

Torque Modeling and Control of a Variable Compression Engine

Master's thesis
performed in **Vehicular Systems**

by
Andreas Bergström

Reg nr: LiTH-ISY-EX-3421-2003

29th April 2003

Torque Modeling and Control of a Variable Compression Engine

Master's thesis

performed in **Vehicular Systems**,
Dept. of Electrical Engineering
at **Linköpings universitet**

by **Andreas Bergström**

Reg nr: LiTH-ISY-EX-3421-2003

Supervisor: **Per Andersson**
Linköpings Univeristet

Examiner: **Assistant professor Lars Eriksson**
Linköpings Universitet

Linköping, 29th April 2003

	Avdelning, Institution Division, Department Vehicular Systems, Dept. of Electrical Engineering 581 83 Linköping	Datum Date 29th April 2003
	Språk Language <input type="checkbox"/> Svenska/Swedish <input checked="" type="checkbox"/> Engelska/English <input type="checkbox"/> _____	Rapporttyp Report category <input type="checkbox"/> Licentiatavhandling <input checked="" type="checkbox"/> Examensarbete <input type="checkbox"/> C-uppsats <input type="checkbox"/> D-uppsats <input type="checkbox"/> Övrig rapport <input type="checkbox"/> _____
URL för elektronisk version http://www.vehicular.isy.liu.se http://www.ep.liu.se/exjobb/isy/2003/3421/		
Titel Title Författare Andreas Bergström Author	Titel Momentmodellering och momentreglering av en variabelkompressionsmotor Title Torque Modeling and Control of a Variable Compression Engine	
Sammanfattning Abstract <p>The SAAB variable compression engine is a new engine concept that enables the fuel consumption to be radically cut by varying the compression ratio. A challenge with this new engine concept is that the compression ratio has a direct influence on the output torque, which means that a change in compression ratio also leads to a change in the torque. A torque change may be felt as a jerk in the movement of the car, and this is an undesirable effect since the driver has no control over the compression ratio.</p> <p>The aim of this master's thesis work is to develop a torque control strategy for the SAAB variable compression engine. Where the main control objective is to make the output torque behave in a desirable way despite the influence of compression ratio changes.</p> <p>The controller is developed using a design method called Internal Model Control, which is a straightforward way of both configuring a controller and determining its parameters. The controller has been implemented and evaluated in a real engine, and has proved to be able to reduce the effect of compression ratio disturbance.</p>		
Nyckelord Keywords Torque control, SVC, Variable compression, IMC, MVEM		

Abstract

The SAAB variable compression engine is a new engine concept that enables the fuel consumption to be radically cut by varying the compression ratio. A challenge with this new engine concept is that the compression ratio has a direct influence on the output torque, which means that a change in compression ratio also leads to a change in the torque. A torque change may be felt as a jerk in the movement of the car, and this is an undesirable effect since the driver has no control over the compression ratio.

The aim of this master's thesis work is to develop a torque control strategy for the SAAB variable compression engine. Where the main control objective is to make the output torque behave in a desirable way despite the influence of compression ratio changes.

The controller is developed using a design method called Internal Model Control, which is a straightforward way of both configuring a controller and determining its parameters. The controller has been implemented and evaluated in a real engine, and has proved to be able to reduce the effect of compression ratio disturbance.

Keywords: Torque control, SVC, Variable compression, IMC, MVEM

Preface

This master's thesis has been performed at Vehicular Systems, Linköping University during fall/winter 2002/2003.

Thesis Outline

The work done during this thesis is described in the following chapters.

- Chapter 1 *Introduction*. Introduction to the thesis
- Chapter 2 *Theory*. An overview of the basic theory for combustion engines, model building and control theory.
- Chapter 3 *The Variable Compression Concept*. Here the ideas behind the variabel compression concept is presented.
- Chapter 4 *Engine Model*. Presentation of the developed model and the simulation results.
- Chapter 5 *Preliminary Study*. Short investigation of which signals that are suitable to use and calculations of important transfer functions.
- Chapter 6 *Control Algorithms*. Presentation of the developed control algorithm and the results from the evaluation.
- Chapter 7 *Conclusions and Future Work*. Conclusions and topics for future studies

Acknowledgment

I would like to thank everyone at Vehicular Systems for a nice and stimulating time, especially my supervisor *Per Andersson* for his encouragement and valuable ideas. Further, I would like to thank *Martin Gunnarsson* for all help in the lab, and also *Christer Rosenquist* for help with the real time system.

Andreas Bergström
Linköping, April 2003

Notation

Nomenclature

Symbol	Quantity	Unit
$(\frac{A}{F})$	Air-to-fuel ratio	-
M	Engine Torque	Nm
M_{net}	Net Torque	Nm
M_c	Combustion torque	Nm
M_i	Ignition angle torque	Nm
M_f	Friction torque	Nm
M_p	Pumping torque	Nm
\dot{m}_{ac}	Air flow into cylinder	kg/s
\dot{m}_{at}	Air flow past throttle	kg/s
\dot{m}_f	Fuel flow into cylinder	kg/s
\dot{m}_i	Change of mass in the intake manifold	kg/s
N	Engine speed	rpm
p_e	Pressure in the exhaust manifold	Pa
p_i	Pressure in the intake manifold	Pa
q_{hv}	Heating value	J/kg
R	Gas constant	$J/(kg \cdot K)$
r_c	Compression ratio	-
T_i	Intake manifold temperature	K
V_i	Intake manifold volume	m^3
V_d	Displaced volume	m^3
V_c	Clearance volume	m^3
γ	Ratio of specific heat	-
η_{vol}	Volumetric efficiency	-
θ_{ign}	Ignition Angle	degree
λ	Air -to-fuel equivalence ratio (normalized $(\frac{A}{F})$)	-
τ_{th}	Throttle time constant	s
τ_r	Compression ratio time constant	s

Abbreviations

Abbreviation	Explanation
BDC	Bottom Dead Center
IMC	Internal Model Control
MEP	Mean Effective Pressure
MVEM	Mean Value Engine Model
SI	Spark Ignited
SVC	SAAB Variable Compression
TDC	Top Dead Center

Contents

Abstract	v
Preface and Acknowledgment	vii
Notation	viii
1 Introduction	1
1.1 Objectives	2
1.2 Methods	2
1.3 Target Group	3
2 Theory	5
2.1 Engine Introduction	5
2.1.1 Four stroke cycle	6
2.2 Model Building	7
2.3 Control Theory	7
2.3.1 The Transfer Functions of the Closed Loop System	8
3 The Variable Compression Concept	9
3.1 Downsizing	9
3.2 Supercharging	10
3.3 Variable compression	10
3.4 SAAB Variable Compression Engine	11
4 Engine Model	13
4.1 Model Overview	13
4.2 Main throttle	14
4.3 Intake Manifold	15
4.3.1 Dynamic Pressure Model	15
4.3.2 Engine Mass Flow Model	16
4.3.3 Fuel Injection	19
4.4 Cylinders	19
4.4.1 Combustion	20
4.4.2 Pumping losses	21

4.4.3	Friction losses	21
4.4.4	Ignition Angle	22
4.5	Compression Ratio	22
4.6	Model Summary	22
4.7	Model Implementation	24
4.8	Model Validation	25
4.8.1	Stationary Validation	25
4.8.2	Dynamic Validation	26
5	Preliminary Study	29
5.1	Controller Signals	29
5.1.1	Air-Flow Past Throttle	29
5.1.2	Ignition Angle	29
5.1.3	Compression Ratio	30
5.1.4	Torque	30
5.1.5	Summary	30
5.2	Transfer Function	31
6	Control Algorithm	33
6.1	Control Demands	33
6.2	Control Structure	34
6.3	Control Design	34
6.3.1	How to Choose the Design Method?	35
6.3.2	Internal Model Control (IMC)	35
6.3.3	Controller Calculations	37
6.4	Observer	40
6.5	Evaluation	40
6.5.1	Evaluation of Observer	41
6.5.2	Evaluation of the Controller	42
7	Conclusions and Future Work	47
7.1	Summary	47
7.2	Accomplishments and Conclusion	48
7.3	Future Work	48
	References	51
	A Ignition Angle Measurements	53
	B Model Parameters	55
	C SIMULINK Implementation	57

Chapter 1

Introduction

SAAB has developed a new engine concept called SVC (SAAB Variable Compression), this new engine concept enables fuel consumption to be radically cut, but without impairing engine performance. The SVC engine has variable compression i.e. variable size of the clearance volume. This alone adds a new control input that affects every aspect of engine control. The engine is also equipped with a compressor for supercharging, with some additional associated control inputs.

One challenge with this new concept is that the compression ratio has a direct influence on the output torque. This is a problem because the engine then can change its characteristics under operation, and this influences the driveability of the car. A big change in the compression ratio can even be felt by the driver as a jerk in movement. Another problem is that the connection and disconnection of the mechanical supercharger also have a direct influence on the engine torque, and sudden connection and disconnection can also be felt as jerks in movement.

So in order to fully take advantage of the benefits of this new engine concept, it is necessary to have some sort of torque control that can keep a constant torque regardless of these disturbances. This thesis concentrates on finding a solution to the problem with the compression ratio changes.

The SVC engine makes it possible to change the compression ratio between 8 and 14. Figure 1.1 shows a step in the compression ratio from 8 to 14 which results in an increased torque. A compression ratio step from 8 to 14 increases the output torque with about 20 %, and such a significant torque change can be felt as a jerk in the car movement. The goal for this thesis is to develop a torque controller that damps or eliminates the effect of compression ratio on the output torque.

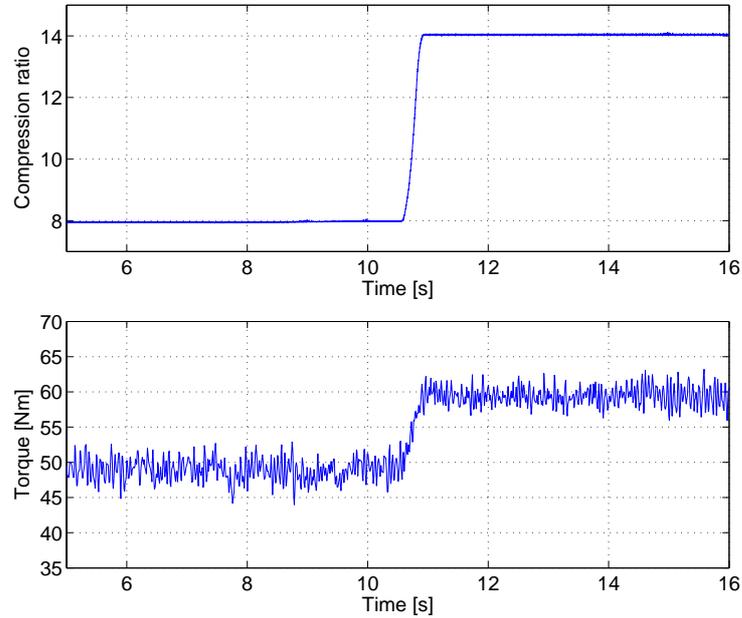


Figure 1.1: Shows how the output torque changes when the compression ratio changes from 8 to 14, at 2000 *rpm*. The top plot shows the compression ratio and the bottom plot shows the output torque.

1.1 Objectives

There are three objectives for this thesis. The objectives are to:

- Develop a torque model for the the SVC engine with the input signals air mass flow past throttle, compression ratio and ignition angle.
- Develop and evaluate one or more torque control strategies in the model.
- Evaluate the control strategies on a real SVC engine in the engine lab.

1.2 Methods

The model and control strategies have been developed and implemented in an MATLAB and SIMULINK environment. The necessary data collection for the model and evaluation of the control strategies has been

performed in an engine test cell in Vehicular Systems' research laboratory, using a measurement system combined with a real time system.

1.3 Target Group

This thesis is aimed for engineers and students, with basic knowledge in the areas of vehicular systems and control theory.

Chapter 2

Theory

In this chapter some theoretical background to the subject areas in this thesis will be given. An introduction to how a four cycle combustion engine works is presented in section 2.1. In section 2.2 and 2.3 some basic principles for model building and control theory is covered.

