Examensarbete

Force Feedback Control of a Semi-Active Shock Absorber

Examensarbete utfört i Reglerteknik vid Tekniska högskolan vid Linköpings universitet av

Per Svennerbrandt

LiTH-ISY-EX--14/4786--SE

Linköping 2014
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Semi-active suspension systems promise to significantly reduce the necessary trade-off between handling and passenger comfort present in conventional suspension systems by enabling active chassis and wheel control. Öhlins Racing AB have developed a semi-active suspension technology known as CES, Continuously controlled Electronic Suspension, based on solenoid control valves which are integrated into specially designed hydraulic dampers, and are currently developing control and estimation systems which will enable their application in advanced motorcycle suspensions. In these systems an important aspect is being able to accurately control the forces produced. Öhlins’ current system uses an open loop control strategy in which currents sent through the solenoid valves, to achieve the requested damping force under the prevailing circumstances, is calculated using experimentally derived static lookup tables.

In this thesis a new closed loop control system, based on the direct measurement of the damper force, is developed and its performance is evaluated in comparison to the old one’s. Sufficient understanding of the system requires extensive modeling and therefore two different models have been developed; a simpler one used for model based control design and a more extensive, high fidelity model used for high accuracy simulations. The developed simulation model is the first of its kind that is able to capture the studied systems behavior with satisfactory accuracy, as demonstrated against real dynamometer measurements.

The valves and damper behave in a highly non linear manner and the final controller design uses a combination of exact linearization, non linear state estimation, dynamical inversion and classical control theory. Simulation results indicate that the new controller reduces the root mean square force tracking error to about 63% of that of the existing controller in the evaluation scenarios used.

Cascaded within the system is also closed loop current controllers. A developed model based controller is shown to reduce the rise time to less than 30% of that of the existing PID-controllers, reduce the overshoot and provide online estimates of the winding series resistance, providing the basis for future solenoid diagnosis and temperature tracking systems.
Abstract

Semi-active suspension systems promise to significantly reduce the necessary trade-off between handling and passenger comfort present in conventional suspension systems by enabling active chassis and wheel control. Öhlins Racing AB have developed a semi-active suspension technology known as CES, Continuously controlled Electronic Suspension, based on solenoid control valves which are integrated into specially designed hydraulic dampers, and are currently developing control and estimation systems which will enable their application in advanced motorcycle suspensions. In these systems an important aspect is being able to accurately control the forces produced. Öhlins’ current system uses an open loop control strategy in which currents sent through the solenoid valves, to achieve the requested damping force under the prevailing circumstances, is calculated using experimentally derived static lookup tables.

In this thesis a new closed loop control system, based on the direct measurement of the damper force, is developed and its performance is evaluated in comparison to the old one’s. Sufficient understanding of the system requires extensive modeling and therefore two different models have been developed; a simpler one used for model based control design and a more extensive, high fidelity model used for high accuracy simulations. The developed simulation model is the first of its kind that is able to capture the studied systems behavior with satisfactory accuracy, as demonstrated against real dynamometer measurements.

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Acknowledgments

I would like to express my sincerest thanks to Martin Lugnberg, David Bolander, Matteo Pelosi and Geir Lindblad at Öhlins Mechatronic Systems for all your valuable inputs, insights and design support during this project.

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Last but not least my greatest thanks to Kristina for your relentless support while I spent weekends and late nights finishing up the remaining bits and pieces of this thesis after the point at which I got employed and moved on to other things within Öhlins.

Stockholm, June 2014
Per Svennerbrandt
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### Abbreviations

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<th>Meaning</th>
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<tr>
<td>ARX</td>
<td>Auto-Regressive model with eXternal input</td>
</tr>
<tr>
<td>CES</td>
<td>Continuously controlled Electronic Suspension</td>
</tr>
<tr>
<td>CAN</td>
<td>Controller Area Network</td>
</tr>
<tr>
<td>ECU</td>
<td>Electronic Control Unit</td>
</tr>
<tr>
<td>LPV</td>
<td>Linear Parameter Varying</td>
</tr>
<tr>
<td>LQR</td>
<td>Linear Quadratic Regulator</td>
</tr>
<tr>
<td>MPC</td>
<td>Model Predictive Control</td>
</tr>
<tr>
<td>ODE</td>
<td>Ordinary Differential Equation</td>
</tr>
<tr>
<td>PCB</td>
<td>Printed Circuit Board</td>
</tr>
<tr>
<td>PID</td>
<td>Proportional, Integral, Differential (regulator)</td>
</tr>
<tr>
<td>PWM</td>
<td>Pulse Width Modulation</td>
</tr>
<tr>
<td>cmp</td>
<td>Compression</td>
</tr>
<tr>
<td>reb</td>
<td>Rebound</td>
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## Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Quantity</th>
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<tbody>
<tr>
<td>$A$</td>
<td>Area</td>
</tr>
<tr>
<td>$B_{eff}$</td>
<td>Effektive Bulk Modulus</td>
</tr>
<tr>
<td>$C$</td>
<td>Constant</td>
</tr>
<tr>
<td>$c_q$</td>
<td>Flow Coefficient</td>
</tr>
<tr>
<td>$F$</td>
<td>Force</td>
</tr>
<tr>
<td>$i$</td>
<td>Current</td>
</tr>
<tr>
<td>$m$</td>
<td>Mass</td>
</tr>
<tr>
<td>$p$</td>
<td>Pressure</td>
</tr>
<tr>
<td>$\Delta p$</td>
<td>Differential Pressure</td>
</tr>
<tr>
<td>$q$</td>
<td>Flow</td>
</tr>
<tr>
<td>$r$</td>
<td>Radius</td>
</tr>
<tr>
<td>$\rho$</td>
<td>Oil Density</td>
</tr>
<tr>
<td>$t$</td>
<td>Time</td>
</tr>
<tr>
<td>$u$</td>
<td>Voltage</td>
</tr>
<tr>
<td>$U_{batt}$</td>
<td>Battery Voltage</td>
</tr>
<tr>
<td>$x$</td>
<td>Position</td>
</tr>
<tr>
<td>$\dot{x}$</td>
<td>Derivative of $x$ (or other variable) with respect to time</td>
</tr>
<tr>
<td>$\ddot{x}$</td>
<td>Double derivative of $x$ with respect to time</td>
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This report is the result of a Master’s thesis project performed at Öhlins Racing AB with the aim of developing a new type of control strategy for their electronically controlled shock absorbers.

1.1 Background

An automotive shock absorber is usually an assembly of two components, namely the main spring and a hydraulic device that resists velocities called the damper. The function of the spring is basically to support the weight of the vehicle while still providing flexibility that allows the wheels to closely follow the road’s surface without unduly transferring its roughness into the vehicle’s chassis. However, a pure frictionless spring-mass system will oscillate back and forth around the equilibrium position in an undesirable manner in response to any disturbances, and hence a controlled amount of “friction” is added through the fitting of a hydraulic damper.

The vast majority of automotive dampers are currently built as so-called passive systems; with static characteristics that are fixed during manufacturing for the service life of the damper. These fixed characteristics has to be chosen based on the contradictory requirements of comfort and safety/road holding and this inevitable results in a significant compromise between the two requirements.

Öhlins Racing AB is a world leading supplier of high end suspension components such as shock absorbers, motorcycle front forks and steering dampers. Among other things the company designs and sells dampers whose characteristics can be set via externally adjustable so-called click valves (while the vehicle is at a standstill). These enable the damping properties to be chosen specifically for each
separate driving occasion, thereby better accounting for prevailing circumstances such as road conditions, rider/driver preferences, etcetera. This often means that significant performance gains can be achieved through fine tuning compared to a fixed damper. Significant compromises still remains though since the setup still has to be suitable for the whole stint and can not be tuned separately for individual events, such as braking, cornering, hitting a certain bump, etcetera.

Because of the inherent compromises in conventional suspension system design there is a lot of interest in what is known as active and semi-active suspensions. A semi-active system contains actuators that have the ability to control the forces opposing the suspensions movements whereas a fully active suspension have actuators that can provide forces independently of the suspensions movement. Semi active systems have been shown to provide performance close to that of fully active systems with lower weight, complexity and power consumption.

Öhlins has developed a technology called CES (Continuously controlled Electronic Suspension). Briefly stated the system consists of hydraulic valves through which the flow resistance for the oil being forced into or out of the damper, as a result of the piston movements, and thus the damping characteristics, can be controlled electronically through an ECU (Electronic Control Unit); individually in each damper and with high bandwidth.

A lot of academic research has been conducted about how to design the control system that calculates the forces to request from the actuators based on the vehicle’s estimated movements and results such as Skyhook control, clipped LQR, LPV $\mathcal{H}_\infty$ and MPC treatments are well publicized and known, see e.g. Koch [2011] and Poussot-Vassal et al. [2012] and the references within for an overview. Most such studies however treat the dampers themselves as perfect “black boxes” that are always able to match the forces requested by them. Not much work has been done about the actual force control problem itself.

Öhlins’ current system uses a open loop control strategy with static speed-force lookup tables produced through testruns of the dampers in a dynamometer.

1.2 Problem Description

The main task for this thesis has been to design a new control system that uses a force sensor to close the loop and investigate what kind of performance that can be achieved using that approach.

1.3 Previous Work

Two previous theses Loman [2010], Söderberg [2011] have attempted to improve the force controller by modeling the system with black-box methods (nonlinear ARX models); but without achieving satisfactory model fit except for particular data sets recorded at singular stroking frequencies.
Another former thesis Johansson and Kvaldén [2011] achieved improved force tracking by introducing an acceleration-based feedforward; but also in that case it proved difficult to get good performance at more than singular stroking frequencies. The biggest improvement achieved was through improved signal processing for estimating the damper speed in the form of a stationary Kalman filter that fuses measurements from a linear displacement sensor with those from an accelerometer.

