Institutionen för systemteknik
Department of Electrical Engineering

Examensarbete

Modeling and Control of an Electro-Pneumatic Actuator System Using On/Off Valves

Examensarbete utfört i Fordonssystem
vid Tekniska högskolan i Linköping
av

Klas Håkansson och Mikael Johansson

LITH-ISY-EX--07/3971--SE

Linköping 2007

Department of Electrical Engineering
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Linköping, 9 March, 2007

Avdelning, Institution
Division, Department

Division of Vehicular Systems
Department of Electrical Engineering
Linköpings universitet
SE-581 83 Linköping, Sweden

Datum
Date

2007-03-09

Språk
Language

☐ Svenska/Swedish
☒ Engelska/English
☐ ______

Rapporttyp
Report category

☐ Licentiatavhandling
☒ Examensarbete
☐ C-uppsats
☐ D-uppsats
☐ Övrig rapport
☐ ______

ISBN
—

ISRN
LITH-ISY-EX--07/3971--SE

Serietitel och serienummer
Title of series, numbering

ISSN
—

URL för elektronisk version
http://www.vehicular.isy.liu.se
http://www.ep.liu.se/2007/3971

Titel / Title

Modellering och reglering av ett elektropneumatiskt aktuatorsystem med On/Off ventiler

Modeling and Control of an Electro-Pneumatic Actuator System Using On/Off Valves

Författare / Author

Klas Håkansson och Mikael Johansson

Sammanfattning
Abstract

To control the exhaust gas recirculation (EGR) and the exhaust brake, the position of a butterfly valve connected to a piston inside a pneumatic cylinder is controlled by altering the pressure inside the cylinder. This thesis evaluates the possibility to do this with pulse width modulation (PWM) controlled On/Off valves. The whole electro-pneumatic actuator system is built out of two On/Off valves and a cylinder.

A mathematical model of the system is constructed. The complete system model on state space form consists of nine states and is nonlinear. The model captures the dynamics of the system. The statics of the system is not captured as accurately. The model is still good enough to be used as aid when developing control strategies, since position feedback is available.

Automatic control strategies for the system are first developed and tested in simulation. The first approach is PID control. Because of the nonlinear properties of the system the results from a PID with a constant proportional part is unsatisfactory. To cope with the nonlinearities, a fuzzy controller is constructed; the results prove somewhat better, but not as good as expected due to implementation difficulties.

In a test bench the system is controlled by a P controller with feedforward from position. The feedforward strongly reduces the nonlinear behavior of the system. With this implementation the results that were hoped for with the fuzzy controller are reached.

Nyckelord
Keywords

On/Off solenoid valve, modeling, PID control, fuzzy control

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Acknowledgments

We, the authors, would like to express our gratitude to a number of people who have made this thesis work easier.

First we would like to thank Scania CV AB for the opportunity to conduct this master's thesis work at the company and for the friendly reception from all co workers in Södertälje. In particular, we would like to thank our supervisor Henrik Johansson for the help and engagement he has put into our work. We would also like to thank our supervisor at Linköping University, Johan Wahlström, for constant feedback and good suggestions on our work.

For showing a special interest in our work and for providing our questions with answers, we would like to thank Anders Holmgren, Leif Pudas and Tommy Sahi. For providing us with information about the valves, we would like to thank the valve manufacturer Norgren GmbH.

For sharing their office spaces, we would like to thank the co workers at the Scania departments NMCE and RDR. For making our working days more delightful through chitchat and coffee brakes we would like to thank our fellow thesis students at Scania, Rikard Falkeborn, Samer Hadad, Joakim Hansen, Mary Lam, Jens Molin, Carl Svärd and Henrik Wassén.

Mikael would like to thank Ylva Hilmertz for her constant love and support.

Klas Håkansson & Mikael Johansson
Södertälje, 15 February 2007

Contents

Chapter 1

Introduction

This master's thesis was performed at Scania CV AB in Södertälje. Scania is a manufacturer of heavy duty trucks, buses and engines for industrial and marine use. The department of Powertrain Control System Development develops the control systems for the powertrain of Scania vehicles. Among these systems the control systems for the exhaust gas recirculation (EGR) valve and the exhaust brake valve are found.

In a combustion engine a mixture of air and fuel is ignited. When the temperature of the combustion reach a certain point and oxygen is present some of the nitrogen in the air burns. As a result of this compounds of nitrogen and oxygen, in general called NO_{x}, is formed. NO_{x} is health hazardous and its origin is desired to be limited. This is also regulated by emission legislation. One method used to achieve this is called EGR. As the name implies a portion of the exhausts is recirculated to reduce the amount of oxygen and to lower the peak temperature in the combustion chamber. In order to control the amount of recirculated exhausts a butterfly valve is used. The valve can be controlled by a pneumatic system that consists of two On/Off valves and a cylinder. This thesis work aims to investigate the performance and possibilities of such an electro-pneumatic actuator system used to control the butterfly valve. A process model is produced and control strategies are constructed and discussed.

1.1 Background

The benefits of using On/Off valves are that they are robust and low in cost. On/Off valves are normally considered to be either closed or open. But in this specific configuration the valves open and close so often that their dynamics can not be neglected. Therefor it is interesting to learn more about the valves and use this knowledge to develop control strategies for the system.

1.2 Thesis Objectives

The goals of this thesis work are to gain knowledge about the dynamics in the On/Off valves by constructing a physical model of the system and to find out what performance it is possible to achieve with this specific system configuration.

1.3 Target Group

Target group for this thesis is undergraduate and graduate engineer students and employees at Scania CV AB.

1.4 Related Work

A master's thesis [13] was performed at Scania CV AB on a similar system using a proportional valve instead of On/Off valves. Modeling of similar systems is performed in a number of conference contributions and articles [1, 11, 12, 14, 15, 16]. Control of a similar system is performed in [10].

1.5 Thesis Outline

Chapter 1 An introduction to the thesis where the background and the objectives are presented.

Chapter 2 A description of the system and modeling. A mathematical model is produced and presented on state space form.

Chapter 3 Calibration of the model.

Chapter 4 Validation of the model.

Chapter 5 The discovered and investigated properties of the system are presented.

Chapter 6 A general discussion on control of the system. A PID controller and a fuzzy-I controller are developed and evaluated.

Chapter 7 The results of the thesis are presented and suggestions on future work are given.

1.6 Notation

Parameter	Description
Valves	
l_{coil}	length of the coil
A_{p}	cross sectional area of the plunge
d_{o}	diameter of the orifice
N	number of turns in the coil
R_{c}	coil resistance
L	coil inductance
L_0	initial coil inductance
α_{L}	inductance proportional constant
x_{p}	plunge position
x_0	minimum air gap between plunge and plunge seat
m_{p}	mass of the plunge
k_{p}	spring constant of the plunge spring
b_{p}	friction coefficient between plunge and housing wall
$C_{\mathrm{d_{sup}}}$	discharge coefficient for the supply valve
$C_{\mathrm{d_{exh}}}$	discharge coefficient for the exhaust valve
T_{c}	coil temperature
Cylinder	
L_{cyl}	piston stroke
A_{cyl}	piston effective area
k_{cyl}	spring constant of the cylinder spring
m_{piston}	mass of the piston
x_{cyl}	position of the piston inside the cylinder
Physical	
μ_0	permeability
R	ideal gas constant
T_{air}	air temperature
γ	ratio of specific heat ($= c_{\mathrm{p}}/c_{\mathrm{v}}$)
Common	
P_{sup}	supply pressure
P_{atm}	atmospheric pressure
T	period time

Abbreviation	Meaning
ECU	Engine control unit
EGR	Exhaust gas recirculation
KVL	Kirchhoff's voltage law
PWM	Pulse width modulation

Chapter 2

System Modeling

In order to understand the characteristics of a system it is important to know how the subsystems that form the system work and how they interact. The reason why a deeper knowledge of a system is desired is that this knowledge makes it possible to predict how the system reacts and what outputs can be expected from certain inputs. Even if the information provided by a model is not used explicitly in the controller, the development of the controller is facilitated with the help of a model.

To conduct experiments is one way to collect information about the characteristics of the system. These experiments are often too expensive, too dangerous or sometimes even impossible to conduct. Another way is to build a mathematical model of the system. With inexpensive and powerful computers it is possible to use the mathematical models to *simulate* different scenarios. This is a cheap, safe and practical alternative to conducting experiments. The results of such simulations are completely dependent on the quality of the model that has been built of the system. To use a model can also ease the work of understanding and controlling the system and make it more efficient, which is the case in this thesis.

There are two different basic principles when constructing models. One way is to use known physical relations, such as Newton's laws or Ohm's law, that describe the subsystems and combined describe the complete system. This way of building models is called *physical modeling* and is only possible when the physical characteristics of all subsystems are known. If the physical characteristics for some reason are unknown, it is possible to observe (measure) how the system behave and then fit the parameters of the model to the actual system. This way to build models is called *identification*, resulting in a so called *black box model*.

As mentioned above, it is important to get a concept of how reliable the model of the system is. All models have limitations of how well they describe the system. The investigations of those limitations are made by *verification* or *validation*. Model verifications and validations are made by comparing the models behavior with those of the actual system and then evaluate the differences.

This chapter will describe the modeling of the electro-pneumatic actuator system. The method that is used is physical modeling, where system equations for the different subsystems are derived from known physical relations. More about system modeling can be found in [7].

2.1 System Description

The pneumatic system is built out of two single solenoid On/Off valves and a cylinder with position feedback. The valves are of 2/2 type which means they have two ports (inlet/outlet) and two functions (opened/closed) each. When the valve is open the two ports have the same pressure. One of the valves, from now on referred to as the *supply valve*, has its inlet port connected to supply pressure (8.2 bar) and its outlet port connected to the cylinder and the inlet port of the other valve. The second valve, from now on referred to as the *exhaust valve*, has its inlet port connected to the outlet port of the supply valve and the cylinder. Its outlet port is connected to atmospheric pressure, see Figure 2.1.

The cylinder has one pneumatic connection. Inside the cylinder a piston runs whose position is pressure dependent. When the pressure in the cylinder is lowered a spring pushes the piston back. The piston can mechanically be connected to a butterfly valve. In this way the position of the butterfly valve depends on the pressure in the cylinder. To raise the pressure in the cylinder, the supply valve is opened. To lower it, the exhaust valve is opened. An overview of the system configuration is shown in Figure 2.1.

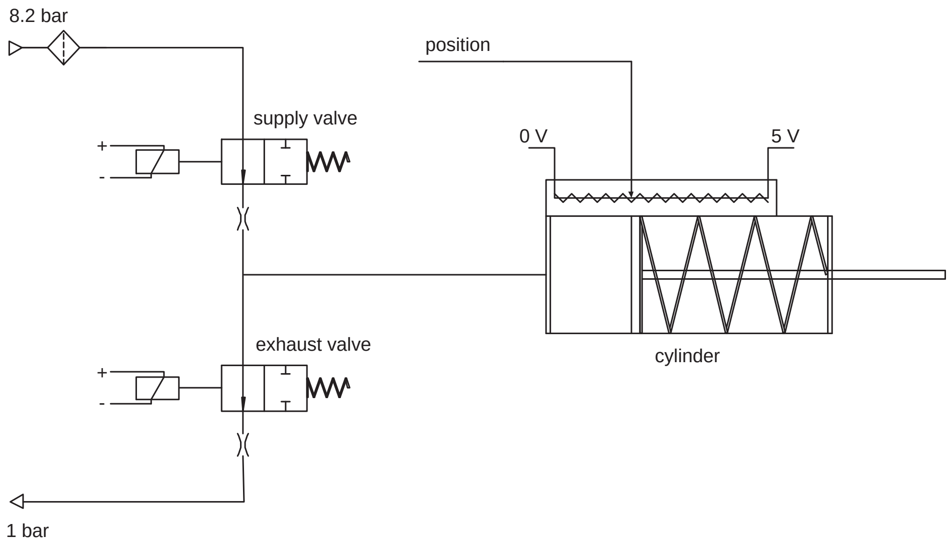


Figure 2.1. Overview of the system configuration with the two On/Off valves and the cylinder.

The On/Off valves are built of a plunge with a spring connected to it, see Figure 2.2. The spring keeps the plunge at an end position which equals to closed valve ($x_{\mathrm{p}} = 0$). The valves are normally closed. The plunge is positioned inside a copper coil connected to an electric power source. When the circuit is closed a magnetic field is induced, applying a force on the plunge which overcomes the force from the spring and moves the plunge to the other end position, resulting in an open valve. Depending on how the valve is connected, higher pressure either works against or together with the spring because a difference in pressure will apply a force on the plunge. This affects closing/opening time of the valve, more about this will be discussed in the sections 2.3.3 and 5.5.

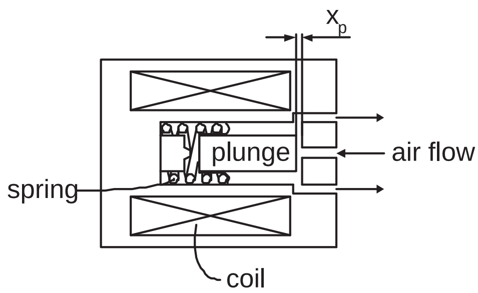


Figure 2.2. Sketch of the On/Off valve.

2.2 The Input Signal

The input signal to a valve is pulse width modulated (PWM) voltage. The frequency of the signal in this application is 50 Hz. This means that the valves open and close every 20 ms when run. There is no point in running the valves faster than 50 Hz since the current through the coil would not have time to change. It would result in a system with a very small regulating range. The reason for this will be discussed in section 5.1. The choice of frequency is a trade off between accuracy and speed. With a high frequency the control signal can be updated more often, with a low frequency the regulating range is larger. The choice of input signal, u, is the choice of for how long of the period the signal should be high and is defined by the quotient

$$u = \frac{T_\mathrm{d}}{T} \tag{2.1}$$

where T_d is the duty cycle and T is the period time. The duty cycle is the time for how long the input signal is high within one period. Assuming the actual applied voltage is denoted u_pwm, it is defined within every period by

$$u_\mathrm{pwm} = \begin{cases} \text{high}, & t \leq uT \\ \text{low}, & t > uT. \end{cases} \tag{2.2}$$

u is often given in percentage, $u \in [0\%, 100\%]$. Three exampels of different PWM signals are shown in Figure 2.3.

