Downshift Speed Control during Shift Phase

An approach using Model Predictive Control

Master's thesis in System, Control and Mechatronics

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Cover: Engine speed visualisation during a gear shift.

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Abstract

Engine control is an important part of developing a vehicle. The technologies around us evolve and the prices are falling. This allows more technologies to be installed in vehicles, more powerful computers become available, sensors get higher accuracy and better opportunities for the vehicle to be connected to the outside world. These factors make it possible to receive more detailed information from both internal sensors and from the surrounding environment. The information obtained can then be calculated with more complex algorithms, due to more processing power.

In this thesis basics on SI-engines and control strategies during gearshift are presented. A linear engine model with both PID and MPC controllers has been created and compared with respect to reduced fuel consumption.

The main achievement of the thesis has been the creation and analysis of two MPC controllers which have been executed in Matlab. The implementation has been done with an input signal from the engine model. Both controllers have shown improvement regarding fuel consumption. The second MPC controller, where a fuel map where acting as a weight, appeared to work best. The simulation time increased due to the additional length of the prediction horizon.

Regarding emissions, CO_2, which can be assumed proportional to the fuel consumption has been reduced.

Keywords: Model Predictive Control, SI-Engine, Break Specific Fuel Consumption.

Acknowledgements

Contents

Abbreviations

A/F	Air and Fuel
BC	Bottom Dead Centre
BMEP	Brake-MEP
BSFC	Brake Specific Fuel Consumption
CO	Carbon-mono-Oxide
CO_2	Carbon Dioxide
EO	Exhaust Valve Opens
FMEP	Friction-MEP
GDI	Gasoline Direct Injection
HC	HydroCarbon
IC	Intake Valve Closes
IGN	Ignition
$IMEP_g$	gross Indicated-MEP
$IMEP_n$	net Indicated-MEP
INJ	Injection
IO	Intake Valve Opens
MAF	Mass Air Flow
MAP	Manifold Absolute Pressure
MBT	Maximum Breaking Torque
MEP	Mean Effective Pressures
MPC	Model Predictive Control
N/A	Naturally Aspirated
NO_x	Various Nitro Oxide
PI	Proportional-Integral
PID	Proportional-Integral-Derivative
PMEP	Pumping-MEP
pV	Pressure Volume
RPM	Revolutions per minute
RPS	Revolutions per second
SI	Spark-Ignition
TC	Top Dead Centre
Vc	Clearance Volume
VCT	Variable Compression Turbo
Vd	Displaced Volume
Vt	Total Cylinder Volume
VVT	Variable Valve Timing

Nomenclature

Variable/parameter	Explanation	Unit
$\lambda = \frac{A}{F}$	Air/Fuel ratio	-
$\dot{m}_f$	Fuel mass flow	g/s
P_b	Break power	kW
η_{losses}	Transmission losses	-
$\eta_{rpm,losses}$	Transmission speed losses	rpm
$Ratio_{gear}$	Gear ratio	-
$Ratio_{final}$	Final gear ratio	-
p_{im}	Intake air pressure	bar
ω	Engine speed	rad/s
N_{wheel}	Wheel speed	rpm
Θ_{ign}	Ignition angle	deg
Θ	Crankshaft angle	deg
D_{im}	Intake air temp	deg
$T_{q,base,req}$	Torque base request	Nm
$T_{q,inst,req}$	Torque instant request	Nm
$T_{q,base}$	Base torque	Nm
$T_{q,inst}$	Torque instant	Nm
$T_{q,rsv}$	Torque reserve	Nm
$T_{q,loss}$	Torque losses	Nm
$T_{q,load}$	Torque load	Nm
u	Manipulated variable	Nm
Δu	Manipulated variable rate	Nm/s
y	Model output variable	rpm
y_{ref}	Reference signal	rpm
J^*	Cost function	-
t_s	Sample time	s
t_f	Simulation time	s
J	Inertia	kg·m^2
N_{ce}	Engine speed	rpm
T_{BMEP}	Break torque	Nm
T_{IMEP}	Indicated torque	Nm
T_{FMEP}	Friction torque	Nm
Pmax	Maximal pressure	bar
P_0	atmospheric pressure	bar

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Introduction

In the recent years, an increasing awareness regarding pollution, greenhouse gases and sustainability together with the introduction of new regulations have pushed vehicle producers towards a new type of research, more oriented to the development of vehicles with high engine efficiency and at the same time a reduced impact on the environment.

EU expects that around 12% of its Carbon Dioxide (CO_2) emission comes from cars. The global demand for cars are still increasing and EU as a power region has therefore legislated emission reduction targets for new vehicles. The CO_2 target for petrol cars the year 2015 correspond to an average fuel consumption of 5.6 l/100km. The CO_2 target for the year 2021 is going to be reduced furthermore, down by 26% compared to 2015 and corresponds to an average petrol consumption of 4.1 l/100km [17]. As the safety regulation also improves, car manufactures has trouble to reduce weight and therefore has to spend time and money to improve the efficiency of the drivetrain.

Along with these new regulations, the processing power and the amount of computers have increased inside vehicles, making it possible to use more sophisticated and complex control algorithms.

Before a control strategy can be applied, inputs to the model have to be collected or created. An example can be the torque signals that cannot be measured and instead are calculated. In general, engine torque is a function made of e.g. intake air pressure, engine speed, air-fuel ratio, ignition angle and intake air temp.

Control strategies play an important role in enhancing an engine's performance, both in terms of power and torque delivered but also in terms of efficiency and fuel consumption. For this reason, one branch of this research is dedicated to the study of new or different control algorithms.

Model Predictive Control (MPC) is being widely studied nowadays, to be imple-

mented in everything from wind turbines to space vehicles. MPC uses a model of the process to predict outputs on the future horizon. At each step, a sequence of control inputs is computed based on the minimisation of a performance index or cost function. Only the first element of the sequence is then applied to the plant and the whole procedure is repeated at the next step. This is called a receding horizon strategy.

MPC mainly differs from ordinary Proportional Integrale Derivative (PID) control because it uses future outputs and inputs to compute the control actions, whereas PID actions are based on past errors. Compared to PID, MPC can exploit additional information given by the future trajectory, if known. This means that the controller is able to foresee changes in the future and therefore the optimisation of the control action is achieved on the future horizon. The possibility of designing a specific reference trajectory will be an advantage for this thesis work, as will be explained later on. At last, MPC allows to include constraints in the calculation of the control sequence to get a better model limitation of the plant and to achieve a physically feasible solution.

1.1   Specification of the issue under investigation

This thesis will focus on engine speed control during shifting phases. In the considered vehicle, the rotation is transmitted to the wheels through an automatic gearbox. The transmission shaft connected to planetary gears is coupled to the engine by a torque converter that has a Lock-Up function, a bypass clutch.

When the transmission is in gear and suitable torque is applied, then the bypass clutch is closed and forces the crankshaft and the input shaft in the transmission to rotate at the same speed. This means that the engine speed is related to the vehicle's speed by the gear ratio. When the gear is shifting, the clutch opens and releases the crankshaft from the input shaft so they stop being synchronised. If the shifting takes place in a short time, the velocity of the vehicle can be considered as constant through the time.

However, since the gear ratio is changed, the engine speed has to change after the shift. This is equivalent to say that a step in the speed reference is applied to the engine. The engine can respond to the step in multiple manners, depending on which controller is implemented. Figure 1.1 illustrates the situation described above. Different responses lead to different fuel consumption. At the present moment, the way the speed adapts to the new reference is not regulated with any considerations regarding fuel consumption.

The idea is to replace the current controller with an MPC that forces the engine to follow a specific trajectory. The trajectory shall be chosen in order to minimise fuel consumption and/or according to other elements that will emerge during the analysis. Brake Specific Fuel Consumption (BSFC) can be used as an indicator of the engine performance. According to the definition, BSFC is a measure of how much fuel is consumed to produce a unity of power.
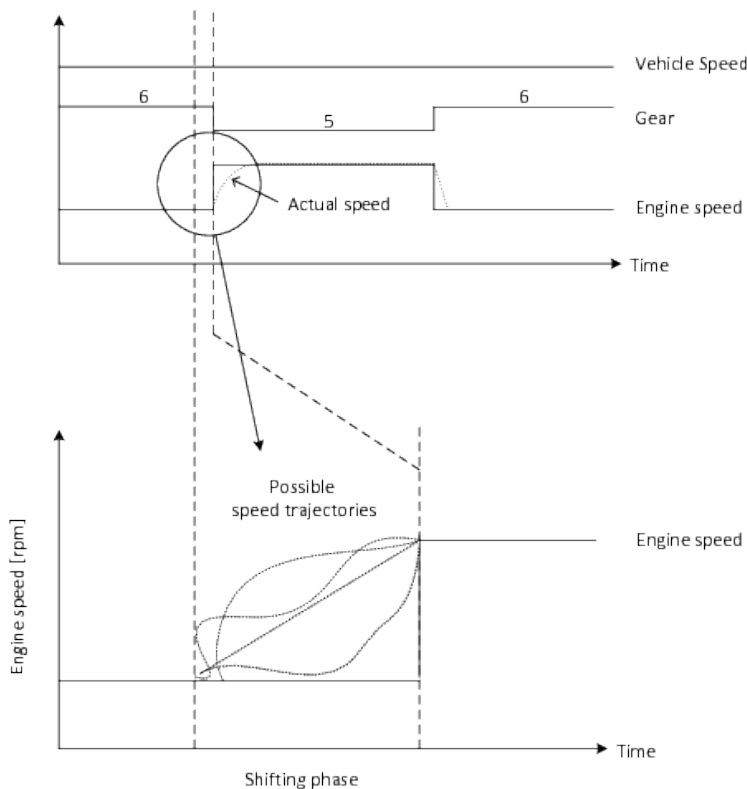
Figure 1.1: An example of a gear shift. The change of gear generates a change of reference speed. The intention for the MPC is to make sure that the engine's speed reaches the new reference speed with an optimal fuel consumption.

1.2 Objectives

The objective of the thesis is to develop a method to control the engine speed in a more optimum way. The purpose is to improve fuel consumption and engine

efficiency. The thesis investigates MPC methods with different cost formulations, tuning and reference trajectories. Three main objectives can be outlined:

- A comprehensive literature study of engine torque modelling and dynamics.
- Design of an MPC controller for controlling engine speed during a gear shift.
- Analysis of different possible reference trajectories and investigate the impact of shift time and fuel consumption.

1.3 Limitations

The work will be carried out on one of the latest drivetrains at Volvo, which contains an 8 speed automatic gearbox connected to a 2.0 litre 4 cylinder single turbocharged engine with a geometrical compression ratio of 10.3:1 [13]. Therefore, adjustments have to be made if this work is to be applied to other powertrain setups, as the parameters considered in these models need to be changed.

