Diagram](image)

(a) Friction on the main poppet  (b) Friction on pilot poppet

Figure 3.8: The considered friction forces

3.2.1 Friction forces

Friction has been approximated using equation (3.23) \[1\]. The formula handles both the velocity dependent viscous friction and the pressure dependent friction. The direction is positive when the main and pilot poppet opens. In the formula the velocity is defined positive when the poppet moves in the same direction as the flow going through the gap. The plots that are shown in this section have normalized axes. The eccentricity \(e\) in the formula is not known and is set to 0 since this gives the lowest velocity dependent friction force.

\[
F_f = \frac{2\pi r \eta l}{h_0 \sqrt{1 - \left(\frac{e}{h_0}\right)^2}} v - \pi r h_0 \Delta p \tag{3.23}
\]

3.2.1.1 Main poppet

The considered friction force for the main poppet is shown in Figure 3.8a. Figure 3.9 shows how the friction force varies with pressure drop over the gap and the main poppet velocity. It can be seen that the pressure drop has higher influence on the friction than the velocity.
It can also be seen that the magnitude of the friction force changes slightly with the stroke direction of the main poppet.

![Figure 3.9: Friction variation on main poppet](image)

3.2.1.2 Pilot stage and solenoid

The considered friction forces for the pilot poppet is shown in Figure 3.8b. The friction forces are calculated using equation (3.23) and then added together. The pressure dependent part of the formula is used for $F_{fd}$ and $F_{fs}$ where there is a significant pressure drop according to the currently used simulation model. In Figure 3.10a the friction force is shown as a function of pressure drop from $p_4$ to $p_T$ and velocity. A high positive constant pressure drop from $p_3$ to $p_4$ is considered that has been taken from simulation. Figure 3.10b shows the same plot except for the pressure drop from $p_3$ to $p_4$ is changed to a high negative constant value also taken from simulation. Off course both the pressure drops changes continuously and therefore the real friction force is hard to capture in one plot. Instead a specific point needs to be studied. In this study only the significant pressure drops are considered. In reality there is a pressure drop coupled to each friction force. The pressure drop from $p_4$ to $p_T$ is
chosen for the plot due to that it varies statically with the applied flow on the valve. The pressure drop from $p_3$ to $p_4$ on the other hand is only present dynamically when a step in either current or flow is taken. Statically this pressure drop is close to 0.

(a) High positive pressure drop from $p_3$ to $p_4$ (b) High negative pressure drop from $p_3$ to $p_4$

Figure 3.10: Friction variation on pilot poppet
3.2.2 Viscous coefficient investigation

In appendix B the velocity dependent flows that are entering and exiting the volumes are investigated. This is done by simplified calculations and assumptions. Figure 3.11 is used for the calculations. In equation (3.24) the damping force for the main poppet is shown. In equation (3.25) the total damping force for the pilot poppet is shown. The term in front of $\dot{x}$ is the velocity dependent friction coefficient. The expressions will not be used for computing exact values due to the assumption of laminar flow. The interesting thing to observe is which parameters that builds up the expressions. If the derivations in appendix B are compared to the detailed derivation of a more simple problem in section 3.1.3.4 one can observe the difference in assuming laminar instead of turbulent flow. The expression for turbulent flow in section 3.1.3.4 is much more complex than the laminar case which indicates that a derivation assuming turbulent flow for the whole valve is not manageable to do by hand. For this reason simulation is a suitable tool for further investigation.

\[
F_{\text{dampmain}} = \left( \frac{A_{mi}l_c + A_{mi}^2}{K_{jd} + l_p + K_{pd}} \right) \dot{x}_m + F_{fm} \text{sign}(\dot{x}_m) \tag{3.24}
\]

\[
F_{\text{damp Pilot}} = \left( \frac{A_{pl_{cp}}}{K_{pd} + K_p + l_{pp}} \right) \dot{x}_p + (F_{fp} + F_{fs} + A_{plunger}(p_4 - p_3)) \text{sign}(\dot{x}_p) \tag{3.25}
\]
Chapter 4

Analysis and measurement methods

4.1 Test equipment

4.1.1 Dynamometer

The dynamometer (shortened dyno), seen in figure 4.1a, is a test stand where the CES valve can be tested in the real application, the damper. The rig consists of a hydraulic servo system and a frame with damper attachments. Two sensors measure force and position on the cylinder. The speed of the cylinder is then derived from the position. When performing this test the CES valve is mounted in the damper. The test methods that can be performed in the dyno are ASR-, PSR- and QSR tests and current sine sweep at constant flow. The methods are explained in the next paragraphs. An advantage with the Dyno is that flow steps can be performed which mimics bumps in the road in a good way.

4.1.2 Flow bench

The flow bench is a test rig where the CES valve is tested separated from the damper. Parts of the test setup is seen in figure 4.1b. The rig consists of two displacement controlled pumps, piping system, safety valves, valve fixture and sensors. Two pumps are used to reduce pump pulsations and to enlarge the working range of the rig. A flow meter is mounted before the CES valve and pressure sensors are mounted both before and after the CES valve. In this test bench it is possible to run ASR and PQ tests. An advantage with the flow bench is that the pilot stage can be measured separately which means that the dynamics can be isolated. The test rig is further examined in [6].
4.2 Test methods

A common way to observe the dynamics of a system is by using step response. The simplest kind of step response begins with applying a step to the system. This means giving some kind of reference signal, e.g. pressure or current, that makes the system react. The system reaction (response) is then compared to the reference signal. Both reference signal and system response is plotted against time and the difference can be observed. The theoretical optimal result would give identical curves. This is though never the case since the system have some kind of dynamics.

The different test methods that can be performed at Öhlins are presented in this section.

4.2.1 Pressure-flow-curves

The pressure-flow-curve-test is shortened PQ-curves and is usually performed in the flow bench. The flow is ramped through the CES valve and the solenoid is held at a constant current. Pressure difference is plotted as a function of flow. The test is repeated for different currents. A typical test graph is shown in figure 4.2a.
4.2.2 Active step response

When using this method a constant flow is set through the CES valve and then steps in solenoid current are taken. Pressure is plotted as a function of time. A typical test graph of a current step is shown in figure 4.2b. Active Step Response is shortened ASR.

![Active Step Response Graphs](image)

(a) PQ result  (b) ASR test result

Figure 4.2: Example of test results

4.2.3 Passive step response

Passive Step Response is shortened PSR and is the inverse of ASR. A constant current level is held while doing a step in flow. The test is repeated for different constant currents and flow steps. This test is suitable to do in the dyno since a flow step is hard to achieve in the flow bench. In the flow bench, the flow meter and pumps are limiting which flow accelerations that are allowed.
Chapter 5

Modelling

In this chapter the work concerning modelling and simulation is presented. The work has been focused around three aspects. In section 5.1 the damping effect of the solenoid has been studied. In section 5.2 the pd orifice is studied. In section 5.3 old and new damping concepts are investigated. The model of the CES8700 has not been dynamically validated fully and therefore the trends, when varying parameters, are studied more closely than exact values of levels, overshoots, rise times and settling times. This is especially true for the concepts which can not be validated since they do not exist physically. The working points and simulation runs are explained in Appendix C.

5.1 The plunger model

When the friction study was made in section 3.2.1 the received value for the viscous dependent friction was found to be to small to make the simulation model stable. This is also discussed in [5]. The question then becomes: Why is the valve stable in reality? To capture what makes the valve stable a simulation model of the plunger was made to investigate if it had any effect on the damping. In Figure 5.1 the created model in AMEsim is shown. The two chamber symbols in the Figure ((A) and (B)) have variable volume and simulates the chambers on each side of the plunger as pointed out in Figure 5.1. The chamber symbols are connected through a block that calculates the viscous friction and flow through a cylindrical leakage gap (D). They are also connected to the blocks that calculates the force generated by the pressure in the chamber ((C) and (E)). A leakage block simulates the leakage over the damping plate (F).
Figure 5.1: The plunger model in AMEsim
5.1.1 Plunger in model of CES8700

In the simulation model of the CES8700 that Öhlin Racing AB has developed, viscous friction has been added in the pilot poppet mass to make the simulation model stable and to make it correspond to reality. When this viscous friction is removed the simulation model get an oscillative behaviour in certain working points shown in Figure 5.2 and 5.4 due to the fact that there is nothing that dampens the model. The oscillations have a frequency of approximately 800 Hertz. When no damping is present in the model it will oscillate as an harmonic oscillator. The flow and current signals used in the simulation are described in appendix C. The oscillations do not occur for all working points as seen in Figure 5.2 and 5.4. Figure 5.3 zooms in on the most oscillative current step. The plunger model was added and the pressure response can be seen in Figure 5.2 and 5.4. It can be seen that the high frequency oscillations are removed and that the plunger model dampens the pressure response.