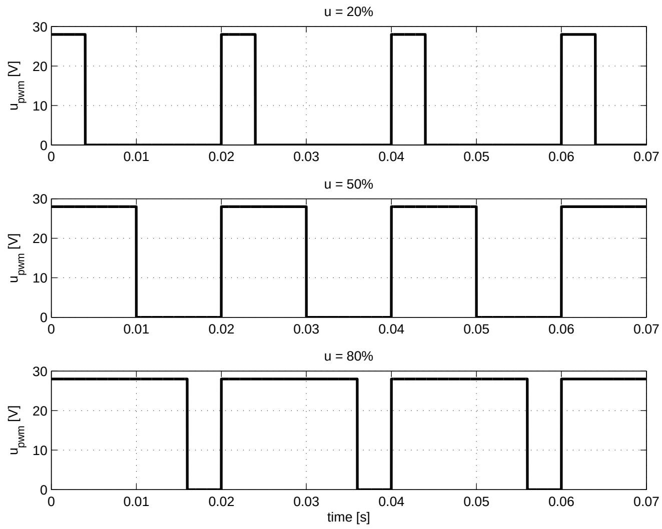


Figure 2.3. Simulated PWM signals with period time $T = 20$ ms where $u_{\text{pwm}} = 28$ V represents high and $u_{\text{pwm}} = 0$ V represents low. From top to bottom: $u = 20\%$, $u = 50\%$ and $u = 80\%$.

2.3 Valve Model

The valve model is divided into four sub models. A block diagram of the valve model is shown in Figure 2.4.

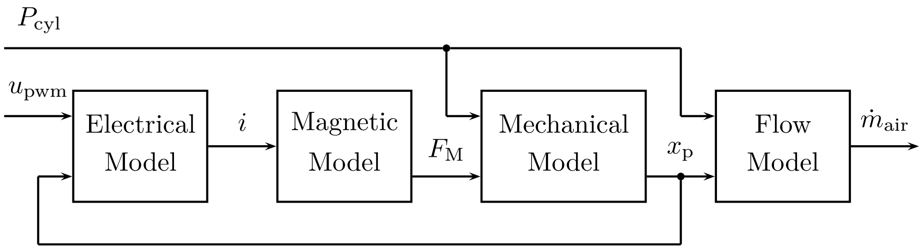


Figure 2.4. Block diagram of the valve.

The blocks Electrical model and Magnetic model form an electromagnet that together can be seen as a voltage-to-force-converter.

In the mechanical model, the plunge movement is modeled.

In the flow model, the flow through the valve is modeled.

2.3.1 Electrical Model

The electrical model determines the current through the coil in the valve, i, and depends on applied voltage, u_{pwm}, and plunge position, x_{p}. A block diagram of the electrical model is shown in Figure 2.5.

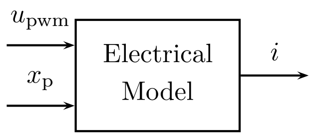


Figure 2.5. Block diagram of the electrical model.

The electric circuit in the electromagnet is modeled as a resistance in series with an inductance, see Figure 2.6.

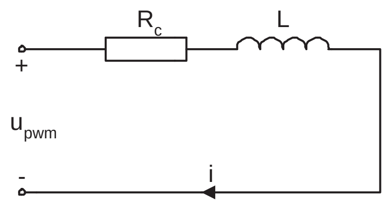


Figure 2.6. Schematic figure of the electric circuit.

The voltage drop in the electric circuit is then given by Kirchhoff's voltage law (KVL)

$$u_{\text{pwm}} = R_{\text{c}} i + V_{\text{L}} \tag{2.3}$$

where u_{pwm} is the input voltage, R_{c} is the coil resistance, i is the current and V_{L} is the voltage drop over the inductance. The inductance in a solenoid is defined by [9]

$$\Phi_{\text{B}} = \frac{Li}{N} \tag{2.4}$$

where Φ_{B} is the magnetic flux, L is the inductance and N is the number of windings in the coil. The voltage drop over the inductance is defined by [9]

$$V_{\text{L}} = N\frac{d\Phi_{\text{B}}}{dt} = \frac{d}{dt}(Li) = L\frac{di}{dt} + i\frac{dL}{dt}. \tag{2.5}$$

Now, the differential equation describing the electric circuit is given by

$$u_{\text{pwm}} = R_{\text{c}} i + L\frac{di}{dt} + i\frac{dL}{dt}. \tag{2.6}$$

The inductance, L, is according to the data sheets provided by the valve manufacturer dependent on if the valve is switched on or off. When the valve is switched on, the plunge moves into the coil resulting in an increasing presence of an iron core and the inductance increases, see Figure 2.2. Therefor, the inductance is modeled as a linear function that increases with plunge position x_p, i.e.

$$L(x_\mathrm{p}) = L_0 + \alpha_\mathrm{L} x_\mathrm{p} \tag{2.7}$$

where α_L is a parameter chosen to fit the data provided by the manufacturer.

Since the inductance is dependent on the plunge position, x_p, the term $i\frac{dL}{dt}$ in equation (2.6) will depend on $\dot{x}_\mathrm{p}$. The magnitude of $\dot{x}_\mathrm{p}$ is however practically impossible to have concept of. It is also separated from zero only when the plunge is in motion, which is a very short period of time. The term $i\frac{dL}{dt}$ is therefor neglected. This transforms the model equation (2.6) together with (2.7) into

$$u_\mathrm{pwm} = R_\mathrm{c} i + (L_0 + \alpha_\mathrm{L} x_\mathrm{p})\frac{di}{dt}. \tag{2.8}$$

2.3.2 Magnetic Model

The magnetic model determines the magnetic force, F_M, affecting the plunge in the valve and depends on the current through the coil, i. A block diagram of the magnetic model is shown in Figure 2.7.

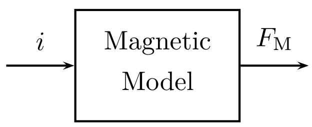


Figure 2.7. Block diagram of the magnetic model.

The real force in a magnetic media is [9]

$$dF = T_\mathrm{Max} dA \tag{2.9}$$

where T_Max is the Maxwell stress tensor. It is defined by [9]

$$T_\mathrm{Max} = T_\mathrm{e} + T_\mathrm{m} \tag{2.10}$$

where

$$\begin{cases} T_\mathrm{e} = \frac{1}{2}\varepsilon_0 E^2 \\ T_\mathrm{m} = \frac{1}{2\mu_0} B^2 \end{cases} \tag{2.11}$$

and ε_0 is the permittivity, E is the electric field, μ_0 is the permeability and B is the magnetic flux density, given by the equation [9]

$$B = \frac{\mu_0 N i}{x_0 - x_\mathrm{p}} \tag{2.12}$$

where $x_0 - x_\mathrm{p}$ is the length of the air gap between the two attracting objects. The size of T_e is neglectable compared the size of T_m, and with $T_\mathrm{Max} dA = T_\mathrm{Max} A_\mathrm{p}$ the equation describing the magnetic force is

$$F_\mathrm{M} = T_\mathrm{m} A_\mathrm{p} = \frac{\mu_0 A_\mathrm{p} N^2 i^2}{2(x_0 - x_\mathrm{p})^2} \tag{2.13}$$

where A_p is the cross sectional area of the plunge.

2.3.3 Mechanical Model

As mentioned in section 2.1 the valves contain a moving plunge. The mechanical model determines the position of the plunge, x_p, which depends on the pressure inside the cylinder, P_cyl, and the magnetic force induced by the electromagnet, F_M. A block diagram of the mechanical model is shown in Figure 2.8.

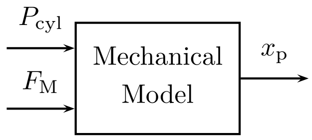


Figure 2.8. Block diagram of the mechanical model.

The plunge is affected by the magnetic force F_M and a spring. The spring has the spring constant k_p and the preload F_pld. Other influences on the plunge are viscous friction (damping with coefficient b_p) and force from pressure, F_prs, calculated as in (2.14), acting together with the magnetic force. Neglectable forces are force of gravity, Coulomb friction and static friction. The equation of motion is retained with the help of a free body diagram of the plunge, see Figure 2.9, and Newton's law of acceleration.

The force that originates from the pressure difference between the upside and downside of the valve is modeled as

$$F_\mathrm{prs} = \pi \left(\frac{d_\mathrm{o}}{2} \right)^2 (P_\mathrm{u} - P_\mathrm{d}) \tag{2.14}$$

where P_u is the upside pressure, P_d is the downside pressure and $\pi(\frac{d_\mathrm{o}}{2})^2$ represents the area affected by the pressure difference. This is a simple model valid when the valve is closed which is when the influence of pressure differences is apparent. The magnetic force then grows big enough to make this force practically neglectable.

The equation of motion is

$$m_\mathrm{p} \ddot{x}_\mathrm{p} = F_\mathrm{M} + F_\mathrm{prs} - F_\mathrm{pld} - k_\mathrm{p} x_\mathrm{p} - b_\mathrm{p} \dot{x}_\mathrm{p} \tag{2.15}$$

where the length of the plunge movement is physically limited, $0 \le x_\mathrm{p} \le x_{\mathrm{p}_\mathrm{max}}$.

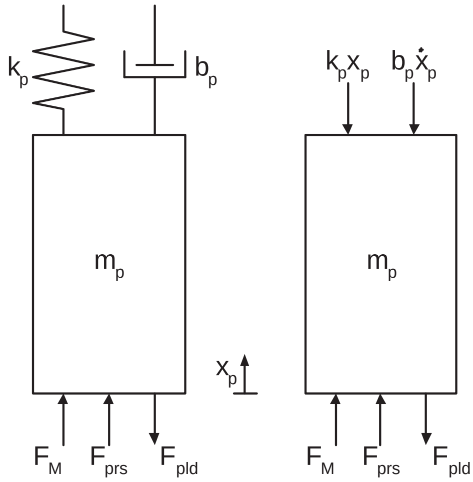


Figure 2.9. Sketch and free body diagram of the plunge.

2.3.4 Flow Model

The flow model determines the air mass flow through the valve, $\dot{m}_{\mathrm{air}}$, depending on the plunge position, x_{p}, and cylinder pressure, P_{cyl}. A block diagram of the flow model is shown in Figure 2.10.

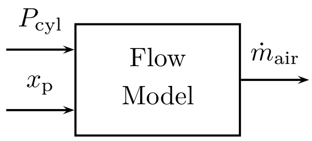


Figure 2.10. Block diagram of the flow model.

The standard equation for mass flow rate of a fluid through an orifice of area A_{v} is presented In [3, 12, 15]. A fluid is in [17] defined as a substance that deforms continuously when acted on by a shearing stress of any magnitude. In this case the fluid is air.

The electro-mechanical part of the valve is used to control the flow of the fluid by moving the plunge and thereby changing the flow area, A_{v}, according to

$$A_{\mathrm{v}} = x_{\mathrm{p}} d_{\mathrm{o}} \pi \tag{2.16}$$

where d_{o} is the diameter of the orifice.

The flow will either be sonic or subsonic depending on the upstream to downstream pressure ratio. The pressure ratio is defined by

$$P_{\mathrm{r}} = \frac{P_{\mathrm{d}}}{P_{\mathrm{u}}}. \tag{2.17}$$

If the pressure ratio is greater than a critical value, the flow is sonic and linearly dependent on the upstream pressure. If the pressure ratio is smaller than the critical value, the flow is subsonic and depends both on the upstream and downstream pressures nonlinearly. The critical ratio of pressure is defined in [3] by

$$P_{\mathrm{cr}} = \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}} \tag{2.18}$$

where γ is the ratio of specific heat. P_{cr} is a fluid constant and is for air determined to $P_{\mathrm{cr}} = 0.528$.

The flow through the valve is described by

$$\dot{m}_{\mathrm{air}} = C_{\mathrm{d}} A_{\mathrm{v}} \frac{P_{\mathrm{u}}}{\sqrt{R T_{\mathrm{air}}}} \Psi\left(P_{\mathrm{r}}\right) \tag{2.19}$$

where

$$\Psi\left(P_{\mathrm{r}}\right) = \begin{cases} \sqrt{\gamma\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{\gamma-1}}}, & P_{\mathrm{r}} \leq P_{\mathrm{cr}} \\ \sqrt{\frac{2\gamma}{(\gamma-1)}\left(\left(P_{\mathrm{r}}\right)^{\frac{2}{\gamma}} - \left(P_{\mathrm{r}}\right)^{\frac{\gamma+1}{\gamma}}\right)}, & P_{\mathrm{r}} > P_{\mathrm{cr}} \end{cases} \tag{2.20}$$

and $\dot{m}_{\mathrm{air}}$ is the air mass flow, C_{d} is the discharge coefficient through the valve, T_{air} is the absolute temperature of the fluid (air) and R is the ideal gas constant. The discharge coefficient, C_{d}, depends on the geometry of the valve and describes that the flow through a restriction contracts on the downstream side resulting in the effective flow area $A_{\mathrm{eff}} = C_{\mathrm{d}} A_{\mathrm{v}}$.

Note that the model of the flow describes the intuitive property that

$$P_{\mathrm{r}} = 1 \Rightarrow \dot{m}_{\mathrm{air}} = 0. \tag{2.21}$$

This is however a problem in simulations since

$$\frac{d}{dP_{\mathrm{r}}}\left(\Psi(P_{\mathrm{r}} \to 1)\right) \to \infty. \tag{2.22}$$

This problem is solved by approximating Ψ with a linear function when P_{r} is close to 1. The function $\Psi(P_{\mathrm{r}})$ is shown in Figure 2.11.

2.4 Cylinder Model

The cylinder model determines the cylinder pressure, P_{cyl}, and the position of the piston inside the cylinder, x_{cyl}, depending on air mass flow into the cylinder, $\dot{m}_{\mathrm{air}}$. A block diagram of the cylinder model is shown in Figure 2.12.

The pneumatic cylinder is modeled in [13], a master's thesis performed earlier to this at Scania CV AB. The cylinder model derived in that thesis is presented here to give a complete picture of the system. A schematic figure of the cylinder is shown in Figure 2.13.

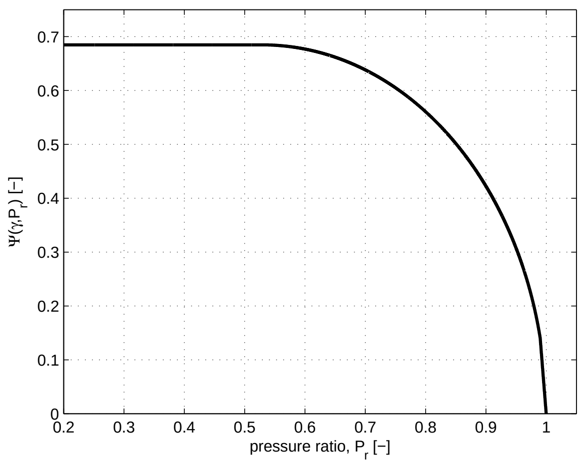


Figure 2.11. The function Ψ, depending on upstream to downstream pressure ratio. When P_{r} is greater than 0.99 Ψ is approximated with a linear function.

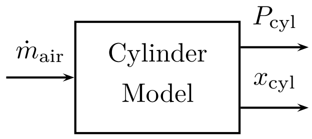


Figure 2.12. Block diagram of the cylinder model.