This study focuses only on reciprocating Spark-Ignition (SI) engines. Rig test scenarios will be limited, at a constant speed of 70 km/h. The number of evaluated cases may be expanded, depending on the outcomes of the work. Within the simulation physical restrictions concerning engine dynamics will be considered.

The assignment is restricted to, in a simple way, investigate if fuel consumption should be a part of future engine control during a gear shift. This is done by minimising the number of input variables and the complexity of the controller, the MPC-controller is only going to be of implicit character.

Due to lack of time, no physical testing of the results in the test rig will be executed.

1.4 Problem formulation

The aim is to minimise the fuel consumption during a shift phase. The generic problem formulation can be written as

$$
\begin{aligned}
\min_{u} \quad & J^* = g(x(t), u(t)) \\
& s.t. \\
& \dot{x}(t) = f(x(t), u(t)) \\
& x(t) \in [0, x_{max}] \\
& u(t) \in [0, u_{max}(x)] \\
& x(0) = x_0
\end{aligned}
\tag{1.1}
$$

where $x(t)$ is the engine speed measured in rad/s and $u(t)$ is the engine torque demanded on the crankshaft, excluding friction losses. The problem can be solved using several techniques involving approximation of the cost function (J^*) described in Chapter 4 Methods.

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Powertrain and rig test modelling

This chapter starts with a small overview describing important parts and continues with explaining how air, fuel and ignition together are needed for combustion to occur and ends up with engine modelling theory. The content of this chapter is mainly based on a selection of good material found in the literature by Heywood [5], Gupta [6], Eriksson & Nielsen [2] and Ljung & Glad [8].

2.1 Engine overview

To meet customers demand and all new regulations a modern engine is very complex and has a small tolerance for error before fault messages will occur. Therefore an introduction to explain the most imported parts to control an engine is necessary.

Let's follow the airflow through the system by starting with cold air passing through the air filter and end up in the exhaust pipe. This is illustrated in Figure 2.1 and described below.
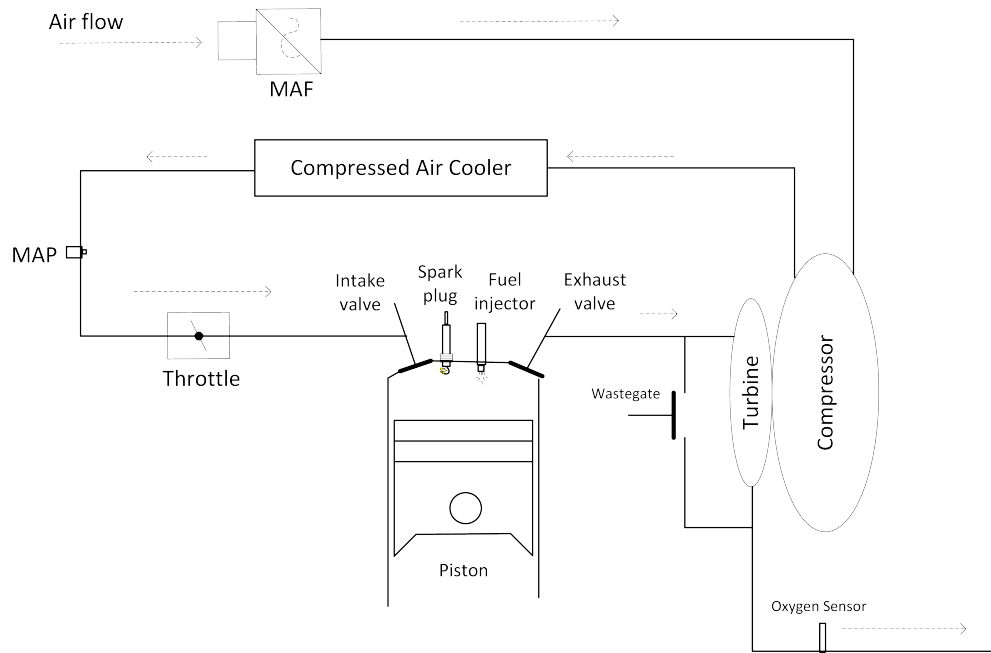
Figure 2.1: Engine complexity, in order for the engine to work properly there is a lot of communication between sensors and actuators involved.

When the air has passed through the air filter, it will reach a Mass Air Flow (MAF) sensor which is needed to inform the engine control system about the air mass, due to the air density fluctuates by temperature and pressure. The power outcome of an engine is limited to the amount of air introduced into the cylinder. With a turbocharger, it is possible to increase the air density before it enters the cylinders. This will result in an increase of maximum power and can be equivalent to a Naturally Aspirated (N/A) engine with a greater displacement volume.

Heat from the conversion from low pressured to high pressured air and by thermal radiation from the turbo turbine is dealt with by sending the air through an air cooler, as is located before the intake manifold. This results in a temperature drop and thereby increasing the air density.

To get a better precision of the Air and Fuel (A/F) ratio and adjusting the ignition timing a Manifold Absolute Pressure (MAP) sensor is used that provides instantaneous information to the engine control system. Most forced induction engines have both MAP and MAF where a N/A engine often employs one or the other.

A throttle valve, mounted on the entrance of the intake manifold, regulates the amount of air available for combustion. By opening or closing the throttle the amount of base torque ($T_{q,Base}$) can be controlled. This part also contains a sensor that informs the engine control system where it is positioned.

Most modern vehicles have two intake valves and two exhaust valves. The idea is to open up as much of the cylinder head as possible to make it easier for the engine to breathe. These valves are opened and closed with camshafts. In general used to improve performance, fuel economy and emissions. Usually one or both of these camshafts have a Variable Valve Timing (VVT) function, which can alternate the timing of the lift event.

The spark plug provides the combustion chamber with a high-voltage discharge which ignites the air and fuel mixture. There are many aspects to consider while designing a spark plug e.g. location, fuel quality and lifespan. A well-suited spark plug will self-clean and maintain a brown light grey colour, which indicates an optimal function during the entire service interval.

During the evolution of SI engines, there have been many solutions about how to insert fuel into the combustion chambers. Modern engines utilise direct injection with a high fuel pressure, which delivers a good dispersion and homogeneity. This lowers the temperature of the cylinder and the piston which permits higher compression ratios and enables earlier ignition timing resulting in higher power output.

In order to control the boost pressure, a common method is to restrict the turbine speed with a wastegate (a flow control valve). It provides the exhaust gases with an alternative passage to the exhaust pipe. Heywood [5] mention that $30\% - 40\%$ of the exhaust gases probably need to be redirected at maximum engine speed and load.

To power the turbine on the turbo, waste energy that otherwise would have been released out through the exhaust pipe is now recycled. To extract the most of the heat energy as possible the turbo needs to be located close to the exhaust manifold.

Modern vehicles have some sort of Variable Compression Turbo (VCT) to adjust the turbine blades for quicker acceleration of the turbine which results in a minimisation of turbo lag.

The last but not the least important sensor is the oxygen sensor, also called lambda sensor, as measures the proportion of oxygen remaining in the exhaust gases. This

sensor signal is important in modern fuel injected vehicles together with data over air volume, temperature entering the cylinders and other variables. The engine control system can then use Look-Up tables to decide how much fuel is needed to run a stoichiometric ratio for obtaining an optimal combustion and for ensuring effective use of the catalytic converter.

2.1.1 Air

The most important factor in order to produce power is air and it responds slowly to torque demand because it is regulated by the throttle. The amount of oxygen trapped in the cylinder, when intake and exhaust valves are closed, is equivalent to the strength of the power stroke can give.

The power created by the back and forth movement of the piston is transmitted to the crankshaft with the assistance of the connecting rod, shown in Figure 2.2. To be aware of the position of the crankshaft it is important for the engine control system to discharge the spark at the adequate moment. The piston moves up and down alternating between Top-Dead-Centre (TC = 0°) and Bottom-Dead-Centre (BC = 180°).

Figure 2.2 also shows a volume change throughout the piston stroke which also can be related to in Figure 2.4, where Vt is the total cylinder volume and can be dividend into Vd and Vc. Vd is the displaced (swept) volume and Vc is the clearance volume.
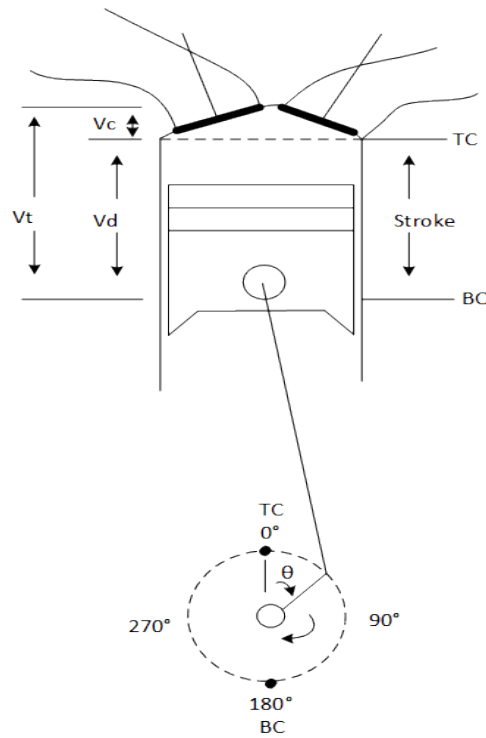
Figure 2.2: Visualising a piston stroke together with important notations in the combustion chamber. Top-Dead-Center (TC) is reached when the piston is in the top of the chamber and Bottom-Dead-Center (BC) is reached when the piston is all the way down. Vt is the total volume and can be divided into Vd, the displacement volume, and Vc, clearance volume, which contains the final volume after the compression. The figure is taken from [5].