Figure 5.2: Controlled pressure when performing ASR with removed viscous friction and with the added plunger model

If the viscous friction is again added in the model and then compared with the model containing the plunger and no extra viscous friction, the graphs become similar. This is shown in Figure 5.5 and 5.6. The two models become different in static pressure level which
Figure 5.3: Enlarged plot of controlled pressure when performing a current step with removed viscous friction and with the added plunger model

Figure 5.4: Controlled pressure when performing PSR with removed viscous friction and with the added plunger model
is also visible in the Figures. This can be explained by the gap that is introduced in the plunger model which connects the upper ((A) in Figure 5.1) and lower ((B) in Figure 5.1) chamber. Since there is a flow through the gap even at static levels it means that there also has to be a static pressure difference between the chambers. The pressure in the upper chamber (A) will be slightly higher and amplify the solenoid force. This will increase the force on the pilot poppet and in turn the pressure inside the main poppet (Volume (B) in Figure 5.2 on page 37). With increased pressure inside the main poppet the pressure outside (Volume (A) in Figure 5.2 on page 37) also needs to increase to maintain force equilibrium. The difference in static pressure level is shown in a Pressure-flow curve in Figure 5.7. The two graphs have somewhat different oscillative characteristics shown in 5.8. From the Figures it can be seen that the initial model (without the plunger model and with added viscous friction) seems to be more oscillative for some working points than the graph that contains the plunger model.

![Pressure response when performing ASR with viscous friction and with the added plunger model](image.png)

Figure 5.5: Pressure response when performing ASR with viscous friction and with the added plunger model
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Figure 5.6: Pressure response when performing PSR with viscous friction and with the added plunger model

Figure 5.7: Pressure-flow curve
The counter acting force that is momentarily created by the plunger when it is added in the model can be seen in Figure 5.9. The force is calculated as $F_{damp} = A_{plunger}(p_B - p_A) + F_{viscous}$ where $F_{viscous}$ is the viscous friction force created in the clearance and $A_{plunger}(p_B - p_A)$ is the pressure drop that is dynamically created. In Figure 5.9 the plunger force is compared with the current dependent solenoid friction. As seen the plunger force is larger in magnitude. Even though the forces are small they have a significant effect on the stability. This is due to the amplification from pilot stage to main stage.

In Figure 5.10 the solenoid force and controlled pressure is shown. It can be seen that the pressure follows the solenoid force with approximately a factor 4 and that this relation remains even after the current step. This is only true if the flow is kept constant.

When the plunger force shown in Figure 5.9 is plotted against the velocity of the plunger a linear behaviour appears. Figure 5.11 shows the linear relation. The slope of the curve then becomes the viscous damping coefficient. If the plot is interpolated with a straight line the slope can be approximated. Since the damping force varies with the unknown parameter eccentricity, the Figure 5.11 shows the damping force at maximum and minimum eccentricity.
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Figure 5.9: Plunger force when performing an ASR simulation

Figure 5.10: Solenoid force relation with controlled pressure
5.1.1.1 Conclusions from the simulations of the plunger model

The simulations presented in this section shows that the solenoid creates a damping force on the pilot poppet that is too large to neglect. The viscous friction that has been added to stabilize the valve in the earlier model of the CES8700 can be substituted for the plunger model. However, dynamic validation of the model is needed. The value that is reached for the viscous damping coefficient is 0.2 times the value that has been used in the older version of the simulation model. When the plunger model is used, the pressure response is somewhat less oscillative compared to the earlier model of the CES8700. The simulated flow through the solenoid creates a pressure drop and therefore the static pressure level is affected.
5.2 The pd orifice

5.2.1 The placement of the pd orifice

In already prototyped versions of the valve, the pd orifice has had another placement than the one investigated in section 6.2. In the CES2000A the pd orifice is placed in the outlet of the pilot stage as seen in Figure 5.12 instead of in front of the pilot poppet orifice as seen in Figure 6.7 in section 6.2. Since the tests that were performed in section 6.1 showed more oscillations for the two valves with the pd orifice placed after the pilot orifice it is interesting to simulate if this is the reason for the oscillations. In Figure 5.13a and 5.13c the reduced pressure that acts in the opening direction on the pilot poppet with the two pd orifice placements can be seen. In Figures 5.13b and 5.13d the different pressure levels created in the outlet chamber shown in Figure 5.12 can be seen. As seen there is a significant pressure level in the case where the pd orifice is placed after the pilot orifice. Despite these internal pressure differences the step responses in controlled pressure are very similar as seen in Figure 5.14.

Figure 5.12: Placement of the pd orifice in the CES2000A
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(a) Pilot poppet opening pressure during ASR  
(b) Pressure in outlet chamber during ASR

(c) Pilot poppet opening pressure during PSR  
(d) Pressure in outlet chamber during PSR

Figure 5.13: Pressure differences when moving the pd orifice

(a) ASR low - mid and low - high current at (b) PSR 0-low, 0-mid, 0-high flow at mid current

Figure 5.14: Controlled pressure with different pd orifice placements
5.2.2 Damping effect of the pd orifice

It is shown in tests in section 6.2 that enlarged pd orifice increases oscillations but decreases overshoots in certain working points. Also, in equation (3.22) in section 3.1.3.4 it can be seen that the orifice area appears in the expression for the viscous friction. It is therefore interesting to investigate the effect of changed dimensions of the pd orifice. In this section the effects of these changes are simulated. In section 3.1.3.4 the following expression is obtained for the dominant damping coefficient (turbulent flow is assumed): \( B_p = \left( \frac{A_p}{(K_t A_o)^2} \right). \) If this expression is used to calculate the damping coefficient of the different sizes of pd orifice that is simulated in this section the following is obtained (the values of \( B_p \) are normalized with respect to the size that is used in the CES8700):

- pd orifice flow area x 0.5 \( \rightarrow B_p = 4 \) [1]
- pd orifice flow area x 1 \( \rightarrow B_p = 1 \) [1]
- pd orifice flow area x 9 \( \rightarrow B_p = 0.012 \) [1]

Where ”pd orifice flow area x 1” is equivalent to the current size of the pd orifice and 9 is the largest dimension that can be fitted into the chosen valve configuration. The value 0.5 is used to investigate the effect of a decreased pd orifice.

In Figure 5.15a the oscillations that occur with enlarged pd orifice is shown. It can also be seen that the rise time decreases and the static pressure level increases with decreased pd orifice. In Figure 5.15b it can be seen that enlarged pd orifice reduces the overshoots and that decreased pd orifice increases overshoots. This occurs especially at low currents when the pilot poppet is fully opened. Overshoots at low currents occur due to the large motion of the main poppet which in turn means a large flow that needs to be displaced through the pd orifice. This creates a larger resistance.

If these oscillations shown in Figure 5.15a are to be reduced or eliminated, damping needs to be introduced. In Figures 5.16 the viscous friction is increased in the pilot and main poppet to investigate where damping must be introduced to reduce oscillations. As seen in the Figures viscous friction added on the pilot poppet has no positive effect on oscillations. In Figures 5.16a and 5.16c the oscillations are damped out but at the same time the overshoots and time lag are increased.
(a) ASR low - mid and low - high current at mid flow
(b) PSR 0-high flow at low current

Figure 5.15: Controlled pressure with different sizes of the pd orifice

(a) ASR low - mid and low - high current at mid flow
(b) ASR low - mid and low - high current at mid flow
(c) PSR 0-low, 0-mid, 0-high flow at mid current
(d) PSR 0-low, 0-mid, 0-high flow at mid current

Figure 5.16: Controlled pressure with added viscous friction in main and pilot poppet
5.3 Already prototyped damping concepts

One part of the thesis is to investigate already prototyped and new damping concepts. This section aims to investigate the damping characteristics of both new and already prototyped designs with help of simulation.

5.3.1 Concept 1: Damping chamber on main poppet

In the model of CES2000A a damping chamber is added on the main poppet. The chamber is marked with two dashed circles in Figure 5.17. The controlled pressure is shown in Figure 5.18 when simulating ASR. As seen the rise time increases for the damped main poppet. In Figure 5.19 the controlled pressure is shown when simulating PSR. As seen the overshoots increases with damped main poppet. The static pressure level is shown in Figure 5.20. The pressure level for the damped main poppet is slightly decreased.

(a) low-mid and low-high current at low flow  (b) low-mid and low-high current at high flow

Figure 5.18: Pressure responses during ASR
(a) 0-low, 0-mid and 0-high flow at low current

(b) 0-low, 0-mid and 0-high flow at high current

Figure 5.19: Pressure responses during PSR

Figure 5.20: Pressure-flow curve with and without main poppet damping.
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The force created by the pressure drop is shown in Figure 5.21. If the plot is interpolated with a straight line the slope can be approximated. Since the damping force varies with the unknown parameter eccentricity, Figure 5.21 shows the damping force at maximum and minimum eccentricity. From the slopes, the damping coefficients are approximated to be in the range of 400 - 1000 Ns/m.