2.1 Engine Introduction

An internal combustion engine uses air and fuel based on hydrocarbons and produces power and emissions. A schematic overview is given in Figure 2.1.

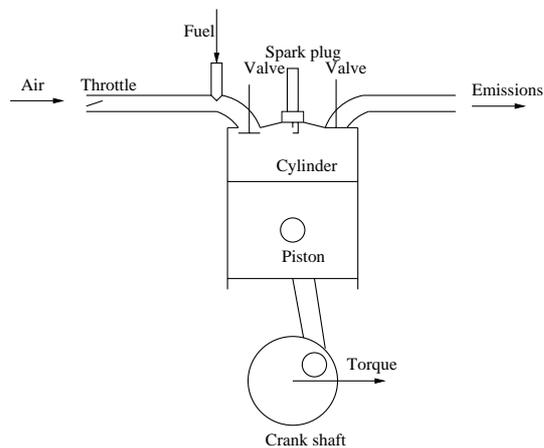


Figure 2.1: An engine takes air and fuel as input and generates torque and emissions.

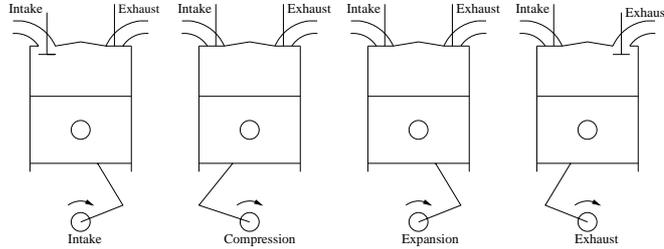


Figure 2.2: The four strokes intake, compression, expansion and exhaust of a four stroke engine.

2.1.1 Four stroke cycle

For a four stroke engine the combustion cycle is divided into four steps, illustrated in Figure 2.2. Note that all the mentioned values of the crank shaft angle are only examples and may vary from one engine to another.

The first stroke is called the *intake stroke* (from top dead center (TDC) to bottom dead center (BDC)). During this, the intake valves is open and while the piston moves downwards the cylinder is filled with air/fuel mixture. Due to the open inlet valves the cylinder pressure remains fairly constant.

A *compression stroke* (BDC-TDC) follows, where the air/fuel mixture is compressed to higher pressure and temperature through mechanical work produced by the piston. Around 25° before TDC a spark ignites the mixture and initiates the combustion, whereas the flame propagates through the combustion chamber and adds heat to the mixture.

The combustion continues into the *expansion stroke* (TDC-BDC) and finishes around 40° after TDC. Work is produced during the expansion stroke when the volume expands. Around 130° after TDC the exhaust valve is opened and the blowdown process starts, where the cylinder pressure decreases as the burned gases is blown out into the exhaust system by the higher pressure in the cylinder.

During the final stroke, *exhaust stroke* (BDC-TDC), the valve is still open and therefore the pressure in the cylinder is close to the pressure in the exhaust system and the rest of the gases is pushed out into the exhaust system as the piston moves upwards. When the piston reaches TDC a new cycle starts with the intake stroke.

There are several textbooks about SI-engines. One book which discusses most aspects of internal combustion engines is [1]. A more concise description of SI-engines is given in [2].

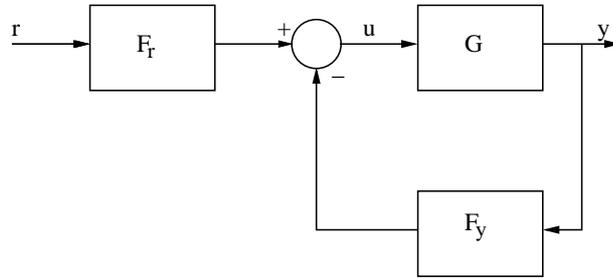


Figure 2.3: A controller with two degrees of freedom.

2.2 Model Building

There are two main ways of building models of systems', *physical modeling* and *identification* [3]. Physical modeling means that the system is divided into subsystems, with known behavior. For technical systems this in general means, that the law of nature are used to describe the subsystems. Identification means that an observation from the system is used to adjust the model properties to the system properties. This principle is often used as a complement to the first one.

One textbook that covers model-building theory is [3].

2.3 Control Theory

A general linear controller can be written as

$$u(t) = F_r r(t) - F_y y(t) \quad (2.1)$$

Where $u(t)$ is the control signal, $r(t)$ is the reference signal and $y(t)$ is a measured output, which should be controlled to desired values. The controller (2.1) is a *two-degrees-of-freedom controller*. It is determined by two independently chosen transfer functions F_r and F_y . See Figure 2.3. A very common special case is $F_r = F_y$ which gives

$$u(t) = F_y (r(t) - y(t))$$

Such a controller is a *one-degree-of-freedom controller*. Often such a controller is first designed, and, if necessary, modified to two degrees of freedom according to

$$u(t) = F_y (\tilde{F}_r r(t) - y(t))$$

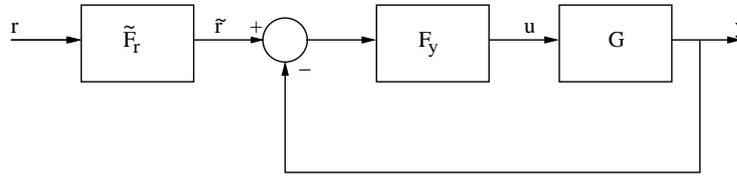


Figure 2.4: A controller with one degree of freedom modified with \tilde{F}_r to a two degrees of freedom controller.

This corresponds to $F_r = F_y \tilde{F}_r$ in (2.1) and can also be seen as a *one-degree-of-freedom controller*, where the reference has been pre-filtered: $\tilde{r} = \tilde{F}_r r$. See Figure 2.4.

2.3.1 The Transfer Functions of the Closed Loop System

If the closed loop system looks like Figure 2.4 the most important transfer functions can be defined as

- The close loop system, G_c

$$G_c = (1 + GF_y)^{-1} GF_y \tilde{F}_r$$

- Sensitivity function, S

$$S = (1 + GF_y)^{-1}$$

- Complementary sensitivity function, T

$$T = (1 + GF_y)^{-1} GF_y$$

Where G_c is the transfer function form r to y . S is the transfer function that describes how the controller reduces model errors and system disturbances. T describes how the controller handles measurement disturbances.

More information about basic control theory can be founded in [4], [5] and [6].

Chapter 3

The Variable Compression Concept

Variable compression is a new engine concept that enables fuel consumption to be radically cut, but without impairing engine performance. The variable compression concept contains three cornerstones *downsizing*, *supercharging* and *variable compression* [7]. The cornerstones are described in section 3.1, 3.2 and 3.3. One example of a variable compression engine is the SAAB Variable Compression (SVC) engine, technical data and pictures of the engine is presented in section 3.4.

3.1 Downsizing

An Otto engine is most efficient and uses the energy in the fuel at its maximum when it is running at high load. A small engine must work harder and must thus run to its almost full load if it is to perform the same work as a bigger engine that utilizes only part of its maximum capacity during normal operation.

One of the reasons is that, under these conditions, the pumping losses are lower in a small engine. The piston in the cylinder is under a slight vacuum during the intake stroke, when it is drawing air into the cylinder. The extra energy needed for pulling the piston down is known as the pumping losses. Since a small engine more frequently runs at full load and the throttle is therefore more often fully open, the pumping losses in the small engine are usually lower than they are in a big engine.

Moreover a small engine is lighter and has lower friction. So a small engine is generally more efficient than a big engine.

3.2 Supercharging

Although a small engine is efficient, it is not powerful enough in practice to be used for anything then powering small, lightweight cars, if it is to give the car acceptable performance. By supercharging, which involves forcing in more air, more fuel can be injected and be burned. The engine then delivers more power for every piston stroke, which results in higher torque and higher engine output. Moreover if the engine is supercharged only at large throttle openings when extra power is really needed, the fuel economy of the small engine can be combined with the performance of a big engine.

3.3 Variable compression

In [1] compression ratio, r_c , is defined as

$$r_c = \frac{V_d + V_c}{V_c}$$

Where V_d is the displaced volume and V_c is the clearance volume. In other words, the amount by which the fuel/air mixture is compressed in the cylinder before it is ignited. The compression ratio is one of the most important factors that determine how efficiently the engine can utilize the energy in the fuel. For an ideal Otto cycle the theoretical efficiency is [1]

$$\eta = 1 - \frac{1}{r_c^{\gamma-1}}$$

Where γ is ratio of specific heats. For an SI-engine operating at stoichiometric mixture $\gamma \approx 1.3$.

As a general rule, the energy in the fuel will be better utilized if the compression ratio is as high as possible. Current engines have a compression ratio around 10, and the question is why is it not higher. This is because [2]:

- If the compression ratio is too high the air/fuel mixture are exposed to higher temperatures, the fuel will auto ignite, giving rise to knocking, which could damage the engine.
- Increased heat transfer to the combustion chamber walls due to small clearance volume.
- Increased friction losses and emission of unburned hydrocarbons.

In a conventional engine, the maximum compression ratio with which the engine can withstand is therefore set by the conditions in the cylinder at high load, when the fuel and air consumption is at maximum. The compression ratio remains the same when the engine is running at low load.

The basic idea of variable compression is simple: Use high compression ratio during low load for high efficiency, and as the load increases the compression ratio is decreased to match the knocking.

3.4 SAAB Variable Compression Engine

The SAAB Variable Compression (SVC) engine is a 5 cylinder 1.6 *L* engine with performance of 225 *Hp*/305 *Nm*. It is supposed to effectively replace a 3.0 *L* natural aspirated engine. A picture of the engine is seen in Figure 3.1.

The SVC engine consists of an upper part comprising a cylinder head with integrated cylinders, which is known as the monohead, and a lower part consisting of the engine block, crankshaft and piston. The compression ratio is varied by tilting the upper part of the engine in relation to the lower part. This alters the volume of the combustion chamber with the piston at top dead center which also changes the compression. In Figure 3.2 the change in compression ratio is shown. Technical data for the SVC engine is presented in Table 3.1. The technical data comes from [7].



Figure 3.1: Picture of the SVC engine.

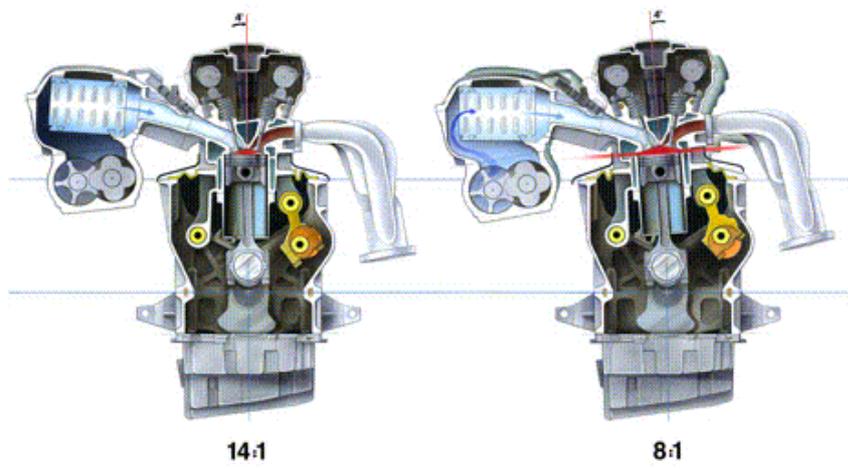


Figure 3.2: Showing the SAAB implementation of the SVC engine in the states of compression ratio 14 and 8.

Engine displacement	1.598 liters
Number of cylinders	5
Maximum power	168 kW
Maximum torque	305 Nm
Cylinder bore	68 mm
Piston bore	88 mm
Compression ratio	8:1 to 14:1
Max. supercharger boost	2.8 bar

Table 3.1: Technical data for the SVC engine.

Chapter 4

Engine Model

In order to develop control algorithms it was decided that a computer model of the system should be developed. This is because:

- Creating a model of the system is a good way of gathering knowledge about the system. The work with the building of the model provides a structured way of gathering the information that is needed for a successful control design.
- Some control designs like Internal Model Control demands a system model.
- If most of trial and error work can be performed on the computer model, much expensive time on the engine test bench can be saved.