The torque and the speed of the rotating crankshaft are then transmitted to the wheel by first a clutch or a torque converter then through a gearbox to better match the road load. An overview can be found in Figure 2.9.
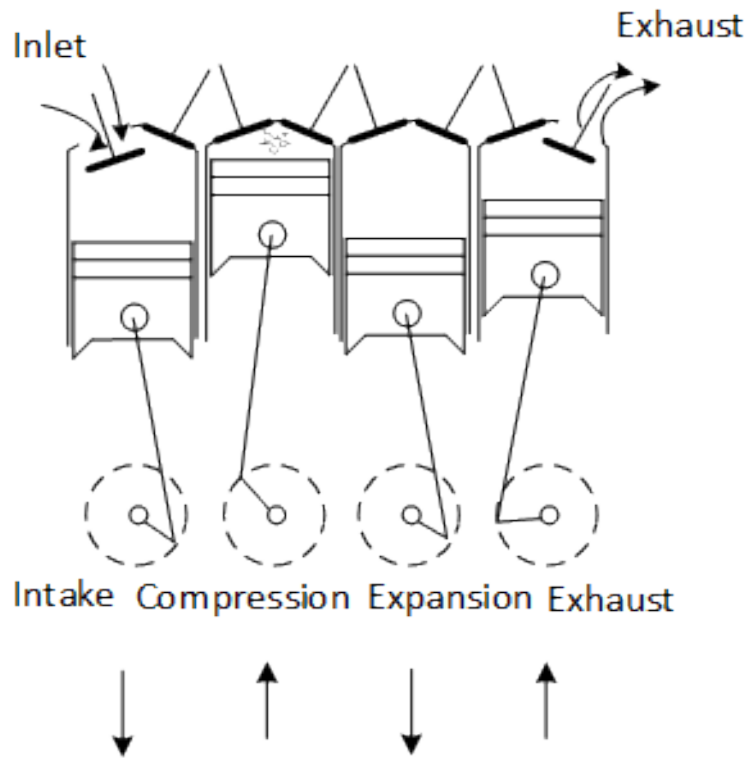
Figure 2.3: An example showing the 4-stroke. The arrows below indicate in what direction the pistons are moving. The figure is taken from [5].

In order to explain 4-stroke cycle Figure 2.3 shows an overview while Figure 2.4 shows a pV-diagram which is a representation of how pressure changes inside the cylinder against a 720° operating cycle. The numbers in the list below indicate the routes taken in Figure 2.4, there is also reference coming from Figure 2.2 which describes volume notations.

The 4-stroke cycle

- **Intake** (1) → (2): The Intake Valve Opens (IO) just before the stroke begins and Intake Valve Closes (IC) slightly after. The volume increases due to that the piston moves from TC towards BC and air is drawn into the cylinder.

- **Compression** (2) → (3): When the piston has changed direction and all valves are closed the volume inside the cylinder decreases and end up as a small fraction of its initial size (see Vc in Figure 2.2). Shortly before the piston reaches TC, fuel gets injected and a spark is discharged (see IGN & INJ in Figure 2.4) which ignites the mixture and the internal pressure rises faster.

- **Expansion** (3) → (4), also called the power stroke: The pressure reaches maximal level (Pmax) and starts pushing the piston down toward BC and forces the crankshaft to rotate. The Exhaust Valve Opens (EO) and the overpressure left in the chamber helps to push out the exhaust gases in the manifold.

- **Exhaust** (4) → (1): The BC has been reached, the piston starts to move towards TC and pushes out the remaining exhaust gases. The overlapping of IO and Exhaust Valve Closed (EC) creates a draft where fresh air gets pulled in due to excess pressure gases moves away.

With Variable Valve Timing (VVT) technology it is possible to change intake- and exhaust-valve strategies which can reduce pumping losses and makes the engine perform better over a wider speed range.
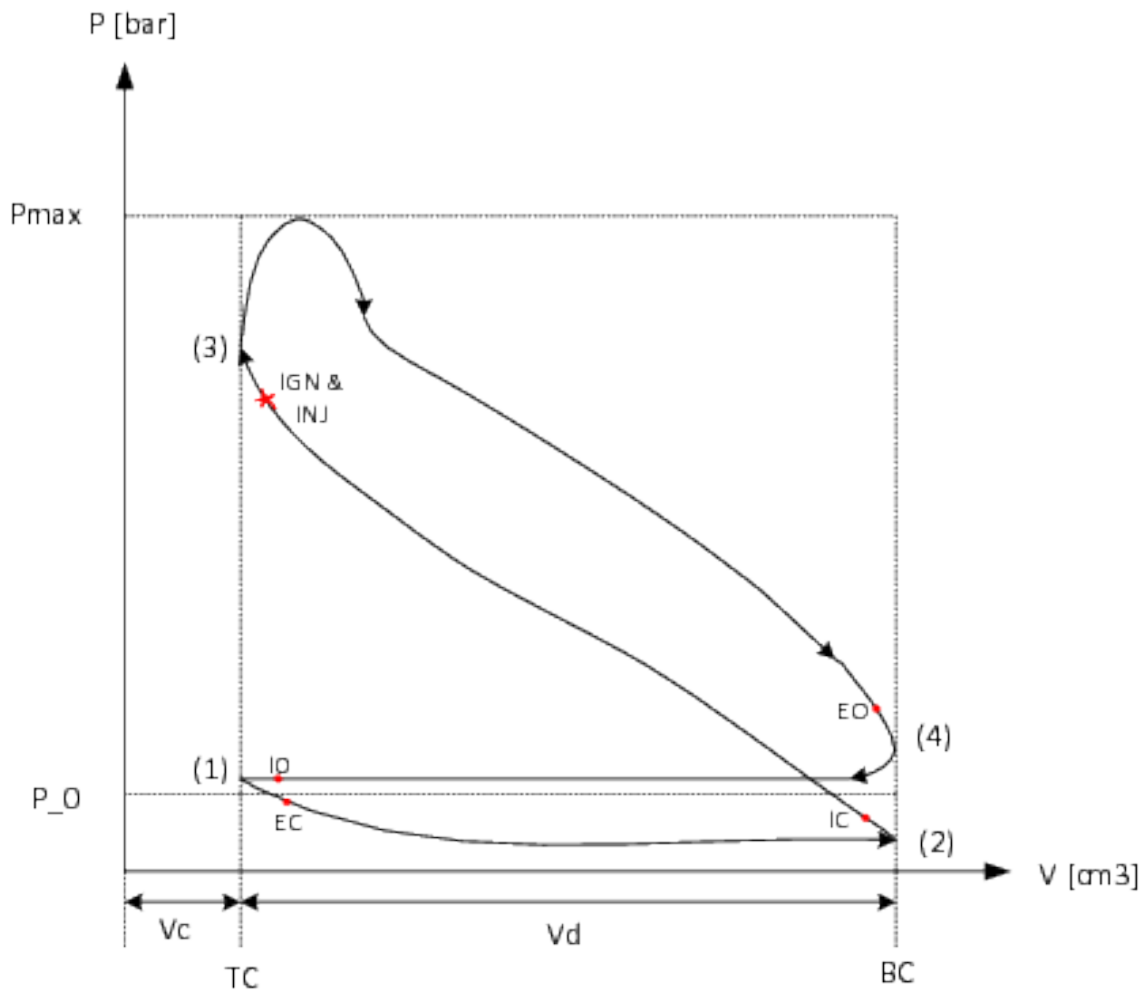
Figure 2.4: A pV-diagram showing a 4-stroke cycle. The parameters are explained like this: IO and IC are intake valve open and closed, EO and EC are exhaust valve open and closed, IGN & INJ is where optimal spark timing occurs and also gives an indication where the fuel injection takes place. P_0 shows the level of atmospheric pressure. The figure is taken from [12].

During the operating cycle, the pressure inside the cylinder changes continuously and use to be analysed in terms of average or Mean Effective Pressures (MEP). MEP can be defined and separated into different parts:

- **net Indicated-MEP** ($IMEP_n$) is a measured or calculated average from the whole operating cycle ($720°$).

- **gross Indicated-MEP** ($IMEP_g$) is measured or calculated from looking at $360°$ of the operating cycle. Only compression and expansion stroke are of

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interest.

- **Brake-MEP** (BMEP) is proportional to engine torque and can be computed by measuring the brake power. It is a practical way to describe engine performance, which allows for a good comparison among different engines.

- **Pumping-MEP** (PMEP) is measured or calculated from looking at 360° of the operating cycle. Only intake and exhaust stroke are of interest and PMEP indicates the cost of move air in and out of the cylinder.

$$\text{PMEP} = \text{IMEP}_g - \text{IMEP}_n$$

- **Friction-MEP** (FMEP), a theoretical pressure loss due to overcome friction.

$$\text{FMEP} = \text{IMEP}_n - \text{BMEP}$$

2.1.2 Ignition

Innovations and control performance have led to direct ignition, which is a popular choice among vehicle manufacturers today. It means that every spark plug has it own high voltage coil and can be controlled by itself. The ability to change the ignition timing makes it possible to immediately lower the output torque from the engine, it is called Instant Torque ($T_{q,inst}$).

Information about engine position comes from the crankshaft- and camshaft-sensor with an accuracy around $\pm 0.01°$ [16], which is processed by the engine control system along with knock- and emission-control in order to discharge at Maximum Breaking Torque (MBT). Engine speed, intake manifold pressure and fuel mixture are some of the variables that impact the optimal spark angle. Figure 2.5 is a general example of how the torque drops in percentage when timing angle varies from its optimal position which gives MBT. The figure also illustrates that if the timing angle is set to far from its ideal position it can result in either engine knocking or misfire, where knocking can ruin the engine and misfire can damage the catalytic converter.
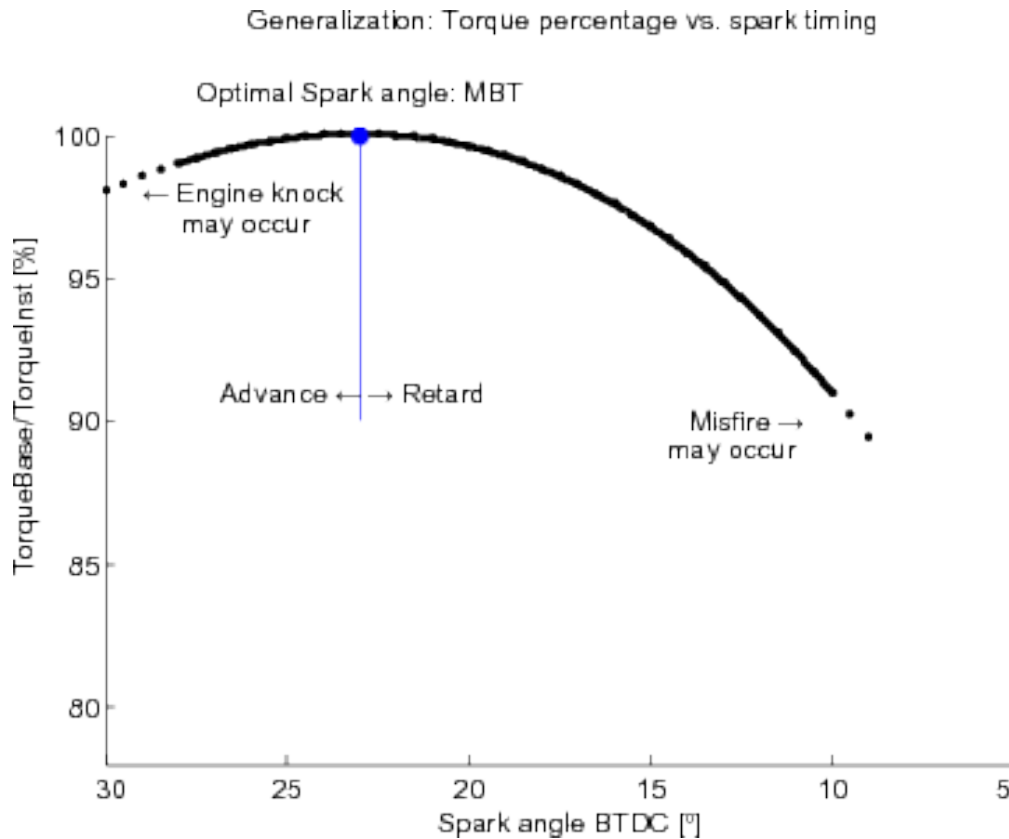
Figure 2.5: Illustration of percentage drop of normalised torque while sweeping ignition timing. The figure is taken from [5].