Figure 5.21: Damping force relation to velocity

In Figure 5.22 two step responses is studied more closely. In Figure 5.22a a step in current is applied. As seen the motion of the poppet creates a pressure difference between chamber A and B ($\Delta p = p_A - p_B$). When the step is applied, the pressure is dynamically higher in chamber A and therefore creates a damping force. This force in turn reduces the acceleration of the main poppet which decreases the main poppet velocity. The rise time in pressure is then decreased. In Figure 5.22b a step in flow is applied. As seen the pressure difference is negative when the step is applied. This is due to the pressure in chamber B increases above the pressure in chamber A. The main poppet again reaches a lower velocity which results in an overshoot in controlled pressure.
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(a) low - high current at mid flow

(b) 0-mid flow at high current

Figure 5.22: Responses in pressure difference, controlled pressure and main poppet position
5.3.2 Concept 2: Damped pilot poppet

In the model of CES4600 a damping chamber is added on the pilot poppet. The chamber is marked with a dashed circle in Figure 5.23a. When modelling the damping concept in the model of the CES8700, which has a different design, compromises have been made in order to investigate the damping contribution from the damping chamber. The damping chamber has been modelled as picture 5.23b. The two pressure lines (A) are connected to the solenoid rod volume making the damping chamber piston statically pressure relieved since the pressure in A and B will be statically of the same magnitude. This separates the damping chamber from being part of the regulating pilot area. By modelling in this way the damping effect of the damping chamber can be isolated and implemented in the model of CES8700. B simulates the volume in the damping chamber and C distributes the force and position to the pilot stage components.

(a) Pilot poppet damping chamber  
(b) Damping chamber in AMEsim

Figure 5.23: Pilot poppet damping chamber

In Figure 5.24 pressure responses can be seen. ASR simulations are shown in Figures 5.24a and 5.24b. It can be seen that the rise time increases. More interesting is the large oscillations occurring at especially medium currents.

PSR simulations can be seen in Figures 5.24c and 5.24d. Overshoots increase at low currents and oscillations increases for medium currents.
In Figure 5.25 a current step is studied more closely. The Figure shows the damping force that is dynamically created by the pilot damping chamber. The force is calculated according to $F_{\text{damp}} = (p_B - p_A)A_{\text{damp piston}} + F_{\text{viscous}}$ where $F_{\text{viscous}}$ is the viscous friction force created by the shearing of the fluid layers in the clearances. $(p_B - p_A)A_{\text{plunger}} >> F_{\text{viscous}}$. As seen, the pilot poppet gains a larger velocity when undamped. In turn the controlled pressure rises faster in the undamped case. The cause for the oscillations are further described in section 7.1.1.3.

The force created by the pressure drop is shown in Figure 5.26. If the plot is interpolated with a straight line the slope can be approximated. Since the damping force varies with the unknown parameter eccentricity, the Figure 5.21 shows the damping force at maximum and minimum eccentricity. From the slopes, the damping coefficients are approximated to be in the range of 225 - 555 Ns/m.
Figure 5.25: low-high current at mid flow

Figure 5.26: Damping force relation to velocity
5.4 New damping concepts

In this section new damping concepts are presented and simulated. The concepts are divided in pilot stage damping and main stage damping. The information regarding concepts 3, 5, 6, 7, 8, and 10 has been reduced due to confidentiality.

5.4.1 Pilot stage damping

5.4.1.1 Concept 3

This concept aims to add damping in the pilot stage. The concept cannot be described in detail due to confidentiality. The pressure responses when varying a design parameter is shown in Figures 5.27 and 5.28. As seen in the figures the pressure level and oscillations increases with decreased design parameter. In Figure 5.29 pressure level is shown in a pressure flow curve.

![Figure 5.27: Pressure response. low - mid and low - high current at high flow](image)

If the flow in the solenoid rod is redirected, the pressure level is not affected. The step responses in pressure is shown in Figures 5.30 and 5.31 and the static pressure level is shown in Figure 5.32. As seen, the dynamic behaviour is similar to Figures 5.27 and 5.28 but the static pressure level is unaffected by the decreased design parameter. It can also be seen that the most beneficial dynamic behaviour is reached with a design parameter between 0.7 and
0.4. At design parameters lower than 0.4 the oscillations in controlled pressure increases. The frequency is of approximately 60 Hertz. (Figure 5.27).

In Figure 5.33 the damping force is seen for different design parameters at an increasing current step. A positive force is defined as the pilot opening direction. What happens first in Figure 5.33a is a force peak that increases with decreased design parameter. This is the damping force that is dynamically created. After the peak a negative force is seen that reaches a steady level until that current step is lowered. This negative force amplifies the solenoid closing force and in turn increases the static pressure level for the whole valve. In Figure 5.33b a similar graph is shown for the case when the flow is redirected. As can be seen, a damping pressure peak initiates the graph. A big difference from Figure 5.33a is that there is no negative force that reaches a steady level. This is the reason for the unaffected static pressure level.
Figure 5.29: Pressure-Flow graph. The figure shows three different currents.

Figure 5.30: Pressure response. low - mid and low - high current at high flow. The flow redirected.
Figure 5.31: Pressure response 0- low, 0 - mid and 0 - high flow at mid current. The flow redirected.

Figure 5.32: Pressure-flow curve. The flow redirected.
(a) low - high current at mid flow

(b) low - high current at mid flow (Flow redirected)

Figure 5.33: Force acting on plunger
5.4.1.2 Concept 4: Orifice in solenoid rod

This concept aims to add damping to the pilot stage without affecting the pressure level. This concept eliminates the flow from the top of the solenoid inside the plunger. The flow path is shown in Figure 5.34 (B). An orifice is added in the top of the solenoid rod (A). When the pilot and solenoid rod moves the volume surrounding the orifice must be displaced. This creates a resistance and in turn a damping force without affecting the leakage circuit. In Figures 5.35 and 5.36 the pressure responses is shown for three different orifice diameters. As seen in Figure 5.35 oscillations and overshoots increase for large current steps when diameter 0.1 mm is used. In Figure 5.36 overshoots and oscillations increase with diameter 0.1 mm for medium and high currents. For low currents the overshoot does not increase since the pilot poppet is not closed. As seen in Figure 5.37 the pressure levels are identical.

Figure 5.34: Damping orifice in solenoid rod

(a) low - mid and low - high current at low flow (b) low - mid and low - high current at high flow

Figure 5.35: Pressure response during ASR
(a) 0 - low, 0 - mid and 0 - high flow at low current

(b) 0 - low, 0 - mid and 0 - high flow at high current

Figure 5.36: Pressure response during PSR

Figure 5.37: Pressure-flow curve with varied orifice diameter
In Figure 5.38 a current step is studied more closely. In the figure, the pressure in chamber A (Figure 5.34) is shown as well as the flow through the rod orifice and the pilot poppet position. The flow through the rod orifice decreases with decreased rod orifice diameter despite that the pilot poppet moves the same distance. The explanation for the decreased flow is the volume in chamber A that compresses, (or decompresses depending on pilot poppet moving direction) leading to a pressure difference over the rod orifice. This change in pressure creates a damping force that counteracts the direction of motion.

Figure 5.38: low - mid current at high flow
5.4.1.3 Concept 5

This concept investigates the importance of pilot poppet velocity when it comes to increasing the pressure level by increased current. The concept is not described in detail due to confidentiality. As seen in Figure 5.39a there is no significant change in the dynamic behaviour. If the current step from low to high (Figure 5.39b) is enlarged, a very small improvement in rise time can be seen. However it is very small and the significance can be questioned.

![Concept 5 Reference valve](image)

(a) low - mid and low - high current at mid flow          (b) Zoom on low - high current

Figure 5.39: ASR with and without the damping concept

In Figure 5.40 the pilot position and controlled pressure is shown when zooming on the current step low - high current. As seen, the pilot poppet closes faster with the investigated concept. However, if observing controlled pressure at the same time it can be seen that the pressure is not affected much by the fact that the pilot moves faster. This implies that the pressure change at a current step is not only set by the speed of the pilot poppet. The pilot poppet is much faster than the pressure build up itself and is therefore not the limiting factor.
Figure 5.40: Pilot poppet position and controlled pressure during a current step
5.4.2 Main poppet damping

If a damping concept is to be successful it needs to improve the valve dynamics in the CES8700. The concept also needs to reach four goals that are often contradicting: low overshoots, fast rise time, short settling time and stabilizing the valve. From the test results in section 6.2 and the simulations in section 5.2.2 it can be seen that enlarged pd orifice decreases overshoots in controlled pressure when performing PSR but increases oscillations. This results in wanting a low damping at flow steps but still to damp out the oscillations occurring in ASR. From the simulations in section 5.2.2 it can also be seen that the damping needs to be added in the main poppet for the oscillations to be damped out. Ideally the damping is also separated from the leakage circuit to not affect the static pressure level.