The model shall describe the torque from the input signals air mass flow past throttle, compression ratio and ignition angle, and will be developed with help of physical model building. The engine model described in this chapter is a mean value engine model (MVEM). MVEM means that no variations within cycles are covered and makes the model valid only for time intervals far greater than one engine cycle.

4.1 Model Overview

A schematic overview of the engine with surrounding devices is given in Figure 4.1. When modeling an SI-engine by using physical modeling, it is beneficial to divide the engine in distinct subsystems.

The amount of air entering the engine is governed primary by the main throttle. The mass flow dynamics over the main throttle is described in section 4.2. After the main throttle the air comes to the by-pass throttle. When the supercharger is on the by-pass throttle

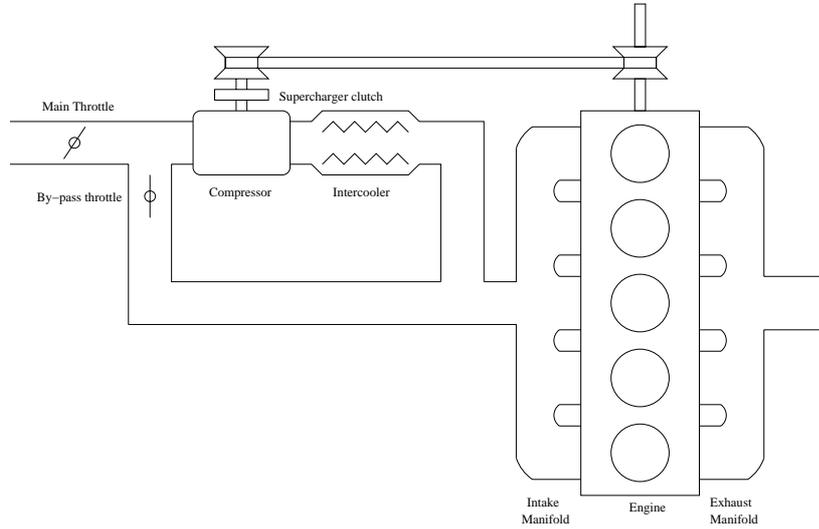


Figure 4.1: Sketch of the SVC engine. The figure illustrates how the engine is divided into subsystems to enable physical modeling of the engine.

is closed and the air enters the compressor and intercooler, before it reaches the intake manifold. When the supercharger is off the by-pass throttle is open, so the air can enter the intake manifold directly. In the intake manifold the fuel is injected. The properties of the intake manifold are discussed in section 4.3. Then the air/fuel mixture goes into the cylinders (denoted as an engine in the figure) where the combustion takes place, and convert the energy in the fuel to power. This will be discussed in section 4.4. It should be stated that the supercharger and subsystems that belong to the supercharger like the intercooler and the by-pass throttle is not modeled in this thesis. This of course makes the model developed only valid when the supercharger is off.

4.2 Main throttle

In gasoline engines a throttle is used to control the air mass flow into the cylinders. The usual throttle model that can be found in for example [2] is quite complex. But after studying step response measurements, it seemed that a rough model of the air flow past throttle, \dot{m}_{at} , could be received as

$$\frac{d\dot{m}_{at}}{dt} = -\frac{1}{\tau_{th}}\dot{m}_{at} + \frac{1}{\tau_{th}} \mathbf{sat} \dot{m}_{atref}$$

where the function *sat* describes a saturation, and defined by

$$\mathbf{sat} \dot{m}_{atref} = \begin{cases} \dot{m}_{max} & \text{if } \dot{m}_{atref} > \dot{m}_{max} \\ \dot{m}_{atref} & \text{if } \dot{m}_{min} \leq \dot{m}_{atref} \leq \dot{m}_{max} \\ \dot{m}_{min} & \text{if } \dot{m}_{atref} < \dot{m}_{min} \end{cases}$$

Where $\tau_{th} = 0.075$, $\dot{m}_{max} = 0.07 \text{ kg/s}$, $\dot{m}_{min} = 0.003 \text{ kg/s}$ and \dot{m}_{atref} is a reference signal for the air flow. Worth noting is that the time constant, τ_{th} , and the saturation boundaries, \dot{m}_{max} and \dot{m}_{min} , actually depends on the engine speed, N . But test shows that $\tau_{th} = 0.075$, $\dot{m}_{max} = 0.07 \text{ kg/s}$ and $\dot{m}_{min} = 0.003 \text{ kg/s}$ give an adequately accurate result. The throttle also contains a time delay of about 0.40 s.

4.3 Intake Manifold

The air transport in the intake manifold depends on

- The throttle and the air flow past it.
- The amount of air that goes into the cylinder.
- The pressure in the intake manifold.

4.3.1 Dynamic Pressure Model

A model of the pressure build-up in the intake manifold is obtained by applying a balance equation expressing the conservation of mass in the manifold [8], see Figure 4.2. The increase or decrease of mass in the intake manifold, \dot{m}_i , is determined by the air flow past the throttle, \dot{m}_{at} , and the air flow into the cylinder, \dot{m}_{ac} . This can be expressed as

$$\frac{dm_i}{dt} = \dot{m}_{at} - \dot{m}_{ac} \quad (4.1)$$

Using the ideal gas law $pV = mRT \Rightarrow m = \frac{pV}{RT}$, on m_i in equation (4.1) can be expressed in terms of the intake manifold pressure, p_i , in following way

$$m_i = \frac{V_i}{RT_i} p_i \quad (4.2)$$

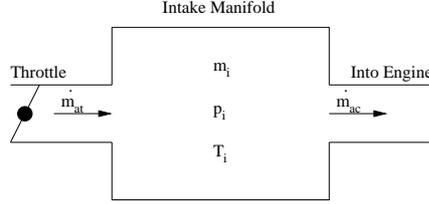


Figure 4.2: A schematic drawing of the intake manifold as a volume that is filled by the throttle flow and emptied by the air flow to the cylinders.

The equation (4.2) is differentiated under the assumption that the intake manifold temperature, T_i , is constant.

$$\frac{dm_i}{dt} = \frac{V_i}{RT_i} \frac{dp_i}{dt} \quad (4.3)$$

The assumption that T_i is constant is not true under transients, but in reality there is only at big steps in \dot{m}_{at} from low engine speeds, N , where this makes any different. Inserting equation (4.3) into (4.1) gives the following differential equation for the intake manifold pressure

$$\frac{dp_i}{dt} = \frac{RT_i}{V_i} (\dot{m}_{at} - \dot{m}_{ac})$$

4.3.2 Engine Mass Flow Model

The air mass flow passing the intake valves, \dot{m}_{ac} , out of the intake manifold into the cylinders depends mainly on engine speed, intake manifold pressure and air temperature.

The volumetric efficiency, η_{vol} , is a measure of the effectiveness of the engine to induct fresh air and is defined as the ratio of actual volume flow rate of air entering the cylinders, $\frac{\dot{m}_{ac}}{\rho_{ai}}$, and the rate at which volume is displaced by the piston, $\frac{V_d N}{2 \times 60}$. The factor 2 in the denominator arises from the fact that the engine only induct fresh air in each cylinder every second revolution. Using the above, the expression for the volumetric efficiency becomes [1]

$$\eta_{vol} = \frac{2\dot{m}_{ac}}{\rho_{ai} V_d N / 60} \quad (4.4)$$

If one again uses the ideal gas law $pV = mRT \Rightarrow \rho = \frac{m}{v} = \frac{p}{RT}$ the air density in the inlet, ρ_{ai} , can be calculated as

$$\rho_{ai} = \frac{p_i}{RT_i} \quad (4.5)$$

If inserting equation (4.5) into (4.4) the airflow into the cylinders, \dot{m}_{ac} , can be expressed as

$$\dot{m}_{ac} = \eta_{vol} \frac{p_i V_d N / 60}{2RT_i} \quad (4.6)$$

Where the volumetric efficiency depends on engine speed, intake manifold pressure and compression ratio, $\eta_{vol}(N, p_i, r_c)$. In order to determine $\eta_{vol}(N, p_i, r_c)$ it is common to measure it and map it over the engine's operating range by running stationary tests. But according to [9] can $\eta_{vol} \cdot p_i$ be described as $s_0 p_i + s_1$ for fixed compressions. If using $\eta_{vol} p_i = s_0 p_i + s_1$ in equation (4.6) the airflow into the cylinders, \dot{m}_{ac} , can be described as

$$\dot{m}_{ac} = (s_0 p_i + s_1) \frac{V_d N / 60}{2RT_i} \quad (4.7)$$

Mass Flow Measurements

In order to determine s_0 and s_1 , experiments were p_i , T_i , N and \dot{m}_{at} was measured stationary for different intake manifold pressures and compression ratios. $\eta_{vol} \cdot p_i$ can be calculated from the measurements as $\eta_{vol} = \frac{\dot{m}_{ac}}{120RT_i}$. Note that for stationary conditions, no mass is stored in the manifold so that the air mass flow into the cylinders equals the air past the throttle and thus $\dot{m}_{ac} = \dot{m}_{at}$. In Figure 4.3 $\eta_{vol} \cdot p_i$ is plotted against p_i for different compression ratios (circles, x-marks, pluses and squares in the figure). From this figure s_0 and s_1 for different compression ratios can be calculated and presented in Table 4.1.

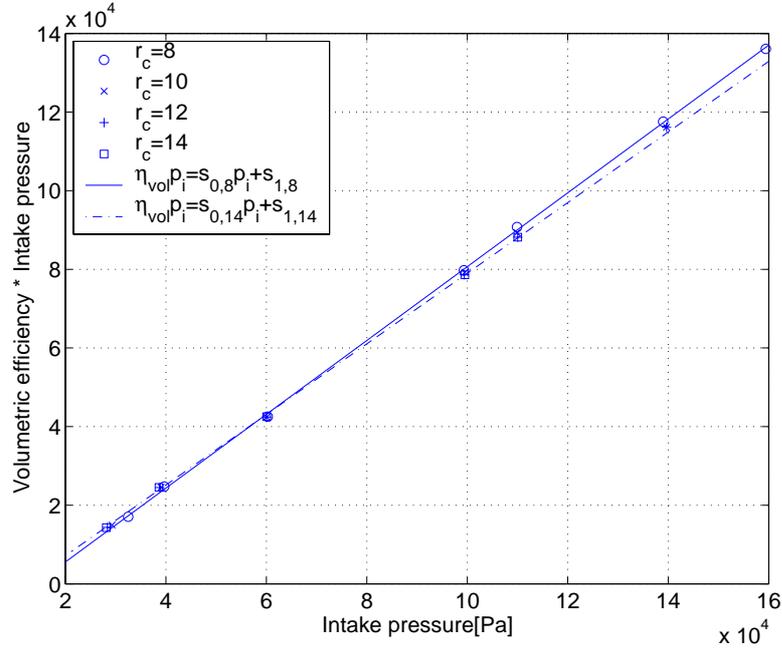


Figure 4.3: Shows $\eta_{vol} \cdot p_i$ plotted against p_i for different compression ratios.

r_c	s_{0,r_c}	s_{1,r_c}
8	0.93899	-13216
9	0.92561	-12336
10	0.91674	-11848
11	0.91342	-11649
12	0.90950	-11495
13	0.89924	-10932
14	0.89822	-10810

Table 4.1: s_0 and s_1 for different compression ratios

From the table it can be seen that both s_0 and s_1 depend on the compression ratio. In Figure 4.3 also $\eta_{vol} \cdot p_i = s_{0,r_c} p_i + s_{1,r_c}$ is plotted for the compressions ratios 8 and 14. If studying the plot it can be seen that in the pressure area which is interesting for this model the maximum difference for $\eta_{vol} \cdot p_i$ for compression ratios 8 to 14 is only about 4%. Therefore if choosing $s_0 = s_{0,11}$ and $s_1 = s_{1,11}$ equation (4.7) will describe the airflow into the cylinder accurate enough for this purpose.