Running at optimal advance was not a problem until regulations forced the manu-factures to expand its view on emissions, not only to reduce CO_2 but also Carbon mono-Oxide (CO), HydroCarbon (HC) and a variate of Nitrogen Oxide (NO_x). The emission dilemma, to lower fuel consumption the engine should operate at highest efficiency which is reached at MBT and its timing is advanced from TC, can be seen in Figure 2.6. Unfortunately, the engine produces more emission when it is running at MBT. In order to reduce the production of NO_x and HC, the timing has to be slightly retarded compared to MBT. The catalytic converter, an after-treatment system, is inefficient until it reaches a temperature above $250°$. This can be done by reducing the timing compared to MBT so more energy ends up in the exhaust system instead of pushing the piston down.
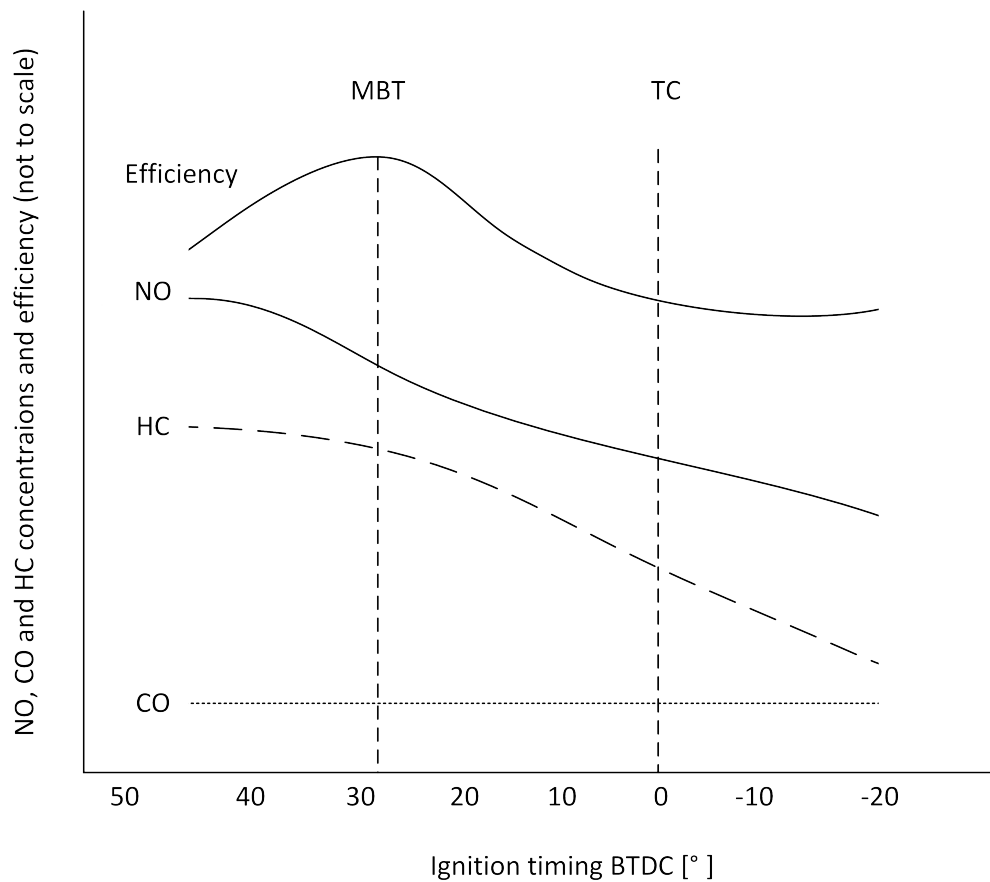
Figure 2.6: Figure showing how ignition timing relates to thermal efficiency and emissions such as Carbon mono-Oxide (CO), HydroCarbon (HC) and Nitrogen Oxide (NO). The curves relate to an engine speed of 3400 rpm. The figure is taken from [11].

2.1.3 Fuel

Air and ignition are primary parameters in torque control. Fuel control is more of a secondary system, because of the amount of injected fuel is decided according to a fuel map pre-programmed in the engine control system. The engine control unit mostly relies on information from MAF- and MAP-sensors to inject fuel. To fine-tune the fuel injection oxygen sensors located in the exhaust give the control system feedback after the combustion.

Fuel injection on cars was introduced back in the 1950's, but the technology has changed a lot since then. The latest addition in fuel delivery is Gasoline Direct Injection (GDI). Where a high-pressure fuel line, called common rail, delivers high-pressure fuel to a solenoid valve that directly inserts the fuel into the combustion chamber during a compression stroke. GDI has both advantages and disadvantages [14]. A few examples are listed below:

Advantages
- Increased fuel economy
- Allows smaller engines to produce more power
- The compression ratios can be higher
- Reduction of cylinder wall temperatures
- Better uniformed A/F-mixture due to smaller droplets
- Lower CO_2 emissions

Disadvantages
- Higher maintenance cost
- Injection time windows can sometimes be in microseconds range
- Higher levels of NO_x and carbon particles (soot), which can be difficult to regulate.

Air and Fuel mixture
At the beginning of engine control, carburettors and distributors got a mechanically fixed fuel mixture at production and became less efficient over time, due to wear and tear but also variations of petrol content regulations. When oxygen sensor (lambda sensor) was discovered it was a major breakthrough when it comes to engine control and universal air quality. This device measures the proportion of oxygen in the exhaust gases. The information is sent to the engine control system to improve anti-pollution or performance tuning. The most common sensor type has been rated in such a way that 1.0 (equivalence ratio) means optimal A/F ratio, another term is the stoichiometric mixture,

$$\lambda_{opt} = \left\{ \frac{A}{F} \right\}_{stoichiometric}.$$

$12 \leqslant A/F \leqslant 18$ is the normal operating range for conventional petrol-driven SI-engines where the desired stoichiometric mixture for optimal combustion is A/F= 14.7, which means for every one gram of fuel needs 14.7 grams of air.

For the engine to run smooth and reliable an optimal or a bit rich fuel mixture is preferable, but as Figure 2.7 shows, a leaner mixture lower both HC and CO

emissions until the mixture becomes too weak and the engine starts to behave rough or misfire which will impact on HC emissions. On the rich side of the scale both CO and HC increases. NO_x give high values around stoichiometric and slightly lean mixture. It is possible to lower NO_x emissions by recirculating exhaust gas into the combustion mixture.
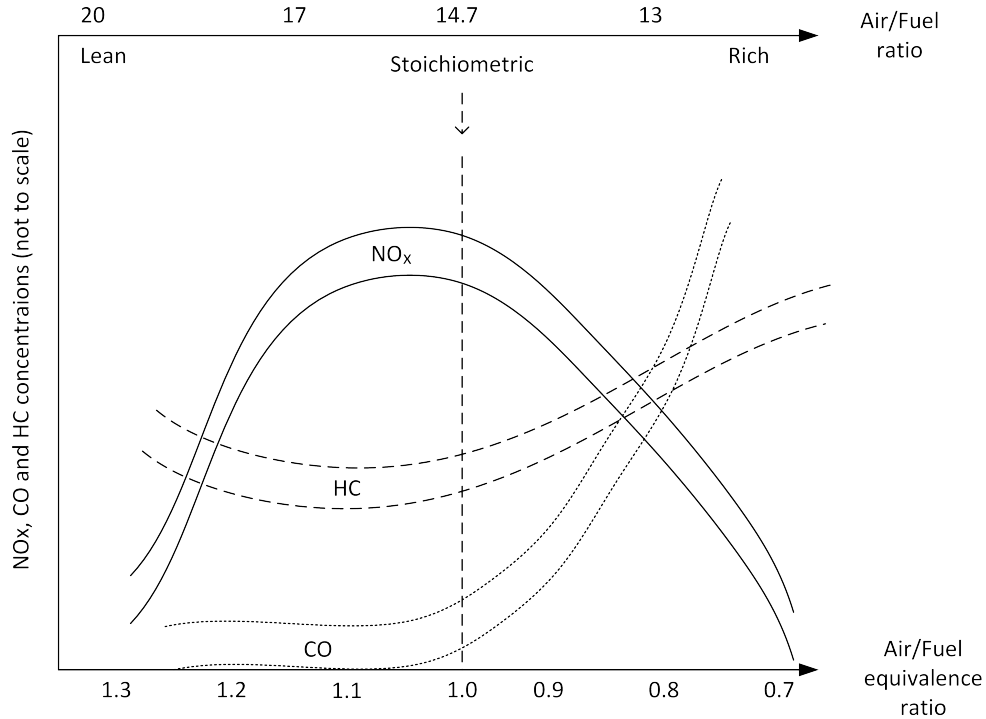
Figure 2.7: Shows how exhaust emission vary with A/F-mixture. The figure is taken from [5].