The cylinder has a piston inside of it which is connected to a spring. The pressure on one side of the piston head results in a force acting on the piston. This pressure is denoted P_{cyl} and is regulated by the On/Off valves. On the other side of the piston head there is atmospheric pressure, denoted P_{atm}. The spring is connected on this side, counteracting the force resulting from P_{cyl}. The spring has a preload. Between the piston and the cylinder walls there is friction. The equation of motion for the cylinder piston is

$$m_{\mathrm{piston}}\ddot{x}_{\mathrm{cyl}} = (P_{\mathrm{cyl}} - P_{\mathrm{atm}})A_{\mathrm{cyl}} - x_{\mathrm{cyl}}k_{\mathrm{cyl}} - F_{\mathrm{friction}} - F_{\mathrm{cylpld}} \tag{2.23}$$

where m_{piston} is the mass of the piston, x_{cyl} is the piston's position (see Figure 2.13), k_{cyl} is the spring constant and F_{cylpld} is the preload in the spring. The length of the movement of the piston is physically limited, $0 \leq x_{\mathrm{cyl}} \leq x_{\mathrm{cyl}_{\mathrm{max}}}$.

The friction, F_{friction}, between the cylinder walls and the piston is of both static and viscous kind. When a force is applied on the piston the static friction has to be overcome in order to move it. Once the piston is in motion, the friction is viscous. The friction properties of the cylinder brings hysteresis. Hysteresis is the property that a systems state depend on its immediate history. With the following

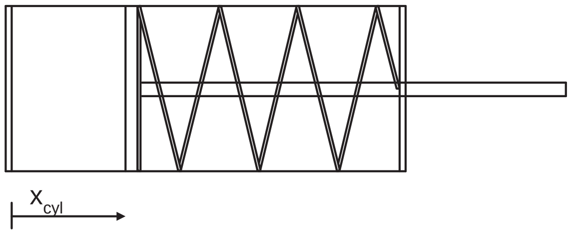


Figure 2.13. Schematic figure of the pneumatic cylinder.

notation

$$
\begin{array}{lcl}
F(v) & = & \text{viscous friction} \\
F_{\mathrm{e}} & = & \text{external forces} \\
F_{\mathrm{s}} & = & \text{static friction}
\end{array}
$$

the friction in (2.23) is determined according to

$$
F_{\mathrm{friction}} = \begin{cases} F(v) & \text{if } v \geq \Delta v \\ F_{\mathrm{e}} & \text{if } |v| < \Delta v \text{ and } |F_{\mathrm{e}}| < F_{\mathrm{s}} \\ F_{\mathrm{s}} sgn(F_{\mathrm{e}}) & \text{if } |v| < \Delta v \text{ and } |F_{\mathrm{e}}| \geq F_{\mathrm{s}} \end{cases} \tag{2.24}
$$

where Δv is used instead of 0 because of simulation difficulties. In order to solve differential equations with $\Delta v = 0$ it would be necessary to detect when $v = 0$ which would require very short step lengths. To solve this Karnopp [5] developed a simulation model where the area with static friction was expanded to an interval where $|v| < \Delta v$.

The cylinder pressure P_{cyl} is retained through the ideal gas law. In order to model the course of events from air mass flow to cylinder pressure the derivative of the ideal gas law is taken with respect to time

$$
\dot{P} = \frac{\partial}{\partial t}\left(\frac{mRT}{V}\right) = RT\frac{\dot{m}V - \dot{V}m}{V^2} \tag{2.25}
$$

where P is the pressure, m is the mass, R is the ideal gas constant, T is the temperature and V is the volume.

Equation (2.25) is valid when the air temperature is constant (isotherm process), which is a simplification. The mass is expressed through the ideal gas law yielding

$$
\dot{P} = RT\frac{\dot{m}V - \dot{V}\dfrac{PV}{RT}}{V^2} = \frac{RT\dot{m} - \dot{V}P}{V} \tag{2.26}
$$

where V is the volume of the complete system consisting of the cylinder volume, $V_{\mathrm{cyl}} = A_{\mathrm{cyl}} x_{\mathrm{cyl}}$, and the constant volume of connecting tubes etc., V_{d}, yielding

$$
\dot{P}_{\mathrm{cyl}} = \frac{RT_{\mathrm{air}}\dot{m}_{\mathrm{air}} - \dot{x}_{\mathrm{cyl}} A_{\mathrm{cyl}} P_{\mathrm{cyl}}}{V_{\mathrm{d}} + x_{\mathrm{cyl}} A_{\mathrm{cyl}}}. \tag{2.27}
$$

2.5 Complete System Model

The complete system model, that consists of all above retained equations, is gathered into the following.

The supply valve equations

$$u_{\text{pwm}_{\text{sup}}} = \begin{cases} 28, & t \leq u_{\text{sup}}T \\ 0, & t > u_{\text{sup}}T \end{cases} \tag{2.28a}$$

$$u_{\text{pwm}_{\text{sup}}} = R_{\text{c}} i_{\text{sup}} + (L_0 + \alpha_{\text{L}} x_{\text{sup}}) \frac{di_{\text{sup}}}{dt} \tag{2.28b}$$

$$F_{\text{M}_{\text{sup}}} = \frac{\mu_0 A_{\text{p}} N^2 i_{\text{sup}}^2}{2(x_0 - x_{\text{sup}})^2} \tag{2.28c}$$

$$F_{\text{prs}_{\text{sup}}} = \pi \left(\frac{d_{\text{o}}}{2}\right)^2 (P_{\text{sup}} - P_{\text{cyl}}) \tag{2.28d}$$

$$m_{\text{p}} \ddot{x}_{\text{sup}} = F_{\text{M}_{\text{sup}}} + F_{\text{prs}_{\text{sup}}} - F_{\text{pld}} - k_{\text{p}} x_{\text{sup}} - b_{\text{p}} \dot{x}_{\text{sup}} \tag{2.28e}$$

$$\dot{m}_{\text{air}_{\text{sup}}} = C_{\text{d}_{\text{sup}}} x_{\text{sup}} d_{\text{o}} \pi \frac{P_{\text{sup}}}{\sqrt{RT_{\text{air}}}} \Psi \left(\frac{P_{\text{cyl}}}{P_{\text{sup}}}\right) \tag{2.28f}$$

The exhaust valve equations

$$u_{\text{pwm}_{\text{exh}}} = \begin{cases} 28, & t \leq u_{\text{exh}}T \\ 0, & t > u_{\text{exh}}T \end{cases} \tag{2.29a}$$

$$u_{\text{pwm}_{\text{exh}}} = R_{\text{c}} i_{\text{exh}} + (L_0 + \alpha_{\text{L}} x_{\text{exh}}) \frac{di_{\text{exh}}}{dt} \tag{2.29b}$$

$$F_{\text{M}_{\text{exh}}} = \frac{\mu_0 A_{\text{p}} N^2 i_{\text{exh}}^2}{2(x_0 - x_{\text{exh}})^2} \tag{2.29c}$$

$$F_{\text{prs}_{\text{exh}}} = \pi \left(\frac{d_{\text{o}}}{2}\right)^2 (P_{\text{cyl}} - P_{\text{atm}}) \tag{2.29d}$$

$$m_{\text{p}} \ddot{x}_{\text{exh}} = F_{\text{M}_{\text{exh}}} + F_{\text{prs}_{\text{exh}}} - F_{\text{pld}} - k_{\text{p}} x_{\text{exh}} - b_{\text{p}} \dot{x}_{\text{exh}} \tag{2.29e}$$

$$\dot{m}_{\text{air}_{\text{exh}}} = C_{\text{d}_{\text{exh}}} x_{\text{exh}} d_{\text{o}} \pi \frac{P_{\text{cyl}}}{\sqrt{RT_{\text{air}}}} \Psi \left(\frac{P_{\text{atm}}}{P_{\text{cyl}}}\right) \tag{2.29f}$$

The cylinder equations

$$m_{\text{piston}} \ddot{x}_{\text{cyl}} = (P_{\text{cyl}} - P_{\text{atm}}) A_{\text{cyl}} - x_{\text{cyl}} k_{\text{cyl}} - F_{\text{friction}} - F_{\text{cylpld}} \tag{2.30a}$$

$$\dot{P}_{\text{cyl}} = \frac{RT_{\text{air}} \dot{m}_{\text{air}} - \dot{x}_{\text{cyl}} A_{\text{cyl}} P_{\text{cyl}}}{V_{\text{d}} + x_{\text{cyl}} A_{\text{cyl}}} \tag{2.30b}$$

$$\dot{m}_{\text{air}} = \dot{m}_{\text{air}_{\text{sup}}} - \dot{m}_{\text{air}_{\text{exh}}} \tag{2.30c}$$

2.5.1 State Space Form

The following states, control signals and measured output are introduced

$$
\begin{array}{llll}
x_1 = i_{\mathrm{sup}} & x_4 = i_{\mathrm{exh}} & x_7 = x_{\mathrm{cyl}} & u_1 = u_{\mathrm{sup}} \\
x_2 = x_{\mathrm{sup}} & x_5 = x_{\mathrm{exh}} & x_8 = \dot{x}_{\mathrm{cyl}} & u_2 = u_{\mathrm{exh}} \\
x_3 = \dot{x}_{\mathrm{sup}} & x_6 = \dot{x}_{\mathrm{exh}} & x_9 = P_{\mathrm{cyl}} & y = x_{\mathrm{cyl}} = x_7
\end{array}
$$

$$
z_1 = \begin{cases} 28, & t \leq u_1 T \\ 0, & t > u_1 T \end{cases} \qquad z_2 = \begin{cases} 28, & t \leq u_2 T \\ 0, & t > u_2 T \end{cases}
$$

The system equations on state space form are

$$\dot{x}_1 = \frac{z_1 - R_{\mathrm{c}} x_1}{L_0 + \alpha_{\mathrm{L}} x_2} \tag{2.31a}$$

$$\dot{x}_2 = x_3 \tag{2.31b}$$

$$\dot{x}_3 = \frac{\dfrac{\mu_0 A_{\mathrm{p}} N^2 x_1^2}{2(x_0 - x_2)^2} + \pi \left(\dfrac{d_{\mathrm{o}}}{2}\right)^2 (P_{\mathrm{sup}} - x_9) - F_{\mathrm{pld}} - k_{\mathrm{p}} x_2 - b_{\mathrm{p}} x_3}{m_{\mathrm{p}}} \tag{2.31c}$$

$$\dot{x}_4 = \frac{z_2 - R_{\mathrm{c}} x_4}{L_0 + \alpha_{\mathrm{L}} x_5} \tag{2.31d}$$

$$\dot{x}_5 = x_6 \tag{2.31e}$$

$$\dot{x}_6 = \frac{\dfrac{\mu_0 A_{\mathrm{p}} N^2 x_4^2}{2(x_0 - x_5)^2} + \pi \left(\dfrac{d_{\mathrm{o}}}{2}\right)^2 (x_9 - P_{\mathrm{atm}}) - F_{\mathrm{pld}} - k_{\mathrm{p}} x_5 - b_{\mathrm{p}} x_6}{m_{\mathrm{p}}} \tag{2.31f}$$

$$\dot{x}_7 = x_8 \tag{2.31g}$$

$$\dot{x}_8 = \frac{(x_9 - P_{\mathrm{atm}}) A_{\mathrm{cyl}} - x_7 k_{\mathrm{cyl}} - F_{\mathrm{friction}} - F_{\mathrm{cylpld}}}{m_{\mathrm{piston}}} \tag{2.31h}$$

$$\dot{x}_9 = \frac{\sqrt{RT}\left(C_{\mathrm{d_{sup}}} x_2 d_{\mathrm{o}} \pi P_{\mathrm{sup}} \Psi\left(\dfrac{x_9}{P_{\mathrm{sup}}}\right) - C_{\mathrm{d_{exh}}} x_5 d_{\mathrm{o}} \pi x_9 \Psi\left(\dfrac{P_{\mathrm{atm}}}{x_9}\right)\right) - x_8 x_9 A_{\mathrm{cyl}}}{V_{\mathrm{d}} + x_7 A_{\mathrm{cyl}}} \tag{2.31i}$$

where
$0 \leq x_2 \leq x_{2_{\mathrm{max}}}$,
$0 \leq x_5 \leq x_{5_{\mathrm{max}}}$
and
$0 \leq x_7 \leq x_{7_{\mathrm{max}}}$.

Chapter 3

Model Calibration

In order to make the model accurate the uncertain model parameters needs to be determined by comparing the model to measured data. In other words, the model needs to be calibrated.

3.1 Method

An approach to perform the calibration is to minimize a cost function $V_{\mathrm{N}}(\theta)$ where θ is a vector containing the parameters of the model. The mathematical software MATLAB has functions for solving this least square problem. The problem is to find the parameter values, θ, that minimizes

$$V_{\mathrm{N}}(\theta) = \frac{1}{2}\sum_{i=1}^{N}(\varepsilon_i(\theta))^2 = \frac{1}{2}\varepsilon^T(\theta)\varepsilon(\theta) \tag{3.1}$$

where

$$\varepsilon_i(\theta) = y(t_i) - \hat{y}(t_i, \theta) \tag{3.2}$$

is the residual, $y(t_i)$ is the measured value and $\hat{y}(t_i)$ is the modeled value at the time $t = t_i$.

MATLAB's function `lsqnonlin` is used together with the model implemented in SIMULINK and measured values collected from a test bench. This nonlinear data-fitting problem is solved using an algorithm of Levenberg-Marquardt type.

Another way to calibrate the model is to manually tune the parameters to fit simulated to measured data. Here both methods are used.

3.2 Parameters

The model parameters, θ, are categorized into two different categories, see Table 3.1. Parameters in terms that consist of variables and constants affect the static value when the system is actuated by a certain pulse. Parameters in terms that consist of derivatives affect the dynamics, i.e. rise/fall time and the time constant of the system.

static case	dynamic case
A_p	L_0
k_p	α_L
F_pld	b_p
N	m_p
x_0	m_piston
d_o	
$C_{\mathrm{d}_\mathrm{sup}}$	
$C_{\mathrm{d}_\mathrm{exh}}$	
A_cyl	
k_cyl	
F_cylpld	

Table 3.1. Parameters in the model.

Since the least square problem grows fast with the number of variables, some parameters will be considered known to make the optimization problem feasible. This however is not a very big sacrifice since some parameters can be measured quite easily and some are provided by the valve manufacturer. These parameters are F_pld, N, A_cyl, k_cyl and F_cylpld in the static case and L_0, m_p and m_piston in the dynamic. This leaves the parameters in Table 3.2 to be determined.

static case	dynamic case
A_p	α_L
k_p	b_p
x_0	m_piston
$C_{\mathrm{d}_\mathrm{sup}}$	
$C_{\mathrm{d}_\mathrm{exh}}$	

Table 3.2. Parameters to determine.

3.3 Data

The measured data from the test bench are the signals that control the system, i.e. $u_{\mathrm{pwm}_\mathrm{sup}}$ and $u_{\mathrm{pwm}_\mathrm{exh}}$, and the system output, i.e. the position of the piston inside the cylinder, x_cyl.