5.4.2.1 Concept 6

The aim of this concept is to have a damping of the main poppet. In turn the pd orifice can be increased. The intent is to decrease the pressure peaks at flow steps and that the rise time at current steps would remain fast. The concept is not described in detail due to confidentiality. In Figure 5.41, step responses in pressure are shown. In the figures the damping concept has been simulated with three different values for a design parameter. In the figures, a model with enlarged pd orifice and no further changes is shown as well as a reference model. The value 0 for the design parameter corresponds concept 1. In Figure 5.41a, it can be seen that the rise time increases when using the damping concept. The rise time increases for two reasons: The pressure build up will be slower. It is also shown in section 5.2.2 that increased pd orifice increases the rise time. The oscillations that occur when increasing the pd orifice remains when using a design parameter = 0.3. In Figures 5.41b, 5.41c and 5.41d PSR is simulated. For the case where the design parameter = 0.3 the overshoots are decreased in most cases but the oscillations at medium currents remains. When a design parameter = 0.1 is used the overshoots increase and are often larger than the overshoots of the reference valve.
(a) ASR low - mid and low - high current at high flow

(b) PSR 0-high flow at low current

(c) PSR 0-high flow at mid current

(d) PSR 0-high flow at high current

(e) Pressure-flow curve

Figure 5.41: Controlled pressure with and without the concept.
In Figure 5.42 a current step is studied more closely. In the figure the damping concept is simulated with different sizes of the design parameter, where 0 corresponds to concept 1. The pressure drop between is what creates the damping force. As can be seen it increases with decreased design parameter which in turn reduces the oscillations in controlled pressure.

Figure 5.42: low - current current at mid flow
5.4.2.2 Concept 7

The aim of this concept is to have a damping on the main poppet but to still be able to reduce overshoots in pressure. The concept is not described in detail due to confidentiality. There is an obvious drawback of the concept since only the overshoots corresponding to the highest pressure level can be eliminated. This is due to a design parameter. If the design parameter is set to a lower value, that value will be the limiting factor for the static pressure level for the whole valve. In Figure 5.43a an example is shown where the design parameter is set so that the overshoots are reduced for medium currents. If only looking at this level it looks like a success. In Figure 5.43b on the other hand it can be seen that all flow steps at higher currents only can have a low static pressure level. This is also seen in a flow pressure curve in Figure 5.44.

![Concept 7 and Reference Valve](image)

(a) 0 - low, 0 - mid, and 0 - high flow at mid current  
(b) 0 - low, 0 - mid, and 0 - high flow at high current

Figure 5.43: PSR with the design parameter set to medium

For this concept to be functional in the whole working range, the design parameter needs to be tuned. With this done and the pd orifice diameter increased, the step responses looks like Figure 5.45. As seen in 5.45a the overshoot is eliminated for the highest current step. On the other hand the rise time is increased which is not beneficial. In Figure 5.45b decreased overshoots for the damping concept due to the increased pd orifice can be seen. In Figure 5.45c no significant difference can be seen. In Figure 5.45d it can be seen that the overshoot for the highest flow step is eliminated.
Figure 5.44: Pressure-flow curve with the design parameter set to medium

Figure 5.45: Step response in pressure with and without the damping concept.
5.4.2.3 Concept 8

The aim of this concept is to obtain a damping of the main poppet. The concept is not described in detail due to confidentiality. The complete concept includes an increased pd orifice (largest possible in the simulated configuration of the CES8700).

In Figure 5.47 the step responses in controlled pressure is shown. The figure includes the modelled damping concept, a reference model and the model of concept 1. As seen in ASR simulations in Figures 5.50a, 5.50b and 5.50c a small time lag is introduced when increasing the current and overshoots are decreased for high currents. The lag occurs due to the slow pressure build up combined with a large pd orifice. In Figures 5.50d, 5.50e and 5.50f it can be seen that the overshoots are decreased for flow steps. A draw back is that a undershoot occurs in Figure 5.50e. This undershoot occurs since the main poppet gets an overshoot in position and then moves to slowly when changing direction to reach the static level.

![Pressure-flow graph with and without the damping concept](image.png)
Chapter 5. Modelling

(a) ASR low-mid, low-high current at low flow
(b) ASR low-mid, low-high current at mid flow
(c) ASR low-mid, low-high current at high flow
(d) PSR 0-high flow at low current
(e) PSR 0-high flow at mid current
(f) PSR 0-high flow at high current

Figure 5.47: Step response in pressure with and without the damping concept.
5.4.2.4 Concept 9: Large volume in main poppet chamber

This concept investigates the effect of a large volume behind the main poppet. The idea behind the concept is to let the large volume compress more easily due to reduced hydraulic stiffness and therefore decrease the overshoots at flow steps. Figure 5.49d shows step responses in pressure with and without enlarged volume. In Figures 5.49a and 5.49b it can be seen that the lag increases but that overshoots and oscillations are reduced. The lag increases do to that the pressure gradient decreases with increased volume. In Figures 5.49c and 5.49d it can be seen that overshoots are drastically decreased and that oscillations disappears. This is the effect of the reduced hydraulic stiffness that smoothes out the pressure peaks.

Figure 5.48: Principle sketch of the large volume placement.

Figure 5.49: Step response in pressure with and without large damping chamber on main poppet.
Chapter 5. Modelling

5.4.2.5 Concept 10

Since all damping concepts that have been presented have increased the rise time when increasing the pressure level in ASR, this concept has been investigated. With this concept, the damping can be tuned within a chosen interval. The damping concept is not described in detail due to confidentiality. As seen in Figure 5.50 all step responses have similar behaviour as for the concept presented in section 5.4.2.3 with one difference: the rise time is decreased. However, the rise time is not decreased compared to the reference model. This is due to the enlarged pd orifice, which increases the rise time. In Figure 5.50d the graphs describing the concept in this section and concept 8 are printed on top of each other since the pressure responses are identical.

Figure 5.50: Step response in pressure with and without the damping concept
In Figure 5.51 a step response in current is studied more closely. The figure examines the difference between the concept presented in this section and concept 8. In the figure the pressure drop that creates the damping force is shown. As seen the rise time is decreased for concept 10.

Figure 5.51: low - high current at mid flow
Chapter 6

Empirical data

6.1 Test of already prototyped damping concepts

In this chapter the results from measurements are presented. ASR and PSR tests has been performed on the CES2000, CES4300A and the CES4600 to compare their damping properties. The CES2000 has damping on the main poppet while the CES4600 has damping on the pilot poppet. The CES4343A has no specific damping concept. The measurements on the CES87005541K are not made specifically for this report but is shown as a reference. Since the valves reaches different static pressure levels, the rise time is not comparable. Instead the pressure gradient is calculated, which is the mean slope from 10% to 90 % of the step. When performing measurements in the dyno, the measured entities are force and velocity. To make these measurements comparable with flow bench measurements and simulations the forces and velocities are converted to pressure and flow by using the damper geometry. In each section the graphs from measurements are shown as well as a table that shows data from the graphs. Since the time has been limited for the thesis work, only two valves of each has been tested. To draw better conclusions, it would be beneficial to perform further testing to investigate the deviations between individuals.
6.1.1 Tests performed in dynamometer

6.1.1.1 Active step response

Figure 6.1: ASR tests 0.3 - 0.7 [A]

Figure 6.2: ASR tests 0.3 - 1.1 [A]
Table 6.1: Active step responses measured in dyno

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<tr>
<th>Valve type</th>
<th>Step</th>
<th>Overshoot [%]</th>
<th>Gradient [bar/ms]</th>
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<td>44.77</td>
<td>3.03</td>
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<td>23.84</td>
<td>7.29</td>
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6.1.1.2 Comments on table 6.1

In table 6.1 it can be seen that the gradient increases with increased current and flow. There is not a major difference in gradient magnitude between the valve models. Overshoots are often largest at a step from 0.3 to 0.7 A at 18.32 l/min and smallest at a step from 0.3 to 1.1 A at 6.1 l/min. The valve that stands out from these statements the most is the CES8700.
6.1.1.3 Passive step response

Figure 6.3: PSR tests 0 - 0.5 [m/s]

(a) 0.38 [A]  
(b) 0.5 [A]

Figure 6.4: PSR tests 0 - 0.5 [m/s]

(a) 0.7 [A]  
(b) 1 [A]
Table 6.2: Passive step responses measured in dyno

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<tr>
<th>Valve type</th>
<th>Step</th>
<th>Overshoot [%]</th>
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</table>

6.1.1.4 Comments on table 6.2

In table 6.2 values for overshoots are shown. Gradients are not calculated since rise time has not been the focus in PSR in this report. Overshoots decreases with increased current for all valves. The CES2000A and CES4600G020K has the highest overshoots, especially at low currents. The CES4343 has the smallest difference between largest and smallest overshoot whereas the CES2000A has the largest.
6.1.2 Tests performed in flow bench

Figure 6.5: ASR tests 0.29 - 0.9 [A]

Figure 6.6: ASR tests 0.29 - 1.6 [A]
Table 6.3: Active step responses measured in flow bench