1.4 Limitations

It has been decided that variability not handled by the current open loop control strategy, such as changes in properties due to temperature, wear etcetera will not be accounted for, apart from those inherent in the force feedback approach itself. Neither will it be assured that the sensor setup is feasible in production.
The shock absorber used during this work has been the Öhlins TTX30 seen in Figure 2.1. It is equipped with a prototype cylinder head containing two electronically controllable CES-valves - one for controlling the flow restriction during compression (that is when the piston is moving into the damper) and the other for controlling the flow restriction during rebound (that is when the piston is moving out of the damper).

*Figure 2.1: Öhlins TTX30 shock absorber with dual external CES-valves.*
2.1 Damper

The main way through which the damper produces its damping force is through the pressure buildup created by careful metering of the flow of oil into and out of the chambers on each side of the piston.

A schematic cross sectional view of the damper is shown in Figure 2.2 below, along the most important nomenclature and designations used for the various components.

![Schematic view of the damper with basic nomenclature defined.](image)

**Figure 2.2: Schematic view of the damper with basic nomenclature defined.**

As the piston moves the volume on both sides of it changes and this in turn causes oil to flow in or out through either the CES-, bleed- or one way valves. A hydraulic diagram showing the flow paths during compression can be seen in Figure 2.3 on the facing page. The compression stroke is characterized by the piston moving into the damper (to the left in Figure 2.3 on the next page). The oil in the compression chamber is forced out through the bleed- and CES valves whereas the compression
side one way valve remains shut. The rebound side is filled by flow coming in through its one way valve.

![Figure 2.3: Schematic view of the damper showing the flow paths during compression.](image)

During rebound, i.e. when the piston is moving out of the damper, the flows are the exact mirror of those during compression, as illustrated in Figure 2.4.

![Figure 2.4: Schematic view of the damper showing the flow paths during rebound.](image)

The bleed valves are used mainly to tune the damping characteristics at low piston speeds, and in the damper used in this work have been mostly closed.

As the piston moves into the damper the total volume available in the chambers decreases due to the displacement of the piston rod. The oil displaced by this movement is taken up by the component known as the accumulator or oil reservoir, which also handles volume changes due to temperature.

The hydraulic oil in the damper always contains a certain amount of dissolved gas/air, even if extensive measures are taken to limit it such as vacuum sucking the
damper before it is filled. To limit the susceptibility to air release and cavitation the low pressure side is pressurized to a level above ambient using nitrogen gas at the back of the accumulator.

## 2.2 CES-valves

A simplified schematic view of a CES-valve can be seen in Figure 2.5. Using hydraulics terminology it is best described as a pressure regulator with a solenoid controlled pilot stage. The flow through the valve is regulated through the position of the main poppet which is given by a force/pressure balance set up by the spring, pushing the main poppet against its seat, and the differential pressure between the main poppet front and back sides.

![Schematic view of the CES-valve.](image)

The main poppet has a small orifice drilled through the middle, connecting the front and back sides, and it is the pressure drop over this orifice which acts as the main poppet differential pressure. The flow out of the chamber behind the main poppet is known as the pilot flow and the restriction for this flow is what is set by the force created by the solenoid force when current is applied through it, acting via the component known as the pilot poppet.

If the solenoid current is increased the flow resistance for the pilot flow is increased, creating a pressure buildup behind the main poppet and hence forcing it to attain
a lower position.

If the solenoid current is decreased the flow resistance for the pilot flow is likewise decreased, leading to a lower pressure at the back of the main popper, and hence making it easier for the main flow to push the main poppet out of the way, meaning it will attain a higher position and hence resist the flow less.

2.3 Sensors

During this work it has been assumed that the following signals are either available as direct measurements, or can be estimated using the other available signals

- Piston displacement/position ($x$)
- Piston velocity ($\dot{x}$)
- Piston acceleration ($\ddot{x}$)
- Solenoid currents ($i$)
- Battery voltage ($U_{batt}$)
- Force ($F_{damping}$)
- Pressures ($p_{cmp}$ and $p_{reb}$)
For the purposes of this thesis two models of different fidelity and complexity has been developed. Both are based on the same physical modeling principles; one aimed at being usable as a mathematical tool during model based control system development and one aimed at being sufficiently accurate to serve as a virtual prototype in simulations.

The model aimed at high accuracy simulations, hereafter referred to as the simulation model, has been implemented in the simulation software AMESim by integrating two preexisting CES-valve models developed at Öhlins Mechatronic Systems, into a damper model built using components from the AMESim standard library.

The CES-valve models are largely based on the work by Gällsjö and Johansson [2012] but has since been reimplemented in AMESim and further refined and verified by Gällsjö and other personnel at Öhlins Mechatronic Systems. The damper model started off as a modification of the one described by Sadeghi Reineh [2012] but has been quite heavily reworked and reparameterized during this work to account for the differences in damper construction, model the hysteresis using a different approach and to include the CES-valves instead of click valves.

The model aimed at control system development, hereafter referred to as the design model, is the result of an effort of trying to capture as much of the physical properties of the damper with as low mathematical complexity as possible, while retaining the same basic physical modeling principles. This has been done by introducing lumped parameters, lookup tables and neglecting minor phenomena wherever possible while retaining suitable levels of accuracy.
3.1 Control System Design Model

The following section gives an overview of the design model. Most of the principles also roughly applies to the simulation model, except when noted. To give a full mathematical description of the valve simulation models would be beyond the scope of this thesis; interested readers are referred to [Gällsjö and Johansson, 2012] and the AMESim documentation.

3.1.1 Forces

The force that the damper produces (when inside of its normal working range in terms of position $x \in [x_{\text{min}}, x_{\text{max}}]$; i.e. has not hit its endstops) is modeled as the sum of the three basic components hydraulic force, friction force and inertial force.

$$F_{\text{damping}} = F_{\text{hydr}} + F_{\text{fric}} + F_{\text{in}}$$ (3.1)

Friction Force

Damper friction can be characterized in a dynamometer by, to the largest extent possible, removing all hydraulic restrictions and then running the damper at various fixed speeds and measuring the resulting forces. Typically it is found that there is a certain force required to get the damper moving at all, called the static, Coulomb or dry friction value, and that beyond that value the friction increases approximately linearly as a function of speed. The damper friction force is hence modeled using the “classical approach”, described in e.g. Olsson et al. [1998], as

$$F_{\text{fric}} = \text{sgn}(\dot{x})F_{\text{static}} + \dot{x}C_{\text{vis}}$$ (3.2)

where the coefficient of viscous friction

$$C_{\text{vis}} = \left. \frac{dF_{\text{damping}}(\dot{x})}{d\dot{x}} \right|_{|\dot{x}|>0}$$ (3.3)

is the slope of the curve at non zero speeds.

Inertial Force

From a rigid body point of view the damper can be thought to consist of two subassemblies; one formed by the piston, pistonrod and end eye, and one formed by all the other damper components including the main body. The force that it takes to accelerate the piston subassembly, in the direction of the damping force, is given by Newtons second law of motion

$$F_{\text{in},p} = \ddot{x}_p \sum_i m_i,$$ (3.4)

where $m_i$ are the masses of the accelerated components, i.e $m_{\text{piston}}$, $m_{\text{pistonrod}}$, $m_{\text{endeye}}$, and $\ddot{x}_p$ their acceleration with respect to an inertial reference frame. The corresponding expression for the main body subassembly is

$$F_{\text{in},b} = \ddot{x}_b \sum_i m_i,$$ (3.5)
with analogous designations. These forces are thus generally functions of the accelerations with respect to an inertial reference frame and are different depending on at which end of the damper the force is measured. For simplicity it has been assumed that the main body is stationary in this work, and hence \( \ddot{x}_b = 0 \) and \( \ddot{x}_p = \dot{x} \), which is true for the dynamometer measurements used and should be a reasonable approximation when the damper is run in a vehicle if both accelerations are not available individually. The inertial forces are typically quite small when the damper is subjected to typical dynamometer signals, and could possibly be neglected if one wishes to simplify the model further, but on the other hand become increasingly important as the excitation frequency/acceleration increases.

**Hydraulic Force**

The forces described this far have mostly been of the nuisance type and are typically sought to be minimized in the mechanical construction of the damper. The main mechanism through which the damper creates its desired damping is through the hydraulic force, \( F_{hudr} \). This force is given by the internal pressures in the compression and rebound chambers, \( p_{cmp} \) and \( p_{reb} \), acting on their respective sides of the piston, and the ambient pressure \( p_{amb} \) acting on the piston rod, according to the equation

\[
F_{hydr} = p_{cmp}A_{cmp} - p_{reb}A_{reb} - p_{amb}A_{pistonrod}. \tag{3.6}
\]

The active pressure areas are given by the geometry of the cross sections according to

\[
A_{piston} = \pi r_{piston}^2, \tag{3.7}
\]
\[
A_{pistonrod} = \pi r_{pistonrod}^2, \tag{3.8}
\]
\[
A_{reb} = A_{piston} - A_{pistonrod}, \tag{3.9}
\]
\[
A_{cmp} = A_{piston}. \tag{3.10}
\]

The internal pressures \( p_{cmp} \) and \( p_{reb} \) are related to the amount of oil entrapped in the respective chambers as described in the following section.