Fuel consumption

Fuel consumption is usually measured in $l/100km$ but a more common unit when it comes to engine testing is fuel mass flow rate ($\dot{m}_f$). As divided by produced power (P_b) gets the brake specific fuel consumption

$$BSFC = \frac{\dot{m}_f}{P_b}. \tag{2.1}$$

A lower BSFC value indicates that less fuel is consumed for a certain power. One way to decrease BSFC is to reduce engine size because of less heat loss between gas and cylinder wall. BSFC increases at higher speed due to efficiency drops and thermal energy which leads to more friction.
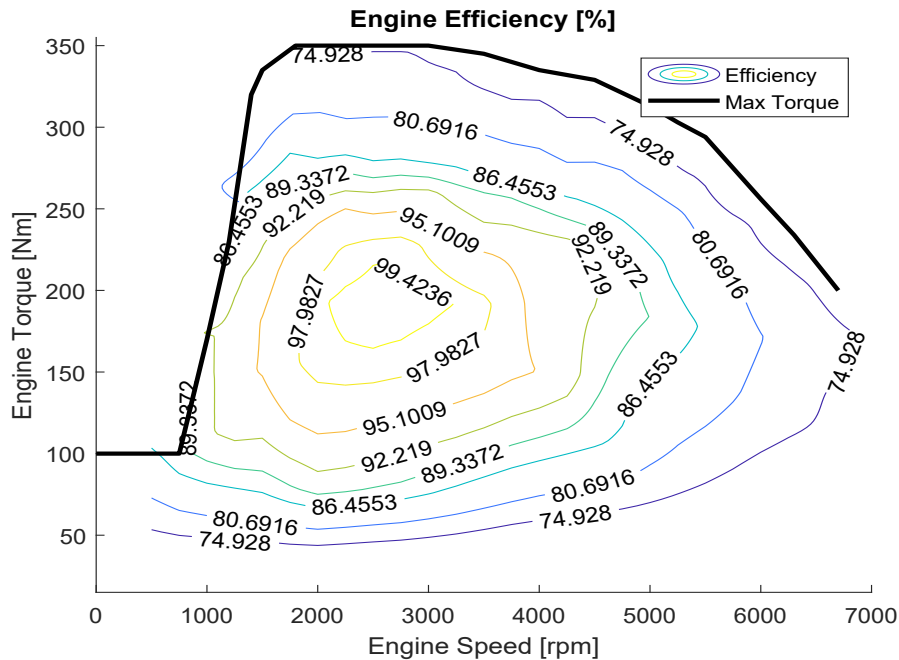
Figure 2.8: Contour plot of the normalised efficiency, a function of engine speed and torque. The black line shows upper torque limit for the engine. Higher numbers in the contour plot symbolise better engine efficiency and less energy go to waste.

The most common way to present BSFC is with help of a contour plot, but in this case it was revealing too much sensitive information. Instead have the real fuel map numbers been converted to display where the engine performs the maximum efficiency. Best numbers can be seen around 2800 rpm with a load of approximately 180 Nm. A fast overview of the Figure 2.8 show which lines are rewarding to cross and which lines to avoid.

2.1.4 Transmission

In this work an 8-speed automatic gearbox has been used. A gearbox transfer power from engine's crankshaft to the drive shafts through a set of gear. Its main purpose is to convert engine speed and torque to improve driving characteristics

at various vehicle velocities and road loads

$$\frac{N_{ce}}{Ratio_{gear} \cdot Ratio_{final}} \cdot \eta_{losses} = N_{wheel},$$

and there will also be losses η_{losses} in the drivetrain coming from e.g. torque converter, worn out clutches and sprocket design. On Volvos homepage [13] it was possible to find information about gear ratios $Ratio_{gear}$ and $Ratio_{final}$ used in their automatic transmissions.

Table 2.1: Gear Ratio for this 8 speed automatic transmission.

1	2	3	4	5	6	7	8	R	Final drive
5.250	3.029	1.950	1.457	1.221	1.000	0.809	0.637	4.015	3.329

Torque converter
Between the engine and the automatic transmission there is a hydrodynamic fluid coupling. Advantages with this set up are e.g. input speed and output speed are decoupled and the torque converter has the capacity to multiply torque by enhancing difference speed on input and output. Because of the inefficiency of this design a locking mechanism has been implemented, when the vehicle has found a suitable gear to compensate torque demand, the Lock-Up function binds engine speed to gearbox input shaft together. The signal that is used contains three stages. When the signal sends number 3 it means the function is locked, if the signal sends the number 1 it is engaging the function and when the signal is sending number 0 it means the Lock-Up function is fully released.

2.2 System representation

Theoretical engine torque, $T_e(p_{im}, \omega, \lambda, \Theta_{ign}, D_m, \cdots)$ where intake air pressure (p_{im}), engine speed (ω), air-fuel-ratio (λ), ignition angle (Θ_{ign}) and intake air temp (D_{im}) are some of the content which needs to be included, although it can be a bit difficult to calculate accurately.

This thesis has instead been for the level of information that is sent between the system control units and can be measured. In order to understand some of the signals used in the vehicle, a short introduction is needed to distinguish the most important torque terms. Torque base ($T_{q,base}$) is used to describe the torque from

the air dynamics, which shows the maximal usable torque at the moment. Whereas torque base request ($T_{q,base,req}$) is the desired value that the engine system control is aiming for. Similar relation can be found related to the ignition timing, where instant torque request ($T_{q,inst,req}$) is desired and instant torque ($T_{q,inst}$) is the actual torque, equivalent to engine output torque (T). $T_{q,inst}$ is limited by the air dynamics and can therefore not be greater than $T_{q,base}$. The difference between these two is called torque reserve $T_{q,rsv}$. An advantage to operate with $T_{q,rsv}$ is to rapidly even out disturbances. Another advantage is the ability to maintain torque level during an increase of $T_{q,base}$, due to the ignition timing movement away from MBT. An disadvantage with $T_{q,rsv}$ is enhanced fuel consumption. Torque losses ($T_{q,loss}$) is a collection of counteracting torques, e.g. friction, which is needed to overcome before the engine can be turned over. To measure the torque load ($T_{q,load}$) a test rig was used in order to create a simulation of e.g. road friction and air resistance.

Moment of inertia
The engine crankshaft dynamics with inertia J is a rotational extension of Newton's 2nd law; $F = ma$, which describes that there will be a change of the force, by changing the weight and/or acceleration. The crankshaft dynamics equation shows the complexity of calculating engine torque.

Inertia comes from the mass of rotational components connected in the engine. There are parts e.g. pistons, crankshaft, balance shaft and even small parts like valves in this internal mixture of masses. External influences can be alternator, compressor, flywheel or flexplate. This means that $T_{q,loss}$ are including a lot of different parts not possible or at least not necessary in the case to separate, also pumping losses are included here.

$$J = \frac{1}{\dot{w}} T_{q,loss}. \tag{2.2}$$

Torque losses can be formulated like this: Produced torque = Load + Losses.

$$T_{q,loss} = T_{q,inst} - T_{q,load}. \tag{2.3}$$

An average value for inertia $J = 0.25 \ [kgm^2]$, but only around 20% of the inertia is coming from powering the engine. Advantages with higher inertia is a method for storing energy to overcome road disturbance, backlashes and other jerks in the drive train. Automatic vehicles have a flexplate instead. As the name indicates this disc can bend a bit but the main inertia contribution in this case comes from the torque converter and the oil pump due to that they whisk around the fluid for the transmission.

2.3 Rig tests and dynamometer torque

There are different ways of performing a rig tests, in this case, the whole drivetrain was included in the test cycle. The purpose of rig testing is to shorten development time and it also provides the ability to increase the complexity of the tests. The vehicle is mounted on dynamometers surrounded by measuring equipment in a sealed room. The closed environment makes it possible to reproduce the tests in same conditions. The wheels were removed and the dynamometers were bolted directly to the driving axes, this made it possible to measure dynamometer torque $(T_{q,load})$.

2.4 Modelling theory

The aim is to use an engine model, as simple as possible, to evaluate if it is sufficient enough to get a proper comparison between different control strategies. The reason why it is interesting with a simple model is that there is a lot of disturbance and error using existing signals, due to many of the signals used for analysing engine behaviour are calculated and created with internal models.

In Figure 2.9 an overview of a drivetrain can be seen, the outcome from the engine is crankshaft torque (T) and crankshaft speed (ω) which are the interesting parameters for this work. Further on in the figure, a clutch is between the engine and the gearbox. In reality, it should be a torque converter but it is now illustrating zero torque losses. The transmission (Gearbox) then provides a speed and torque conversion which through the drive shafts is connected to the test rig where T_{wheel} and ω_{wheel} are measured.
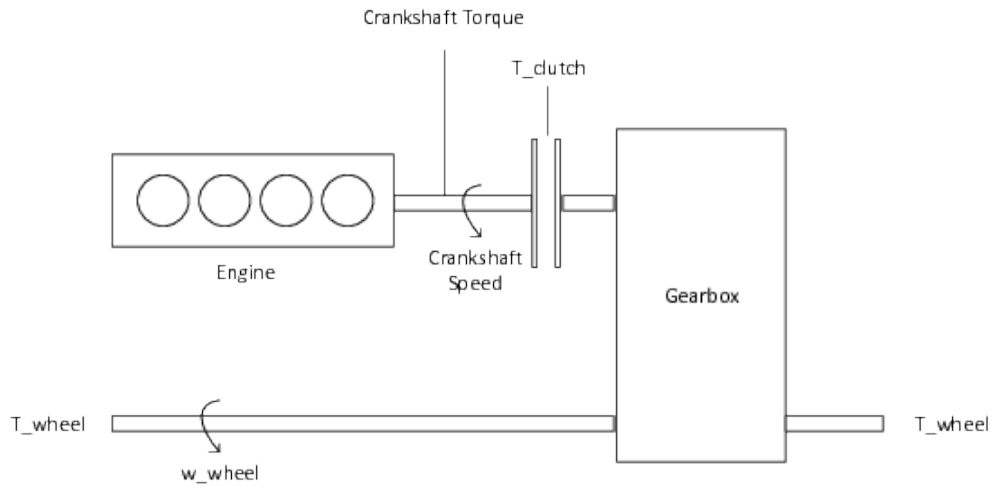
Figure 2.9: Drivetrain overview

This report will use torque as a control signal $u(t) = T_{IMEP}$ and engine speed as a state signal $x(t) = \omega$, the state space representation will be written as:

$$\dot{\mathbf{x}}(t) = \mathbf{A}\mathbf{x}(t) + \mathbf{B}\mathbf{u}(t) \tag{2.4}$$

$$\mathbf{y(t)} = \mathbf{C}\mathbf{x}(t)$$

where **A**,**B** and **C** are determined in Section 4.2 Engine modelling.

3

Control strategies

Control systems have been around for centuries and have evolved from mechanically to pneumatically and now almost all regulators are implemented by means of electronics, through microcomputers and signal processors. The most common controller type in industrial applications is called PID.

Model Predictive Control (MPC) is gaining popularity within the industry, especially within the process industry, because it has been proven to increase output quality or quantity with its superior control compared to techniques that are more conventional. With better control, it is possible to reduce the variance of the output and operate closer to the constraints and thereby improve profits. The content of this chapter is mainly based on a selection of material found in literature by Lennartsson [9], Maciejowski [3], Camacho and Bordons [4], lectures by J A Rossiter [18], Matlabs homepage [19] and the pdf: User's Guide for MPC Toolbox [15].