Since the system behaves differently depending on if the pressure in the cylinder is high or low, the whole pressure spectrum must be included in the data. Also different magnitudes of the control signals are included. Both the supply valve and the exhaust valve are used. The valves are running at 50 Hz.

When determining the parameters affecting the static levels, the data needs to have an output that changes between two different static levels. Here it is important that all states in the model are correctly initiated. This will give the static levels a large weight in the least square problem. Four examples of steps between two different static levels used to calibrate the model are shown in Figure 3.1. The cost function (3.1) is minimized.

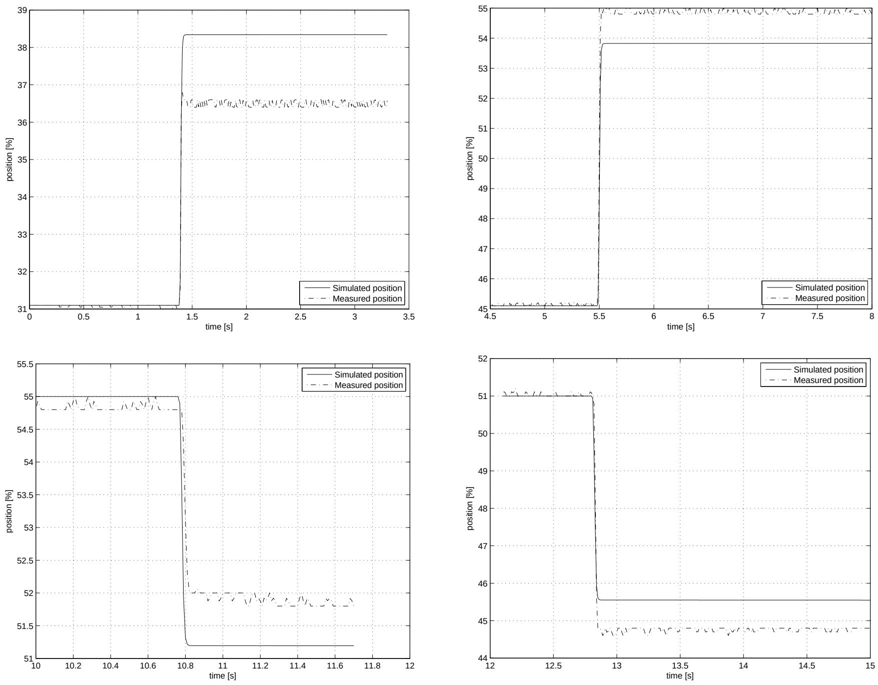
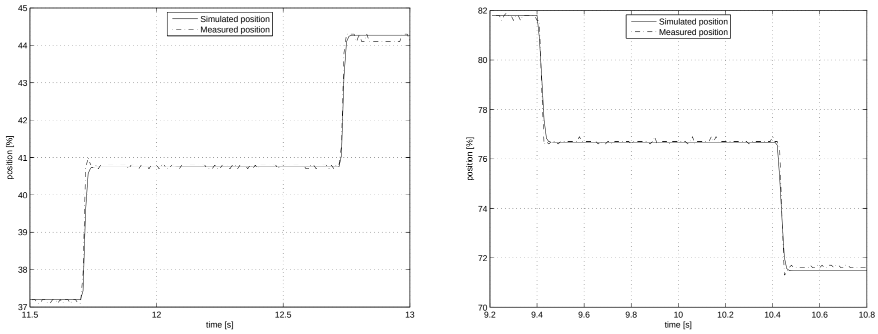


Figure 3.1. Examples of steps between two different levels used to calibrate the static parameters of the model.

The parameters concerning the dynamics of the system are manually tuned by visually comparing simulated and measured data. The reason for this is that the measured data has noise which makes it hard to use an optimization routine and expect a reasonable result. Instead, the dynamic parameters are tuned so that the time constant in a step response is the same in simulations as in experiments, see Figure 3.2(a) and 3.2(b) which show step responses from the supply and the exhaust valve respectively.

3.4 Results

To show the result of the calibration, a train of single PWM pulses with different magnitudes are given as input to the valves. As can be seen in Figure 3.3 and 3.4,



(a) Steps concerning the supply valve.

(b) Steps concerning the exhaust valve.

Figure 3.2. Examples of steps used to calibrate the dynamic parameters of the model.

the magnitudes of the different steps in position are not calculated correctly using the model. On the other hand, the general characteristics for the positions from 20% to 90% are present in the model, and the simulated final values are close to the measured.

The calibration of the dynamic parameters results in a time constant for the simulated step responses that on average fits the measured.

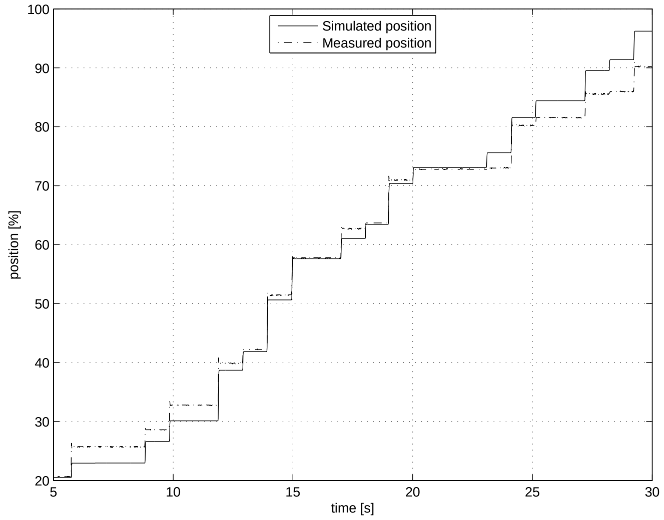


Figure 3.3. Simulated and measured position for the static calibration data for the supply valve.

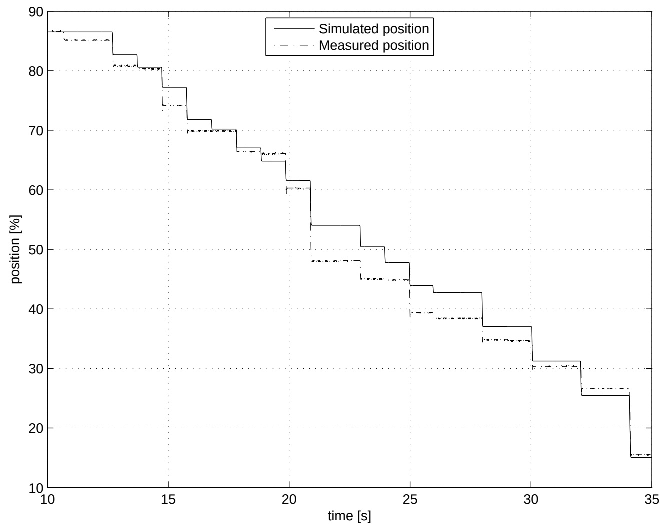


Figure 3.4. Simulated and measured position for the static calibration data for the exhaust valve.

Chapter 4

Model Validation

As mentioned in chapter 2, a model is usable first when the validity of it has been determined. This is done by model validation and will be described in this chapter.

4.1 Method

Two ways to measure the accuracy of a model are to calculate the mean relative error and the maximum relative error given by

$$\text{mean relative error} = \frac{1}{N}\sum_{i=1}^{N}\frac{|y(t_i) - \hat{y}(t_i)|}{|\max y(t_i)|} \tag{4.1}$$

$$\text{maximum relative error} = \max_{1\leq i\leq N}\frac{|y(t_i) - \hat{y}(t_i)|}{|\max y(t_i)|} \tag{4.2}$$

where $y(t_i)$ is the measured value, $\hat{y}(t_i)$ is the simulated value at the time $t = t_i$ and N is the number of samples.

Also, a visual study of the measured data compared to the simulated gives a decent perception on the performance of the model, see Figure 4.1.

4.2 Data

In order to validate the calibrated model a new set of data is needed. The validation data should excite the system as much as possible to really challenge the model. By generating a control signal sequence with random numbers between 0 and 100 with the function `rand` in MATLAB, the whole spectrum of working points and control signals is covered. Both static levels and sequences that excite the dynamics of the system are included in the measured data. The measured system output generated with these control signals is shown in Figure 4.1.

4.3 Results

The modeled piston position is compared with the measured, see Figure 4.1. It is seen that the simulated values are close to the measured, describing the dynamics of the system but not always the static levels of the piston position. Because of these static errors a feedback control loop is necessary when constructing a controller for the system with the help of this model. In this case, a feedback loop is available and therefor the model is accurate enough to use for development of controllers. The mean and maximum relative errors in (4.1) and (4.2) are calculated and presented in Table 4.1.

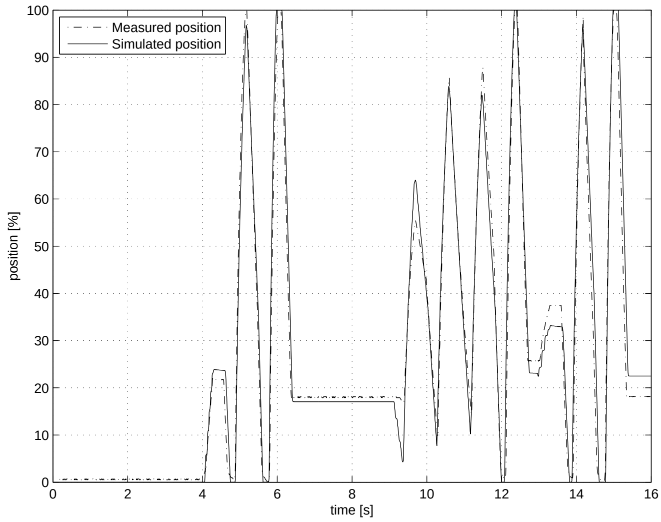


Figure 4.1. Simulated and measured position.

Mean relative error	3.9%
Maximum relative error	26.3%

Table 4.1. Mean and maximum relative error.

4.4 Sources of Errors

The same parameters are used for both the exhaust and the supply valve, however individual spread exists.

The air and coil temperature, supply pressure and voltage are considered constant but may vary during the measurements. This is not considered by the model.

Some parameters are considered to be known but may in fact not be completely accurate, see section 3.

The magnetic model could be made more complex by modeling it as a network of magnetic resistances.

The electrical model contains approximations. The inductance is approximated as a linear function and the derivate of the inductance is neglected. As a result of this some accuracy is likely to be lost.

The friction in the cylinder has great influence on the system output. Since no measurements of the friction has been conducted in this thesis it might be inaccurate.

Chapter 5

System Properties

Several properties of the system has been discovered and investigated when the model has been constructed and through experiments. These are important to have in mind when control methods are developed. Some of the characteristics discussed are present in the model, while some are not. This chapter will discuss the system characteristics from a control point of view.

5.1 Charge and Discharge of the Coil

The control signal for the system is, as mentioned in section 2.2, PWM voltage of which the control system set the duty cycle. Since the coil in the valve is an inductive element; the current is delayed both on rising and falling edge and does not follow the input voltage signal completely. Plots from simulations of both the input voltage and the resulting current through the coil are shown in Figure 5.1.

On the rising edge, the current needs to be big enough to induce a force which overcomes the force of the spring and pressure in the valve. Analogously, on the falling edge the current needs to induce a smaller force than the force pulling it back. It is also important to note that it takes a smaller current to attract an object that is closer. The result of this is that the magnitude of the current needed to induce a force big enough to move the plunge from its initial position ($x_{\mathrm{p}} = 0$) is larger than the magnitude of the current needed to hold the plunge in a position $x_{\mathrm{p}} > 0$, see (2.13) and Figure 5.2.

Because of the inductive proporties of the coil, it is not wise to chose the PWM period any smaller than 20 ms. Otherwise the current never has time to change enough to affect the plunge. For the same reason it is not wise to chose the duty cycle less than $\sim 20\%$ or more than $\sim 80\%$ when running at 50 Hz because the valve would never open/close between two PWM periods. If the PWM period was shorter than 20 ms the control range would shrink even more. The PWM period also decides the frequency of the control system. A control signal is active for one PWM period.

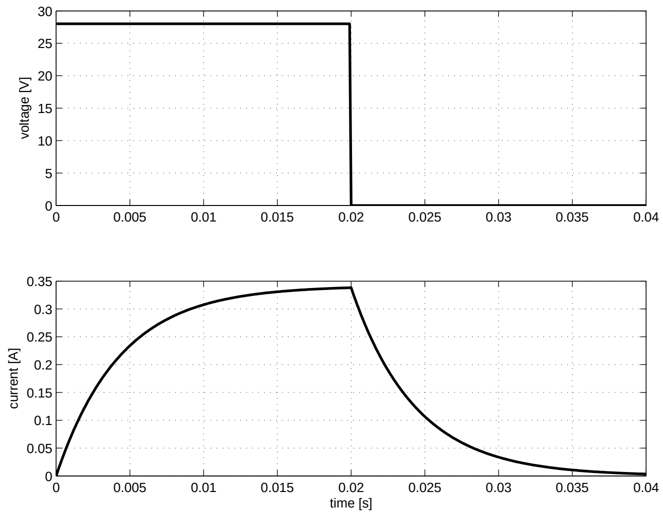


Figure 5.1. Simulation of the input voltage and the resulting current through the coil.

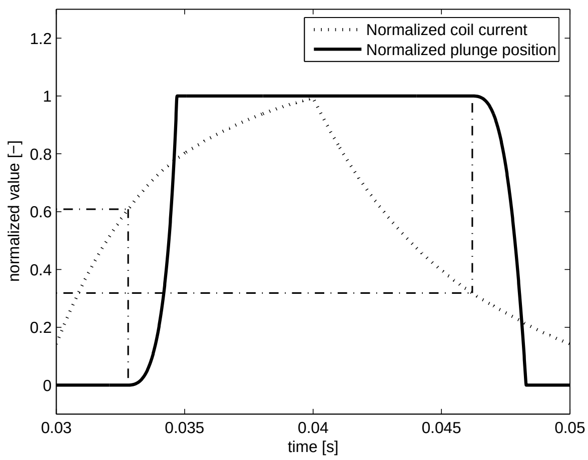


Figure 5.2. Simulation of the current through the coil and the plunge movement, showing at which current levels the plunge starts to move.

5.2 Temperature Dependence

The resistance in a material is dependent on temperature, which also means that the dynamic characteristics of the electromagnet depend on temperature. Since these valves will be operating close to a combustion engine, and the energy loss in an electric resistance becomes heat, the influence of temperature differences can not be neglected. The equation describing the resistance's dependence of temperature is [9]

$$R_c = R_0(1 + \alpha(T_c - T_0)) \tag{5.1}$$

where R_c is the coil resistance, R_0 is the known resistance, α is the temperature constant which is a material constant, T_c is the coil temperature and T_0 is the temperature when the known resistance was measured. For example, a resistance $R = 100\ \Omega$ raises to $R = 143.3\ \Omega$ if the temperature raises from $T_{c1} = 300$ K to $T_{c2} = 400$ K, assuming the material is copper. This is a very important property, since the current in the electric circuit depends on the resistance and the magnetic force depends on the current ($F_M \propto i^2$, see section 2.3.2).