4.3.3 Fuel Injection

In order to inject the correct amount of fuel into the engine, it is necessary to know the theoretical proportions of air and fuel, i.e. there must be enough air to oxidize the fuel perfectly. This ratio is called the stoichiometric air to fuel ratio [1]

$$\left(\frac{A}{F}\right)_s = \left(\frac{m_{ac}}{m_f}\right)_s$$

The petrol used in the laboratory has $\left(\frac{A}{F}\right)_s \approx 15$.

An interesting property is the ratio between the true air to fuel ratio, $\left(\frac{A}{F}\right)$, and $\left(\frac{A}{F}\right)_s$

$$\lambda = \frac{\left(\frac{A}{F}\right)}{\left(\frac{A}{F}\right)_s}$$

It essential to keep λ close to one in order to maintain good catalyst function. It is only possible to have $\lambda = 1$ if $\left(\frac{A}{F}\right) = \left(\frac{A}{F}\right)_s$ and as a consequence we must have the following relation between the mass flow of air into the cylinders, \dot{m}_{ac} , and the mass flow of fuel into the cylinders, \dot{m}_f

$$\dot{m}_f = \frac{1}{15}\dot{m}_{ac}$$

4.4 Cylinders

In the cylinders the combustion process takes place, and convert the energy in the fuel to power. The combustion process is very complicated and a simple of model for the torque is present below. The engine torque depends mainly on:

- How the heat stored in the fuel can be converted into torque.
- Torque lost to pumping when the burned mixture is pumped out.
- Torque lost to friction between the piston and walls and friction losses in the crank configuration.
- Torque lost to accessories such as compressor, servo pumps, electrical generator etc.

If neglect the torque lost to accessories, the net torque, M_{net} , can be expressed as

$$M_{net} = M_c - M_f - M_p$$

Where M_c is the torque delivered from combustion of the fuel, M_f and M_p is the torque lost to friction and pumping.

4.4.1 Combustion

The torque is measure of an engine's ability to do work, power is the rate at which work is done. It follows that the power, P , is delivered by the engine is the product of torque and angular speed [1]

$$P = 2\pi MN/60 \quad (4.8)$$

Ideally all the heat stored in the fuel may be converted to power, in that case the delivered power is given by taking the product of the heating value of the fuel, $q_{hv} = 44.3 \text{ MJ/kg}$ for isooctane, and the fuel mass flow through the cylinders, \dot{m}_f . However there will be losses during the energy conversion, thus we have to multiply with the fuel conversion efficiency, $\eta_{f,i}$

$$P = \eta_{f,i} q_{hv} \dot{m}_f c \quad (4.9)$$

If equation (4.8) and (4.9) are combined a model for the engine torque is given as

$$M_c = \eta_{f,i} \frac{q_{hv} \dot{m}_f c}{2\pi N/60}$$

If the combustion process is approximated as a Otto cycle or constant volume cycle, the $\eta_{f,i}$ can be calculated as, See [1]

$$\eta_{f,i} = \kappa \left(1 - \frac{1}{r_c^{\gamma-1}} \right)$$

Where κ has been introduced because in the calculations above energy losses due to incomplete combustion, heat transfer from gas to the cylinder walls and timing losses has been neglected. From engine measurements of how the total torque changes when the compression, r_c , changes, the value of κ was determined to 0.74.

4.4.2 Pumping losses

The piston in the cylinder is under a slight vacuum during the intake stroke, when it is drawing air/fuel into the cylinder. The extra energy needed for pulling the piston down is known as the pumping losses.

Mean effective pressure (MEP) is define as

$$MEP = \frac{\text{work produced per cycle}}{\text{volume displaced per cycle}}$$

From this the pump mean effective pressure (pMEP) can be calculated as the ratio of the pumping work per cycle, $(p_e - p_i)V_d$, See [1], where p_e and p_i is the pressure in the exhaust and the intake manifold, and the volume displaced per cycle, V_d .

$$pMEP = p_e - p_i \quad (4.10)$$

The mean effective pressure can also be calculated based on work that the engine produces in a dynamo-meter, See [1].

$$MEP = \frac{2\pi M n_r}{V_d} \quad (4.11)$$

From equations (4.10) and (4.11) the torque that is lost to the pumping, M_p , can be calculated as

$$M_p = \frac{V_d}{2\pi n_r} (p_e - p_i)$$

4.4.3 Friction losses

Friction losses depends mainly on engine speed, N . A common way to model the torque that is lost to friction is as follows, see [1]

$$M_f = AN^2 + BN + C$$

In order to decide the values of the constants A, B and C real engine measurements of the torque for different speeds are compared with the torque from the model for the same speeds. The constants A,B and C are then fitted so torque from the model agreed with the torque from the measurements.

4.4.4 Ignition Angle

With ignition angle, θ_{ign} , it is meant the position in crank angles before TDC where the spark discharge occurs. The ignition angle has a direct influence on the engine efficiency and the generated torque. In order to determine how the torque depends on the ignition angle, experiments were performed running the engine at a large number of different spark advances, compression ratios and torque. See Appendix A for plots. From these experiments a three-dimensional look-up table is implemented. The look up table describes the torque, M_i , that is lost for different ignition angle, compression ratios and torque.

$$M_i = \mathbf{Look-up\ table}(\theta_{ign}, r_c, M_{net})$$

The total output torque then can be calculated as

$$M = M_{net} - M_i$$

To keep down the number of measurements needed, the ignition angle is only mapped around operating point 40–80 Nm and at engine speed 2000 rpm . This means that the ignition angle map is only valid around this working point. For simulations at other operating points the ignition angle map can be disconnected and then the model works as if the optimal ignition angle is used at all times.

4.5 Compression Ratio

Studying step response measurements it seems like the compression ratio, r_c , can be modeled as

$$\frac{dr_c}{dt} = \frac{1}{\tau_{rc}}(-r_c + r_{ref})$$

Where $\tau_{rc} = 0.40$ and r_{ref} is a reference signal for the compression ratio.

4.6 Model Summary

The model is summarized in this section to make it easier to get a complete picture of the model equations. In Figure 4.4 there is a schematic overview of the model, where the arrows indicate the flow of information through the model.

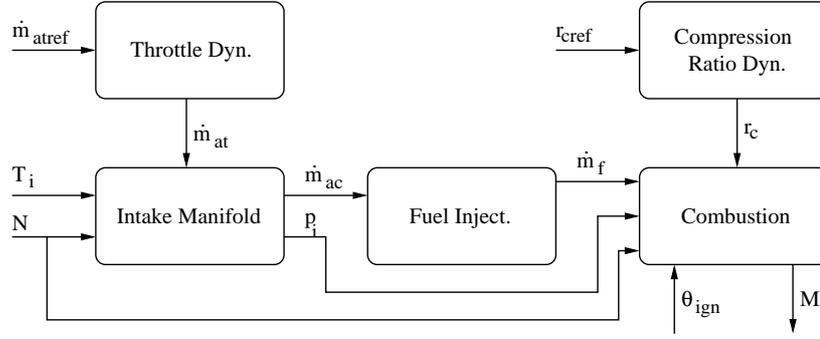


Figure 4.4: Input and output signals from engine subsystems and the interconnection between them.

- Throttle

$$\frac{d\dot{m}_{at}}{dt} = -\frac{1}{\tau_{th}}\dot{m}_{at} + \frac{1}{\tau_{th}} \mathbf{sat} \dot{m}_{atref}$$

where the function *sat* is defined by

$$\mathbf{sat} \dot{m}_{atref} = \begin{cases} \dot{m}_{max} & \text{if } \dot{m}_{atref} > \dot{m}_{max} \\ \dot{m}_{atref} & \text{if } \dot{m}_{min} \leq \dot{m}_{atref} \leq \dot{m}_{max} \\ \dot{m}_{min} & \text{if } \dot{m}_{atref} < \dot{m}_{min} \end{cases}$$

- Intake Manifold

$$\frac{dp_i}{dt} = \frac{RT_i}{V_i}(\dot{m}_{at} - \dot{m}_{ac}) \quad (4.12)$$

Where \dot{m}_{ac} is modeled as

$$\dot{m}_{ac} = (s_0 p_i + s_1) \frac{V_d N / 60}{RT_i n_r} \quad (4.13)$$

- Fuel Injection

$$\dot{m}_f = \frac{\dot{m}_{ac}}{15} \quad (4.14)$$

- Compression Ratio

$$\frac{dr_c}{dt} = \frac{1}{\tau_r}(-r_c + r_{cref}) \quad (4.15)$$

- Combustion

$$M_{net} = M_c - M_p - M_f \quad (4.16)$$

Where M_c is

$$M_c = \frac{1}{2\pi} \kappa \left(1 - \frac{1}{r_c^{\gamma-1}}\right) \frac{q_{hv} \dot{m}_f}{N/60} \quad (4.17)$$

M_p and M_f is

$$M_p = \frac{V_d}{2\pi n_r} (p_e - p_i) \quad (4.18)$$

$$M_f = AN^2 + BN + C \quad (4.19)$$

And then M is calculated as

$$M = M_{net} - M_i \quad (4.20)$$

Where M_i is

$$M_i = \text{Look-up table}(\theta_{ign}, r_c, M_{net}) \quad (4.21)$$

4.7 Model Implementation

The model is implemented in SIMULINK, which is a software package for modeling, simulation and analysis of dynamic systems. SIMULINK is included as a toolbox in MATLAB. Both linear and nonlinear systems can be implemented in continuous and discrete time. In SIMULINK a graphical interface (GUI) is used, so models are built as block diagrams. The model is built up hierarchically, first with larger blocks then with more and more details. The model implementation are shown in Appendix C.

4.8 Model Validation

In the following sections the simulation results are presented and compared with measurements on the real engine in the engine laboratory. The model is validated in terms of both static and dynamic properties. In the first case the measurements are made when all dynamic effects have died out. The engine dynamics are validated by using data from experiments where the throttle and the compression ratio are subject to step changes.

4.8.1 Stationary Validation

The most straightforward way to determine the quality of the model is to compare the modeled torque with the measured torque for certain air mass flows, engine speeds, compression ratios and ignition angles. Another parameter of interest for model validation is the intake pressure, this is because the intake pressure indicates how well the first part of the model works.

The measurements are performed under steady-state conditions, i.e. when all dynamic effects have died out. In order to keep down the number of measurements the test have been performed only at engine speed 2000 *rpm* and at optimal ignition angle. This engine speed has been chosen because is a very common engine speed during driving. The simulation results and measured data are presented in Table 4.2. *Error* stand for the errors made by the model. The errors are simply calculated as

$$\mathbf{Error} = \frac{\mathbf{Simulated\ value} - \mathbf{Measured\ value}}{\mathbf{Measured\ value}}$$

To save space in the tables, the units for the quantities in the tables do not always follow the standard nomenclature in the report. For convenience all units which appear in the tables are listed her, air mass flow [kg/s], pressure [kPa] and torque [Nm].

In Table 4.2 it can be seen that the error for both intake pressure and torque are quit small. But it can be noted that the error in the output torque varies a lot with different air mass flows, this indicate that the model of the combustion is not perfect. Another thing worth noting is that the torque error is about the same for different compression ratios. This indicates that in spite of the problems with the combustion model, its seems to capture how the torque depends on the compression ratio quite well.

Inputs		Measurement		Simulation		Error [%]	
\dot{m}_{at}	r_c	p_i	M	p_i	M	p_i	M
0.0072	8	39.7	10.9	38.3	10.7	-3.52	0.9
0.0072	10	39.7	14.5	38.3	13.5	-3.52	0.9
0.0072	12	39.7	16.3	38.3	15.3	-3.52	-6.0
0.0072	14	39.7	17.1	38.3	17.0	-3.52	-0.5
0.0126	8	60.2	41.9	57.3	39.0	-4.8	-6.9
0.0126	10	60.2	46.9	57.3	43.8	-4.8	-6.6
0.0126	12	60.2	49.8	57.3	46.7	-4.8	-6.2
0.0126	14	60.2	51.8	57.3	46.8	-4.8	-9.6
0.0237	8	99.3	95.9	96.4	97.9	-2.9	2.1
0.0237	10	99.3	105.0	96.4	106.6,9	-2.9	1.5
0.0237	12	99.3	110.4	96.4	113.2	-2.9	2.5

Table 4.2: The table shows steady state validation at $N = 2000$ rpm.