3.1   PID basics

Often an open loop system like the plant model has not the performance to meet the requirements. To improve the result and compensate for various errors can be done by letting the system making adjustments of its own. A solution can be a feedback control through a PID controller. This is a simple and effective start and fits a wide array of applications. PID controller is the most commonly used type of controller in industrial applications. PID stands for Proportional, Integral and Derivative. Each of these parts treats the error term separately and then become added together and sent into the plant.

The ideal PID controller can be describe as

$$u(t) = K_P \cdot e(t) + K_I \int_{t_0}^{t} e(\tau)d\tau + K_D \frac{de(t)}{d(t)} \qquad (3.1)$$

where $u(t)$ is the input for the engine model it will control and $e(t)$ is the error, i.e. the difference between reference value and out value from the plant. In general a big value on K_P gives faster response it might also worsen the stability. The integrating part can improve the output but at the expense of stability. Derivation part is used to improve stability but it is sensitive to disturbance and therefore often gets excluded.
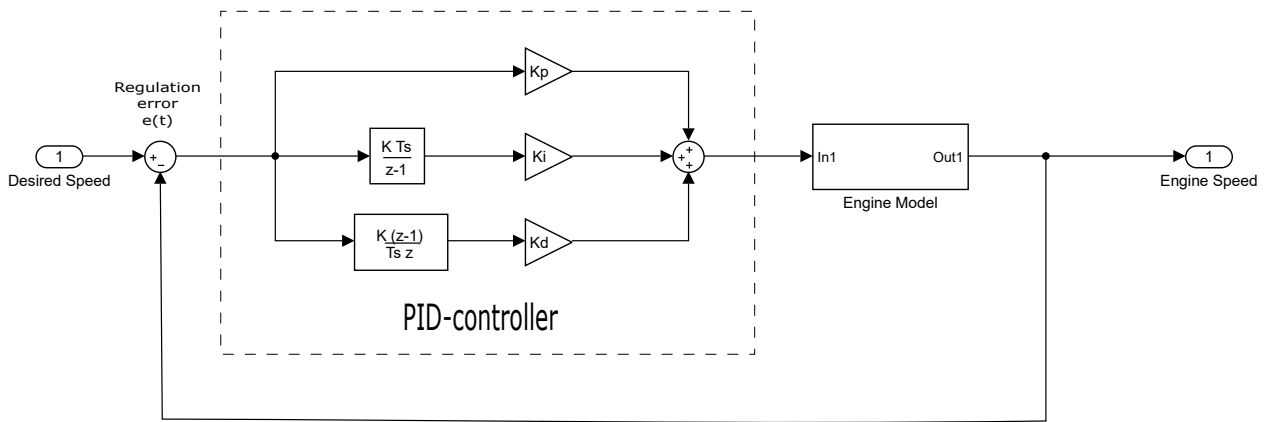
Figure 3.1: The figure shows how the PID controller interacts with the engine model.

3.2   Example, stabilising engine model with PID

Assume the setup from Figure 3.1, the PID controller is called G_c in the example below and an arbitrary engine model is presented as G_p and will look like this:

$$G_p(s) = \frac{1}{s-2}. \tag{3.2}$$

Closed-Loop Transfer function:

$$T = \frac{G_c G_p}{1 + G_c G_p}. \tag{3.3}$$

And the control function can be rewritten like this:

$$G_c = K_P + \frac{1}{s}K_I + sK_D = \frac{K_D s^2 + K_P s + K_I}{s}. \qquad (3.4)$$

By adding these three 3.2, 3.3, 3.4 equations together the new closed loop transfer function will look like this:

$$T = \frac{K_D s^2 + K_P s + K_I}{(K_D + 1)s^2 + (K_P - 2s) + K_I}. \qquad (3.5)$$

To make this system stable assume the poles is in the left hand plane, $P_1 = P_2 = -1$, and from this construct suitable values to fit the controller. With these poles a new equation comes to use:

$$(s + 1)^2 = s^2 + 2s + 1. \qquad (3.6)$$

By comparing the denominator in Equation 3.5 to Equation 3.6 it is possible to extract values on K_P, K_I, and K_D

$$As^2 + Bs + C = (K_D + 1)s^2 + (K_P - 2)s + K_I. \qquad (3.7)$$

Where

$$A = (K_D + 1)$$
$$B = (K_P - 2)$$
$$C = K_I$$

3.3 MPC basics

Model Predictive Control is a control strategy, which combine prediction, optimisation and receding horizon implementation. A well-designed algorithm gives an optimal stability and robustness. By using constraints and cost in the analysis it is possible to penalise movement close to "no go"-zone. Algorithm design and tuning require a clear understanding of how MPC works. Silly mistakes easily lead to a poor implementation that often ends up with bad results.

3.3.1 Prediction

Humans often use prediction naturally without thinking about it, e.g. when driving a car. If the visibility is good and no dangerous situations are noticeable the speed of the vehicle will probably be higher than if the weather and road condition would be the opposite. This is because one always predict beyond the safe braking

distance otherwise it is not possible to give a reasonable guarantee to not cause a crash. Prediction underpins practical human behaviour and therefore seems logical to incorporate into an automated control strategy. One should always predict beyond the key dynamics of a process or else the part that not been observed could become trouble.

3.3.2 Receding horizon

When referring to receding horizon it means that the algorithm continually updates the prediction and decision making so it can use the most recent target and measured data. The outcome of this is that the prediction horizon is moving along with the latest position. As the time goes and one moves forward so does also the end point of the prediction. The update of prediction together with the latest measured data introduces a feedback and by making decisions based on most recent measurement closes the loop.

3.3.3 Modelling

An aim for the model is that it should be easy to form predictions, ideally to use linear equations. The main goal is to accomplish accurate predictions but not more complex than necessary. In practice, the predictions can differ around $10 - 20\%$ in steady state and still be effective enough.

The importance is to capture the key dynamic changes during transients. Later the continuous feedback will correct for small modelling errors. Thereby it is rarely beneficial to spend an unreasonable amount of effort to create a high order model for improving accuracy without major impact on behaviour. It is just enough to make sure that the model has the ability to give good long-range predictions.

3.3.4 Performance index

The use of performance index is to numerically define which input trajectory is best. This might be a bit arbitrary but it gives an opportunity for a comparison between different strategies. To avoid corresponding risks with the complexity it is important to keep a cautious use of performance index if the model is of low quality. This often leads to a better conditioned and smoother optimisation. Typically quadratic performance indices are used, which give an optimisation with a unique minimum. When adapting to a more complex strategy it is highly likely to make a mistake and also more likely to get stuck. Greater performance comes with a cost, less robustness or ability to deal with uncertainty.

3.3.5   Degrees of freedom

This is a description of the complexity of the input predictions and the degrees of freedom shares the issues with the performance index. The useful numbers of degrees of freedom are connected to how accurate a prediction is made. If the model is poor there is no point utilising high numbers of degrees of freedom and a highly demanding performance index, since the result would be meaningless.

3.3.6   Constraint handling

Most control strategies treat constraints as an afterthought. That is why it is a great advantage with MPC that can have constraints systematically embedded into strategy development. This information is critical to get a robust and effective closed-loop behaviour without any proposed input that allows overshooting. It may result in slightly slower rise time, but will be optimised. Without active constraints, a well-tuned law may work effectively in a small range and can be disastrous over a wider operating range.

3.3.7   Multivariable

Processes with numerous inputs and outputs need an effective control law that considers all inputs and outputs simultaneously, due to changing one input changes all outputs. Classical control techniques such as PID can cope with $2 - 3$ input/outputs without struggling. An advantage with MPC is its framework that automatically takes account of all of the usable variables.

3.4   Signals and tuning variables

By using an MPC Controller Block in Matlab/Simulink some input and output channels have to be connected and other parameters inserted through the script. The list that follows describes some of the inports in this block:

- **Measure Output (MO)**: provides the controller with a feedback signal from the plant and sends out a vector with dimension n_{MO}.
- **Reference Signal (y_{ref})**: the controller prediction needs a reference target or a set point target. This input value is crucial for the first step in the control prediction horizon. The value holds for the entire horizon.
- **Manipulated Variable (MV)**: the default output is calculated and sent to the plant for each of the control instants. It is updated at regular intervals and sends out a vector with dimension n_{MV}.

- **Sample Time** (t_s): controls interval and is defined as t_s in the MPC object for every time unit the controller samples the input and updates the output signal.
- **Measured Disturbance (MD)**: this input is optional if the plant has a known disturbance. It makes sense to consider this information in the prediction. If not MD is provided the controller will, by default, add white noise. This can also be removed.

3.4.1 Input/Output limits and weights

An advantage with MPC is the ability to set boundaries, which the controller can be regulated within. It would also have been possible to add restrictions separately within input/output if the size would have been greater than one, as it is in this work. The notations $u_{\min}$ and $u_{\max}$ have been used as limits to add bounds on the measured variable

$$u_{\min} \leqslant u(t) \leqslant u_{\max}.$$

It was also possible to define the maximal measured variable increment $\Delta u_{\max}$. The notation $y_{\min}$ and $y_{\max}$ have been used as limits to put bounds on the plant output

$$y_{\min} \leqslant y(t) \leqslant y_{\max}$$

and on the reference signal y_{ref}. These bounds work together with suitable scale factors and values for constraints softening. To add a penalty or remove some of the tension from the boundaries, parameters called weights are used. These parameters are used in both types of the MPC algorithms, found in this thesis, but added in different ways. In algorithm 1, the

$$\text{weights} \begin{cases} W_u \\ W_{\Delta u_{\max}} \\ W_y \end{cases}$$

are implemented directly in the MPC model whereas in algorithm 2, the

$$\text{weights} \begin{cases} \boldsymbol{Q} \\ R \\ Ł \end{cases}$$

are written into the cost function which is a part of the CustomCostFcn template.

3.5 Finding maximal torque rate

To find a realistic maximum on how fast it is possible for the engine to gain torque real measurements were used. To test this, a fixed speed was set and the only change during data logging was the throttle that was going from 0 to 20% thus kick-down is not of interest here. Figure 3.2 is showing the Base Torque and the Instant Torque.
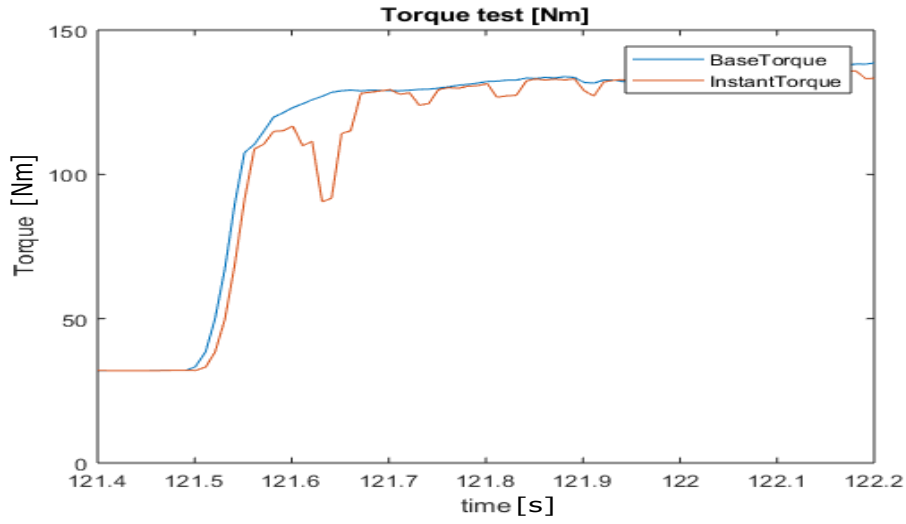
Figure 3.2: Analysing the derivative of torque on real data. The blue line is $T_{q,base}$ and the red line is $T_{q,inst}$. These measurement signals are not from a gear change, they were obtained with full throttle for a short period of time without external load.