**3.1.2 Pressures**

The pressures in the compression and rebound chambers, \( p_{cmp} \) and \( p_{reb} \) respectively, have been modeled as state variables whose time derivative is given by the linearized state equation for a liquid (see e.g. Merritt [1967])

\[
\dot{p} = \frac{B_{eff}}{V} \left( \sum_i q_i + \dot{V} \right), \tag{3.11}
\]

where

\[
B_{eff} = -V \frac{dp}{dV} \tag{3.12}
\]

is a lumped parameter approximating the oils bulk modulus (including effects from dissolved gases) and the stiffness of the enclosing chambers.
$V$ is the volume of the enclosing chamber

\[ V_{reb} = V_{0,reb} + A_{reb}x \quad (3.13) \]

\[ V_{cmp} = V_{0,cmp} + A_{cmp}(x_{max} - x) \quad (3.14) \]

and hence

\[ \dot{V}_{reb} = A_{reb}\dot{x} \quad (3.15) \]

\[ \dot{V}_{cmp} = -A_{cmp}\dot{x}. \quad (3.16) \]

The sum $\sum q_i$ is the net flow of oil, i.e. the flow through the CES and bleed valves, defined positively into the chamber. The ambient pressure $p_{amb}$ is assumed to be atmospheric pressure at sea level. For control purposes the pressure in the accumulator is assumed to have a fixed value $p_{accu}$ whereas in the simulation model it is modeled using the standard AMESim accumulator component.

### 3.1.3 Fluid Flows

As shown in Section 2.1 on page 6 the main flow paths for the oil is through the bleed valves, one way valves and CES valves. In the literature fluid flow is commonly modeled as proportional to the pressure differential if the flow is thought to be mostly laminar, i.e

\[ q \propto \Delta p \quad (3.17) \]

and proportional to the square root of the pressure differential if the flow is thought to be mostly turbulent,

\[ q \propto \sqrt{|\Delta p|} \text{sgn}(\Delta p). \quad (3.18) \]

The flow through an orifice generally starts out as being laminar at low pressure differentials/flows, and then transitions into being turbulent as the pressure differential/flow increases. To predict whether a particular flow is mostly laminar or turbulent the so called Reynolds number and rules of thumb are commonly used. The proportionality factor is commonly expressed in terms of the flow area $A$, the fluid density $\rho$ and the factor $C_q$, known as the flow coefficient, meaning that (3.18) can spelled out as

\[ q = C_q A \sqrt{\frac{2}{\rho} |\Delta p|} \text{sgn}(\Delta p). \quad (3.19) \]

Empirically developed formulas for $C_q$ for different types of geometries exist, but it is also widely acknowledged that a fixed value of about 0.7 often is a good baseline approximation if no flow measurements are available.

AMESim uses a kind of hybrid approach and models most flows using (3.19) with the flow coefficient given by the expression

\[ C_q = C_{q,max} \tanh \left( f(\ldots) \sqrt{\frac{2}{\rho} |\Delta p|} \right), \quad (3.20) \]

where $f(\ldots)$ is a function of various user specifiable fluid parameters, mean-
ing that $c_q$ approaches $c_{q,max}$ asymptotically as $\Delta p$ increases and that the flow smoothly tapers off at decreasing differential pressures. Importantly it also means that $\frac{dq}{dp}$ is smooth and well behaved which is crucial for efficient numerical integration of (3.11) at low pressures/flows.

### Bleed Valves

The flow through the bleed valves was modeled as strictly turbulent and hence by the equation

$$q_{\text{bleed}} = C_q A \sqrt{\frac{2}{\rho}} |\Delta p| \text{sgn}(\Delta p)$$

(3.21)

$\Delta p = p - p_{\text{accu}}$

where $A$ is the effective area of the orifice and $C_q = 0.7$ the flow coefficient.

### One Way Valves

The one way valves consist of a thin metering shim that is pushed against four openings by a preloaded spring. During mechanical design the aim is to make the valve behave as close as possible to an ideal one way valve and hence the shim is made light and the spring quite soft to minimize the valves dynamics. The consensus within Öhlins seems to be that the exact behavior of the one way valve, for various reasons, is quite complex at low pressures and that the transition between closed and open can be quite complicated.

The approach chosen during this work was to model the metering shim as massless, the spring using a fixed spring rate $k$ and preload force $F_0$. The differential pressure on the shim was assumed to act on an area $A$, which was taken to be the same as the area of the four openings. Since the shim is modeled as massless it will instantly attain an opening height $h$, $0 < h < h_{\text{max}}$, where the force from the spring, $kh + F_0$ by Hooke’s law, exactly counteracts the force from the pressure difference, $A\Delta p$. Solving for $h$ and taking account for the end stops gives

$$h = \frac{1}{k} (A\Delta p - F_0)$$

$$\bar{h} = \begin{cases} 0 & h < 0 \\ h & 0 \leq h \leq h_{\text{max}} \\ h_{\text{max}} & h > h_{\text{max}} \end{cases}$$

$\Delta p = p_{\text{accu}} - p$.

The flow was modeled as being turbulent and passing radially outwards through a cylindrical section with diameter $d$, which gives the one way valve flow model

$$q_{\text{owv}} = C_q \bar{h} \pi d \sqrt{\frac{2}{\rho}} |\Delta p|.$$  

(3.22)

Dynamometer comparisons indicate that this kind of model gives adequate model fit for control purposes, at least at the frequencies studied. This is valuable since it means that the flow through the one way valve can be approximated as zero.
as soon as $A\Delta p < F_0$, and that e.g. the one way valve can be assumed to be completely closed at the pressurized side of the damper.

**Piston Leakage Flow**

The seal between the piston and cylinder is never completely tight which means that there is a small leakage flow from chamber with the highest pressure to the one with a lower pressure. For control purposes this flow has been assumed to be negligible. In the simulation model the piston leakage flow is modeled using the standard AMESim component with the same name and the same geometrical parameters as those chosen by Sadeghi Reineh [2012].

**CES-valves**

To accurately model the hydraulic section of the CES-valves requires accounting for a number of internal pressures, positions, flows and forces, each of which depend of various physical properties such as geometry, spring rates, flow characteristics etcetera.

The approach chosen for the design model has been to build a “virtual flow-bench” and map the valves behavior in stationarity over a fine grid of $(\Delta p, q, i)$-values covering the valves working range. This results in lookup tables where the value of one of the variables is given as a function of the other two using linear interpolation, i.e.

$$q_{CES,static} = f(\Delta p_{CES,static}, i_{CES,static}) \quad (3.23)$$

$$i_{CES,static} = f(\Delta p_{CES,static}, q_{CES,static}) \quad (3.24)$$

$$\Delta p_{CES,static} = f(q_{CES,static}, i_{CES,static}) \quad (3.25)$$

A plot of (3.24) for the compression valve can be seen in Figure 3.1 on the next page.

The simulation model includes two instances of a detailed model based on the work described in [Gällsjö and Johansson, 2012] which has since been reimplemented in AMESim and further refined and verified by Gällsjö and other personnel at Öhlins Mechatronic Systems. Each of these has 15 states and hundreds of configuration parameters, covering it in detail would be beyond the scope of this thesis. The interested reader is referred to [Gällsjö and Johansson, 2012] which still roughly applies.

**3.1.4 Drive Electronics and Solenoids**

The electromagnetic circuit of the CES valves consists of coil wire wound around a plastic bobbin which is then inserted around the central core formed by the pilot poppet and its housing. The valves outer ferromagnetic shell completes the magnetic circuit.

The solenoid electrical circuits were modeled as an inductor with inductance $L$ in series with the wire resistance $R$. The current dynamics is hence given by the first
3.1 Control System Design Model

order ODE

\[ u(t) = R \dot{i}_{CES}(t) + L \frac{d}{dt} i_{CES}(t) \]  \hspace{1cm} (3.26)

where \( u(t) \) is the driving (input) voltage and \( i_{CES}(t) \) the resulting current. The series inductance and resistance, \( L \) and \( R \) respectively, are generally functions of armature position, frequency, temperature etcetera but in this work it is assumed that a constant value for \( L \), that gives sufficient model fit for the observed PWM-waveforms and working conditions, can be found through the parameter optimization

\[ (\hat{L}, \hat{R}) = \arg \min_{L,R} \sum_{k=0}^{N-1} (i_{k,pred}(L,R) - i_{k,meas})^2 \]  \hspace{1cm} (3.27)

and that the value of \( R \) can thereafter be estimated online.

The positive solenoid terminal is alternatingly connected to the battery voltage \( U_{batt} \), through the transistor in the high side PWM-driver, and ground, through a flyback diode. This switching was modeled as instant and lossless and \( u(t) \) is therefore given by

\[ u(t) = \begin{cases} U_{batt} & \text{for } kT \leq t < (k + r_k)T \\ 0 & \text{for } (n + r_k)T \leq t < (k + 1)T \end{cases} \]  \hspace{1cm} (3.28)

where \( r_k \in [0,1] \) represents the PWM duty ratio in the PWM period \( k \) of fixed length \( T \).
3.2 Simulation Model Overview

As stated previously the simulation model was implemented in the simulation software AMESim by integrating two preexisting CES-valve models developed at Öhlins Mechatronic Systems, into a damper model built using components from the AMESim standard library.

A screenshot of the graphical representation of the model is shown in Figure 3.3.

![Figure 3.3: The simulation model.](image)

The various enumerated components are
1. Compression side piston model (pressure $\rightarrow$ force, velocity $\rightarrow$ flow)
2. Piston leakage model
3. Rebound side piston model (pressure $\rightarrow$ force, velocity $\rightarrow$ flow)
4. Piston assembly dynamics model (mass, friction and endstops)
5. Position output to SIMULINK
6. Velocity input from SIMULINK
7. Force output to SIMULINK
8. Compression volume model (pressure dynamics)
9. Compression side one way valve model
10. Rebound side one way valve model
11. Rebound volume model (pressure dynamics)
12. Compression pressure output to SIMULINK
13. Compression side CES-valve inlet restriction and volume models
14. Compression side bleed model (differential pressure $\rightarrow$ flow)
15. Rebound side bleed model (differential pressure $\rightarrow$ flow)
16. Rebound side CES-valve inlet restriction and volume models
17. Rebound pressure output to SIMULINK
18. Compression side solenoid current input from SIMULINK
19. Compression side CES-valve model
20. Rebound side CES-valve model
21. Rebound side solenoid current input from SIMULINK
22. Gas accumulator model
23. Low pressure output to SIMULINK

3.3 Parameter Tuning and Model Validation

The parameter space through which the models behavior is decided is vast and there is a considerable risk to get lost trying to make the models predictions fit the observed behavior of the real hardware if one is not careful. An effort was therefore made to make sure that as many of the model parameters as possible were physically measurable or deducible and hence could be constrained to their real world values.
Three parameters were tuned manually, namely the effective bulk modulus $\beta_{\text{eff}}$ in the main cylinder volumes, the effective flow area/flow coefficient of the bleed valves and the CES-valves pilot diameter.