4.8.2 Dynamic Validation

The dynamic qualities of the model can be established from step experiments. Step changes in air mass flow e.g. throttle angle and compression ratio provide a straightforward method to get a good picture of the dynamics.

The step response for an air mass flow step are given in Figure 4.5 through 4.8, and the response for a compression step are given in Figure 4.9 through 4.12. All these plots show experiments at 2000 rpm.

According to these figures the model seems to capture the dynamic characteristics well. Figure 4.6 shows a overshoot not captured by the model, but this is not important for the model. Of higher importance is the intake pressure and output torque that are shown in Figures 4.7 and 4.8. It is worth to note that the overshoot in has little effect on the dynamic response of the intake pressure and output torque, and these are well described by the model.

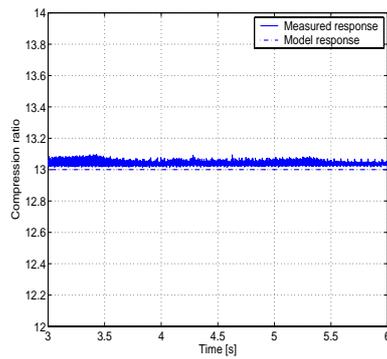


Figure 4.5: The compression ratio response for an air mass flow step at 2000 rpm.

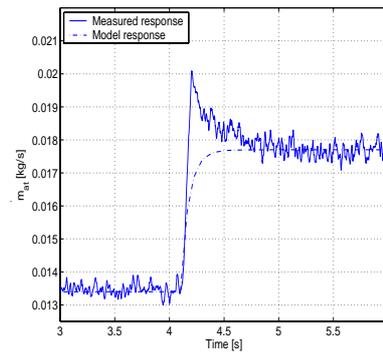


Figure 4.6: The air mass flow response for an air mass flow step at 2000 rpm.

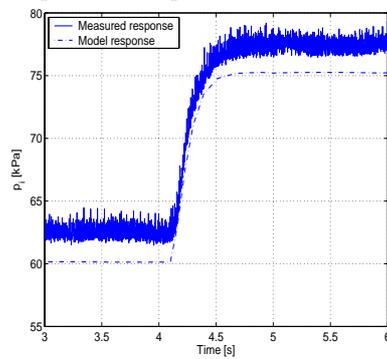


Figure 4.7: The intake pressure response for an air mass flow step at 2000 rpm.

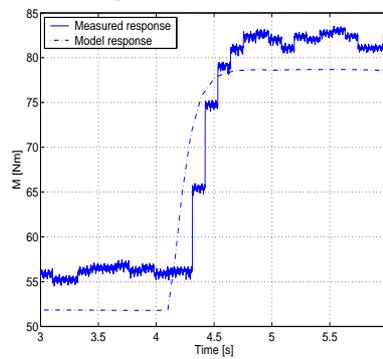


Figure 4.8: The torque response for an air mass flow step at 2000 rpm.

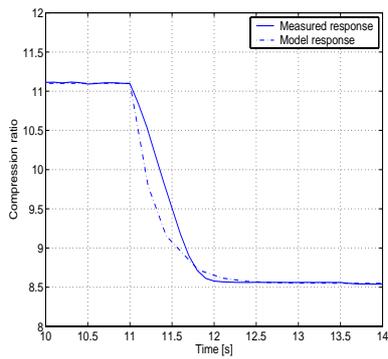


Figure 4.9: The compression ratio response for a compression ratio step at 2000 *rpm*.

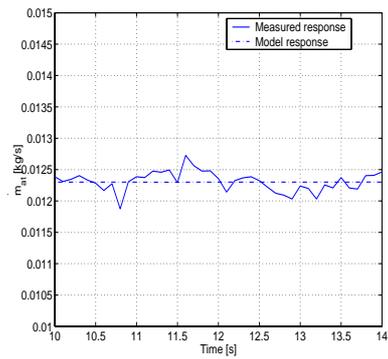


Figure 4.10: The air mass flow response for a compression ratio step at 2000 *rpm*.

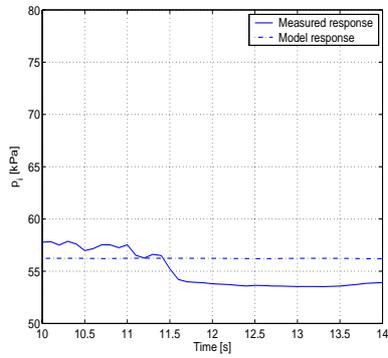


Figure 4.11: The intake pressure response for a compression ratio step at 2000 *rpm*.

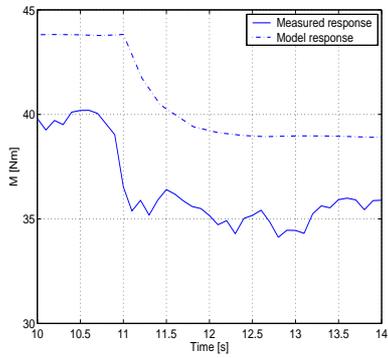


Figure 4.12: The torque response for a compression ratio step at 2000 *rpm*.

Chapter 5

Preliminary Study

In order to decide the main structure of the controller it is important to gain as much knowledge as possible about the system that is to be controlled. The aim of this chapter is to gathering some of the information about the system that emerges during the model building. Two questions that are particular important to answer is:

- Which signals can be used as output signals from the controller?
- Which signals can be used as input signals to the controller?

There can also be very good to have a knowledge of important transfer functions such as the transfer function from air mass flow to output torque.

5.1 Controller Signals

Some of the signals that are interesting for torque control purposes is discussed bellow.

5.1.1 Air-Flow Past Throttle

Air flow past throttle (or actually throttle angle) is the signal which the driver normally uses to control the torque in a car and therefore a natural chose as output signal from the controller. The time constant for the air flow past throttle to torque is around 0.50 s.

5.1.2 Ignition Angle

With the ignition angle its possible to control the torque much faster than with the air mass flow. The time constant for ignition angle

to torque is in order of one engine cycle, and therefore instantaneous in our point of view. An alternative would be to use this signal as a complementary output signal. But drawbacks with this signal is however that it have a direct influence on the efficiency of the engine, which means that changing it from its optimal value can give the engine a poor fuel economy. Another thing that makes the ignition angle unsuitable for this purpose is that changing it from its optimal value can make the engine start knocking, and knocking is extremely dangerous for an engine. In addition to this the ignition angle can not be controlled with the computer system used for controller implementation. For these reasons the idea with the ignition angel as a complementary have been rejected.

5.1.3 Compression Ratio

The compression ratio is controlled by the engine's control system and can in our case be seen as a measurable disturbance. An alternative can be to feed forward the compression ratio to the controller. But since this will result in a more complex controller structure this alternative has not been tested in this thesis.

In the previous chapter the time constant for compression ratio to torque was derived to 0.40 s.

5.1.4 Torque

The most natural choice when designing a controller for the torque is of course to feed back the engines output torque, and let the error between a set point torque and the output torque govern the controller. But on a production engine the output torque is not measured. Note that in the engine lab there is possible to measure the torque. This means that to be able to use a feedback structure according to Figure 2.4 in Chapter 2, an observer that can estimate the torque from measurable signals have to be developed.

5.1.5 Summary

According to this investigation a control structure with the input signals reference torque and estimated torque, and the output signal air mass flow, would be the best choice.

The problem with only being able to use the air flow signal as output signal is that this signal is slower then the disturbance. This means that the effects of the disturbance never can be completely gone, but there is reason to believe that this control structure can reduce the effect of the disturbance.

5.2 Transfer Function

When developing a control algorithm it is good and sometimes necessary to know how the transfer function for different signals looks like. The most important transfer function in this thesis is the air mass flow to torque.

If the throttle saturation, throttle time delay and ignition angle is neglected in the model in Chapter 4, a model for fixed compression ratios can be described as

$$\begin{aligned}
 \frac{d\dot{m}_{at}}{dt} &= -\frac{1}{\tau_{th}}\dot{m}_{at} + \frac{1}{\tau_{th}}\dot{m}_{atref} \\
 \frac{dp_i}{dt} &= \frac{RT_i}{V_i}\dot{m}_{at} - \frac{V_d N/60s_0}{2V_i}p_i - \underbrace{\frac{V_d N/60s_1}{2V_i}}_{\alpha} \\
 M &= \left(\frac{\kappa q_{hv} V_d s_0}{2\pi 15 RT_i 2} \left(1 - \frac{1}{r_c^{(\gamma-1)}} \right) + \frac{V_d}{2\pi n_r} \right) p_i + \frac{\kappa q_{hv} V_d s_1}{2\pi 15 RT_i n_r} \left(1 - \frac{1}{r_c^{(\gamma-1)}} \right) \\
 &\quad - \frac{V_d p_e}{2\pi n_r} - AN^2 - BN - C
 \end{aligned}$$

This is still a nonlinear system, and can not be described with a transfer function. But by neglecting α in the equations above a linear system that still capture most of the dynamics is received as

$$\begin{aligned}
 \frac{d\dot{m}_{at}}{dt} &= -\frac{1}{\tau_{th}}\dot{m}_{at} + \frac{1}{\tau_{th}}\dot{m}_{atref} \\
 \frac{dp_i}{dt} &= \frac{RT_i}{V_i}\dot{m}_{at} - \frac{V_d N/60s_0}{V_i n_r}p_i \\
 M &= \left(\frac{\kappa q_{hv} V_d s_0}{2\pi 15 RT_i 2} \left(1 - \frac{1}{r_c^{(\gamma-1)}} \right) + \frac{V_d}{2\pi 2} \right) p_i + \underbrace{\frac{\kappa q_{hv} V_d s_1}{2\pi 15 RT_i 2} \left(1 - \frac{1}{r_c^{(\gamma-1)}} \right)}_{\beta} \\
 &\quad - \underbrace{\frac{V_d p_e}{2\pi 2}}_{\delta} - \underbrace{AN^2 - BN - C}_{\psi}
 \end{aligned}$$

β , δ and ψ are only static displacements, therefore they can be neglected. The transfer function from air mass flow to torque can now be derived as

$$G(s) = \frac{\frac{RT_i}{V_i \tau_{th}} \left(\frac{\kappa q_{hv} V_d s_0}{2\pi 15 RT_i 2} \left(1 - \frac{1}{r_c^{\gamma-1}} \right) + \frac{V_d}{2\pi 2} \right)}{\left(s + \frac{1}{\tau_{th}} \right) \left(s + \frac{V_d N/60}{V_i 2} \right)} \quad (5.1)$$

Equation (5.1) clearly shows that one of the poles depends on the engine speed. See also Figure 5.1. This implies that there would be a good idea to design different controllers for different engine speed, and then change between these as the engine speed changes.

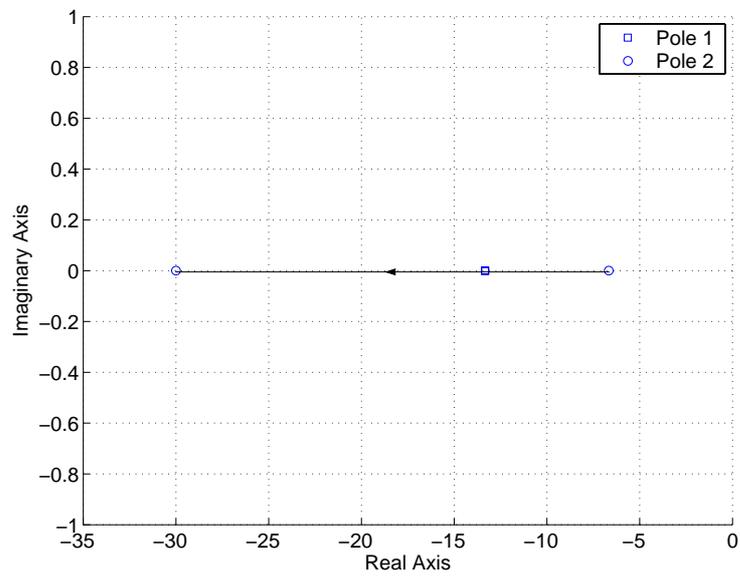


Figure 5.1: A plot that shows how one of the poles moves with different engine speeds, that pole is marked with a circle and moves from right to left along the real axis as the engine speed increases. The engine speed changes from 1000 *rpm* to 4500 *rpm*. The other pole, independent of operating characteristics, is marked with a square.