In Equation 3.8 are t^1 represent a time coordinate on $T_{q,base}$ and t^2 represent a time coordinate on $T_{q,inst}$, the times t_x are arbitrarily chosen to represent a real scenario. By looking at the interval where the signals are almost longitudinally parallel to each other it is possible to see the time difference. This can be seen as a delay and is taken into consideration while calculating the torque rate

$$\Delta u = \frac{T_{q,inst}(t_2) - T_{q,base}(t_1)}{t_{delay}},$$
(3.8)

where the

$$t_{delay} = t_1^2 - t_1^1.$$

By using Equation 3.8 and the data in Figure 3.2 presenting the relation between instant torque and base torque, makes it possible to receive values on maximum torque rate and time delay.

$$\Delta u = 1242 \quad [Nm/s]$$
$$t_{delay} = 0.0107 \quad [s]$$

This can be set in a contest with the outcome from the restricted simulation found in Figure 5.4 where the gradient $\Delta u = 269$ Nm/s.

3.6   Create a reference signal

With a predictive controller it is possible to use signals, that otherwise are not taken into consideration, to create a reference signal y_{ref}. To find the optimal gear for current load, the control system in the vehicle cross-reference with a load-dependent gear map. When the control system detects that a gear change is needed information is sent to other control units. To create y_{ref} one can look on the signal for Lock-Up function, then calculate a suitable engine speed from the fixed gear ratio, and add some speed to compensate the losses in the torque converter.

Assume the vehicle is travelling with an engine speed of $y_{ref,start} = 1500$ rpm on the 6th gear and the load increases pass the limits and forces the gearbox to gear down to the 5th gear. y_{ref} can be constructed as a transient with start value and end value which fits the new gear ratio. To calculate the speed ($y_{ref,end}$) after the gear change can be possible through

$$y_{ref,end} = \frac{y_{ref,start} \cdot Ratio_{gear-1}}{Ratio_{gear}} + \eta_{rpm,losses} \tag{3.9}$$

By applying numbers to the Equation 3.9

$$y_{ref,end} = \frac{1500 \cdot 1.221}{1.000} \approx 1900 \quad [rpm]$$

can a new $y_{ref,end}$ be found. Gear ratios can be found in Table 2.1. Torque converter losses ($\eta_{rpm,losses}$) is not analysed because of the nonlinear behaviour of the torque converter and in this case assumes to be $\eta_{rpm,losses} = 68.5$ rpm.

The Lock-Up signal is up-scaled 500 times to be visible on the engine speed graph. More information about the Lock-Up function can be found in Chapter 2. By looking at Figure 3.3, a time delay of almost $300ms$ can be noticed between unlocking of the converter and increasing of engine speed. This time can be spent on predictive calculations and to increase speed earlier. In a good scenario, improvement of fuel consumption can be done as well as achieving faster gear changes.
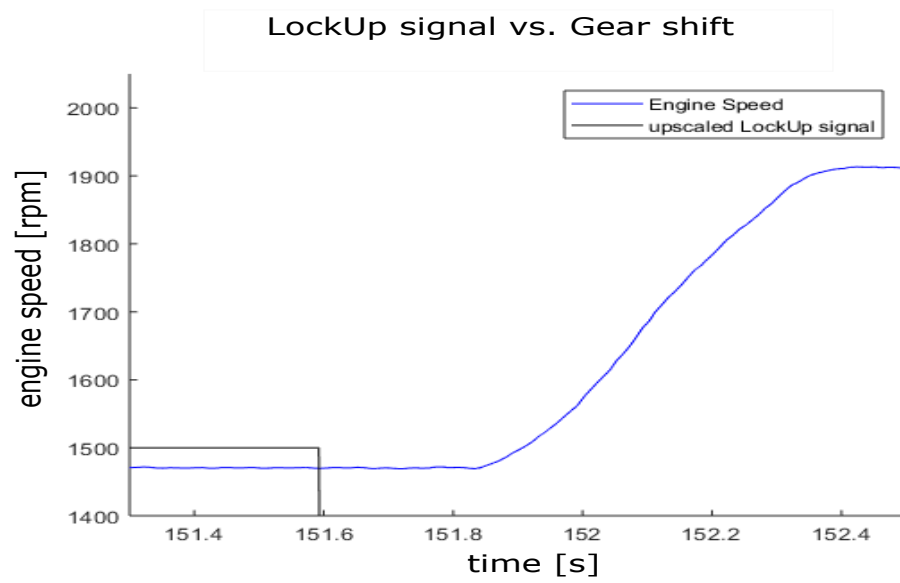
Figure 3.3: Evaluating time delay, the blue curve is a gear change displayed in contrast to the release of the Lock-Up function shown in black. The Lock-Up signal is multiplied with 500 to fit this figure, and moves from locked to released.

4

Methods

4.1 Rig tests

Old data from previous test runs were first examined to see if these could be used to start the modelling. Unfortunately, these old tests were sometimes missing important signals or had collected data with wrong intentions. Therefore, new rig tests had to be made. It was anyway a great start to sort out a lot of new questions about signals, abbreviations and test programs needed to proceed.

4.1.1 First data collection

The problem with the first set of tests, an example shown in Figure 4.1, was that they showed negative torque values and this led to weird load spikes during gear changes. The test rig was forcing the engine to rotate when the throttle was held at 0%. This also created problems for the transmission and the usage of the torque converter.
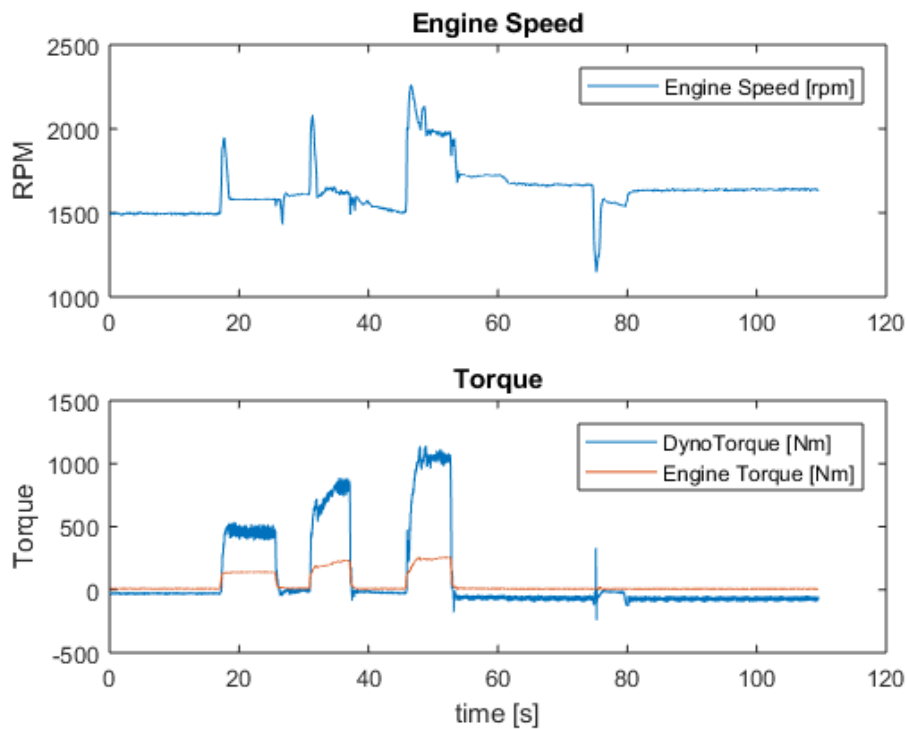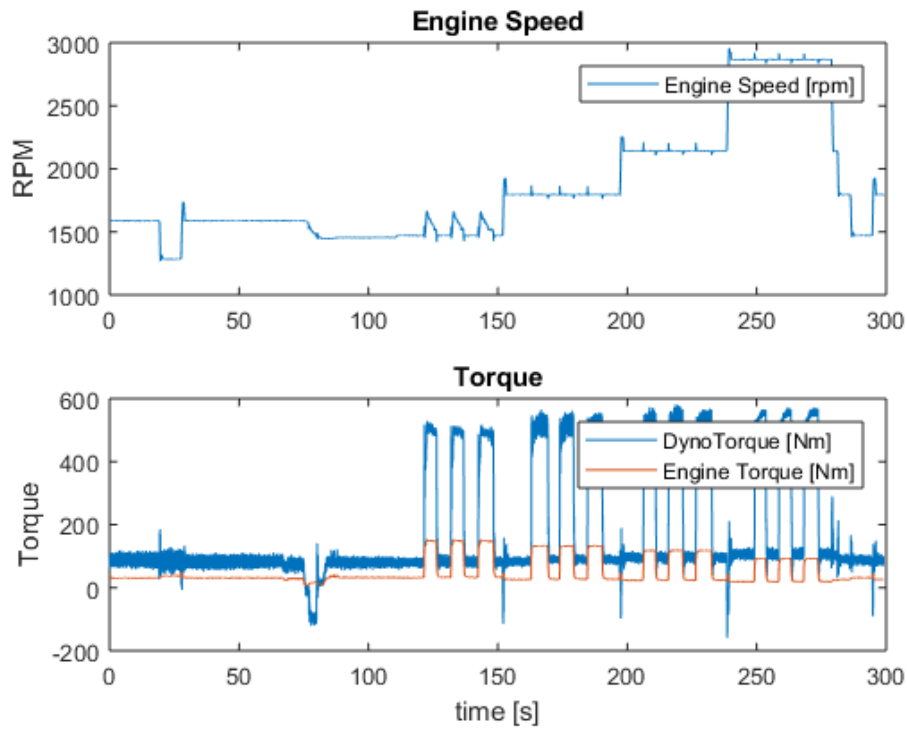
Figure 4.1: First rig test showing throttle increase and a change of gear. The top figure displays the engine speed and the second figure shows how the orange engine torque curve relates to the blue measured dynamometer torque from the rig test.