Changing the effective bulk modulus changes the pressure dynamics, and increasing it has the effect of increasing the width of the hysteresis loop seen in damper dynamometer plots. Increasing the bleed valve flow area/coefficient affects the slope of the curve at low speeds and lowers the curve evenly slightly at high speeds. With these two parameters tuned the model was observed to give slightly to much slope at high currents and slightly to little at low, indicating that the flow predicted through the CES-valve models was slightly to high at low currents and slightly to low at low. To avoid having to change the structure of the CES-valve model this mismatch was corrected by decreasing the pilot diameter parameter in the compression valve by 0.03 mm and that in the rebound valve by 0.13 mm.

As long as the right parameters were adjusted, tuning at one particular frequency and current gave improvements at all others as well, which gives confidence that the physics of the damper is well captured by the model; something which has not been true during previous modeling attempts.

The parameters of the simulation model were tuned until the model was deemed to provide sufficiently accurate predictions of measurement data from dynamometer runs performed using fixed currents between 300 and 1800 mA, in 200 mA steps, and sinusoidal velocity inputs at 4 and 16 Hz.

Plots of the simulated and measured force values, as a function of velocity, after this tuning had been performed can be seen in Figures 3.4 and 3.5 on the next page and on page 22.

The level of model fit achieved has to be considered very good and is much better than what previous modeling attempts has been able to achieve.

### 3.4 Co-Simulation

Since the controller has been developed in SIMULINK and the simulation model in AMESim simulation of the complete system has been done through co-simulation, meaning that both software are run simultaneously and are made to exchange input and output signals throughout the simulation.
**Figure 3.4:** Comparison of simulated and measured values.
Figure 3.5: Comparison of simulated and measured values.
4.1 System Architecture

A block diagram of the controller can be seen in Figure 4.1. To facilitate the integration into Öhlins’ existing system an early design decision was taken to implement a cascaded control architecture where the force and current tracking parts of the controller are implemented as separate subsystems. The force controller receives the force reference $F_{ref}$, the measured/estimated damper states such as the force $F_{damping}$, velocity $\dot{x}$, and position $x$ and based on those calculates current reference values which are sent to the current controllers. The current controllers receives the reference values along with measured electrical states such as...
as current, battery voltage and estimated solenoid resistance and in turn calculates the PWM-duty cycle for the solenoid drivers.

Another early design decision was to implement the force controller such that the valve that is actively controlled using force feedback is chosen based on whether the damper is undergoing compression ($\dot{x} > 0$) or rebound ($\dot{x} < 0$). The CES valves are not built to operate on reversed flow so for the chamber which is growing most of the oil goes through the one way valves and the CES valve have very little influence over the pressure.

## 4.2 Solenoid Current Estimation and Control

Given the cascaded architecture it is natural to start the design/analysis of the inner most current loop and work outwards since its characteristics potentially could have a big impact on the design of the outer loops. The initial approach was that the current controllers should be left as is, if possible, and the emphasis put on hydraulic control, but it was later discovered that additional electrical performance was likely achievable and desirable.

### 4.2.1 Classical PID Approach

Öhlins’ excising implementation contains PID current controllers, implemented in the firmware of the ECU, which select the PWM duty ratio $r(t)$ based on a discretized implementation of the control law

$$r(t) = K_P e(t) + K_I \int_0^t e(\tau) d\tau + K_D \frac{d}{dt} e(t)$$

(4.1)

where $e(t) = i(t) - i_{ref}(t)$, together with simple anti-windup protection.

To enable comparison against the model based controller described in the following sections the exact C-code of this PID-controller was implemented in Embedded MATLAB and run in simulation.

### 4.2.2 One Step Ahead Current Predictor

During time intervals when the drive transistor is conducting, and the solenoid input voltage $u(t)$ hence equals $U_{batt}$, (3.26) has the explicit solution

$$i_{on}(t) = \frac{U_{batt}}{R} - \left( \frac{U_{batt}}{R} - i_0 \right) e^{-\frac{R}{L} t}$$

(4.2)

where $i_0$ represents the initial value for the current, at the start of the interval.

Solving (3.26) when the input voltage $u(t) = 0$, that is when the transistor is not conducting and the flyback diode is clamping the voltage, gives the solution

$$i_{off}(t) = i_0 e^{-\frac{R}{L} t}.$$

(4.3)
4.2 Solenoid Current Estimation and Control

Since the current at the end of an interval becomes the initial value for the next these solutions can be cascaded to give the solution over an arbitrary \{on, off\} sequence. Setting $t = r[k]T$ in (4.2) and putting the resultant in place of $i_0$ in (4.3), followed by the substitution of $t = (1 - r[k])T$, gives the one PWM-step ahead current predictor

$$i_{k+1} = \left( \frac{U_{\text{batt}}}{R} - \left( \frac{U_{\text{batt}}}{R} - i_k \right) e^{-\frac{R}{\tau} r_k T} \right) e^{-\frac{R}{\tau} (1-r_k)T}$$

$$= i_k e^{-\frac{R}{\tau} T} + \frac{U_{\text{batt}}}{R} e^{-\frac{R}{\tau} T} \left( e^{\frac{R}{\tau} T r_k - 1} \right). \quad (4.4)$$

The PWM-frequency is quite high though and ideally one would like to have a predictor for an entire control/estimation time step without having to raise the execution rate of the controller/estimator to that level. By applying (4.4) recursively one, after some algebra, realizes that the $n$-PWM-step ahead current predictor (for an $n$-step fixed PWM duty ratio $r_k: k+n$) is given by the expression

$$i_{k+n} = i_k e^{-\frac{R}{\tau} Tn} + \frac{U_{\text{batt}}}{R} e^{-\frac{R}{\tau} Tn} \left( e^{\frac{R}{\tau} T r_{k:k+n} - 1} \right) \sum_{j=0}^{n-1} e^{\frac{R}{\tau} Tj}, \quad (4.5)$$

where $n$ can be chosen as $n = \frac{f_{\text{PWM}}}{f_{\text{est/ctrl}}}$, i.e. the number of PWM periods within one estimation/control period,

By introducing $\tau = \frac{R}{L} T$ and utilizing the fact that $\sum_{j=0}^{n-1} (e^\tau)^j$ is a geometric series this can be simplified to

$$i_{k+n} = i_k e^{-\tau n} + \frac{U_{\text{batt}}}{R} e^{-\tau n} \left( e^{\tau r_k:k+n} - 1 \right) \frac{e^{\tau n} - 1}{e^\tau - 1}$$

$$= i_k e^{-\tau n} + \frac{U_{\text{batt}} (1 - e^{-\tau n})}{R (e^\tau - 1)} \left( e^{\tau r_k:k+n} - 1 \right) \quad (4.6)$$

4.2.3 Model Based Current and Resistance Estimator

If one models the solenoid series resistance $R$ and inductance $L$ as random walk processes (4.6) can be extended into a statistical context as

$$\begin{bmatrix} i_k \\ R_k \\ L_k \end{bmatrix} = \begin{bmatrix} i_{k-1} e^{-\tau n} + \frac{U_{\text{batt}} (1-e^{-\tau n})}{R_{k-1} (e^\tau - 1)} \left( e^{\tau r_{k-1}} - 1 \right) \\ 0 \\ 0 \end{bmatrix} + \begin{bmatrix} w_{k-1} \end{bmatrix} \quad (4.7)$$

and the Extended Kalman Filter can then be used to estimate the complete state vector. This leads to quite high computational complexity since the expressions for $\frac{\partial f}{\partial R}$ and $\frac{\partial f}{\partial L}$ are fairly complicated (see Appendix A on page 55), so instead a hybrid approach was adapted where $i$ is estimated with an Extended Kalman Filter with covariance scheduling, $R$ with an adaptive filter and $L$ is assumed to
be constant. The top term of $\frac{\partial f}{\partial i}$, from now on referred to as $\frac{\partial f_i}{\partial i}$, simply becomes $e^{-\tau n}$, which can be calculated quickly.

This gives the update procedure:

1. **Predict the solenoid current using the model.**
   
   $$\hat{i}_{k|k-1} = i_{k-1}e^{-\tau n} + \frac{U_{batt}(1 - e^{-\tau n})}{R_k(e^\tau - 1)}(e^{\tau r_k} - 1)$$  \hspace{1cm} (4.8)

2. **Calculate the a priori estimate variance** $(\sigma_{k|k-1})$. Account for the additional uncertainties that stem from the uncertainty in $L$ by adding a variance term that's proportional to the square of the change in current $(\Delta i)$. (That is, express that the model should be trusted less and the measurements more when there are big changes in the current.)
   
   $$\sigma_{k|k-1} = \left(\frac{\partial f_i}{\partial i}\right)^2\sigma_{k-1|k-1} + \sigma_{ss} + c\Delta i(\Delta i)^2$$  \hspace{1cm} (4.9)

   The term $\sigma_{ss}$ is the process variance in steady state conditions, $c\Delta i$ a tuning constant and $\Delta i_k = \hat{i}_{k|k-1} - \hat{i}_{k|k-1}$.