<table>
<thead>
<tr>
<th>Valve type</th>
<th>Step</th>
<th>Overshoot [%]</th>
<th>Gradient [bar/ms]</th>
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<td>CES4600G020K</td>
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<td>1.49</td>
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6.1.2.1 Comments on table 6.3

In table 6.3 values for overshoots and gradients are shown. The gradients follows the same pattern as in table 6.1. They are however smaller in magnitude which is explained by the larger volume in the flow bench. The largest overshoot occur at a step from 0.29 to 0.9 at 20 l/min. The overshoots decrease with increased current.
6.2 The pd orifice

In Figure 6.7 the location of the pd orifice is shown in the latest valve version, CES8700. The pd orifice creates a pressure drop in the pilot flow path. According to an internal report [15] at Öhlins Racing AB the pd orifice has a great influence on the valve dynamics. The report contains test data regarding the CES8700 with and without the pd orifice. In Figure 6.8 two current steps is shown where the valve without the pd orifice is named CES8700N5541K57 and the valve with the pd orifice is named CES8700N5541K. As seen, the CES8700N5541K57 is highly oscillative compared to CES8700N5541K. It can also be seen that the rise time increases when the pd orifice is enlarged. In Figure 6.8 two flow steps at different currents are shown. One can see in Figure 6.9a that the overshoot is highly reduced for the CES8700N5541K57 at a low solenoid current. In Figure 6.9b it can be seen that the valve get an oscillative behaviour when the solenoid current is increased. Also the overshoots is now similar in size between the two valves as seen in Figure 6.9b.

![Figure 6.7: Placement of the pd orifice in the CES8700](image-url)
Chapter 6. Empirical data

(a) Current step 0.3 to 0.7 [A] at 0.3 [m/s]  
(b) Current step 0.3 to 1.1 [A] at 0.3 [m/s]

Figure 6.8: Current steps with and without the pd orifice

(a) Flow step -0.1 to 0.5 [m/s] at 0.3 [A]  
(b) Flow step -0.1 to 0.5 [m/s] at 0.7 [A]

Figure 6.9: Flow steps with and without the pd orifice
Chapter 7

Result

7.1 Damping in a two stage valve

7.1.1 The step response

When it comes to investigating step responses in the CES8700 one soon discovers that there is a significant difference in flow and current steps and whether a specific component or the complete valve is considered. In Figure 7.1 examples of step responses is shown. The figure shows the pressure response during ASR and PSR simulations with different currents and flows. The interesting thing to observe is the different dynamics when comparing the current and the flow step. In Figure 7.1a long rise time and no significant overshoot can be seen when simulating a low flow. In Figure 7.1c on the other hand fast rise time and large overshoot can be seen. In Figures 7.1b and 7.1d the differences is not as apparent but still it can be seen that the rise time is larger for Figure 7.1b than for Figure 7.1d.

When the flow step is applied and the main poppet is forced to open, there will be a pressure build up in front of the main poppet (\(P_1\) in Figure 3.11a section 3.2.2) due to the fact that the poppet is not infinitely fast. This is due to the volume behind the poppet (\(V_2\) in Figure 3.11a section 3.2.2) that must be evacuated. When the current step is applied the pressure changes inside the main poppet (\(V_2\) in Figure 3.11a section 3.2.2) and the poppet needs to be accelerated in the closing direction. Now the resistance acts in the opposite direction than before, meaning that the pressure in front of the main poppet (\(P_1\) in Figure 3.11a section 3.2.2) will be increased more slowly.

These difference in response creates difficulties when it comes to measuring the damping from pressure response.

For these reasons the two step response methods needs to be separated and studied separately and can not be mixed when it comes to measuring the damping. Also, damping of a separate component in the valve needs to be separated from damping in pressure response.
Chapter 7. Result

7.1.1.1 Damping effect of the pilot shim

Two current steps are studied more closely in Figure 7.3. The current steps are simulated with two different pilot shims. The pilot shim decides which solenoid force that is required to fully close the pilot orifice (to go from bleed to pressure regulator). In Figures 7.3a and 7.3b the pilot position (where 0 means closed) and controlled pressure is shown. In order to make the two shims comparable the pilot seat have been tuned to match the static pressure levels.

Figure 7.1: Examples of step responses

Figure 7.2: Sketch of the pilot shim in CES8700
In Figure 7.3a it can be seen that when a weak shim (low spring coefficient) is used, the pilot orifice closes when the current step is applied. When the pilot orifice is closed the pressure builds up quickly which results in fast rise time, overshoot and oscillations. If the stiff shim (high spring coefficient) is studied it can be seen that the pilot orifice does not close and in turn the pressure builds up slowly. This results in higher rise time and no overshoot or oscillations. Figure 7.3b shows a current step large enough to close the pilot orifice independently of pilot shim. This time the step responses are similar.

In Figure 7.4 the pd orifice has been enlarged to create an oscillative step response. However, when the stiff pilot shim is used the oscillations disappear. These simulations show that the pilot shim stiffness is important for the valve dynamics and the damping of the pressure response. By increasing the pilot shim stiffness the pressure response can be stabilized. The pilot seat diameter can then be used to tune the static pressure level.

(a) low - mid current at high flow. Stiff shim opened and weak shim closed. (b) low - high current at high flow. Both shims opened and weak shim closed.

Figure 7.3: Two current steps with two different pilot shims
This section aims to investigate the effect of increasing the viscous friction and in turn the
damping in the main stage. The simulation program LMS AMEsim has the possibility to
add viscous friction to a mass without having a specific damping design. The step responses
studied in this section are according to appendix C. Figure 7.5 shows the pressure response
when increasing the viscous friction on the main poppet.

Figures 7.5a and 7.5b shows four current step. As seen the time lag is increased with
increased viscous friction.

Figures 7.5c and 7.5d shows six flow step. As seen the overshoot increases and the time
lag is not affected. The overshoots are larger at low currents since a larger volume needs to
be displaced from the main poppet backside due to a large motion of the main poppet. The
oscillations and overshoots increase at high currents and large flow steps. The reason for the
smaller overshoots on high currents is that the main poppet moves a smaller distances and
also reaches a lower velocity in order to reach the final level and. The damping force on the
main poppet is then lower due to the lower velocity according to equation (3.22) in section
3.1.3.4. In Figure 7.6 a flow step is studied more closely. In the Figure controlled pressure
is shown as well as main poppet position. One can see that with increased viscous friction
the main poppet moves more slowly and therefore the pressure builds higher. This results
in a higher overshoot than with no viscous friction on the main poppet.
(a) ASR low - mid and low - high current and constant low flow
(b) ASR low - mid and low - high current and constant high flow
(c) PSR 0-low, 0-mid, 0-high flow and constant low current
(d) PSR 0-low, 0-mid, 0-high flow and constant high current

Figure 7.5: Pressure responses when changing the viscous friction in the main stage
Figure 7.6: PSR 0 - high flow at low current
7.1.1.3 Viscous friction in pilot stage

This section aims to investigate the effect of increasing the viscous friction and in turn the damping in the pilot stage. The step responses studied in this section are according to appendix C. Figure 7.7 shows the pressure response when increasing the viscous friction on the pilot stage.

(a) ASR low - mid and low - high current and constant low flow
(b) ASR low - mid and low - high and constant high flow
(c) PSR 0-low, 0-mid, 0-high flow and constant low current
(d) PSR 0-low, 0-mid, 0-high flow and constant high current

Figure 7.7: Pressure responses when changing the viscous friction in the pilot stage

Figure 7.7a remains unchanged with increased viscous friction. When the flow is increased in Figure 7.7b the oscillations increase with increased viscous friction. There is a small (relative the damped main poppet) overshoot in Figure 7.7c. In Figure 7.7d, the overshoot and oscillations increase with increased flow step. The rise time in the PSR simulations are unaffected.

To understand why the oscillations increase with increased viscous friction on the pilot poppet (fore some working points), a current step is studied more closely in Figure 7.8. If the graph with low viscous friction is studied one can see that the maximum pressure (1)
occurs at the same time as the maximum pilot opening (2) occurs. This means that as a response to the pressure build up, the pilot poppet opens with no significant lag. If the graph with high viscous friction is studied one can see that when the maximum pressure is reached (3) the pilot poppet has not reached its maximum opening yet (4). When the pressure has decreased approximately 20 bars (5) the pilot poppet reaches its maximum opening (6). In other words a phase shift is introduced between pilot position and pressure build up. When pilot position and pressure build up lags in this way they will counteract each other and continue oscillating. By increasing the viscous friction on the pilot poppet, the response to a pressure build up is slowed down i.e the bandwidth of the pilot poppet has decreased below the bandwidth of the main stage. This results in lowered resonance frequency for the whole valve.