Chapter 6

Control Algorithm

In this chapter the development and evaluation of the control algorithm is described. The work flow can be summarized as follows:

1. *Design and Configuration.* Determine a controller structure and its parameters. This work is done with help of computer simulations.
2. *Implementation.* When the controller performs well on the simulated system it should be tested on the real system. To be able to do this the selected controller must be implemented on the actual system. The controller is implemented with help of SIMULINK Real-Time Workshop with generates code to a real time platform.
3. *Testing and Evaluation.* The evaluation of the controller is the last step. If the controller does not fulfill the demands the designer must restart at step 1.

The controller developed in this chapter is designed to work around engine speed, 2000 *rpm*. To get a controller that works over the entire engine operating range, the controller should be extended with gain scheduling that change parameters of the controller for different engine speeds. But this has not been done in this thesis.

6.1 Control Demands

The main goal with the control design is of course to develop a controller that increases the driveability of a car that is equipped with a variable compression engine. A good driveability can be obtained if the controller is design so:

- Torque changes due to changes in compression ratio are substantially reduced. In other words system disturbances shall have little influence on the output.
- The controlled variable is good at following the reference signal.

One problem is that these two demands stand in conflict with each other [5], which means that if a controller is designed to follow the reference signal perfectly, its usually not so good at dealing with model errors and system disturbances. Therefore there is important here to find a design that give a good compromise between these two demands.

6.2 Control Structure

It is useful to treat the closed loop properties (i.e. sensitivity and robustness) separately from the servo properties (i.e. the transfer function G_c from r to y). The key to this separation is that the sensitivity function S and complementary sensitivity function T depend only on F_y , and not on \tilde{F}_r . A natural approach is thus to first choose F_y so that S and T get the desired properties. If this does not give acceptable servo properties for closed loop system the filter \tilde{F}_r is modified. This filter can be used to "soften" fast changes in the reference signal. \tilde{F}_r then is of low pass character which means that the peak values of the controller input decrease.

This results in the controller structure in Figure 6.1. Note that during the design of the controller the torque signal is treated as a measurable signal, and later it is replaced by the estimated torque.

Input signal is the error between engine torque and the reference torque. Where the driver controls the reference torque. Output signal is the air flow past throttle. Worth noting is that this control structure involves a different way of control the torque. Normally the driver adjusts the torque with the air flow (or actually the throttle angle). But with this method the driver controls the torque with a torque reference signal.

6.3 Control Design

In modern control systems the controllers are almost exclusively digitally implemented in computers, signal processors or dedicated hardware. This means that the controller will operate in discrete time, while the controlled physical system naturally is described in continuous time. With today's fast computers the sampling in the controller can be chosen very fast, compare to the time constant of the controlled system. Then the controller approximately can be considered as a time

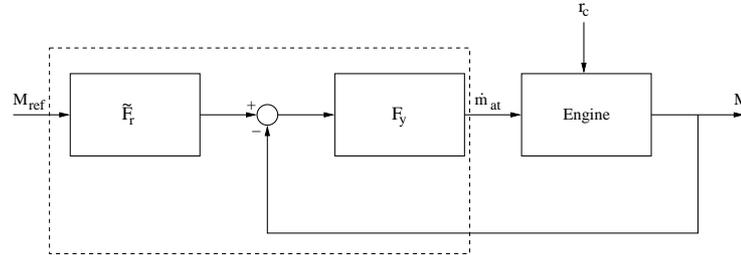


Figure 6.1: Showing the control structure used. The box surrounds the blocks that belong to the controller. The figure also shows important signals.

continuous system. The design can be carried out in continuous time, and the discrete time theory is needed only in the final implementation phase, when the controller is transformed to discrete time [5].

The real time system that has been used for the implementation allows continuous time blocks, therefore all the design work will be performed in continuous time.

6.3.1 How to Choose the Design Method?

Literature like [4], [5] and [6] suggest a number of different design methods, and an important question is how to choose the "best" method for a given problem. The basic principle, as in all engineering work, is "to try simple things first". If a simple PID-controller gives satisfactory behavior, there is no need to bring in for example a H_∞ -machinery.

In this thesis a design method called Internal Model Control, IMC, will be used. A big advantage with the IMC method is that it provides a way of both configuring a controller and determining the most of its parameters and leaving only one parameter left to determined. This means that a controller designed with IMC is much easier to tune in then a PID where three parameters needs to be determined.

6.3.2 Internal Model Control (IMC)

The basic idea in IMC is to only feed back the new information. Consider control of a stable system with model G . The model may very well differ from the true system G_0 . If there were no model errors and no disturbances, there would be no reason for feedback. To pinpoint this, it is natural to focus the feedback on the information in the output that is new, i.e. the part that originates from disturbances, measurement errors, and model errors.

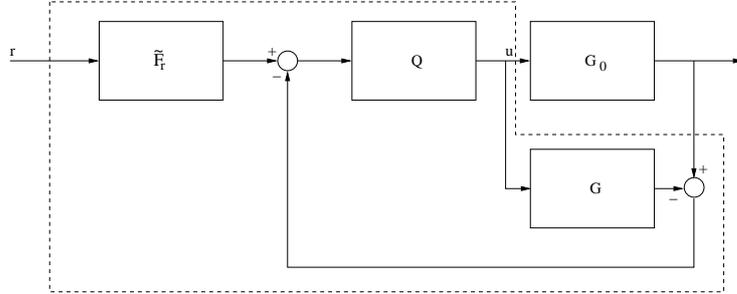


Figure 6.2: Feedback from model error.

$$y - Gu$$

This new information is fed back to the control input by a transfer function Q .

$$u = -Q(y - Gu)$$

See Figure 6.2. Including the link from the reference signal we obtain

$$u = -Q(y - Gu) + Q\tilde{F}_r r \quad (6.1)$$

Expression for Some Important Transfer Functions

Bases on 6.1 the essential transfer functions that describe the closed loop system. The transfer function from y to u is obtained as

$$u = -\frac{Q}{1 - QG}y \text{ i.e. } F_y = \frac{Q}{1 - QG} \quad (6.2)$$

The closed loop system in the nominal case $G_0 = G$ is

$$G_c = GQ\tilde{F}_r$$

The complementary sensitivity function is

$$T = GQ$$

The sensitivity function

$$S = 1 - GQ$$

Design Based on IMC

How to choose Q? The ideal choice $Q = G^{-1}$, which would make $S = 0$ and $G_c = 1$ is not possible since it corresponds to $F_y = \infty$, but is still important guidance. What makes this choice impossible, and how can Q be modified to a possible and good choice:

1. G has more poles than zeros, and the inverse cannot be physically realized. Use $Q(s) = \frac{1}{(\lambda s + 1)^n} G^{-1}(s)$ with n chosen so that $Q(s)$ can be realized (numerator \leq denominator). λ is a design parameter that can be adjusted to desired bandwidth of the closed loop system.
2. G has instable zero (it is non-minimum phase) which would be canceled when GG^{-1} is formed. This would give an unstable closed loop system. There are two possibilities if $G(s)$ has the factor $(-\beta s + 1)$ in the numerator.
 - Ignore $(-\beta s + 1)$ when Q is formed.
 - Replace $(-\beta s + 1)$ with $(\beta s + 1)$ when Q is formed. This only gives a phase error and no amplitude error in the model.

In both cases the original G is used when the controller is formed according to 6.2.

3. G has a time-delay, i.e. the factor $e^{-s\tau}$. There are two possibilities.
 - (a) Ignore $e^{-s\tau}$ when Q is formed, but not in 6.2.
 - (b) Approximate $e^{-s\tau}$ with $\frac{1-s\tau/2}{1+s\tau/2}$ and use step 2. If necessary use the same approximation of $e^{-s\tau}$ in equation 6.2.

All the information and design rules for IMC can be found in [5].

6.3.3 Controller Calculations

In the previous section it was clear that in order to calculate a controller with IMC an internal model, G , in transfer form is needed. In section 5.2 the model in chapter 4 is approximated with a transfer function as

$$G(s) = \frac{\frac{RT_i}{V_i \tau_{th}} \left(\frac{\kappa q_{hv} V_d S_0}{2\pi 15 RT_i n_r} \left(1 - \frac{1}{r_c^{\gamma-1}} \right) + \frac{V_d}{2\pi n_r} \right)}{\left(s + \frac{1}{\tau_{th}} \right) \left(s + \frac{V_d N / 60}{V_i n_r} \right)} \quad (6.3)$$

With equation 6.3 as internal model, G , an IMC controller can be calculated according to the design rules in section 6.3.2.

Using equation 6.3 and values from Appendix B, the transfer function G can be calculated at point $r_c = 11$ and $N = 2000$ rpm. Where a fix compression ratio can be chosen because changes in compression ratio results only in small dynamic changes in G . Engine speed, 2000 rpm, is the operating point which the controller is designed to work around.

$$G(s) = \frac{9.422 \cdot 10^5}{s^2 + 26.67s + 177.8} \quad (6.4)$$

If the time delay in throttle is taken into account transfer function G looks like

$$G(s) = \frac{9.422 \cdot 10^5}{s^2 + 26.67s + 177.8} e^{-0.4s}$$

Suppose the G can be written as $G(s) = G_1(s)e^{-0.4s}$. Approach 3(a) for handling time delays implies that the time delay is ignore when Q is formed but not in 6.2.

$G(s)$ has 2 more poles then zeroes, therefore according to rule 1 $Q(s)$ shall be chosen as

$$Q(s) = \frac{G(s)^{-1}}{(\lambda s + 1)^2} = \frac{s^2 + 26.67s + 177.8}{9.422 \cdot 10^5 (\lambda s + 1)^2}$$

Using equation 6.2 and compensate for the time delay gives the controller

$$F_y(s) = \frac{Q(s)}{1 - Q(s)G_1(s)e^{-s\tau}}$$

Let $F_y^0 = Q/(1 - QG_1)$ be the controller designed for the system without time delays. Simple manipulations then give the controller

$$F_y = \frac{F_y^0(s)}{1 + (1 - e^{-s\tau})F_y^0(s)G_1(s)}$$

where G_1 is

$$G_1 = \frac{9.422 \cdot 10^5}{s^2 + 26.67s + 177.8}$$

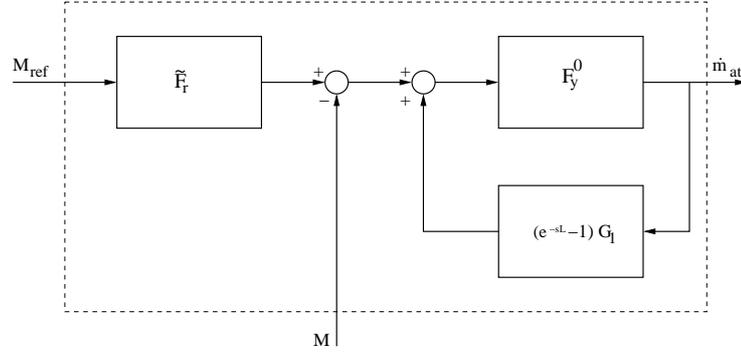


Figure 6.3: A block diagram of the controller.

And F_y^0 can be derived as

$$F_y^0 = \frac{Q(s)}{1 - QG_1} = \frac{s^2 + 26.67s + 177.8}{9.422 \cdot 10^5(\lambda^2 s^2 + 2\lambda s)}$$

Such a controller is called *dead-time compensation according to Otto Smith*. Figure 6.3 shows a block diagram of the controller

λ is a design parameter that determine the bandwidth for the closed system. λ were first approximated using the computer model of the system, and then adjusted further to $\lambda = 0.07$ when the system was implemented in the engine.