3. **Calculate the innovation/residual.**
   
   $$e_k = i_{k,meas} - \hat{i}_{k|k-1}$$  \hspace{1cm} (4.10)

4. **Calculate the innovation/residual variance.**
   
   $$\sigma_{e_k} = \sigma_{k|k-1} + \sigma_{i,meas}$$  \hspace{1cm} (4.11)

   where $\sigma_{i,meas}$ is the variance of the current measurement.

5. **Calculate the Kalman gain.**
   
   $$K_k = \frac{\sigma_{k|k-1}}{\sigma_{e_k}}$$  \hspace{1cm} (4.12)

6. **Update the current estimate with the new information.**
   
   $$\hat{i}_{k|k} = \hat{i}_{k|k-1} + K_ke_k$$  \hspace{1cm} (4.13)

7. **Update the variance estimate.**
   
   $$\sigma_{k|k} = (1 - K_k)\sigma_{k|k-1}$$  \hspace{1cm} (4.14)

8. **Calculate the resistance estimate adaptation gain.** The main idea is to adapt the most when we have had small changes in current, since in those cases the prediction error should be a fairly good indication of the error in $\hat{R}$ (and not stem from any inaccuracies in $L$).
   
   $$\mu = -c_\mu e^{\frac{(\hat{i}_{k|k} - \hat{i}_{k-1|k-1})^2}{c^2}}$$  \hspace{1cm} (4.15)
9. Update the resistance estimate in the direction that decreases the prediction error.

\[ \hat{R}_k = \hat{R}_{k-1} + \mu e_k \quad (4.16) \]

For a description of the theory behind Extended Kalman Filtering see Gustafsson [2010], and for adaptive filtering Gustafsson et al. [2010].

### 4.2.4 Model Based Solenoid Current Control

The basic requirement for the control of the solenoid currents in the cascaded architecture is that they should follow the given current references as fast and accurately as possible, meaning that there is no speed/energy trade off involved in choosing the value for the control input as long as it is within the control input limitations. Explicitly solving (4.5) for the PWM duty ratio \( r_{k:k+n} \) that should be used to hit a given current \( i_{k+n} = i_{\text{ref}} \) after \( n \) PWM-steps, gives

\[ r_{k:k+n} = \frac{1}{\tau} \ln \left( \frac{R(e_{\tau} - 1)}{U_{\text{batt}}(1 - e^{-n})} \left( i_{\text{ref}} - i_k e^{-n} \right) + 1 \right). \quad (4.17) \]

Taking the control input limitations \( 0 \leq r_{k:k+n} \leq 1 \) into account gives

\[ r_{k:k+n} = \begin{cases} 1 & \text{if } i_{\text{ref}} \geq i_{\text{max}}(i_k) = e^{-\tau n}i_k + \frac{U_{\text{batt}}}{R}(1 - e^{-\tau n}) \\ r_{k:k+n} & \text{if } i_{\text{min}}(i_k) < i_{\text{ref}} < i_{\text{max}}(i_k) \\ 0 & \text{if } i_{\text{ref}} \leq i_{\text{min}}(i_k) = i_k e^{-\tau n} \end{cases} \quad (4.18) \]

This control law, together with the estimator described in the previous section, can be seen either as an internal model controller, where the error signal is used to adapt the parameters of the internal model, or as an adaptive dynamic inversion controller where the value of \( R \) is adapted in the direction that decreases the tracking error.

It is also the solution to the optimization problem

\[ r_{k:k+n} = \text{arg min}_{r_{k:k+n}} |i_{k+n}(r_{k:k+n}) - i_{\text{ref}}| \]

s.t.

\[ 0 \leq r_{k:k+n} \leq 1. \]

since \( |i_{k+n}(r_{k:k+n}) - i_{\text{ref}}| \) is zero for \( i_{\text{min}}(i_k) < i_{\text{ref}} < i_{\text{max}}(i_k) \) and monotonically increasing/decreasing for \( i_{\text{ref}} \leq i_{\text{min}}(i_k) \) and \( i_{\text{ref}} \geq i_{\text{max}}(i_k) \), respectively.

Within a classical context the controller could be seen as a sort of gain scheduled proportional controller that operates on the “predicted error signal”, \( i_{\text{ref}} - i_k e^{-n} \), (should the controller set \( r_{k:k+n} = 0 \)), and within such a context it is interesting to note that \( I \)-action is provided by continuously adjusting the value of \( \hat{R} \) so that the controller gives zero tracking error in steady state.
4.3 Damper Position and Velocity Estimator

The Kalman filter developed by Johansson and Kvaldén [2011] to fuse accelerometer and position sensor readings assumes that the accelerometer strictly measures the acceleration experienced by the piston rod relative to the cylinder head. The real accelerometer however also measures things such as the acceleration of the complete damper, (parts of) the gravity vector and, as most sensors, is never completely free of bias. A new Kalman filter was therefore developed where the state vector was augmented with a third state $\delta$ which represents the above mentioned accelerometer offset, which was modeled as a slowly varying random walk process, and the filter was made to estimate this state as well. The state space model then becomes

$$\frac{d}{dt} \begin{bmatrix} x \\ \dot{x} \\ \delta \end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 \\ 0 & 0 & 1 \\ 0 & 0 & 0 \end{bmatrix} \begin{bmatrix} x \\ \dot{x} \\ \delta \end{bmatrix} + \begin{bmatrix} 0 \\ 1 \\ 0 \end{bmatrix} \ddot{x} + \mathcal{N}(0, Q)$$  \hspace{1cm} (4.19)

$$z = \begin{bmatrix} 1 & 0 & 0 \end{bmatrix} \begin{bmatrix} x \\ \dot{x} \\ \delta \end{bmatrix} + \mathcal{N}(0, \sigma_x^2)$$  \hspace{1cm} (4.20)

and the standard form of linear Kalman Filter update equations can be applied.

4.4 Force Controller

The force controller receives the force reference $F_{\text{ref}}$ as an input and should, based on that and measured/estimated damper states such as the force $F_{\text{damping}}$, velocity $\dot{x}$, and position $x$, calculate the solenoid current reference value that is sent to the current controllers.

4.4.1 Classical Linear Feedback Approach

To check the feasibility of implementing a simplistic control scheme, where the current reference is chosen simply based on the force error, attempts were made with PID-controllers of the form

$$i_{\text{ref}}(s) = \left( K_P + K_I \frac{1}{s} + K_D \frac{s}{s^2 + 1} \right) F_e(s)$$  \hspace{1cm} (4.21)

where $F_e(s) = F_{\text{damping}}(s) - F_{\text{ref}}(s)$ and $s$ the Laplace variable. Despite dedicated tuning efforts and implementation of various features such as anti-windup, reset of the integrator at state transitions and different low pass filtering of the derivative those attempts however never led to satisfactory performance. It seems that the gain can not be made large enough to give good tracking while at the same time maintaining system stability, and the nonlinearity of the system makes the analysis difficult.
4.4.2 Exact Linearization Approach

Expanding ODE (3.11), which governs the internal pressures in the respective chambers during compression and rebound, gives the nonlinear state space model

\[
\dot{p}_{cmp} = \frac{B_{eff}}{V_{cmp}} \left( A_{cmp} \dot{x} - C_q A_{bleed} \text{sgn}(\Delta p_{cmp}) \sqrt{\frac{2}{\rho} |\Delta p_{cmp}|} - q_{CES,cmp} \right) \tag{4.22}
\]

\[
\dot{p}_{reb} = \frac{B_{eff}}{V_{reb}} \left( -A_{reb} \dot{x} - C_q A_{bleed} \text{sgn}(\Delta p_{reb}) \sqrt{\frac{2}{\rho} |\Delta p_{reb}|} - q_{CES,reb} \right) \tag{4.23}
\]

where \( \Delta p_{cmp} = p_{cmp} - p_{accu} \) and \( \Delta p_{reb} = p_{reb} - p_{accu} \). It can be observed that by choosing the CES-valve flows as

\[
q_{CES,cmp} = \frac{A_{cmp} \dot{x} - C_q A_{bleed} \text{sgn}(\Delta p_{cmp}) \sqrt{\frac{2}{\rho} |\Delta p_{cmp}|} + V_{cmp}}{B_{eff}} \tag{4.24}
\]

\[
q_{CES,reb} = -\frac{A_{reb} \dot{x} - C_q A_{bleed} \text{sgn}(\Delta p_{reb}) \sqrt{\frac{2}{\rho} |\Delta p_{reb}|} + V_{reb}}{B_{eff}} \tag{4.25}
\]

and treating \( q_{corr,cmp} \) and \( q_{corr,reb} \) as new inputs one is left with the linearized system

\[
\dot{p}_{cmp} = -q_{corr,cmp} \tag{4.26}
\]

\[
\dot{p}_{reb} = -q_{corr,reb}. \tag{4.27}
\]

This method of choosing the inputs such that the resulting system becomes linear is known as exact linearization and is described in e.g. Glad and Ljung [2003].

These two new inputs are not strictly flows per se; but become flows when scaled with the factor \( V/B_{eff} \) and it is hence useful to mentally treat them as such since they effect the system in much the same way.

Since the system now approximately behaves in a linear manner the inputs \( q_{corr,cmp} \) and \( q_{corr,reb} \) can be calculated using linear control principles. Various types of control structures were tried until eventually settling with PD-controllers of the form

\[
q_{corr}(s) = \left( K_P + K_D \frac{s}{\frac{1}{\tau} s + 1} \right) p_e(s) \tag{4.28}
\]

where

\[
p_e(s) = \frac{1}{A} (F_{ref}(s) - F_{damping}(s)) . \tag{4.29}
\]

The \( D \)-term was observed to help with system stability and hence enable slightly higher proportional gains to be chosen. Increasing \( K_d \) also decreases the height of overshoots, up to a point. By studying (4.26) and (4.27) it is made clear that, within a state space framework, the \( D \)-term represents an estimate of the current
control "flow".