![Graph showing pressure and pilot poppet position over time]

Figure 7.8: ASR low - mid current at high flow
7.1.2 Dry friction

7.1.2.1 Main poppet

One way to measure the dry friction on the main poppet is to observe the hysteresis when performing a pre stroke test. This test is done when assembling the valve. A pre stroke test is shown in Figure 7.9. From the plot it is possible to observe the dry friction by dividing the vertical distance between the curves by 2. From the plot it is clear that this vertical distance deviates throughout the stroke and if several of these plots are examined it is clear that the deviations are large. It is therefore hard to receive a constant value for the dry friction. If friction is added in the simulation model that is in the same size range as in the pre stroke graph it is hardly noticeable as seen in Figure 7.10. This is due to the friction force being extremely small in comparison to the forces acting on the main poppet due to the surrounding pressures.

Figure 7.9: Pre stroke test example
Chapter 7. Result

7.1.2.2 Pilot stage

In the current model of the CES8700 dry friction is modelled as a function of current due to the unsymmetrical design of the solenoid plunger. This friction has been measured and added to the model as the only dry friction. In section 3.1.2.1 coulomb friction is said to have a damping but not stabilizing effect. In Figure 7.11 step responses in pressure is shown with and without dry friction. One can clearly see that the dry friction introduces more overshoots and oscillations for all simulations. Using the knowledge from section 7.1.1.3 the step response for an over damped pilot stage should, for some working points, result in higher overshoot and more oscillations. If this definition is used together with Figure 7.11 it seems that added dry friction increases the damping in the pilot stage resulting in increased oscillation in pressure.

(a) ASR low - mid current at mid flow
(b) PSR 0 - mid flow at mid current

Figure 7.10: Simulation with and without dry friction in the main poppet

(a) ASR low - mid, low - high current at high flow
(b) PSR 0 - low, 0 - mid, 0 - high flow at mid current

Figure 7.11: Simulation with and without dry friction in the pilot poppet
Chapter 8

Conclusions

8.1 Conclusions based on simulation

The base model that have been used in the simulations has been statically validated at Öhlins. However, the model has not been fully dynamically validated against step responses. For this reason the results and conclusions based on the models created in this thesis are focused on trends and comparisons to the reference valve model (created at Öhlins). Exact levels and values are not considered. Instead the overall characteristics is observed.

- If a damping mechanism is introduced in a leakage path, the static pressure level is affected. This is shown in sections 5.1.1 5.2.2 5.3.1 5.4.1.1 5.4.2

- Damping of a separate component does not have to mean damped valve behaviour. This is shown in sections 5.1.1 5.2.2 5.4.1.1 5.3.2 5.4.1.2 and 7.1.1.3

- The damping created by the the solenoid is crucial for the valve stability. This is shown in section 5.1.1

- When damping the pilot stage, it has no major significance where in the pilot stage damping is introduced. The valve behaviour becomes similar for all pilot damping concepts. This is shown in sections 5.4.1.1 5.3.2 5.4.1.2 and 7.1.1.3

- Increased damping in the pilot stage does not improve the dynamic behaviour of the valve. This is shown in 5.4.1.1 5.3.2 5.4.1.2 and 7.1.1.3

- Decreased pd orifice decreases rise time in ASR but increases overshoots in PSR. This is shown in section 5.2.2

- Current dependent dry friction in the pilot stage introduce more oscillations, both in ASR and PSR simulations. This is shown in section 7.1.2.2
• It has no major dynamic impact if the pd orifice is placed in the pilot stage outlet (after the pilot orifice) instead of the placement in CES8700 (before the pilot orifice). This is shown in section 5.2.1.

• Physical measurements of damping in separate components are not recommended. This would demand a large number of sensors in a very small space, which would have to be mounted without affecting the valve geometry. Since the damping differs with working point and configuration, all configurations would need to be measured "in action". This would make physical measurements costly and hard to analyse.

• Minimising overshoots often conflicts with stabilizing the valve since low overshoots is benefiting from low damping. This is shown in 5.2.2 and 6.2.

• A stiff pilot shim reduces oscillations in controlled pressure more than a weak pilot shim. This is shown in section 7.1.1.1.

8.2 Damping concept summary

In this section the damping concepts presented in sections 5.3 and 5.4 are summed up and rated. The scores used in the tables are not calculated from data. The score is an indicator to show if the concept has improved or impaired the valve performance compared to the reference valve model. In the tables the concepts are rated in the categories static pressure level, overshoot, rise time and settling time for all simulated working points.

The following scores are used:

1. Impaired performance compared to the reference valve.
2. Slightly impaired performance compared to the reference valve.
3. Similar performance compared to the reference valve.
4. Slightly improved performance compared to the reference valve.
5. Significantly improved performance compared to the reference valve.

The static pressure level will be graded from 1 to 3 since a changed pressure level is defined as undesirable. 3 denotes an unchanged pressure level, 2 slightly affected and 1 is significantly affected.

The concept that only has grades between 3 and 5 is concept 5. The concepts that have grades between 2 and 5 are concepts 5, 9 and 10. The concepts that reduces the overshoots the most is concepts 8, 9 and 10. Only concept 5 reduces the rise time.
<table>
<thead>
<tr>
<th>Concept 1</th>
<th>Concept 2</th>
<th>Concept 3</th>
<th>Concept 4</th>
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Figure 8.1: Table of concept ratings in Active Step Response

Chapter 8. Conclusions
## Passive step response

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Figure 8.2: Table of concept ratings in Passive Step Response
Chapter 9

Discussion

9.1 Optimal valve damping

From the result there is a few criteria that characterize the optimal damping. First the optimal design is when the damping mechanism is separated from the leakage circuit. With this criteria satisfied one can tune the damping without affecting the static pressure level of the valve. Also, to optimize the damping three different aspects of the step response needs to be taken care of: overshoots, rise time and settling time. To eliminate overshoots in pressure when performing flow steps, the main poppet needs to be able to open quickly. But in between the rapid flow steps a slow motion of the main poppet can be beneficial since it gives a comfortable ride in the car. In the other case, when performing current steps, the main poppet should again be able to move quickly in order to reach the desired pressure level as fast as possible (minimize rise time). A more complex behaviour is the settling time which is particularly long for medium currents. This behaviour is possibly related to the valve resonance frequency and might benefit from another working strategy than to add damping. Instead of adding damping to reduce these oscillations (which is one solution, but it have negative effects on rise time and overshoot in all working points) one should aim for changing the valve resonances frequency by other means. Other means can be e.g increasing or decreasing chamber volumes and masses of the moving parts.

In addition there should also be a suitable damping of the pilot poppet. From the simulations in 7.1.3 and 5.1.1 it seems that the valve behaviour is not improved when increasing or decreasing the viscous friction. With decreased damping the valve get an oscillative or even unstable behaviour due to the fact that no damping is present. When increasing the damping, the valve also get an oscillative behavior. This time the reason is a phase shift between pressure build up and pilot poppet position. In other words the bandwidth of the pilot poppet is decreased below the bandwidth of the main stage. In addition it can be seen in section 7.1.2.2 that the current dependent friction increases oscillations. The positive
With this in mind it seems that the damping magnitude of today is a suitable pilot damping except for the current dependent dry friction in the solenoid. The damping should be kept as small as possible but large enough to prevent that the pilot poppet oscillates freely.

9.2 Simulations vs empirical tests

The aim of this report has not concerned validation of simulations models. However, empirical testing has been performed in order to compare the dynamic characteristics of the four main generations of the CES valve. These results can only partly be compared to the simulations performed since the simulations have always had the CES8700 as a base, whereas the empirical tests has compared four completely different valve designs. A few things are observed:

- Overshoots increase in PSR when valve damping is introduced.
- Rise time increases with increased pd orifice in both empirical tests and simulations.
- Rise time, when performing ASR, is not comparable between empirical tests and simulations since the tested valves reaches different static pressure levels. However, the initial lag that occurs when increasing the damping in simulations can also be seen in the test performed in dynamometer.

9.3 Simulation

9.3.1 Damping concepts

To meet the demands on damping that have been presented in the report and are summarised in section 9.1 a number of damping concepts have been simulated. As all the demands on optimal valve damping are hard to satisfy, compromises have been made in the damping concepts. Concepts 3,5,6,7,8 and 10 is not discussed in detail due to confidentiality.

9.3.1.1 Main poppet damping concepts

- **Concept 1: Damping chamber on main poppet**
  This concept belongs to the first generation of CES valves and has therefore all ready been prototyped. The simulations of the concept clearly shows that rise time increases in ASR and overshoots increases in PSR. The simulations give similar results as for the empirical tests performed in chapter 6.
• **Concept 6**
  This concept showed that this kind of damping does not solve the oscillation problem when increasing the pd orifice. The concept had a too low damping at mid currents.

• **Concept 7**
  This concept shows ideal characteristics for a certain working point when it comes to eliminating overshoots. The obvious drawback is that a design parameter decides which overshoots that can be eliminated. This results in only eliminating overshoots that are higher than the design parameter value. In the other working points it has the same drawbacks as concept 1. The exception from concept 1 is that the pd orifice can be increased and therefore the overshoots at low currents are still reduced.