To show the effect of temperature differences, the current is simulated with different resistances measured at different temperatures. The results of those simulations are shown in Figure 5.3. It can be seen that a higher temperature and thereby also a higher resistance, results in a lower current. This results in a smaller magnetic force affecting the plunge and that the plunge starts to move later at a higher temperature, resulting in a smaller air mass flow. Looking at this from the input signal point of view, a signal of $\sim$ 30% PWM at a certain temperaure has the same effect as a signal of $\sim$ 40% PWM at another (higher) temperature.

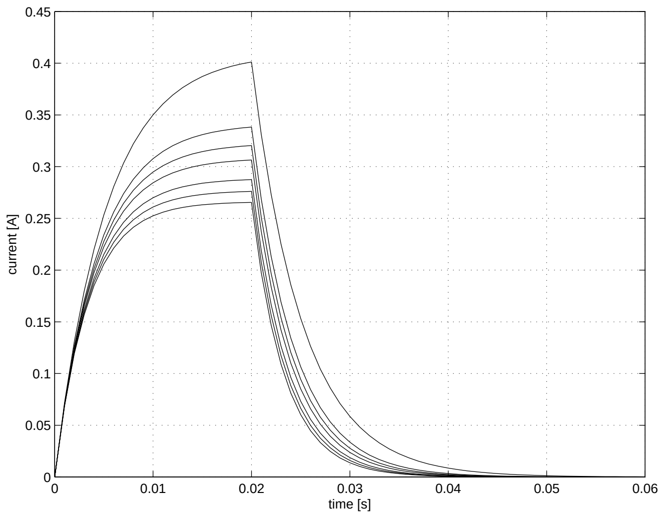


Figure 5.3. Simulated current in the electric circuit at different temperatures. From top to bottom: 0°C, 20°C, 40°C, 60°C, 80°C, 100°C and 120°C.

To handle this the controller needs to be either very robust or adaptive. This means the parameters of the controller either are chosen in such a way that the controller works even with varieties in temperature or that the difference in resistance is taken in consideration when calculating the control signal. To do this the resistance has to be known. One way to calculate the resistance is to, when the position of the piston is at its minimum or maximum, apply a voltage over the coil in the valve that is not active (supply valve at maximum and exhaust valve at minimum) and log the current using the engine control unit (ECU). This provides the information necessary to calculate the coil resistance using Ohm's law.

Once the resistance is known it can be used to calculate a feedforward signal, see Figure 5.4, where u_R is the feedforward control signal, calculated from the coil resistance (which is dependent on temperature), and u_e is the control signal calculated from the control error.

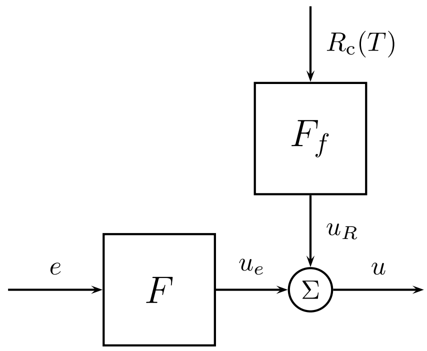


Figure 5.4. The basic structure of feedforward control from temperature.

5.3 Thermodynamic Effects

The purpose of the actuator system is to place a butterfly valve in a certain position. Once in position no regulation should be required. However thermodynamic effects and differences in exhaust gas flow may make regulation necessary. When the air passes through the supply valve the temperature of the air increases and thereby also the volume (ideal gas law). Inside the cylinder the air might cool down depending on outside temperature. The volume then decreases and the pressure drops, causing a movement of the piston. Analogously, when the air passes through the exhaust valve, the temperature of the air inside the cylinder decreases, when it later heats up by its surrounding the pressure raises. Difference in exhaust gas pressure cause differences in the force acting on the butterfly valve which might change the position of it.

Figure 5.5 shows the existence of thermodynamic effects. When a control signal is applied to the exhaust valve, the position of the piston in the cylinder and the pressure changes. This can be seen in Figure 5.5 as the steep changes. Between these steep changes, no control signal is applied and the valve is closed. This means no air is removed/added to the cylinder. However, the pressure still raises. The reason for this is thermodynamic effects. The air heats up after the pressure drops, increasing the pressure. The piston position however is unchanged. The change in pressure does not create a force big enough to overcome the static friction in this case.

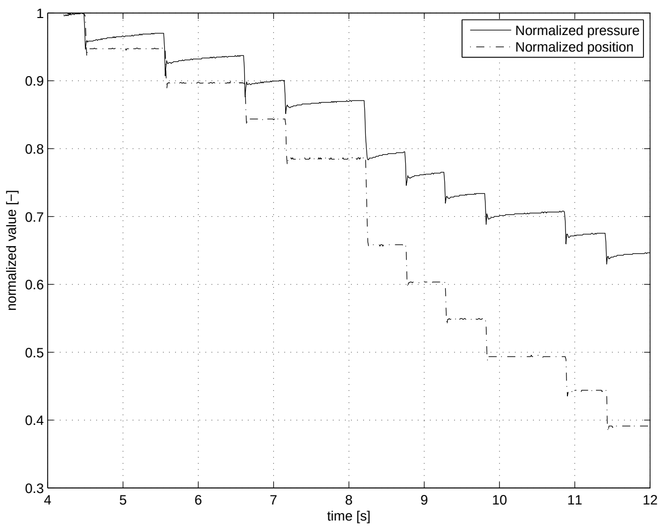


Figure 5.5. The pressure changes while the piston is at a constant position without any applied control signal. This shows the presence of thermodynamic effects.

5.4 Integrating System

The system is integrating, meaning a certain constant control signal does not have a corresponding steady state of the system. Instead, for any constant control signal, the output will just keep rising/falling. A typical example of a system like this is a water tank where the water level is the output and the flow to/from it is the control signal. In this particular system the cylinder works the same way as the water tank where the pressure in the cylinder raises/falls when a control signal is applied. This property also means the system lacks steady states. The only steady states are when both valves are closed or when the piston in the cylinder is at an end position. However static friction can make the position of the piston unchanged even though the pressure has been changed. This could perhaps in some way also be seen as a steady state.

To show this a constant control signal is first applied to the supply valve and then to the exhaust valve. The position of the piston in the cylinder first raises and then decreases, see Figure 5.6. The delay in position change between the time 0.4 to 0.5 s when the exhaust valve is run depends on that the pressure in the cylinder has to be lowered to a certain level before the piston's position is affected. When the supply valve is switched off, the pressure in the cylinder is the same as the supply pressure. As can be seen in Figure 5.6, it takes about five openings to lower the cylinder pressure enough to move the piston in the cylinder.

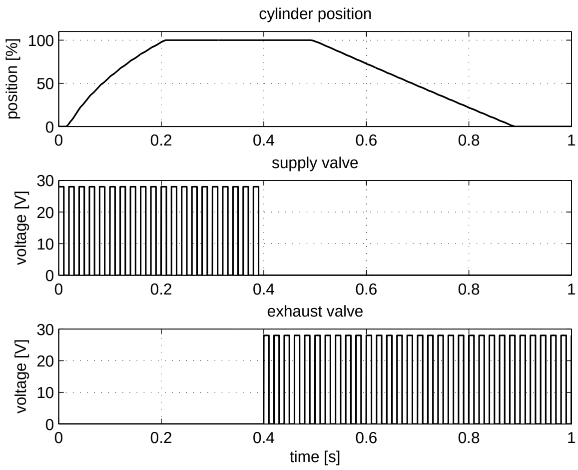


Figure 5.6. Measured piston position when given a constant input signal of 50% first to the supply valve, then to the exhaust valve, showing that the system is integrating.

5.5 Nonlinear Position Change

What happens when a valve is opened is that a portion of air flows through it causing a change in cylinder pressure and thereby a change in piston position. In other words, the magnitude of the change depend on the air mass flow. As described in section 2.3.4, the mass flow depend on the pressure ratio. Therefor a certain control signal has different effects depending on the current pressure ratio. Another factor that enhances this effect is that the force on the plunge, F_{prs}, in (2.15) is larger when the pressure difference is high. The consequence of this is that the magnetic force needed to open the plunge and thereby also the time a valve is open during one PWM period is different depending on the pressure difference.

This can be seen in Figure 5.7 where a constant control signal is applied first to the supply valve, then to the exhaust valve. Between two control signal outputs there is a pause to make the influence of each control signal visible in the piston position. By looking at the length of each position change, it can be seen that they become shorter and shorter when the position increases.

However, this does not show when decreasing the position. By looking at each step in position change it can be seen that they all have roughly the same length. The reason for this is that the cylinder pressure is about twice the size of atmospheric pressure when the piston is fully subtracted ($x_{\text{cyl}} = 0$), making the pressure ratio small enough to make the flow sonic at all times when decreasing the position of the piston in the cylinder, see (2.20).

One way of counteracting this is to add an element in the control signal that depends on the pressure difference and compensates for this. Just as in the case with the influence from temperature.

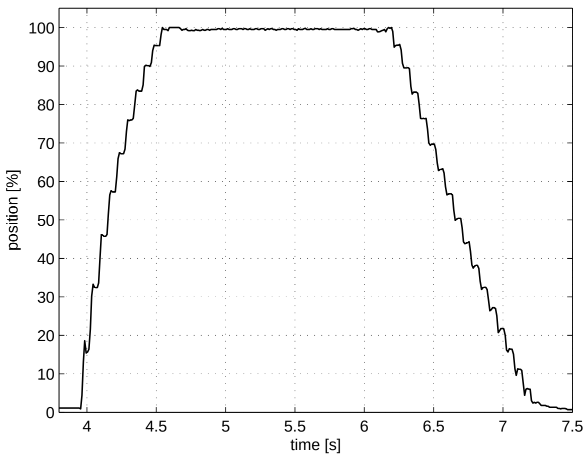


Figure 5.7. Different measured lengths in movement when increasing the piston position, dependent on start position. Constant length when decreasing it.

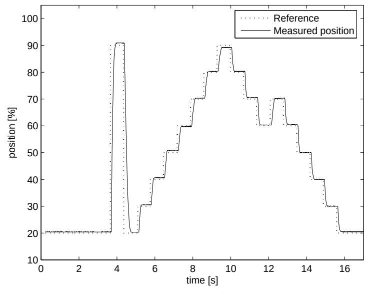
5.6 Individual Spread

Individual spread between different valves exist. The consequence of this is that a controller tuned to fit a certain set of valves does not work as good on other valves. This problem also affects the accuracy of the model, decreasing the accuracy of a controller developed with the help of the model.

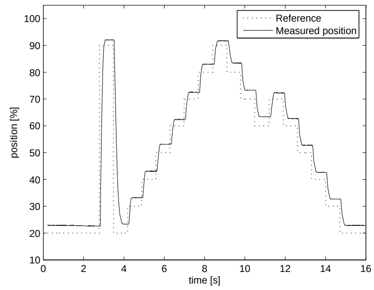
In Figure 5.8(a) the performance of a controller tuned on the same valves as are used is shown. In Figure 5.8(b) the exact same controller is used to follow the same reference on an other set of valves. As can be seen, the performance is worse.

5.7 Free Wheel Diode on Output Circuit

The output circuits on the ECU are protected by free wheel diodes. A free wheel diode is introduced to slow down the discharge of the coil in the valve, otherwise all energy stored in the coil is absorbed by the ECU when the circuit is opened. This energy absorption results in heating of the ECU and its components which could damage the ECU and is of course not acceptable. When using a free wheel diode, it is connected in a circuit parallel to the coil in the electric circuit, modeled in section 2.3.1. The parallel diode is connected according to Figure 5.9.



(a) The valves which the controller is tuned on.



(b) Different valves from which the controller is tuned on.

Figure 5.8. A controller tuned to fit a certain set of valves tested on those valves and another set of valves, showing the existence of individual spread between the valves.

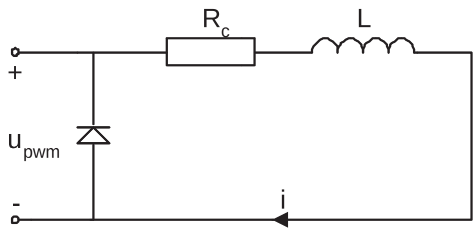


Figure 5.9. Schematic figure of the electrical circuit with a free wheel diode connected to the ECU's output circuit.

The voltage drop over the diode is always 0.7 V. This is why the coil is discharged slower with the free wheel diode. It results in a high current for a longer time, making the magnetic force larger, keeping the valve open for a longer time when the control signal is low.

Since this thesis work aims to evaluate the abilities of the pneumatic system, the free wheel diode has been neglected since its effect on the system is a property that has to do with the ECU. While the experiments has been conducted, an output circuit that removes the effect of the free wheel diode has been used.

Although, when using a free wheel diode its effects has to be considered when, for instance, a control strategy is developed. This should be done when the system is controlled in a test bench and in an engine test cell.

Chapter 6

System Control

The *control problem*, as described in [8], is to determine, with guidance of the system outputs, an input so that the system purpose is fulfilled. The purpose of a system can be of various kinds but two common ones are either to maintain a steady output despite the influence of disturbances, or to follow a reference.

Many different methods are possible to use when controlling a system, some more appropriate than others depending on the system. In [4, 6] the theory and implementation of different controllers are described. The methods discussed here are PID control and fuzzy control.

6.1 Control Objectives and Limitations

This master's thesis aim is to study what ability in performance this system configuration has. In terms of settling time and acceptable control error, the aim is to make the system as fast and as accurate as possible.

Since the lifespan of the valves is limited and dependent of the number of closings and openings, it is important to reach the desired output from a minimum number of openings. From this perspective it is good to use large control signals which also brings the advantage of a fast system. Although this of course is the real challenge anyhow.

The diameter of the valves sets a limit in settling time since it limits the air mass flow rate. The sampling time also sets a limit.

6.2 The Control Signal

As described in section 2.2 the input signal to each valve is PWM voltage with an amplitude of 28 V. Since an applied signal to the supply valve extracts the piston inside the cylinder (increasing x_{cyl}) while an applied signal to the exhaust valve subtracts it (decreasing x_{cyl}), it is necessary to define positive and negative control signal, u. If the control signal is positive, it aims to extract the piston and therefor the supply valve should be used while a negative aims to subtract it and therefor the exhaust valve should be used. Formally this is given by

$$\begin{cases} u_{\text{sup}} = u \\ u_{\text{exh}} = 0 \end{cases}, u \geq 0 \tag{6.1}$$

$$\begin{cases} u_{\text{sup}} = 0 \\ u_{\text{exh}} = |u| \end{cases}, u < 0. \tag{6.2}$$

6.3 PID Control

PID control is a common method which is simple to implement and does not require much calculations. A basic feedback control loop is shown in Figure 6.1, where the PID controller is in the block F. In this section, a PID controller is implemented to control the On/Off valves.