I-action in the controller was found to be of negligible benefit and hard to implement robustly since a damper very seldom runs in steady-state and due to the fast switching between the compression/rebound controller every time the piston switches direction, and was hence not included.

The calculated ces-valve flows are used to lookup the corresponding current values in the CES-valve models, i.e.

\[
i_{ref,cmp} = i_{CES,cmp,static} (\Delta p_{cmp}, q_{CES,cmp}) \tag{4.30}
\]

\[
i_{ref,reb} = i_{CES,reb,static} (\Delta p_{reb}, q_{CES,reb}) , \tag{4.31}
\]

and the calculated currents are sent as reference values to the current controllers.

The pressure drop values \( \Delta p \) should theoretically be

\[
\begin{align*}
\Delta p_{cmp} &= p_{cmp} - p_{accu} \tag{4.32} \\
\Delta p_{reb} &= p_{reb} - p_{accu} \tag{4.33}
\end{align*}
\]

but it has been observed that choosing them as

\[
\begin{align*}
\Delta p_{cmp,ref} &= \frac{1}{A_{reb}} (F_{ref} - F_{fric} + A_{reb} p_{reb} + A_{rod} p_{amb}) - p_{accu} \tag{4.34} \\
\Delta p_{reb,ref} &= \frac{1}{A_{reb}} (F_{ref} - F_{fric} - A_{cmp} p_{cmp} + A_{rod} p_{amb}) - p_{accu} \tag{4.35}
\end{align*}
\]

which represent the values that the pressure drops should have according to the model if the force tracking was perfect, gives better results, indicating that it is better to use a prediction of what the pressure drop will be once the current has changed and the valve has begun to settle at its new position, than to use the present time values, and hence those are the expressions that are used in the controller.

4.4.3 Valve and Electrical System Simulation Study

When tuning the controller one is inevitable faced with the question of how the downstream system, comprising of the lookup table, valve and current controller behaves from the force controllers point of view. To investigate this simulation experiments were conducted at fixed pressure drop values while the input current was varied in a step like fashion to levels chosen by propagating flow reference values through the \( i_{CES,cmp,static} (\Delta p_{cmp}, q_{CES,cmp}) \) map. The results can be seen in Figure 4.2 on the next page. The same type of experiment was then performed with fixed flow instead of pressure, using the same current signal, while the pressure was monitored. The results can be seen in Figure 4.3 on the facing page. A zoomed in view of the obtained pressure drop signal can be seen in Figure 4.4 on page 32.

Two things are particularly clear when looking at these plots. One is that the valve’s dynamics differs quite significantly based on whether the current is increasing or decreasing, with faster dynamics for decreasing currents (in this plot perhaps
Figure 4.2: The results from simulation study with fixed pressure difference.

Figure 4.3: The results from simulation study with fixed flow rate.
Figure 4.4: The results from simulation study with fixed flow rate, zoomed in.
exaggerated by the fact that the upward steps are bigger in terms of current, but it also seems to be true in general.) The other thing is that the valve seems to have some sort of internal stiction phenomena that means that the flow or pressure drop can differ quite significantly for one and the same steady state current based on in what state the valve happened to “get stuck”. These two phenomena are probably the main limiters in terms of how big gains that can be chosen in the force controller without going into instability.

It is natural to think that the valve should dynamically behave as a second order system (perhaps with different parameters throughout the state space) since the flow resistance is set by the position of the main poppet, which is essentially a spring/mass system driven by the acceleration forces created by the main spring and the front to back pressure difference (with damping provided by the friction between the main poppet and its surrounding walls and by the oil). In reality it seems to be quite difficult to fit some kind of reduced model to the valve’s behavior.
Simulation Results

The developed control and estimation systems described in Chapter 4 were implemented in MATLAB/SIMULINK and iteratively developed and evaluated in closed loop together with the AMESim-model using co-simulation. During the development considerations were also taken to make sure that efficient C-code for Öhlins ECU could be generated from the SIMULINK-schematic, but unfortunately a complete hardware implementation could not be tested within the time frame of the thesis, due to the hysteresis in the developed force sensor described in Chapter 6. While testing using a real damper would have been preferable the simulation model is believed to be of sufficient accuracy to make evaluation using simulated performance meaningful.

5.1 Evaluation of Developed Current and Resistance Estimator

The solenoid current and estimation algorithm described in Section 4.2.3 was evaluated in simulation with additive normally distributed measurement noise with a standard deviation of 1 mA. A plot showing true and estimated current when the true current follows a step wise pattern is shown in Figure 5.1 and the estimated solenoid resistance in Figure 5.2 on the following page. Even though the initial guess for the resistance is intentionally made to be more than 50% off the true value, which is 4.6 Ω in this simulation, both the current and the resistance estimates can be observed to converge and track the true values very well.
Figure 5.1: Plot of the estimated and true current, $\hat{i}$ and $i$, during a simulation with normally distributed current measurement noise with a standard deviation of 1 mA.

Figure 5.2: Plot of how the estimated resistance value $\hat{R}$ converges to the true value (4.6 $\Omega$) when starting with an initial guess of 2 $\Omega$ while subjected to a normally distributed current measurement noise with a standard deviation of 1 mA.
5.2 Evaluation of Developed Current Controller

To evaluate the new model based current controller's performance in comparison to the old one both controllers ability to follow various reference signals was evaluated in simulation. The response to a step in reference signal can be seen in Figure 5.3. A plot from a simulation where the controllers follow a sampled white noise reference is given in Figure 5.4 on the following page.

To test the controllers’ robustness against modeling errors in the inductance $L$ simulation studies were performed and it was found that the controller could tolerate large such errors without degrading significantly. Over estimates of $L$ leads to bigger overshoots and under estimates to slower tracking, as expected, and both the state estimator and controller remain stable.

![Controller comparison – 0.2 [A] step response](image)

**Figure 5.3:** Comparison of step responses produced by the existing PID-current-controller and the new model based one.
Figure 5.4: Plot of a simulation result where the controllers follow a sampled white noise reference signal.
5.3 Evaluation of Developed Force Controller

The developed force controller was evaluated using different speed and reference signals. The step response at a fixed speed of 0.1 m/s, when going from 1000 N to 1100 N with the reference, is shown in Figure 5.5. As can be seen the controller achieves quite good steady state reference tracking but also produces quite a significant overshoot. This overshoot is believed to be mostly due to the characteristics of the valve since the overshoot produced by the closed loop controller is actually smaller than the one produced in open loop. The size of the overshoot can be reduced slightly by increasing the derivative gain, and this has been done to the largest extent found possible. It can be reduced further by going to lower proportional gains, but at quite a high cost in reduced speed/bandwidth. The results shown in Figure 5.5 represent what is believed to be a reasonable trade off between speed and overshoot, but more aggressive tunings in either direction could also be found. In open loop the system can be seen to oscillate slightly at this operating point and also give a steady state tracking error.

![Step Response](image)

**Figure 5.5:** Step response produced by the developed force controller.

The simulation result when tracking a clipped sine wave, while undergoing a sinusoidal velocity profile can be seen in Figure 5.6 on the next page. The force oscillates a bit after the sharp edge in the reference, but the oscillations get progressively smaller. The closed loop controller manages to reduce the root mean square tracking error to a factor of 61.4% of that of the existing controller.

The simulation result when tracking a perfect sine wave, while undergoing a sinusoidal velocity profile can be seen in Figure 5.6 on the following page. The
Figure 5.6: Tracking of a clipped sine force reference while undergoing a sinusoidal velocity profile.
root mean square tracking error for the closed loop controller is 62.2\% of that of the open loop controller. The biggest tracking errors for both controllers occur around the zero crossings, where the damper is unable to produce the requested damping force even at maximum current.

\begin{figure}
\centering
\includegraphics[width=\textwidth]{fig5_7.jpg}
\caption{Tracking of a smooth sine force reference while undergoing a sinusoidal velocity profile.}
\end{figure}
In this section work performed to enable the measurement of the force in the real damper is described. Three approaches were identified as feasible and studied further.

1. Integration of a commercial load cell somewhere inline with the piston rod.

2. Fitting of compression and rebound side pressure sensors and development of estimation algorithms which estimate the force using the measured pressures.

3. Application of strain gauge sensor elements onto the piston rod.

Out of these approach number one would have been the most straightforward if the mechanical integration could be worked out, but that proved to be quite challenging. Since it is desirable to measure the damping force in isolation from the spring force the load cell would have had to been mounted between the end of the piston rod and the end eye, inside of the spring platform, which would have required the construction of both a new piston rod and end eye, which would have been resource and time consuming.

Approach number two would have been interesting from a signal processing and modeling perspective but also resulted in an unknown uncertainty in the force signal due to the indirect nature of the measurement, and also would have required modifications to the prototype cylinder head.

Approach number three promised to provide the neatest installation while requiring the least amount of modifications, seemed to be an interesting challenge and was therefore chosen.
6.1 Strain Gauge Measurement Principle

A strain gauge consists of a precisely etched pattern, typically similar to the one shown in Figure 6.1, of resistive metal foil on a flexible backing. If the pattern is elongated in the direction of measurement its resistance $R$ increases due to the lengthening and narrowing of the individual traces, and similarly if the pattern is compressed the resistance decreases. This enables the measurement of the strain, $\epsilon = \Delta L/L$ i.e. the relative change in length, as $\epsilon = c \Delta R$, where $c$ is a constant, by gluing strain gauges to the surface of the structure and observing the resistance through sensitive electronics. In components such as the piston rod, which are subjected to stresses below the yield strength of the material, the force can be linearly related to the strain according to $F = EA \epsilon$, where $E$ is the Young’s modulus of the material and $A$ is the cross sectional area of the structure at the measurement position.