• **Concept 8**
  This concept shows good characteristics when it comes to reducing overshoots and at the same time stabilizing the valve. The concept also proves that this kind of damping is enough to eliminate the oscillations caused by enlarged pd orifice. A drawback is that the rise time for ASR increases. Another drawback is the undershoot that occurs for PSR at mid currents.

• **Concept 9: Large volume in main poppet damping chamber**
  This concept investigates the effect of introducing a large volume in the damping chamber. The concept uses the oils ability to compress as a spring and absorbs in this way overshoots in controlled pressure. PSR simulations show very good properties in terms of oscillations and overshoots. In ASR simulations the concept shows a slower rise time.

• **Concept 10**
  This concept is a more complicated variant of Concept 8 and shows similar characteristics. The difference is that the damping can be tuned within a chosen interval. The rise time is improved compared to concept 8 but still impaired compared to the reference valve.

### 9.3.1.2 Pilot poppet damping concepts

Even though one result of this report implies that no further damping is needed in the pilot stage, it is still part of the thesis to investigate new pilot poppet damping concepts. Both the concepts show clearly that increased damping affects the pressure response in a negative way.

• **Concept 2: Damped pilot poppet**
  This concept does not improve the dynamic behaviour in any way when implemented
in the model of CES8700. Since there is already a significant damping in the solenoid in the model of CES8700 the added model of the damping chamber increases the damping above the level of what is needed. In the CES4600 (where the pilot damping chamber was used) the damping in the solenoid could vary heavily due to gas mixed into the oil and therefore the damping concept was needed. The concept has a complex design compared to concept 3 and 4 and is therefore not recommended for future use.

- **Concept 3**
  This concept shows good possibilities to add damping without affecting the pressure level. When a large design parameter is used the valve oscillates with a high frequency since the damping is to low. With decreased design parameter the step response gets more and more oscillative (especially for mid currents) due to large damping.

- **Concept 4: Orifice in solenoid rod**
  This also is a good way of adding damping without affecting the pressure level. The dynamic behaviour is very similar to the "Plate on plunger"-concept. An orifice diameter around 0.1 mm starts to affect the pressure response negatively.

- **Concept 5**
  This concept increases the speed of the pilot poppet when increasing the pressure level. The concept does not affect the static pressure level. However, the rise time is not affected much which shows that rise time is more dependent on the pressure gradient behind the main poppet. The concept shows that to decrease rise time, other measures than increasing pilot poppet speed needs to be considered.

### 9.3.2 Measurement of damping

When it comes to measuring the damping it is important to separate damping in pressure response and damping of a component. The report clearly shows that increased damping of a component can decrease the damping of the pressure response. If comparing empirical step responses to simulated, one can conclude that the reality is not always as ideal as the simulation environment. Since the measurement methods have to work for both empirical tests and for simulation, and since a certain pressure response behaviour can not be derived to damping of a certain component, all pressure responses needs to be handled in the same way. From simulation one can distinguish certain behaviours that occur for a specific valve configuration and working points. However, these behaviours are not always visible in the empirical tests. Therefore the measurement method needs to be valid in the whole working range. I.e it is not possible to couple precisely a pressure response behaviour to a certain damping, even if the working point and configuration is known.

Two things can however be be stated:
• Rise time increases with increased valve damping in ASR tests and simulations. Rise time is then a good measurement of valve damping in ASR tests and simulations. By measuring the time it takes to rise from 0% to 63% the initial lag is also included.

• Overshoot increases with increased valve damping in PSR tests. Percentage overshoot is then a good measurement of valve damping in PSR tests and simulations.

By using these simple measurement methods it is easy to get a quick and intuitive comparison number between different designs.

Damping ratio is not recommended to use as measurement method for damping since it has limitations and is not intuitive. Damping ratio is a measurement of the logarithmic decay which doesn’t always give a clear view of overshoot, rise time and settling time. The two formulas that are presented in section 3.1.1.1 have limitations as how and when they can be used. As said earlier the measurement method needs to be functional for all working points and for this reason the damping ratio fails.

If the aim is to damp the pressure response, then the view of damping needs to be extended past the point of adding damping to separate components. However, if the damping of a specific component or part of the valve needs to be investigated, simulation is the best tool. As shown when simulating prototyped concepts in section 5.3 the damping constant can be found if the damping force can be isolated and plotted against the velocity of the moving part. The slope of that curve is then the damping constant.
Chapter 10

Suggestions for future work

As a continuation of the work concerning valve damping the following is suggested:

- Dynamic and static validation of the plunger model.

- Perform tests with removed D-shape to eliminate the current dependent dry friction. In section 7.1.2.2 it is shown that the current dependent dry friction impairs the dynamic valve behaviour.
Bibliography


[10] AMESim documentation, *Using the power train library*.


Appendix A

Nomenclature

$F$                     Force
$F_c$                   Coulomb friction force
$F_{brk}$               Break away friction force
$F_{app}$               Applied force
$F_L$                   Load force
$F_{spring}$            Spring force
$F_{fm}$                Friction force main poppet
$F_{dampmain}$          Damping force on the main poppet
$F_{damppilot}$         Damping force on the pilot poppet
$F_{fp}$                Friction force on pilot poppet
$F_{fs}$                Current dependent friction force on solenoid
$F_s$                   Solenoid force
$F_{fp}$                Friction force on plunger
$F_{fb}$                Friction in pilot bearing
$F_{fd}$                Friction in damping plate
$v$                     Velocity
$x$                     Position
$\dot{x}$               Velocity
$\ddot{x}$              Acceleration
$z_c$                   Car vertical position
$\dot{z}_c$             Car vertical velocity
$\ddot{z}_c$            Car vertical acceleration
Appendix A. Nomenclature

\( u_r \)  
Road vertical position

\( \dot{u}_r \)  
Road vertical velocity

\( q \)  
Flow

\( q_{lm} \)  
Leakage flow from main poppet

\( q_{pump} \)  
Flow due to volume change in the main poppet

\( q_{pd} \)  
Flow through the pd restriction

\( q_{jd} \)  
Flow through the jd restriction

\( q_{pp} \)  
Flow through pilot poppet variable restriction

\( q_p \)  
Leakage flow through solenoid

\( q_{la} \)  
Flow between the two solenoid chambers

\( p \)  
Pressure

\( \dot{p} \)  
Time derivative of pressure

\( V \)  
Volume

\( A_{piston} \)  
Piston area

\( A_c \)  
Cylinder area

\( A_{mo} \)  
Outer main poppet area

\( A_{mi} \)  
Inner main poppet area

\( A_p \)  
Pilot poppet regulator area

\( A_{plunger} \)  
Area on plunger

\( c_l \)  
Piston clearance

\( l_c \)  
Velocity dependent leakage coefficient for main poppet

\( l_p \)  
Pressure dependent leakage coefficient for main poppet

\( l_{cp} \)  
Velocity dependent leakage coefficient for pilot poppet

\( l_{pp} \)  
Pressure dependent leakage coefficient for pilot poppet

\( r_m \)  
Main poppet radius

\( h_0 \)  
Mean clearance on radius

\( l \)  
Contact length

\( e \)  
Eccentricity on radius of poppet

\( K_{pd} \)  
Laminar flow coefficient for the pd restriction

\( K_{jd} \)  
Laminar flow coefficient for the jd restriction

\( K_p \)  
Laminar flow coefficient for the pilot restriction

\( K_{pl} \)  
Laminar flow coefficient for the restriction between the plunger chambers

\( K_{laminar} \)  
Laminar flow coefficient
<table>
<thead>
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<th>Symbol</th>
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<tr>
<td>$B_{pp}$</td>
<td>Viscous damping coefficient for the pilot poppet</td>
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<td>Car mass</td>
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<td>$m_p$</td>
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<td>$k_m$</td>
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Appendix B

Viscous friction coefficient calculation

In this section the result of a simplified calculation of the viscous friction coefficient is presented. The calculations concern the main and pilot poppet and involves the flows that is entering and exiting the volumes. The focus of the calculation is the velocity dependent flows and their addition to the velocity dependent force that is involved in the force equilibrium. These calculations are simplified in order to be able to do them by hand.

In the following calculations all flows are assumed to be laminar. This assumption is not true in reality but the simplification needs to be made to be able to handle the equations by hand. Flow forces are also neglected for the same reason. The aim of the calculation is not to be able to compute exact values but instead see which parameters that affect the damping. To simplify further all the constants in each flow equation have been merged to one.

B.0.2.1 Main poppet

The following calculations are based on the sketch seen in Figure B.1a.