The only block left to determine is the prefilter, \tilde{F}_r . Simulations shows that the IMC controller with $\lambda = 0.07$ already gives sufficient servo properties, therefore the prefilter can be chosen as $\tilde{F}_r = 1$.

The complete controller for working point $N = 2000 \text{ rpm}$ can be described as

$$\tilde{F}_r = 1 \quad (6.5)$$

$$F_y = \frac{F_y^0(s)}{1 + (1 - e^{-0.4s})F_y^0(s)G_1(s)} \quad (6.6)$$

Where G_1 and F_y^0 looks like

$$G_1 = \frac{9.422 \cdot 10^5}{s^2 + 26.67s + 177.8} \quad (6.7)$$

$$F_y^0 = \frac{s^2 + 26.67s + 177.8}{4.616 \cdot 10^3 s^2 + 1.319 \cdot 10^5 s} \quad (6.8)$$

6.4 Observer

As the torque is not measurable the alternative is to estimate the torque from measurable signals. The observer is developed from the already existing torque model, by excluding the throttle dynamic and the compression ratio dynamic, since the sensors already measure the true dynamic in this signals. Besides the throttle and compression ratio models also the ignition angle map is excluded. In other words the observer is derived using equations (4.13), (4.14), (4.15), (4.17), (4.18), (4.19) and (4.20) from chapter 4. The observer can be derived as

$$\begin{aligned} \frac{d\hat{p}_i}{dt} &= \frac{RT_i}{V_i} \dot{m}_{at} - \frac{V_d N / 60 s_0}{V_i n_r} \hat{p}_i \\ \widehat{M} &= \left(\frac{\kappa q_{hv} V_d s_0}{2\pi 15 RT_i 2} \left(1 - \frac{1}{r_c^{(\gamma-1)}} \right) + \frac{V_d}{2\pi 2} \right) \hat{p}_i + \frac{\kappa q_{hv} V_d s_1}{2\pi 15 RT_i 2} \left(1 - \frac{1}{r_c^{(\gamma-1)}} \right) \\ &\quad - \frac{V_d p_e}{2\pi 2} - AN^2 - BN - C \end{aligned}$$

The observer estimates the intake pressure, \hat{p}_i , and the torque, \widehat{M} , from the measurable signals air mass flow, \dot{m}_{at} , intake manifold temperature, T_i , compression ratio, r_c , and engine speed, N . It is worth to note that the intake pressure actually is a measurable signal, but since the measured intake pressure is a very noisy signal, the observer instead estimates the intake pressure from the air mass flow signal. The rest of the parameters are constants and can be found in Appendix B.

Figure 6.4 shows how the observer estimates the torque and how the estimated torque is fed back to the controller. In Appendix C the SIMULINK implementation can be seen.

6.5 Evaluation

The final evaluation of the controller and observer has been done in an engine test cell. In the engine test cell, the engine is connected to a dynamometer. The dynamometer can both brake the engine and drive the engine. The later is useful for simulating driving in downhill conditions.

The controller and observer were implemented with help of SIMULINK Real-Time Workshop which generates code to real time platform. The real time platform communicates with the engine via a measurement system connected to the engine. A big advantage with this Real-Time Workshop is that for example a controller can be implemented as SIMULINK blocks, this makes it very easy to test different controllers.

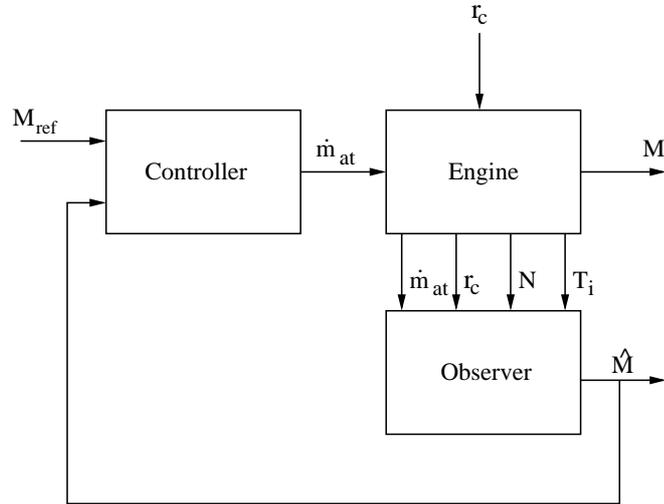


Figure 6.4: The observer estimates the torque from measurable signals and feedback the estimated torque to the controller.

6.5.1 Evaluation of Observer

To evaluate the observer, it has been implemented in the real time system and experiments has been performed were the measured torque is compared with the estimated torque. Figures 6.5 and 6.6 show the results from a compression ratio step respectively a air flow step. Unfortunately the torque were only sampled with 10 Hz during the compression ratio step, and this is the reason for the time displacement in Figure 6.6.

From these figures it can be seen that the observer works very well at air mass flow steps and compression ratio steps from 8 to 14. But at compression ratio steps from 14 to 8 there are some dynamics that is not capture by the observer (this can be seen as a glitch in the measured torque). Possible explanation to this is that the glitch has something to do with the ignition angle or the fuel injection, which are not taken into account by the observer. In order to examine this problem closer more validation measurements where the ignition angle and lambda are measured needs to be done. Limited access to the engine test laboratory have unfortunately made it impossible to examine this future in this thesis.

Figures 6.5 and 6.6 also indicates that there is a stationary error between the measured torque and the estimated torque. This stationary error is hard to compensate for since there are no signal that can be fed backed to the observer.

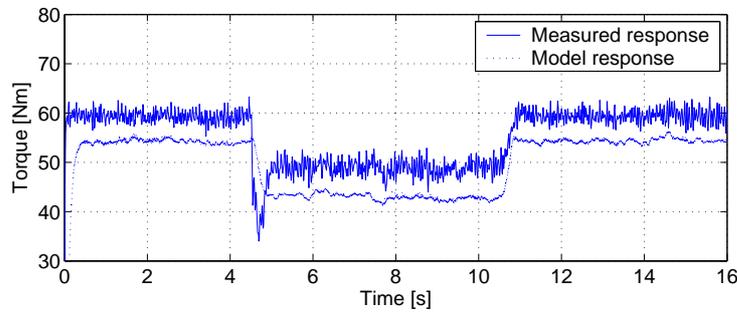


Figure 6.5: Shows measured torque and the estimated torque response for a compression ratio step from 14 to 8 and 8 to 14 at 2000 *rpm*.

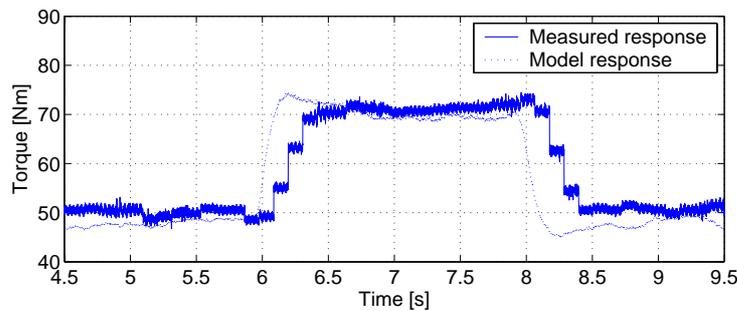


Figure 6.6: Shows measured torque and the estimated torque response for a air mass flow step at 2000 *rpm*.

6.5.2 Evaluation of the Controller

During this final test the complete structure with both the controller and the observer is tested. Two characteristics are studied when evaluating the controller.

- The controllers step response characteristics.
- The ability to handle compression ratio disturbances.

The step response characteristics is evaluated by studying the response time, T_r , the settling time, T_s , and the overshoot, M .

T_r is the time needed for the output to proceed from 0.1 of its final to 0.9 of its final value.

T_s is defined as the smallest time, t , which satisfies $1 - p \leq y(t) \leq 1 + p$ for all $t > T_s$. $y(t)$ is the output and $p = 5\%$. In this case $y(t)$ has the final value 1.

M is simply measurement of the overshoot in %.

All of these definitions can be found in [4]. The ability to handle compression ratio disturbances is evaluated by studying M and T_s . Where T_s now is defined as the time from the start of the disturbance until the output satisfies the $1 - p \leq y(t) \leq 1 + p$ for all t after the start of the disturbance.

Description of the Experiments

Two different experiments were made to validate the controller.

- *Test 1* The controllers ability to handle compression ratio disturbance are tested by changing r_c from 8 to 14 and from 14 to 8 at a constant output torque.
- *Test 2* The step response characteristics are tested with a torque reference step from 60 Nm to 70 Nm and from 70 Nm to 60 Nm at both $r_c = 8$ and $r_c = 14$.

Results

The results from the test are presented in Figures 6.7, 6.8, and in Table 6.1.

Test 1 shows that the controller reduces the effects of compression ratio changes, and within 1 second the torque is at the same level as before the compression step. The glitches, that still can be seen in the torque when the compression ratio changes, are of course a problem. A probable cause is the time delay in the throttle that delays the controller's reaction and makes the controller to slow to completely eliminate the impact of the disturbance. A possible solution to this problem can be to use the compression ratio in the controller as a feed forward. The big advantage with feed forward is that it starts to counteract the disturbance before it reveal it self in the output signal.

Figure 6.7 shows that compression ratio step from 14 to 8 result in a bigger glitch then compression steps from 8 to 14. The reason for this is that the observer misses the the big dip that arises during compression ratio changes 14 to 8, see section 6.5.1. Because the dip not can be seen in the estimated torque the controller have no chance to compensate for it and therefore this dip can not be eliminated by the controller. A possible solution to this is to improve the observer to be able to better estimate the torque at compression steps from 14 to 8.

The results from *Test 2* shows that the controller handles reference steps very well, the response is fast and there is no overshoot. It can be seen in Figure 6.8 that there is a stationary error. This stationary error is inherited from the observer and described in section 6.5.1. A good thing is that the stationary error is at the same level before and after the reference step, this means that a change in the reference signal also results in an equally large change in the torque.

Test	T_r [s]	M [%]	T_s [s]
N=2000			
Torque step, 60 Nm to 70 Nm at $r_c = 8$	0.28	0	0
Torque step, 70 Nm to 60 Nm at $r_c = 8$	0.20	0	0
Torque step, 60 Nm to 70 Nm at $r_c = 14$	0.37	0	0
Torque step, 70 Nm to 60 Nm at $r_c = 14$	0.14	0	0
Compression ratio disturbance 8 to 14	-	14	0.47
Compression ratio disturbance 14 to 8	-	42	0.56

Table 6.1: Results from the experiments.

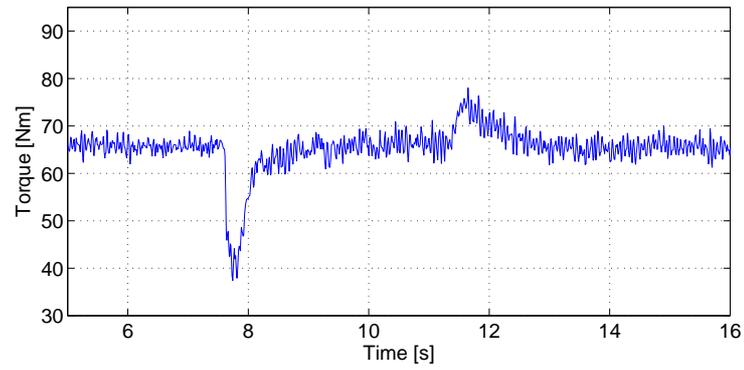


Figure 6.7: Torque measurements of how the controller handles a compression ratio step from 14 to 8 and 8 to 14 at 2000 *rpm*.