Combined this gives the expression

$$F = EA c \Delta R.$$ 

\textbf{Figure 6.1: Illustration showing a typical strain gauge pattern.}

6.2 Modified Piston Rod

To enable the strain gauges to be glued to the part of the piston rod which is inserted into the mounting hole in the end eye a slight indentation was machined off at the end of it. Since the original rod was deemed to be unnecessarily stiff a hole was also drilled through the middle of the measurement area to increase the strain levels.

Four strain gauges were then glued to the measurement area and wired up according to the illustration in Figure 6.3 in a configuration known as a full Wheatstone bridge. Four gauges at 90 degree angles were used, as opposed to one, to provide increased gain, temperature and cross load compensation.
Figure 6.2: Drawing showing the modified piston rod. (Design kindly provided by Geir Lindblad).

Figure 6.3: Illustration showing the stain gauge application.
6.3 Electronics

Even using four strain gauges in the Wheatstone bridge the sensor still only provides a full swing signal of a couple of millivolts and a high gain, low noise amplifier is therefore needed before the signal can be fed into a typical 0-5 V A/D-converter such as the one in the used ECU.

A suitable circuit was designed around an Analog Devices AD8556 instrument amplifier with digital gain control and offset compensation. To provide sufficient anti-alias filtering a first order R/C filter was integrated in addition to the second order filter present in the ECU. The resulting schematic can be seen in Figure 6.4.

![Schematic of the developed strain gauge signal amplifier and filtering electronics.](image)

To route the signal wires out of the damper could have been quite a challenge and ideally the amplifier should be fitted as close as possible to the strain gauges to minimize noise pickup. The printed circuit board seen in Figure 6.5 is designed to fit around the piston rod, just over the strain gauges, and rest against the top of the end eye. (If it can not be guarantied that the damper will never hit the end stop, such as when mounted to a bike for instance, care should be taken to fasten the PCB to the end eye securely and pod it for protection.)

A driver which enables the gain and offset of the AD8556 to be set via its serial interface through the ECU was then developed, along with software which reads the sensor values and transmits them over the CAN bus.

6.4 Results

An illustration of the constructed force sensor can be seen in Figure 6.6.

To evaluate the sensors performance it was mounted in a dynamometer with a high quality load cell and the readings were compared. The results can be seen in Figure 6.7. The sensor was observed to basically perform as expected, with good
Figure 6.5: Printed circuit board layout of the developed strain gauge signal amplifier and filtering electronics. The piston rod is inserted through the hole in the middle and the wires from the strain gauges are soldered to the four terminals adjacent to it.

linearity, low noise level and low drift. Unfortunately it was also found to suffer from about ±50 N of hysteresis when the force switches from being increasing to decreasing and vice versa. While this only represents 0.5% in relation to the complete measurement range it could pose a problem when used for feedback control. To not risk this affecting the results obtained with the force controller it was decided that the controller evaluation should be performed using simulations.
Figure 6.6: Picture of the constructed force sensor.
Figure 6.7: Force sensor reading attained from the sensor plotted against those measured by the high quality load cell in one of Öhlins damper dynamometers.
It is possible to achieve improved force tracking using a closed loop damping controller. The improvements found are mostly in terms of steady state force tracking performance and increased robustness against modeling inaccuracies such as those caused by wear, temperature changes, etcetera, and not so much in terms of increased force tracking bandwidth. The system has to be modeled to a greater extent than before and it has not been feasible to completely eliminate the usages of “black box” static maps in the modeling.

One advantage of the new controller is that the static maps that are used are only utilized to describe the CES-valves flow characteristics and not the whole damping system, as in the existing solution. This means that the mapping process potentially can be made only once, for the CES-valves themselves, and then applications to different dampers can be done through the mechanical knowledge of the individual dampers. This solution could be implemented also without the force feedback portion of the developed controller.

After having had the chance to study typical damper velocity and force request profiles it is the opinion of the author that the performance benefits that can be had using force feedback control does not fully motivate the increases in complexity and cost of such a system. In practice the biggest force tracking errors tend to be either very transient in nature or occur when there are firm hardware limitations present; such as when requesting damping forces which are outside of the range of what the damper can physically achieve at a prevailing velocity (in practice typically high damping forces at low damper velocities). Force feedback control can only have a minor effect in those circumstances and there is a distinct trade off between aggressively trying to follow the force reference and becoming increasingly sensitive to rapid future changes in the damper velocity. A typical
Conclusions and Future Work

The damper velocity profile in a real motorcycle application is similar to pink noise in nature and only weakly correlated to the damper force requests which are applied to minimize unwanted chassis movements. This means that the damper seldom runs in steady state circumstances where force feedback control has been shown to give the biggest benefits.

It has been possible to significantly increase the performance of the solenoid current controller while at the same time also providing an estimate of the solenoids series resistance which could be useful also for other purposes such as diagnosis and estimation of solenoid temperature. Implementing this in the real ECU could possibly enhance the performance of the system for a comparatively small amount of work and would be highly interesting.

The TTX-CES damper model developed during this work provides the first ever implementation of such a model capable of capturing the damper’s behavior with a satisfactory level of accuracy. The physical modeling approach used means that the model parameters to a large extent are real measurable quantities and therefore enables interesting design and parameter optimization studies to be performed in the future.

Unfortunately it has not been possible to benchmark the new force controller on actual damper hardware because of limitations of the constructed force sensor and the ECU platform, which would have been too time consuming to resolve during a single Master’s thesis. Verifying the performance in practice represents interesting future work.

It would be highly interesting to simply log the signal from the developed force sensor piston rod, when fitted in a damper in a real bike during a test scenario, and compare that to the given reference, even if still using the existing controller. The evaluation could provide interesting insights into the system performance during real world usage, which is still a bit of an unknown.

Another interesting future possibility would be to investigate whether the damper- and/or CES-valve model, using a suitable formulation or map, could be iteratively adapted based on the feedback provided by the force sensor, since that could possibly enhance the accuracy of the critical feedforward part of the controller during varying conditions, which could potentially increase both the performance and robustness.
Appendix
\[
\frac{\partial f}{\partial R} = -\frac{inTe^{-\frac{nRT}{L}}}{L} + \frac{U \left( 1 - e^{-\frac{nRT}{L}} \right) \left( e^{\frac{rRT}{L}} - 1 \right)}{R^2 \left( e^{\frac{RT}{L}} - 1 \right)} + \frac{nTUe^{-\frac{nRT}{L}} \left( e^{\frac{rRT}{L}} - 1 \right)}{LR \left( e^{\frac{RT}{L}} - 1 \right)} + \frac{rTU \left( 1 - e^{-\frac{nRT}{L}} \right) e^{\frac{rRT}{L}}}{LR \left( e^{\frac{RT}{L}} - 1 \right)} - \frac{TUe^{\frac{RT}{L}} \left( 1 - e^{-\frac{nRT}{L}} \right) \left( e^{\frac{rRT}{L}} - 1 \right)}{LR \left( e^{\frac{RT}{L}} - 1 \right)^2}
\]

\[
\frac{\partial f}{\partial L} = \frac{inRTe^{-\frac{nRT}{L}}}{L^2} + \frac{TUe^{\frac{RT}{L}} \left( 1 - e^{-\frac{nRT}{L}} \right) \left( e^{\frac{rRT}{L}} - 1 \right)}{L^2 \left( e^{\frac{RT}{L}} - 1 \right)^2} - \frac{nTUe^{-\frac{nRT}{L}} \left( e^{\frac{rRT}{L}} - 1 \right)}{L^2 \left( e^{\frac{RT}{L}} - 1 \right)} - \frac{rTU \left( 1 - e^{-\frac{nRT}{L}} \right) e^{\frac{rRT}{L}}}{L^2 \left( e^{\frac{RT}{L}} - 1 \right)}
\]
function [i_hat_out, var_i_hat_out, R_hat_out] = solenoid_est(i_meas, i_pred, params)

% Constants
T_PWM = single(1/params.solenoid_est_and_ctrl.PWM_frequency);
T_s = single(1/params.solenoid_est_and_ctrl.exec_frequency);
N = single(T_s/T_PWM);

L = single(params.solenoid.inductance);
R_hat_0 = single(params.solenoid.resistance);

var_i_dot_ss = single((1e-2)^2);
var_i_dot_deltai_coeff = single(1e3);
var_i_hat_0 = single((1)^2);
var_i_meas = single((1e-3)^2);

% Initialisation
persistent i_hat var_i_hat R_hat;
if isempty(i_hat)
    i_hat = single(0);
    var_i_hat = var_i_hat_0;
    R_hat = R_hat_0;
end

% Kalman filter update for i (the solenoid current)
i_hat_prev = i_hat;
f_i = exp(-(R_hat/L)*T_PWM*N);
% Account for the additional uncertainties that stem from the uncertanty
% in L (the solenoid inductance) by adding a variance term that's
% proportional to the square of the change in current. (That is, express
% that the model should be trusted less and the measurements more when
% there are big changes in the current.)
delta_i = i_pred - i_hat_prev;
var_i_dot = var_i_dot_ss + var_i_dot_deltai_coeff*delta_i^2;
var_i_hat = f_i^2*var_i_hat + var_i_dot*T_s; % A priori estimate variance
var_i_resid = var_i_hat + var_i_meas; % Modeled residual variance
i_inov = i_meas - i_pred; % Innovation
k_i = var_i_hat/var_i_resid; % Kalman gain
i_hat = i_pred + k_i*i_inov; % Filter estimate
var_i_hat_out = var_i_hat;

% Adaptive filter update for R
% Main idea: Adapt the most when we have had small changes in current,
% since in those cases the prediction error should be a pretty good
% indication of the error in R (and not stem from any inaccuracies in L).
mu = -1*exp(-(i_hat-i_hat_prev)^2/(0.01^2)); % Adaptation gain for R
R_hat = R_hat + mu*i_inov;
R_hat_out = R_hat;
end
% Model predictive current controller and for CES solenoid
% Per Svennerbrandt, 2013
function [r, i_pred] = solenoid_mpc(i_ref, i_hat, R_hat, U_batt, params)