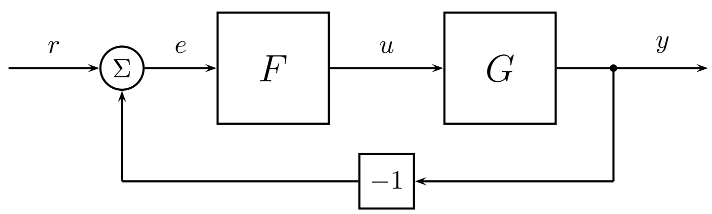


Figure 6.1. The structure of a basic feedback control loop.

6.3.1 Principle of PID Control

PID is short for Proportional Integral Derivative. These three parts are taken with respect to the control error and they all contribute to the control signal. A PID controller can e.g. be

$$u(t) = K \left(e(t) + \frac{1}{T_{\text{i}}} \int_{t_0}^{t} e(\tau)\, d\tau + T_{\text{d}} \frac{d}{dt} e(t) \right), \tag{6.3}$$

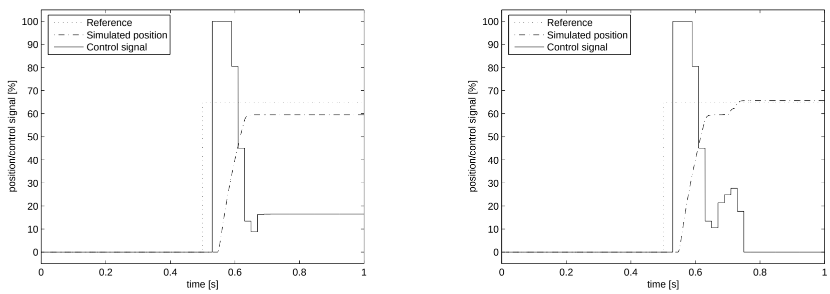
where K is the gain, $e(t) = r(t) - y(t)$ is the control error, $r(t)$ is the reference, $y(t)$ is the measured output, T_{i} and T_{d} are constants with time as unit. These parameters determine the weight of the different three contributions to the control signal, $u(t)$. The proportional part determines the speed of the system. A large proportional part can make the system unstable. The integral part is used to minimize the steady state error. The derivative part is used to increase the stability of the system.

In this particular system the proportional part will contribute most to the control signal in most cases. This part will roughly erase the control error.

The derivative part is used because the system is sluggish i.e. it has a dead time. The derivative part can then reduce overshoots and increase stability.

The integral part is usually used when a control signal is necessary to keep a steady output. This is not the case on this system since it is integrating, see section 5.4. However an integrating element could still be used to remove steady state errors because of the property of the inductive element in the electric circuit. This results in, as discussed in section 5.1, that the control signal needs to reach a certain value to affect the plunge inside the valve. This could be accomplished with the help of an integrating element.

In Figure 6.2(a) no I-part is used. A steady state error is present and the control signal is proportional to the control error. But it is not big enough to affect the system since the current through the coil never has time to grow strong enough to move the plunge in the valve. In Figure 6.2(b) an I-part is introduced. The control signal gradually grows until the valve opens, removing the steady state error.



(a) Without I-part. (b) With I-part.

Figure 6.2. The difference between a controller with an I-part compared to one without in simulation.

6.3.2 Discrete Implementation of the PID Controller

Since the controller will be digitally realized on an ECU it needs to be discrete. A discrete PID is given by

$$u_{\mathrm{n}} = K\left(e_{\mathrm{n}} + \frac{T_{\mathrm{S}}}{T_{\mathrm{i}}}\sum_{k=0}^{n} e_{\mathrm{k}} + T_{\mathrm{d}}\frac{e_{\mathrm{n}} - e_{\mathrm{n\text{-}1}}}{T_{\mathrm{S}}}\right) \tag{6.4}$$

where T_{S} is the sampling time and e_{n} is the sampled control error at sample n.

6.3.3 The I-part

Since the pneumatic system is integrating, see section 5.4, the implementation of the integrating element in the PID controller will be a bit special.

In order to use the integral part only for minimization of the steady state error, it should only be separated from zero when the output is steady and an error is present. To realize this the sum in (6.4) is only updated when the control error is outside the tolerance limits and within an outer limit, less than what the proportional part can handle.

If the described situation is not the case, the integral part is set to zero. If the integral part were not set to zero, the sum would keep adding up and the integral part could make $u_\mathrm{n} \neq 0$ even though the control error is zero. In a non integrating system, this would not be a problem since a certain control signal would equal a certain system state. In this system, when the desired state is reached, the control signal needs to be zero to keep the system in this state.

Since the control signal can be saturated it is also necessary to implement the I-part so that integrator windup [4] is avoided.

6.3.4 Tuning the PID

The PID parameters are tuned through simulations with the help of the knowledge about the system provided by the model and experiments. Different PID parameters are suitable for the supply and exhaust valve. A smaller control signal is needed for the supply valve making the proportional part smaller when the supply valve is used, i.e. the control error is positive.

An example on how the knowledge about the system is used to tune the parameters is if the reference is 60% and the actual position is 50%, making $e = r - y = 10\%$. Experience from measurements and simulations says a suitable control signal would be $u \approx 30\%$. Through (6.4) a suitable value for the gain K is calculated to $K = 3$.

First the gain K is set for both valves to roughly erase the error, see Figure 6.3. However steady state errors are present and some overshoots exist when increasing the reference.

To reduce the overshoots, a derivative part is added. In order to reduce the steady state errors, an integral part is also added, see Figure 6.4.

6.3.5 The Resulting PID

When decreasing the reference it is followed fairly well even with a P controller, although steady state errors are present. When increasing the reference the system property of nonlinear position change, described in section 5.5, becomes visible. When the reference in position of the piston in the cylinder is small the control signal is too large, causing an overshoot. When the reference is large the control signal is too small. With the I-part and D-part added the reference is better followed. However the steady state error is removed pretty slow. A larger I-part would cause overshoots and is not an option. The problem is that different proportional parts are suitable for different states of the system. Some kind of gain scheduling seems to be necessary in order to reach better results.

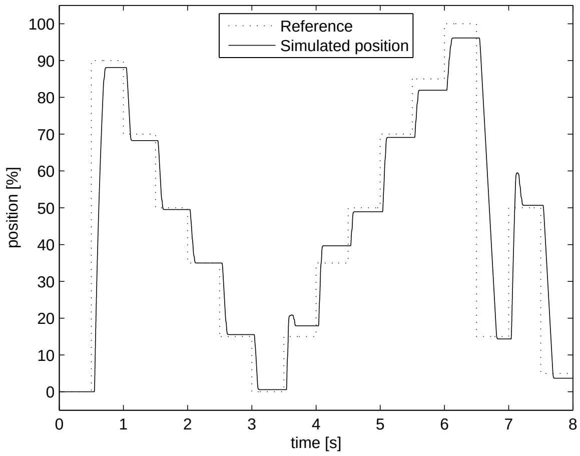


Figure 6.3. The pneumatic system simulated with a P-controller.

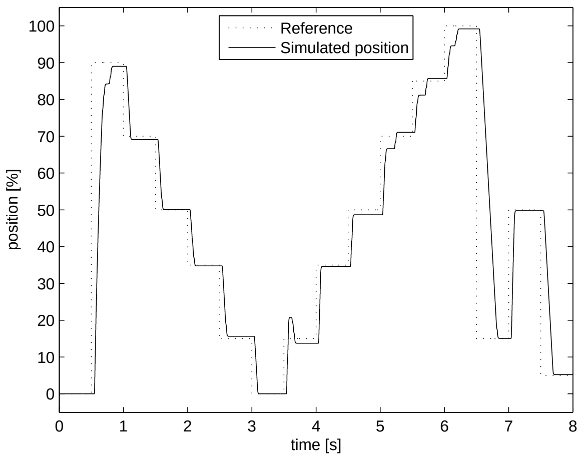


Figure 6.4. The pneumatic system simulated with a PID-controller.

6.4 Fuzzy Control

In order to cope with the fact that the system behaves differently depending on piston position (and cylinder pressure) it is interesting to evaluate what performance a fuzzy controller can achieve. It is possible to construct a fuzzy controller where the controller output is dependent on both control error and piston position. Fuzzy controllers are interesting because they are able to cope with nonlinearities and can be constructed to avoid under- and overshoots. In Figure 6.5 a block diagram of the fuzzy controller implemented in simulations is shown. The block $Fuzzy$ is the fuzzy controller, depending on r and y, and the block I is the integral part of the controller.

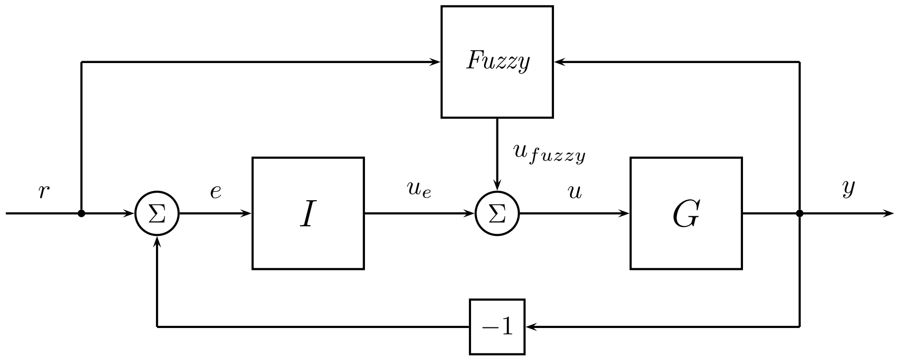


Figure 6.5. The structure of the fuzzy-I controller implemented in simulation.

In [10] a successful fuzzy controller for a similar system configuration is implemented. An introduction to fuzzy control and its theory is given in [2].

6.4.1 Membership Functions

Fuzzy controllers are in [4] described as verbally defined controllers. The name "fuzzy" originates from "fuzzy sets" which are verbally defined mathematical sets. An example of how a fuzzy set can be defined is

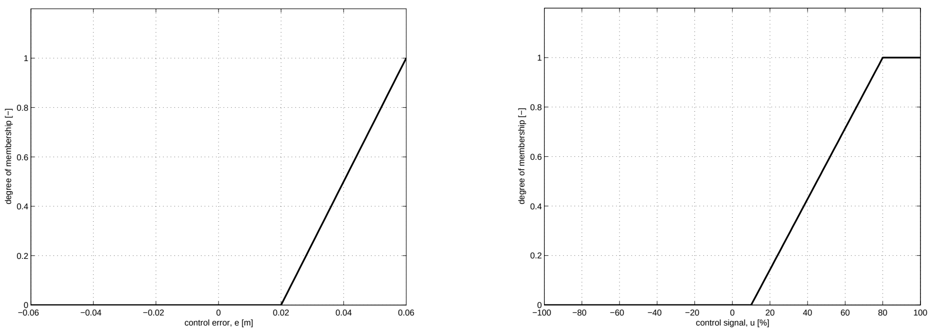
"If the control error is big and positive,
the control output should be big and positive" (6.5)

Now, it has to be decided in what degree a certain control error is a member of the statement that "the control error is big and positive". This is done by a *membership function*. Suppose the control error is $e = r - y$, and $-0.06 \leq e \leq 0.06$. A possible membership function is shown in Figure 6.6(a). So, if the control error is $e = 0.06$, it is definitely a member of the statement, while if the control error is $e = 0$ it is not. The sliding scale of membership when $0.02 \leq e \leq 0.06$ is interesting and results in that a control error of $e = 0.04$ is a member of the statement in the degree of 0.5. The membership function in Figure 6.6(a) is given by

$$\mu_{\mathrm{A}}(e) = \begin{cases} 25(e-0.02), & 0.02 \le e \le 0.06 \\ 0, & \text{else.} \end{cases} \tag{6.6}$$

It is also necessary to define the membership functions for the controller output. For the statement “the control signal is big and positive”, a possible membership function is presented in Figure 6.6(b), defined by

$$\mu_{\mathrm{B}}(u) = \begin{cases} 1, & u > 80 \\ (u-10)/70, & 10 \le u \le 80 \\ 0, & \text{else.} \end{cases} \tag{6.7}$$



(a) Possible membership function, $\mu_{\mathrm{A}}(e)$, for the statement “the control error is big and positive”.

(b) Possible membership function, $\mu_{\mathrm{B}}(e)$, for the statement “the control signal is big and positive”.

Figure 6.6. Examples of membership functions for the control input, $\mu_{\mathrm{A}}(e)$ defined by (6.6), and output, $\mu_{\mathrm{B}}(e)$ defined by (6.7).

6.4.2 Fuzzy Union, Intersection and Complement

In order to use the fuzzy sets, it is necessary to define a few usable operations. The operations union, intersection and complement from the classical theory of sets need to be translated into correspondences for fuzzy sets. Those are defined as [2]

$$\begin{aligned} \text{fuzzy union:} &\quad \mu(e) = \max(\mu_{\mathrm{A}}(e), \mu_{\mathrm{B}}(e)) \\ \text{fuzzy intersection:} &\quad \mu(e) = \min(\mu_{\mathrm{A}}(e), \mu_{\mathrm{B}}(e)) \\ \text{fuzzy complement:} &\quad \mu(e) = 1 - \mu_{\mathrm{A}}(e). \end{aligned} \tag{6.8}$$

6.4.3 Conditional Implications

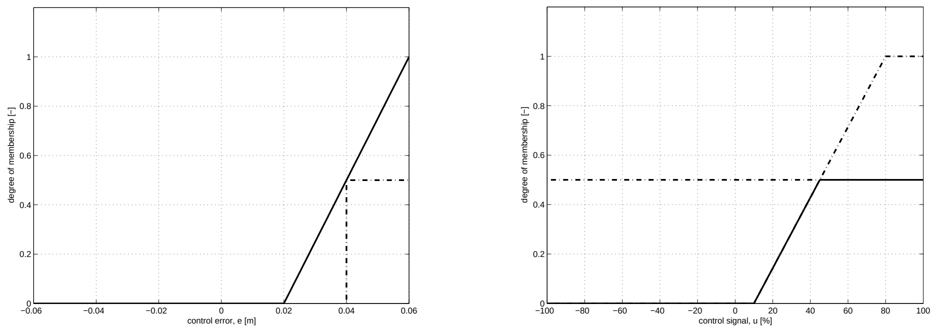
In order to connect the input to the output it is necessary to have a mathematical approach to calculate the output. Conditional implications, like

$$\text{“If } e \text{ is A, } u \text{ should be B”} \tag{6.9}$$

is for the measured value $e = e_0$ calculated with fuzzy intersection which results in the membership function

$$\nu(u) = \min(\mu_{\mathrm{A}}(e_0), \mu_{\mathrm{B}}(u)). \tag{6.10}$$

Using the same example as above, assuming there is a measured value $e_0 = 0.04$, the membership function $\nu(u)$ is calculated as depicted in Figure 6.7. Assuming



(a) $\mu_{\mathrm{A}}(e_0)$ (b) $\nu(u) = \min(\mu_{\mathrm{A}}(e_0), \mu_{\mathrm{B}}(u))$

Figure 6.7. Computation of the membership function $\nu(u)$.

there are several conditional implications affecting the same output, the following fuzzy sets can be used

$$\begin{array}{l}\text{“If } e \text{ is A, } u \text{ should be B”} \\ \text{“If } r \text{ is C, } u \text{ should be D”.}\end{array} \tag{6.11}$$

If the signals e and r have the values e_0 and r_0, the fuzzy sets with membership functions according to (6.10) becomes

$$\begin{aligned}\nu_1(u) &= \min(\mu_{\mathrm{A}}(e_0), \mu_{\mathrm{B}}(u)) \\ \nu_2(u) &= \min(\mu_{\mathrm{C}}(r_0), \mu_{\mathrm{D}}(u)).\end{aligned} \tag{6.12}$$

Since both conditional implications should be valid at the same time it is natural to calculate the union of the two fuzzy sets [4], i.e. the fuzzy set with membership function

$$\nu(u) = \max(\nu_1(u), \nu_2(u)). \tag{6.13}$$

6.4.4 “Defuzzification”

The computation of the output signal $\hat{u}$ from the membership functions is called “defuzzification” [4] and a usual method to do this is to calculate the center-of-gravity of the area below the function $\nu(u)$, i.e.

$$\hat{u} = \frac{\int u\nu(u)du}{\int \nu(u)du}. \tag{6.14}$$

In [2] more examples of how to calculate the output are presented. So, if all linguistic rules that describe the controller are calculated according to (6.10) and (6.13) and the output signal is "defuzzificated" according to (6.14) a control output, $\hat{u}$, is received.