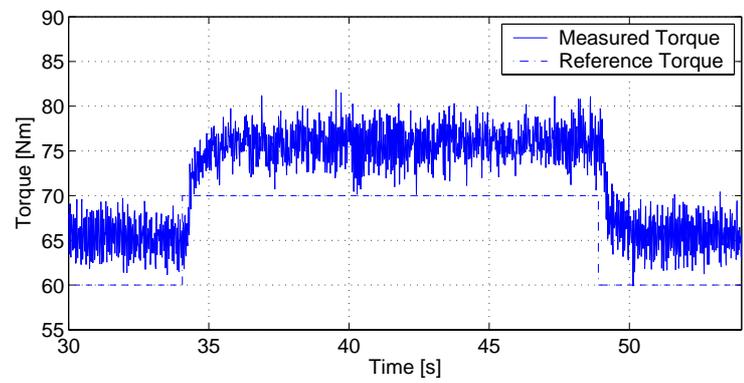


Figure 6.8: Torque measurements of how the controller handles a torque reference step from 60 *Nm* to 70 *Nm* and 60 *Nm* to 70 *Nm* at 2000 *rpm*.

Chapter 7

Conclusions and Future Work

This chapter consists of a summary of the work done during this thesis together with a restatement of the conclusions and a discussion of possible further work.

7.1 Summary

The aim with this thesis project has been to first develop a simple simulation model which can be used for control design, and then develop and test a torque control strategy. The work done during this thesis can roughly be divided into 5 steps.

1. *Gathering knowledge.* The first step, when designing a control strategy was to gain as much knowledge as possible about the system that is to be controlled. The quality of the control structure is greatly dependent on the designer's experience and knowledge of the system. In order to obtain this a lot of documentation about SI-engines have been gathered and studied.
2. *Construct a model of the system.* In order to efficiently control the system its preferable to have a system model that can be used for simulations. An MVEM that includes information about typical disturbances and reference signals was developed with help of physical modeling.
3. *Designing the control system.* The design of the control system starts with deciding a main structure of the controller. When a suitable control structure have been chosen the design of the controllers it self can commence.

4. *Implementation of the controller* When the controller performs well on the simulated system it should be tested on the real system. To be able to do this the controller must be implemented on the actual system. The controller is implemented with help of SIMULINK Real-Time Workshop which generates code to a real time platform.
5. *Evaluation of the controller performance.* The last step was to test and evaluate the controller. This has been done with help of real measurements in a engine test cell.

7.2 Accomplishments and Conclusion

The modeling of the torque was an important part of this thesis and much effort was put into learning how the system behaved. It was found that the system is very complex and has many dependencies. The model that was developed worked well and was as accurate as one can expect given the simplifications made. The model proved to be a great help in the design work.

Because the torque not is a measurable signal in a ordinary engine, an observer that could estimate the torque from measurable signals was developed. The observer was developed from the already existing torque model, by excluding the throttle- and the compression ratio dynamics, since the sensors already measure the "true" dynamic in this signals. The observer was proved to work better at compression steps from 8 to 14 than 14 to 8, were a big dip at the output torque arises. In addition to this the lack of feedback in the observer made so there was a static error at estimated torque.

The controller it self was designed using IMC. A big advantage with the IMC method was that its both designs a controller and determine most of its parameters leaving only one parameter left to determine. Witch makes it very easy to adjust the controller. The controller was proved to reduce the effects from compression ratio changes. But a problem is that the problems with the observer propagates to the controller. Its also reason to believed that a feed forward from the desired compression could make the controller performance even better.

7.3 Future Work

Possible future work with the model and the control algorithm are

- In order to further develop and test new control strategies the model should be extended with models of the supercharger and the intercooler.

- The controller should be extended with a feed forward from the desired compression ratio.
- More work need to be put down to get the observer work better. This is really important since the controller performance is highly dependent on the observer performance.
- The controller should be designed and tested for different engine speeds.
- More experimental validations of the observer and the controller needs to be done.

References

- [1] J. Heywood. *Internal Combustion Engine Fundamentals*. McGraw-Hill, 1988.
- [2] L. Eriksson L. Nielsen. *Vehicular Systems*. Linköping, Sweden, 2001. Course material, Linköpings Universitet, Sweden.
- [3] L. Ljung T. Glad. *Modellbygge och simulering*. Studentlitteratur, Lund, Sweden, 1991. In Swedish.
- [4] L. Ljung T. Glad. *Reglerteknik. Grundläggande teori*. Studentlitteratur, Lund, Sweden, 2nd edition, 1989. In Swedish.
- [5] L. Ljung T. Glad. *Reglerteori. Flervariabla och olinjära metoder*. Studentlitteratur, Lund, Sweden, 1997. In Swedish.
- [6] T McKelvey A Stenman T. Glad, L. Ljung. *Digital styrning kurskompendium*. Linköping, Sweden, 2000. Course material, Linköpings Universitet, Sweden.
- [7] SAAB. *New unique engine concept for high preformance and lower fuel consumption*. <http://media.gm.com>, February 2000.
- [8] S. Sorenson E. Hendricks. *Mean Value Modelling of Spark Ignition Engines*. SAE Paper 900616. SAE, 1990.
- [9] Elbert Hendricks, Alain Chevalier, Michael Jensen, and Spencer C. Sorenson. *Modelling of the Intake Manifold Filling Dynamics*. SAE Paper 960037. SAE, 1996. SAE 960037.

Appendix A

Ignition Angle Measurements

Below plots of how the output torque depends on the ignition angle and compression ratio is shown.

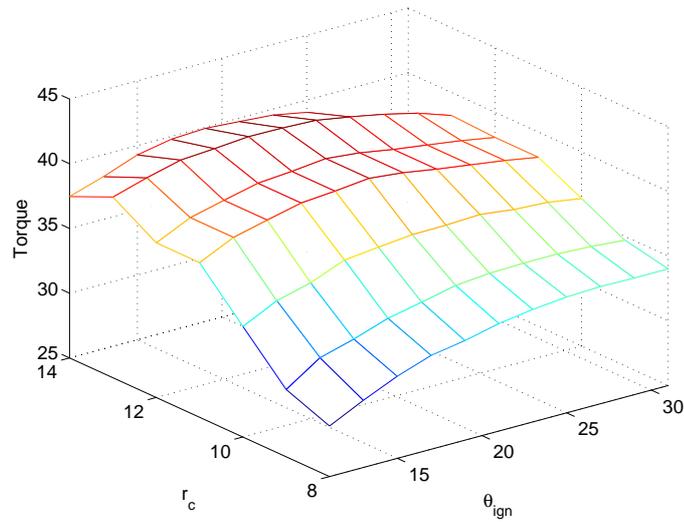


Figure A.1: Measurements of torque as a function of compression ratio and ignition angle, at working point $40 Nm$ and $2000 rpm$

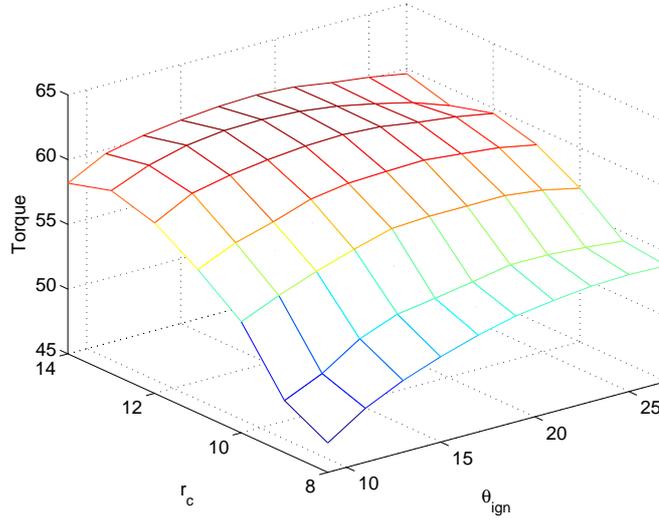


Figure A.2: Measurements of torque as a function of compression ratio and ignition angle, at working point 60 Nm and 2000 rpm

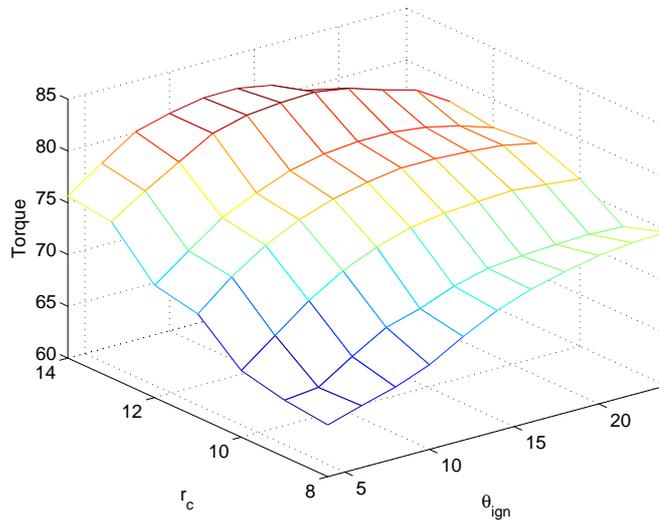


Figure A.3: Measurements of torque as a function of compression ratio and ignition angle, at working point 80 Nm and 2000 rpm

Appendix B

Model Parameters

Symbol	value	unit
A	$1.454 \cdot 10^{-6}$	-
B	$-3.599 \cdot 10^{-3}$	-
C	$1.6262 \cdot 10^1$	-
γ	1.3	-
κ	0.74	-
n_r	2	-
p_e	$1 \cdot 10^5$	Pa
q_{hv}	$44.3 \cdot 10^6$	J/kg
R	287	$J/(kg \cdot K)$
s_0	0.91342	-
s_1	-11649	-
T_i	300	K
τ_{rc}	0.40	-
τ_{th}	0.075	-
V_d	0.0016	m^3
V_i	0.002	m^3

Table B.1: The model parameters

Appendix C

SIMULINK Implementation

Below are the implementation of the engine model and the observer showed. For convenience, only the top level of SIMULINK models is shown.

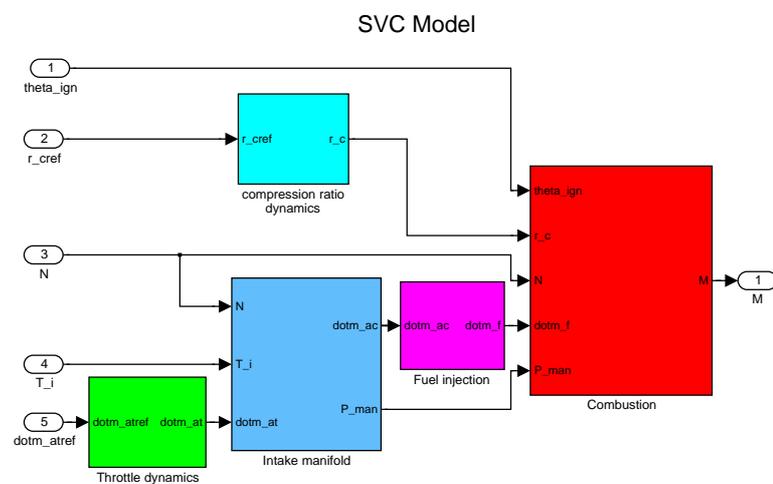


Figure C.1: The figure shows the top level of the SIMULINK model.

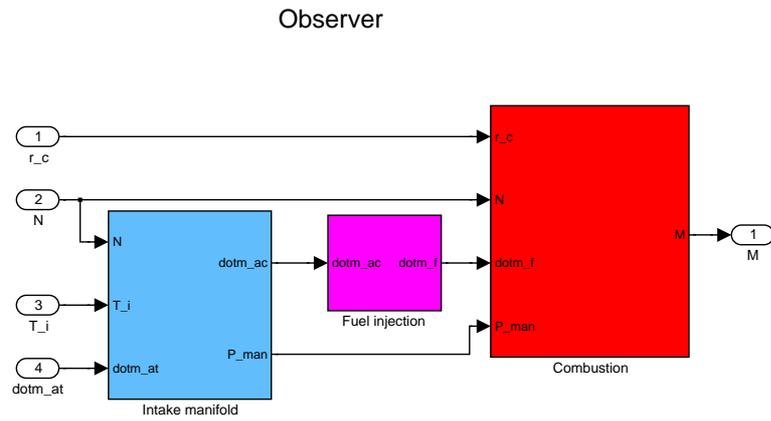


Figure C.2: The figure shows the top level of the observer.