%#codegen
% Constants
% Period of the PWM signal [s]
T_PWM = single(1/params.solenoid_est_and_ctrl.PWM_frequency);
% Delta time between executions of this code [s]
T_s = single(1/params.solenoid_est_and_ctrl.exec_frequency);
% Nr of PWM periods during the control horizon
N = single(T_s/T_PWM);

% Nominal/initial parameter values
% Coil inductance [Henry]
L = single(params.solenoid.inductance);

% One step prediction model based controller
i_max = U_batt/R_hat + (i_hat - U_batt/R_hat)*exp(-(R_hat/L)*T_PWM*N);
i_min = i_hat*exp(-(R_hat/L)*T_PWM*N);
if i_ref >= i_max
    r = single(1);
i_pred = i_max;
elseif i_ref <= i_min
    r = single(0);
i_pred = i_min;
else
    r = L/(R_hat*T_PWM)*log((i_ref - i_hat*exp(-(R_hat/L)*T_PWM*N))/...(U_batt/R_hat)*exp(-(R_hat/L)*T_PWM*N)*sum(exp((R_hat/L)*T_PWM*(single(0):single(T_s/T_PWM)-1))));
i_pred = i_ref;
end
/* ad8556.h */

#ifndef AD8556_H
#define AD8556_H

/* Functions that should be called at regular intervals to generate 
the appropriate serial interface waveforms. */
void ad8556_pulse_start(void);
void ad8556_pulse_after_50ns_before_10us(void);
void ad8556_pulse_after_50us(void);

void ad8556_init(void);
void ad8556_set_first_stage_gain(Uint16 gain_code);
void ad8556_set_second_stage_gain(Uint16 gain_code);
void ad8556_set_output_offset(Uint16 offset_code);

void ad8556_dryrun(void);

#endif /* AD8556_H */

/*
ad8556.c - Utility functions for handling an Analog Devices AD8556. 
Written by Per Svennerbrandt, 2013
*/

#include "DSP2833x_Device.h"
#include "digital_io.h"

/*
AD8556 mini intro:
*/
Single supply instrumentation amplifier with digitally programmable gain and offset (via a single wire serial interface).

The single wire serial interface uses a pulse width based protocol:
- A logical zero (0) is written as a pulse (low-high-low) that’s between 50ns and 10us long.
- A logical one (1) is written as a pulse that’s over 50us long.
- Time between pulses should be 10 us minimum.

Serial command structure:

<table>
<thead>
<tr>
<th>Field No.</th>
<th>Bits</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0 to 11</td>
<td>12-Bit Start of Packet 1000 0000 0001</td>
</tr>
<tr>
<td>1</td>
<td>12 to 13</td>
<td>2-Bit Function</td>
</tr>
<tr>
<td></td>
<td>0</td>
<td>00: Change Sense Current</td>
</tr>
<tr>
<td></td>
<td>1</td>
<td>01: Simulate Parameter Value</td>
</tr>
<tr>
<td></td>
<td>10</td>
<td>10: Program Parameter Value</td>
</tr>
<tr>
<td></td>
<td>11</td>
<td>11: Read Parameter Value</td>
</tr>
<tr>
<td>2</td>
<td>14 to 15</td>
<td>2-Bit Parameter</td>
</tr>
<tr>
<td></td>
<td>0</td>
<td>00: Second Stage Gain Code</td>
</tr>
<tr>
<td></td>
<td>1</td>
<td>01: First Stage Gain Code</td>
</tr>
<tr>
<td></td>
<td>10</td>
<td>10: Output Offset Code</td>
</tr>
<tr>
<td></td>
<td>11</td>
<td>11: Other Functions</td>
</tr>
<tr>
<td>3</td>
<td>16 to 17</td>
<td>2-Bit Dummy 10</td>
</tr>
<tr>
<td>4</td>
<td>18 to 25</td>
<td>8-Bit Value</td>
</tr>
<tr>
<td></td>
<td>0</td>
<td>Parameter 00 (Second Stage Gain Code): 3 LSBs Used</td>
</tr>
<tr>
<td></td>
<td>1</td>
<td>Parameter 01 (First Stage Gain Code): 7 LSBs Used</td>
</tr>
<tr>
<td></td>
<td>10</td>
<td>Parameter 10 (Output Offset Code): All 8 Bits Used</td>
</tr>
<tr>
<td></td>
<td>11</td>
<td>Parameter 11 (Other Functions)</td>
</tr>
<tr>
<td></td>
<td>0</td>
<td>Bit 0 (LSB): Master Fuse</td>
</tr>
<tr>
<td></td>
<td>1</td>
<td>Bit 1: Fuse for Production Test at Analog Devices</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>Bit 2: Parity Fuse</td>
</tr>
<tr>
<td>5</td>
<td>26 to 37</td>
<td>12-Bit End of Packet 0111 1111 1110</td>
</tr>
</tbody>
</table>
struct ad8556_command_fields
{
    Uint64 pkg_end:12; /* End pattern 0111 1111 1110 */
    Uint64 value:8;
    Uint64 dummy:2; /* Dummy pattern 10 */
    Uint64 parameter:2;
    Uint64 function:2;
    Uint64 pkg_start:12; /* Start pattern 1000 0000 0001 */
};

#define AD8556_NUM_COMMAND_BITS 38
#define AD8556_START_PATTERN (1 << 11) | 1
#define AD8556_DUMMY_PATTERN (1 << 1)
#define AD8556_END_PATTERN ~(AD8556_START_PATTERN)

union ad8556_command
{
    Uint64 all;
    struct ad8556_command_fields fields;
};

enum ad8556_function_id
{
    CHANGE_SENSE_CURRENT,
    SIMULATE_PARAMETER_VALUE,
    PROGRAM_PARAMETER_VALUE,
    READ_PARAMETER_VALUE
};

enum ad8556_parameter_id
{
    SECOND_STAGE_GAIN_CODE,
    FIRST_STAGE_GAIN_CODE,
    OUTPUT_OFFSET_CODE,
    OTHER_FUNCTIONS
};

static union ad8556_command ad8556_current_command;
static volatile int ad8556_current_bit_pos = -1;
static volatile int ad8556_transmission_in_progress = 0;

static inline int bit_is_clear(Uint64 var, int bit_pos)
{
    return !(var & (1ULL << bit_pos));
}

#pragma CODE_SECTION(ad8556_pulse_start, "ramfuncs");
void ad8556_pulse_start(void)
{
    if (ad8556_current_bit_pos >= 0)
    {
        /* Set the output high if there's a bit to transmit. */
        digital_output_a_set_high();
        ad8556_transmission_in_progress = 1;
    }
if (ad8556_transmission_in_progress &&
    bit_is_clear(ad8556_current_command.all, ad8556_current_bit_pos))
{
    /* Set the output low at this point if we're transmitting a
     * zero. */
    digital_output_a_set_low();
}

if (ad8556_transmission_in_progress)
{
    /* Unconditionally set the output low at this point. */
    digital_output_a_set_low();
    /* We're now done with this bit - move on to the next. */
    ad8556_current_bit_pos--;
}

int ad8556_send_command(enum ad8556_function_id function,
                        enum ad8556_parameter_id parameter,
                        Uint16 value)
{
    if (ad8556_transmission_in_progress)
    {
        /* Simply refuse to add the new command and return failure if a
         * previous transmission is still in progress (since otherwise we
         * would overwrite the partially written command mid transmission
         * and likely corrupt it). This function succeeds when (at least)
         * AD8556_NUM_COMMAND_BITS * 0.1ms = 3.8ms has elapsed since the
         * previous invocation. */
        return 0;
    }
    else
    {
        /* No transmission is currently in progress. Copy the supplied
         * command to the data structures read by the transmit functions
         * and let them begin to generate the appropriate waveforms. */
        ad8556_current_command.fields.pkg_start = AD8556_START_PATTERN;
        ad8556_current_command.fields.function = function;
        ad8556_current_command.fields.parameter = parameter;
        ad8556_current_command.fields.dummy = AD8556_DUMMY_PATTERN;
        ad8556_current_command.fields.value = value;
        ad8556_current_command.fields.pkg_end = AD8556_END_PATTERN;
        ad8556_current_bit_pos = AD8556_NUM_COMMAND_BITS - 1;
        return 1;
    }
void ad8556_set_first_stage_gain(Uint16 gain_code) {
    ad8556_send_command(SIMULATE_PARAMETER_VALUE,
                        FIRST_STAGE_GAIN_CODE,
                        gain_code);
}

void ad8556_set_second_stage_gain(Uint16 gain_code) {
    ad8556_send_command(SIMULATE_PARAMETER_VALUE,
                        SECOND_STAGE_GAIN_CODE,
                        gain_code);
}

void ad8556_set_output_offset(Uint16 offset_code) {
    ad8556_send_command(SIMULATE_PARAMETER_VALUE,
                        OUTPUT_OFFSET_CODE,
                        offset_code);
}

void ad8556_wait_until_ready(void) {
    while (ad8556_transmission_in_progress || ad8556_current_bit_pos >= 0) {
        /* Simply busy-loop until any ongoing transmission is complete. */
    }
}

void ad8556_init(void) {
    /* Set second stage gain to 100 */
    ad8556_set_second_stage_gain(5);
    ad8556_wait_until_ready();

    /* Set first stage gain to 5.145 */
    ad8556_set_first_stage_gain(100);
    ad8556_wait_until_ready();

    /* Set output offset to 0V */
    ad8556_set_output_offset(0);
    ad8556_wait_until_ready();
}

void ad8556_dryrun(void) {
    ad8556_set_second_stage_gain(5);
    while (ad8556_current_bit_pos >= 0) {
        ad8556_pulse_start();
        ad8556_pulse_after_50ns_before_10us();
        ad8556_pulse_after_50us();
    }
}


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