6.4.5 Dynamics in a Fuzzy Controller

So far, the fuzzy sets presented have only been concerning static nonlinearities. Though, it is possible to define signals that represents the dynamics in the system. For instance, define

$$\begin{aligned} y_1(t) &= e(t) - e(t-1) \\ y_2(t) &= y(t-1) + e(t) \end{aligned} \tag{6.15}$$

and y_1 can be considered the derivative of the control error, and y_2 the integral of the control error. If fuzzy sets are defined for those signals, a fuzzy-PID controller can be constructed. Although, it can be difficult to have a concept of those fuzzy sets, why an I-part and a D-part may be easier to implement as in a conventional PID controller, with proportional gains in both the integral- and derivative part of the controller.

6.4.6 Knowledge Based Rules for the Actual System

Since there is knowledge about the system that is hard to implement in a simple controller this knowledge is wise to use. By just defining a few simple knowledge based rules, a nonlinear controller is constructed. The rules, together with their formal membership functions according to (6.10), are

"If the error is big and positive,
the control output should be big and positive":
$\nu_1(u) = \min(\mu_{\text{ePB}}(e), \mu_{\text{uPB}}(u))$

"If the error is medium and positive,
the control output should be medium and positive":
$\nu_2(u) = \min(\mu_{\text{ePM}}(e), \mu_{\text{uPM}}(u))$

"If the error is zero,
the control output should be zero":
$\nu_3(u) = \min(\mu_{\text{eZ}}(e), \mu_{\text{uZ}}(u))$

"If the error is medium and negative,
the control output should be medium and negative":
$\nu_4(u) = \min(\mu_{\text{eNM}}(e), \mu_{\text{uNM}}(u))$

"If the error is big and negative,
the control output should be big and negative":
$\nu_5(u) = \min(\mu_{\text{eNB}}(e), \mu_{\text{uNB}}(u))$

"If the error is medium and positive and the position is big,
the control output should be big and positive":
$\nu_6(u) = \min(\mu_{\text{yP}}(y), \mu_{\text{uPB}}(u))$.

Here, the last linguistic rule describes the property in (2.20) as discussed in section 5.5, i.e. that the valve needs to be open for a longer time if the pressure difference (between supply and cylinder pressure) is small and where y is the measured position.

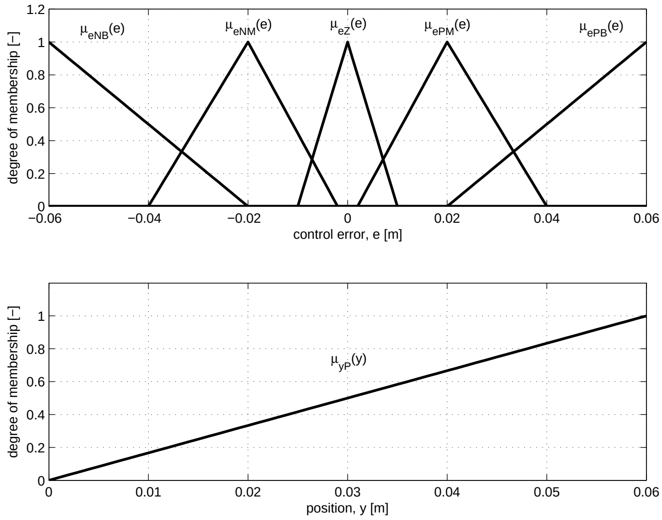


Figure 6.8. Suggested membership functions for the controller input.

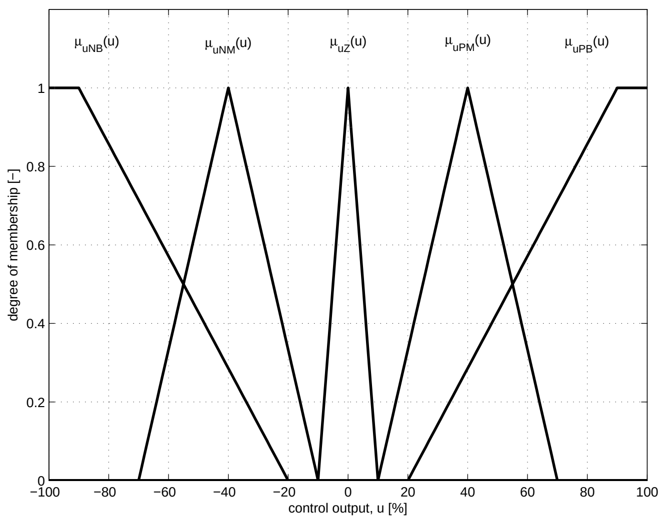


Figure 6.9. Suggested membership functions for the output signal.

In Figure 6.8, the suggested membership functions for the controller input is shown. Figure 6.9 shows the membership functions for the output.

So, now it is possible to calculate the controller output assuming measured values of e and y are available. By using (6.13), the resulting membership function becomes

$$\nu(u) = \max(\nu_1(u), \nu_2(u), \nu_3(u), \nu_4(u), \nu_5(u), \nu_6(u)). \tag{6.16}$$

Finally, the "defuzzification" in (6.14) computes the actual output, $\hat{u}$. In Figure 6.10, a surface plot of the controller outputs for some of all possible inputs are shown. Note the increasing output signal for a larger positive value of y while the control error, e, is held constant, described by ν_6.

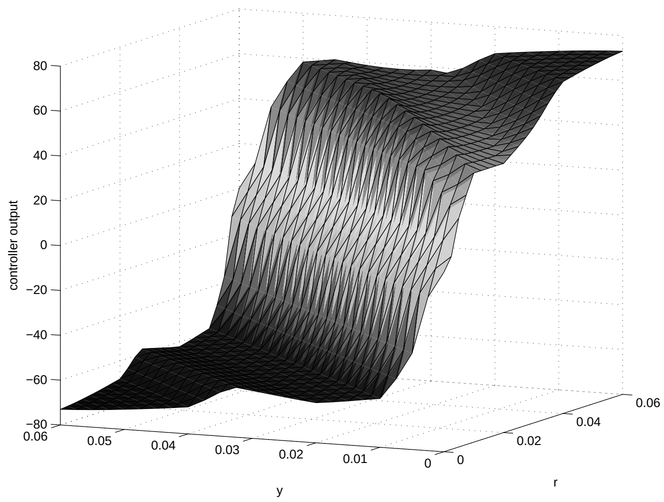


Figure 6.10. Surface plot of the controller output.

6.4.7 The Resulting Fuzzy Controller

The resulting fuzzy controller will follow a reference well, see Figure 6.11. Some problems occur when a steady state error exists, why an integrating element of the same size as in the PID controller is added, making it a fuzzy-I controller.

6.4.8 Discrete Implementation of the Fuzzy Controller

When implementing the fuzzy controller in the ECU it is done by interpolation of a matrix like the one plotted in Figure 6.10. The integrating element of the controller is implemented as in the PID, with proportional gain on the integrator sum.

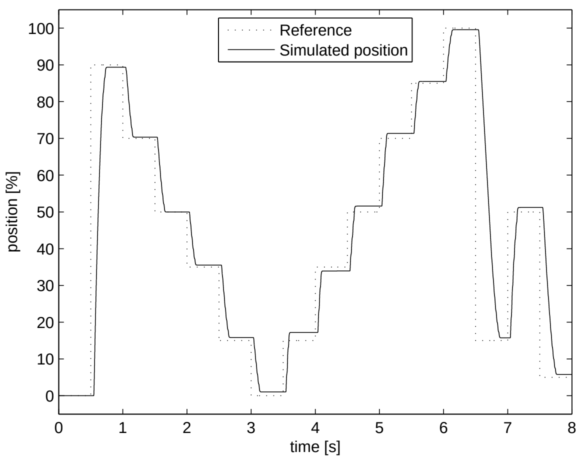


Figure 6.11. The pneumatic system simulated with a fuzzy controller.

6.5 PID Compared to Fuzzy-I

When testing the performance of the control of the EGR valve in simulation, a reference measured from a common driving case is suitable to use. Such a reference of 30 minutes length is used here. Samples of the reference for the piston position are shown in Figure 6.12, 6.13, 6.14 and 6.15 together with the simulated output from a PID controller and a fuzzy-I controller.

Through a visual study of the plots in Figure 6.14 and 6.15 some conclusions can be drawn. Figure 6.14(a) and 6.15(a) shows that the PID controller tends to have a bit more overshoots than the fuzzy-I controller. Figure 6.14(b) and 6.15(b) shows that the controllers perform equally good when the reference is rapidly increased. This is also the case when the reference is rapidly decreased, as seen in Figure 6.14(c) and 6.15(c), where it also can be seen that the controllers have equal settling times. When the reference is moderately alternating, the PID controller tends to change the output more often than the fuzzy-I controller does, as can be seen in Figure 6.14(d) and 6.15(d). Overall the fuzzy-I controller is a bit more moderate than the PID.

The mean relative error between the reference and the actual position is calculated according to (4.1). This is for a common driving case simulated over 1800 s. The results indicates the PID and the fuzzy-I controller being equally good, see Table 6.1. However the data contains lots of time periods with zero as reference. If the mean relative error is calculated for the sequences shown i Figure 6.12 and 6.13, the fuzzy-I controller is slightly better, see Table 6.2. The reason for this is that the fuzzy-I controller takes position into consideration and thereby causing smaller overshoots than the PID controller.

Another important property for the controllers to have is to keep down the

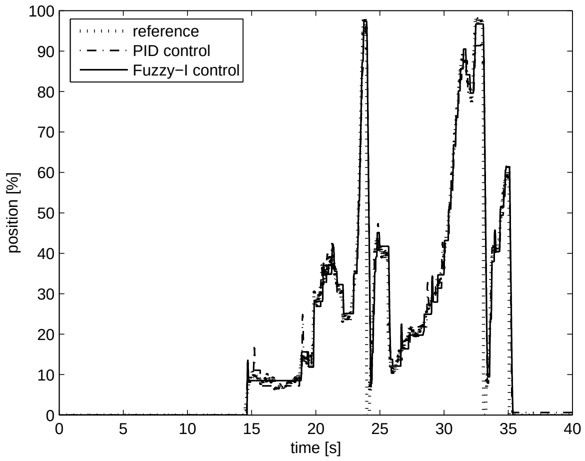


Figure 6.12. A reference for the EGR valve with the simulated output retrieved with a PID controller and a fuzzy-I controller.

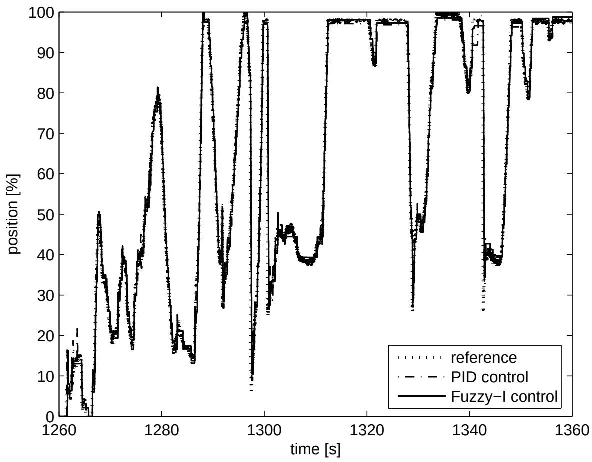
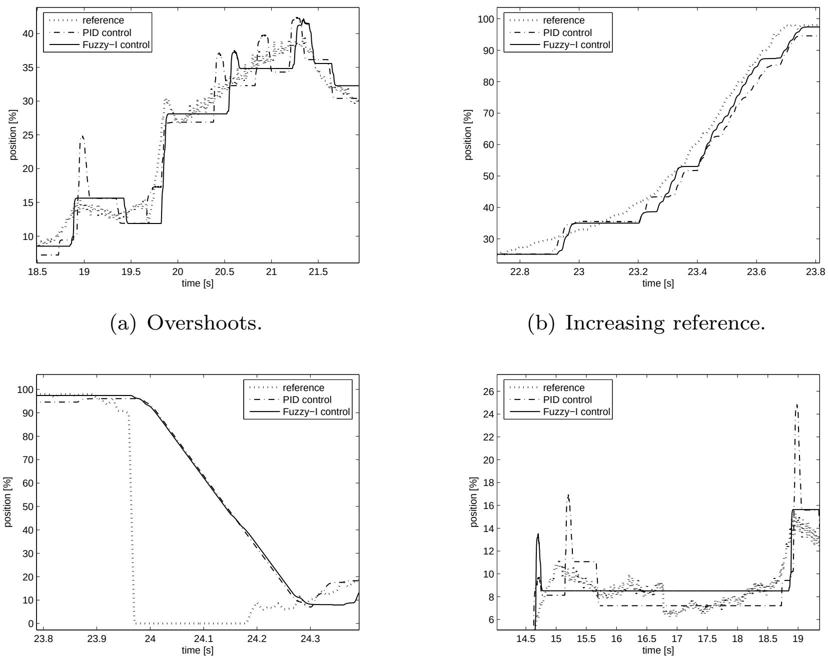


Figure 6.13. A reference for the EGR valve with the simulated output retrieved with a PID controller and a fuzzy-I controller.