The force equilibrium become:

\[ m_m \ddot{x}_m + k_m x_m = p_1 A_{mo} - p_2 A_{mi} - F_{fm} \text{sign}(\dot{x}_m) \]  \hspace{1cm} (B.1)

The flow equations become:

\[ q_{lm} = -l_c \dot{x}_m + l_p (p_2 - p_T) \]  \hspace{1cm} (B.2)

where \( l_c = \pi v r_m h_0 \) and \( l_p = \frac{\pi r_m h_0^3}{6 \eta_f} \left[ 1 + 1.5 \left( \frac{\dot{x}_m}{h_0} \right)^2 \right] \) according to [I].

\[ q_{pump} = -A_{mi} \dot{x}_m \]  \hspace{1cm} (B.3)

\[ q_{pd} = K_{pd} (p_2 - p_3) \]  \hspace{1cm} (B.4)
Appendix B. Viscous friction coefficient calculation

Figure B.1: Sketch used for calculations

\[ q_{jd} = K_{jd}(p_1 - p_2) \]  
\[ (B.5) \]

The flows in and out from volume \( V_2 \) become:

\[ q_{jd} + q_{pump} = q_{im} + q_{pd} - \frac{V_2 - A_{mi}x_m}{\beta_e} \dot{p}_2 \]  
\[ (B.6) \]

\[ K_{jd}(p_1 - p_2) = -l_c \dot{x}_m + l_p(p_2 - p_T) + K_{pd}(p_2 - p_3) - A_{mi} \dot{x}_m - \frac{V_2 - A_{mi}x_m}{\beta_e} \dot{p}_2 \]  
\[ (B.7) \]

When simplified and solved for \( p_2 \) the expression becomes:

\[ p_2 = \frac{K_{jd}p_1 + l_c \dot{x}_m + l_p p_T + K_{pd}p_3 + A_{mi} \dot{x}_m - \frac{V_2 - A_{mi}x_m}{\beta_e} \dot{p}_2}{K_{jd} + l_p + K_{pd}} \]  
\[ (B.8) \]

If the equation \((B.8)\) is put into \((B.1)\) the following is received:

\[ m_m \ddot{x}_m + k_m x_m = p_1 A_{mo} - A_{mi} \left( \frac{K_{jd}p_1 + l_c \dot{x}_m + l_p p_T + K_{pd}p_3 + A_{mi} \dot{x}_m - \frac{V_2 - A_{mi}x_m}{\beta_e} \dot{p}_2}{K_{jd} + l_p + K_{pd}} \right) - F_{fm \text{sign}(\dot{x}_m)} \]  
\[ (B.9) \]
Appendix B. Viscous friction coefficient calculation

When simplified and put on standard form the expressions become:

\[
m_m \ddot{x}_m + \left( \frac{A_{mi} l_c + A_{mi}^2}{K_{jd} + l_p + K_{pd}} \right) \dot{x}_m + \left( k_m + \frac{A_{mi}^2}{\beta_c (K_{jd} + l_p + K_{pd})} \dot{p}_2 \right) x_m = p_1 A_{mo} - A_{mi} \left( \frac{K_{jd} p_1 + l_p p_T + K_{pd} p_3 - \frac{V_2}{\beta_c} \dot{p}_2}{K_{jd} + l_p + K_{pd}} \right) - F_{fm} \text{sign}(\dot{x}_m) \]  

(B.10)

The expression in front of \( \dot{x} \) is identified as the viscous damping coefficient according to equation (B.11). It can be seen that the orifice areas is gathered in the denominator. If these areas are made smaller the viscous damping coefficient will increase.

\[
B_{pm} = \frac{A_{mi} l_c + A_{mi}^2}{K_{jd} + l_p + K_{pd}} \]  

(B.11)

The total damping force is expressed as equation (B.12). The expression is built up by a velocity dependent friction force and a constant friction force. These forces will counter act the direction of motion.

\[
F_{damp\text{main}} = \left( \frac{A_{mi} l_c + A_{mi}^2}{K_{jd} + l_p + K_{pd}} \right) \dot{x}_m + F_{fm} \text{sign}(\dot{x}_m) \]  

(B.12)

B.0.2.2 Pilot poppet

The following calculations are based on the sketch seen in Figure B.1b.

The force equilibrium on the pilot poppet becomes:

\[
m_p \ddot{x}_p + (x_{pmax} - x_p) k_p = -p_3 A_p - (F_{fp} + F_{fs}) \text{sign}(\dot{x}_p) + F_s - |A_{plunger} (p_4 - p_3)| \text{sign}(\dot{x}_p) \]  

(B.13)

Where the force created from \( |A_{plunger} (p_4 - p_3)| \text{sign}(\dot{x}_p) \) will appear when the plunger is given a velocity. This force will act opposite to the direction of motion and increase with velocity due to the time dependent pressure build up. The force will therefore act as a damping force.

The flow equations become:

\[
q_{pp} = K_p (p_3 - p_T) \]  

(B.14)

\[
q_{lp} = -l_{cp} \dot{x}_p + l_{pp} (p_3 - p_T) \]  

(B.15)

\[
q_{la} = K_{pl} (p_4 - p_3) \]  

(B.16)
The dynamics of the flow between the plunger chambers when the pilot is closing is described by equation (B.17).

\[ K_{pl}(p_4 - p_3) = -\frac{\partial V_{p2}}{\partial t} + \frac{V_{p2} - A_{plunger} x_p}{\beta_e} \dot{p}_4 \]  

(B.17)

The flows through the pilot poppet and the solenoid has the following relations:

\[ q_p = q_{pp} + q_p \]  

(B.18)

\[ q_{pd} = q_p + \frac{V_3}{\beta_e} \dot{p}_3 \]  

(B.19)

\[ K_{pd}(p_2 - p_3) = K_p(p_3 - p_T) - l_{cp} \dot{x}_p + l_{pp}(p_3 - p_T) + \frac{V_3}{\beta_e} \dot{p}_3 \]  

(B.20)

When simplified and solved for \( p_3 \) equation (B.21) becomes.

\[ p_3 = \frac{K_{pd}p_2 + K_p p_T + l_{cp} \dot{x}_p + l_{pp} p_T + \frac{V_3}{\beta_e} \dot{p}_3}{K_{pd} + K_p + l_{pp}} \]  

(B.21)

The pressure \( p_3 \) will be dependent on both the pressures before and after the volume. This pressure is interesting from a damping point of view because it decides the flow magnitude between main and pilot poppet. If \( p_T = 0 \) the expression becomes:

\[ p_3 = \frac{K_{pd}p_2 + l_{cp} \dot{x}_p + \frac{V_3}{\beta_e} \dot{p}_3}{K_{pd} + K_p + l_{pp}} \]  

(B.22)

If equation (B.21) is put into (B.13) and simplified, the expression becomes:

\[ m_p \ddot{x}_p + \left( \frac{A_p l_{cp}}{K_{pd} + K_p + l_{pp}} \right) \dot{x}_p + (x_{pmax} - x_p) k_p = \]  

\[ - A_p \left( \frac{K_{pd}p_2 + K_p p_T + l_{pp} p_T + \frac{V_3}{\beta_e} \dot{p}_3}{K_{pd} + K_p + l_{pp}} \right) - (F_{fp} + F_{fs}) \text{sign}(\dot{x}_p) + F_s \]  

(B.23)

The viscous damping coefficient for the pilot poppet becomes:

\[ B_{pp} = \frac{A_p l_{cp}}{K_{pd} + K_p + l_{pp}} \]  

(B.24)

The total damping force becomes:

\[ F_{damp\text{pilot}} = \left( \frac{A_p l_{cp}}{K_{pd} + K_p + l_{pp}} \right) \dot{x}_p + (F_{fp} + F_{fs} + |A_{plunger}(p_4 - p_3)|) \text{sign}(\dot{x}_p) \]  

(B.25)
Appendix C

Simulation step signals

In the simulations that have been performed, a standard test run has been used to cover most of the working points. In this appendix the three standard simulation test runs are presented.

C.1 Active step response

When performing an active step response the flow through the valve is kept constant while performing a step in solenoid current. The steps that have been used is shown in Table C.1 and the test run graph is shown in Figure C.1.

![Active step response simulation signal](image)

Figure C.1: Active step response simulation signal
Table C.1: Active step response specifications

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<th>Active step response</th>
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<td></td>
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<tr>
<td></td>
<td></td>
<td>high - low</td>
</tr>
</tbody>
</table>

C.2 Passive step response

When performing a passive step response the solenoid current is kept constant while performing a step in solenoid flow. The steps that have been used is shown in Table C.2 and the test run graph is shown in Figure C.2.

C.3 Pressure-flow curves

When performing a pressure-flow curve the solenoid current is kept constant while the flow through the valve is ramped up during 10 seconds and then down during 10 seconds. The specifications that have been used for the simulations is shown in Table C.3 and a graph of the signal is shown in Figure C.3.
### Table C.2: Passive step response specifications

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<thead>
<tr>
<th></th>
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### Table C.3: Pressure-flow curve specifications

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Appendix C. Simulation step signals

Figure C.2: Passive step response simulation signal

Figure C.3: Pressure-flow curve simulation signal