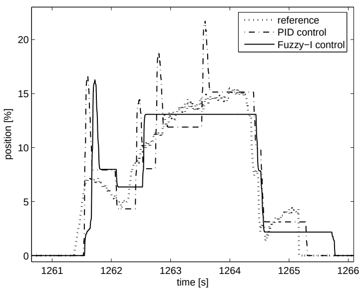
(a) Overshoots.

(b) Increasing reference.

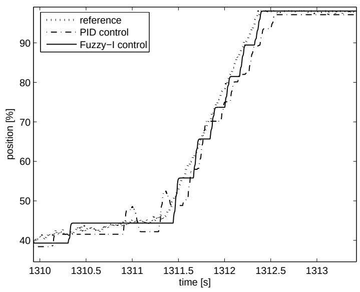
(c) Decreasing reference.

(d) Fairly constant reference.

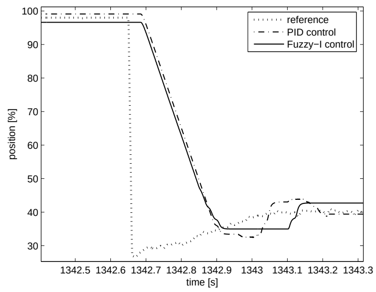
Figure 6.14. Zooms from Figure 6.12.



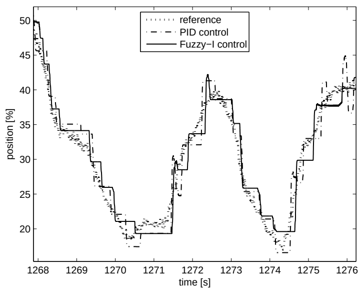
(a) Overshoots.



(b) Increasing reference.



(c) Decreasing reference.



(d) Alternating reference.

Figure 6.15. Zooms from Figure 6.13.

number of openings of the valves. In simulations this can be measured. The total number of openings of both valves together during a simulation over 1800 s is presented in Table 6.3.

	Mean relative error
PID	2.04%
fuzzy-I	2.03%

Table 6.1. The mean relative error between reference and actual position with PID control and fuzzy-I control in simulation with a common driving case over 1800 s.

	Mean relative error	
time window	0-40 s	1260-1360 s
PID	2.22%	5.01%
fuzzy-I	2.10%	1.73%

Table 6.2. The mean relative error between reference and actual position with PID control and fuzzy-I control in simulation with time windows from driving cases with few static levels.

	Valve openings		
Valve	supply	exhaust	$\sum$
PID	8318	5734	14052
fuzzy-I	8240	7032	15272

Table 6.3. The total number of valve openings with PID control and fuzzy-I control in simulation of a common driving case over 1800 s.

6.6 Control in Test Bench

The system is set up in a test bench consisting of a pneumatic cylinder together with two On/Off valves and an ECU. The implemented control system is a P controller together with a feedforward loop from position. This simple implementation is made to evaluate if there is an increase in performance in a controller that takes piston position into consideration. The effects of the free wheel diode in the ECU is taken into account.

A block diagram of the control system implemented in the test bench is shown in Figure 6.16, where the block G is the system, F is the P controller and F_f is the feedforward control from position.

6.6.1 Feedforward from Postion

Because of different pressures inside the cylinder, it takes different PWM signals to open the valves, as discussed in section 5.5. The reason for this is the force denoted

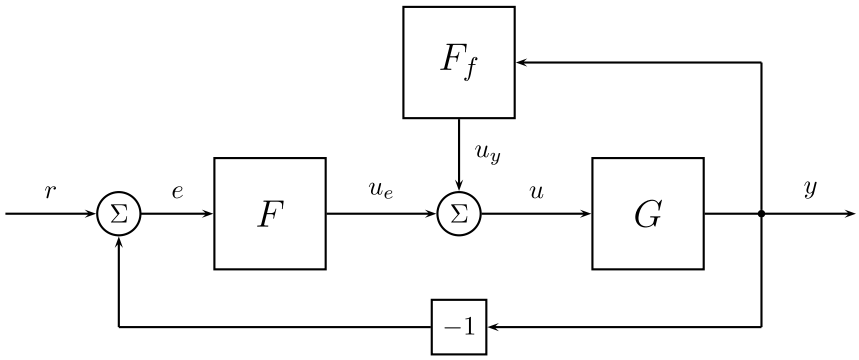


Figure 6.16. The structure of the controller implemented in the test bench.

F_{prs} in (2.15). This force grows linearly with the pressure inside the cylinder and a different pressure therefor acts on the plunge with different magnitudes. This results in that a larger PWM signal needs to be applied on the supply valve when the position of the piston is larger because the electromagnet needs time to induce a current big enough to overcome the force from the pressure inside the cylinder. The opposite goes for the exhaust valve.

E.g., when the piston position is close to its maximum level (high pressure) a PWM signal of $\sim 30\%$ needs to be applied on the supply valve before the plunge starts to move, but while it is close to its minimum position (low pressure) a PWM signal of $\sim 20\%$ is big enough. When it comes to moving the plunge of the exhaust valve a $\sim 30\%$ PWM needs to be applied when the piston position is close to its minimum while a $\sim 20\%$ PWM is needed if the piston position is close maximum. To reduce this nonlinearity a feedforward loop from position is implemented so that the plunge always is at the verge of moving. Then a P controller is applied which is proportional to the control error, giving the same PWM signal (almost) the same effect, independent of piston position.

Which feedforward control signals are needed for certain pistion positions are measured and a function is fitted to those measured values.

6.6.2 Results

With a reference from a common driving case, the controller described above follows it better than the simulated controllers. A plot of the reference and the measured position is shown in Figure 6.17 and zoomed in Figure 6.18.

This way to feedforward a control signal from piston position is dependent on supply pressure. When the supply pressure is changed, the performance of a controller tuned for a certain pressure will therefor decrease.

This controller, where the plunge always is at the verge of opening, will have many valve openings.

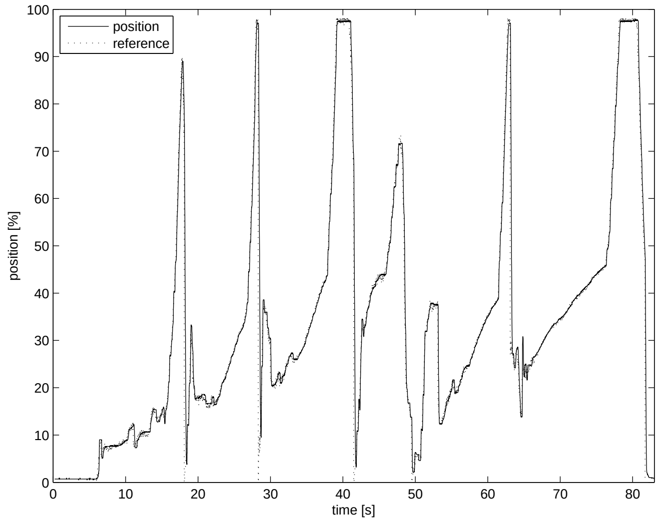


Figure 6.17. Reference for the piston position from a common driving case and measured piston position in a test bench.

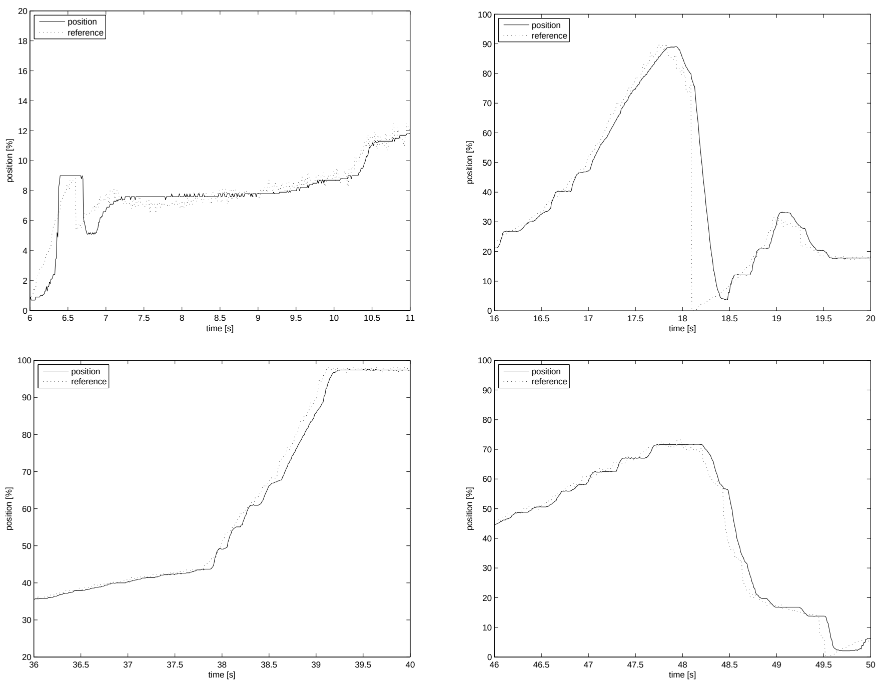


Figure 6.18. Zooms from Figure 6.17.

6.7 Control in Engine Test Cell

The same control system that was implemented in a test bench, presented in section 6.6.1, is also implemented in an engine test cell. The engine is run on a common driving case over 1800 s.

6.7.1 Results

The results from the engine test cell are similar to the results from the the test bench. A plot of the reference and the measured output is shown in Figure 6.19.

It is also interesting to compare the On/Off valve configuration to the system used to control the EGR valve today, which is the same pneumatic cylinder controlled with a proportional valve. As presented in the introducing section 1.1, the On/Off valves are more robust and lower in cost than today's proportional valve and therefor it is interesting to compare those two systems. The same driving case is run on the same engine using the proportional valve. A comparison of the reference and the measured output is shown in Figure 6.20. As can be seen when comparing Figure 6.19 and Figure 6.20, the performance of both systems are similar, indicating that the On/Off valve system has potential.

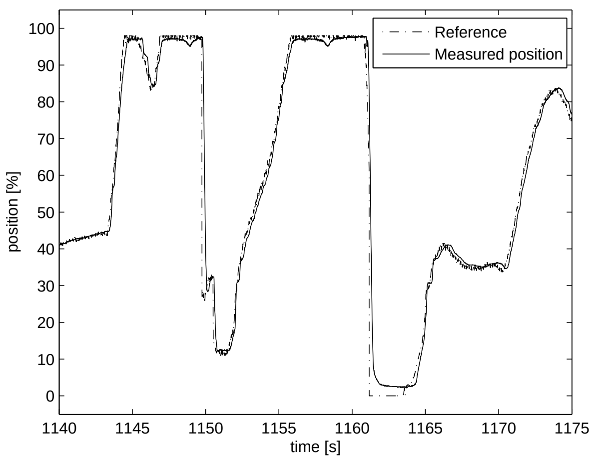


Figure 6.19. Reference for the piston position from a common driving case and measured piston position in an engine test cell with On/Off valves.

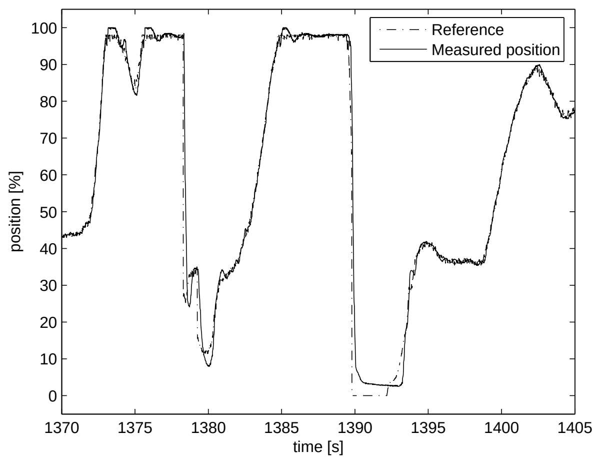


Figure 6.20. Reference for the piston position from a common driving case and measured piston position in an engine test cell with the EGR valve control system used today.

Chapter 7

Results and Future Work

This chapter will discuss the results of this master's thesis and what conclusions can be drawn from those results. It will also suggest future work.

7.1 Results

Here the results of the thesis are presented.

7.1.1 System Model

The mathematical model of the complete system including two On/Off valves and a cylinder consists of nine states. It has several nonlinearities.

The model captures the dynamics of the system, i.e. the rise/fall time, well. The static levels reached from a certain input signal is not always correct. Since position feedback is available, it is still concluded that the model is valid to be used as aid when developing and evaluating different control methods.

7.1.2 Properties of the System

The coil in the valve is an inductive element. As a result of this it is charged and discharged, causing time delays. A certain current level has to be reached in order to open the valve. To close the valve, the current has to drop below a certain level. Therefor the control signal needs to be chosen between certain values to have effect.

The resistance of the coil is temperature dependent. A certain control signal will therefor have different effect dependent on the coil temperature.

The position of the butterfly valve controlling the exhaust gas flow depends on the pressure inside the pneumatic cylinder. When the piston position is altered the pressure and the temperature in the cylinder changes (ideal gas law). Once in position the air in the cylinder takes on the same temperature as its surrounding, changing the pressure inside it.

Any control signal big enough to open the valve and overcome the static friction in the cylinder will alter the position of the piston inside of the cylinder. The system is thereby an integrating system.

The length of a piston movement caused by a certain input signal is different dependent on initial position. The reason is that the pressure difference on the upside and downside of the valve influence the opening/closing time of the valve, and the mass flow rate through it.

Individual spread between different valves exists. A certain set of parameters will therefor not fit all valves.

7.1.3 System Control

A PID controller is developed and evaluated in simulations. It is discovered that different proportional parts, depending on piston position, is needed because of the nonlinear behavior of the system.

To deal with the nonlinearities and to include the piston position in the choice of control signal, fuzzy control is introduced. It is implemented as a map. The control signal is interpolated between different map values dependent on control error and piston position. The results are better than with the PID, but not as good as hoped for. The map implementation is a likely reason since it limits the precision when choosing control signal.

In a test bench, the nonlinearities are held back through feedforward from position. The feedforward signal is chosen so that the valves are always at the verge of opening. A P controller then has almost the same effect independent of position. The result is a reference better followed than the simulated controllers.

In an engine test cell the same results as in the test bench are achieved with the same controller. The performance of the electro-pneumatic actuator system using On/Off valves are compared to the system used today. The results are similar, indicating that the On/Off valve system has potential.

7.2 Future Work

The model could be improved by modeling the magnetic force as a network of magnetic resistances, instead of using Maxwell's magnetic force model.

A larger map will probably improve the performance of the fuzzy controller.

The test bench results could be improved by making the P part also position dependent. Feedforward from temperature/resistance could also be introduced. Adaption to set the feedforward from position for each individual valve would make the controller general.

The robustness of the system should be evaluated.

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