The outcome from this set of tests made it possible to calculate a value for inertia just under $J \approx 0.25$ [kg·m^2] using Equation 2.2. It showed to be reasonable compared to the numbers found in Vsim. Because of different values depending on what kind of gearbox that is in use, a general inertia value was used in the model calculations.

After working with these first test results was the conclusion to re-test.

4.1.2 Second data collection

The next set of tests was made with new knowledge and the conclusion was that the results were useful for developing a PID controller. In Figure 4.2 a dyno torque can be seen which overall gives a positive value. The big difference between engine and dyno torque is due to that the gear ratio has not been compensated.

Figure 4.2: The second rig test showing throttle increase at different gears. The top figure displays the engine speed and the second figure shows how the engine torque relates to the measured data from the rig test.

For the latest measurements, it was possible to pick out parts that were relevant. An example is the recorded gear change presented in Figure 4.3.
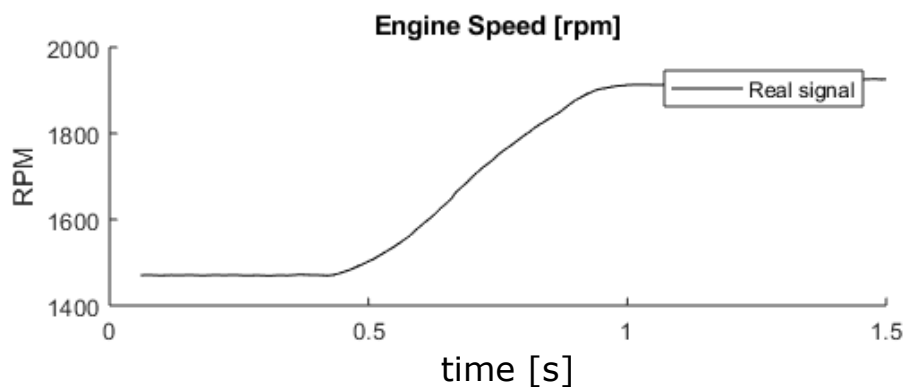

Figure 4.3: Engine speed during gear change from rig test

4.2 Engine modelling

There are many ways to describe a real system through a model, especially when the system is as complex as a modern engine. There are many sensor signals collected and forwarded to be processed in the engine control system. Each signal contains a portion of disturbances and many of the signals are converted by internal models. Disturbance creates additional uncertainty to the signal before it reaches the control unit and gets recalculated into e.g. a torque demand, which is needed to control the powertrain. The conclusion was to create a model depending on the brake specific engine torque (T) as a control signal and the speed of the engine (ω) as the only state,

$$x(t) = \omega, x \in \mathbb{X}$$
$$u(t) = T, u \in \mathbb{U},$$

where the engine torque is limited by the current operating point and the engine speed.
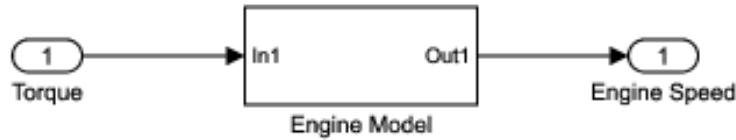
Figure 4.4: An example showing input and output in the engine model.

The engine speed model is calculated by assuming a mechanical model including inertia, see Equation 2.2. The model is described according to

$$T = J \cdot \dot{\omega}(T), \tag{4.1}$$

and can be rewritten

$$\dot{\omega}(T) = \frac{1}{J} \cdot T, \tag{4.2}$$

to solve this equation. In order to use this equation a value for the moment of inertia had to be decided and a description of the meaning of engine torque had to be explained. In general when an engine is tested the notation *break* is used,

$$T = T_{BMEP}, \tag{4.3}$$

and Brake Mean Effective Pressure (BMEP) can be calculated from break torque that is commonly used in rig testing. The engine torque is measured at the crankshaft and is a result of produced torque minus losses from e.g. mechanical friction. To continue using pressure notation, explained in the previous chapter, the new expression can be written like this

$$T_{BMEP} = T_{IMEP_n} - T_{FMEP}, \tag{4.4}$$

to simplify friction torque it is assumed that T_{FMEP} will increase proportionally to the engine speed

$$T_{FMEP} = k \cdot \omega. \tag{4.5}$$

This is a good assumption to describe reality in the speed region concerning this thesis. The value of k is set to 0.1 which means that at $\omega = 150$ rad/s the torque needed to overcome friction and maintain engine speed is $T_{FMEP} = 15$ Nm.
By adding the Equations 4.2 - 4.5 together in the steady state model, found in Equation 2.4, the result

$$\dot{\omega}(T) = \frac{-k}{J}\omega + \frac{1}{J}T_{IMEP_n}, \tag{4.6}$$

is the engine model that has been used in this project.

The value for inertia is set to $J = 0.25$ kg·m² and this is a general assumption [10]. Inertia can be divided into two parts

$$J = J_{ce} + J_c. \tag{4.7}$$

J_{ce} is connected to the weight of internal parts often around $J_{ce} = 0.05$ kg·m². The other part in the inertia, J_c, comes from external weights, like adding a flywheel on manual vehicles or from the torque converter and oil pump on automatic vehicles. This value usually are around $J_c = 0.20$ kg·m². Total Inertia (J) has also been compared to the data measurement log from the rig tests where a value of just under $J_{rig} \approx 0.25$ kg·m² was calculated.

By adding the discussed values to Equation 4.6 the engine model

$$\dot{\omega}(T) = -0.4 \cdot \omega + 4 \cdot T_{IMEP_n}, \tag{4.8}$$

was then inserted into Simulink, as shown in Figure 4.4, with torque as an input and engine speed as an output.

4.3 Fuel consumption calculations

In order to find a more efficient routing to gain speed one need to be able to define and penalise cost. In this case, petrol is the energy source needed to be reduced. Among the internal signals of the vehicle it is possible to find fuel related signals but, to save time and rig test cost, a pre-designed fuel map from Volvo where used.
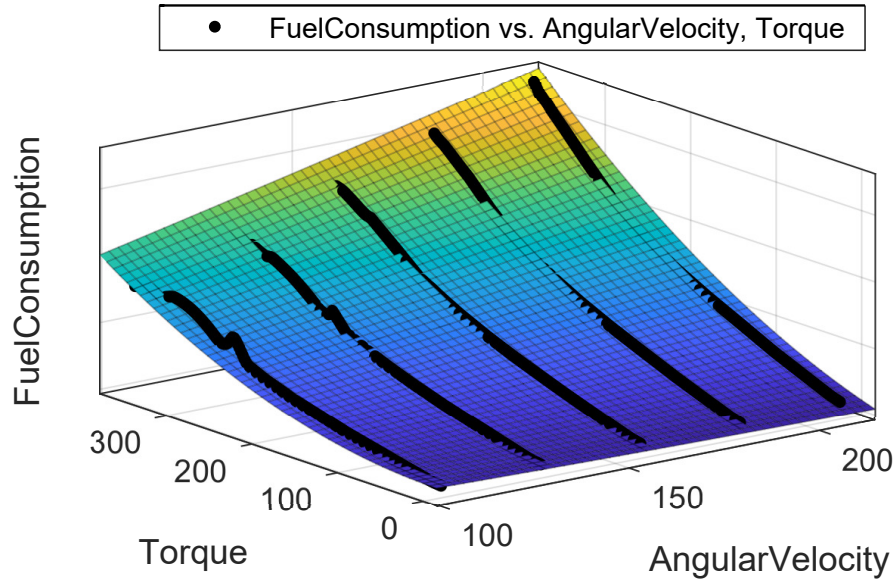
Figure 4.5: Plotted fuel consumption at specific angular velocity and torque. FuelConsumption axis is unlabelled due to sensitive information, but one can see that higher speed and/or torque gives higher consumption. The black lines are real values and the colourful plane is the representation.

To be able to include fuel consumption as a strategy parameter into the MPC algorithm a torque and engine speed depended equation had to be made. This was created with Matlab Curve Fit and with some customisation of operating range seen in Figure 4.5. The curve is limited within operating engine speed and only consider positive torque. It would be possible to limit the maximal torque according to engine specification, torque limits can be seen in Figure 2.8. This was not needed because the accuracy is acceptable and in this thesis the ability to reach maximal torque is not desired. A polynomial equation was created with 99% accuracy

$$f(x, y) = \alpha_1 + \alpha_2 \cdot x + \alpha_3 \cdot y + \alpha_4 \cdot x^2 + \alpha_5 \cdot x \cdot y + \alpha_6 \cdot y^2 \tag{4.9}$$

where $x = \omega$ = Angular velocity [rad/s] and $y = T$ = Torque [Nm]. The values on α_i is no information that Volvo is comfortable to make public.

4.4   Optimising method

To start with the MPC controller hard constraints need to be established, bound-
aries for output signal not to cross and also boundaries for the predicted input
signal that should not be breached. When these are in place it is time to analyse
what to accomplish. The goal is to reach a reference value without overshoot in
less time than it takes for the real signal seen in Figure 2.5. Due to not compro-
mising customer experience a rise-time of between 200 and 500 ms seems like a
fair goal. By stretching out the rate bad sound and jerks from the power-train will
be overcome.

4.4.1   MPC without fuel consumption penalty

To begin designing an MPC controller for engine speed at gear shift a basic con-
troller was created using Matlab MPC-toolbox. The purpose of this was to study
behaviour and see how this type of controller would perform in comparison with
other simulations. As a reminder of the wanted outcome: not only fuel con-
sumption is important but also simulation time and engine limitations need to be
considered.

The first thing to do in the MPC-toolbox application in Simulink was to define an
internal model. It is the same as the plant model found in equation 4.8. Next step
was to add several parameters to make the constructed controller perform in a
desired way. The parameters that have been tampered with are e.g. sample time,
testing different plant and control horizons, set points, weights and constraints.

The sample time is set to $t_s = 10$ ms, compared to the existing sample time of the
idle controller which has a $t_s = 16$ ms [1]. Many of the vehicles internal system
also have a changing sample time which varies along with the engine speed. A
quick calculation also provides some guidance, presume an engine speed of 2000
rpm, which equals to

$$\frac{2000[rpm]}{60} \approx 33.3 \quad [rps],$$

in every engine rotation there are 2 ignitions. This means that there is approxi-
mately a new ignition every 0.015 second at the speed. If this thesis also would
have concerned to control the ignition timing to manage a torque reserve, 10 ms
of sample interval should have been enough.

The ability to decide the length of the prediction and control horizon makes it pos-
sible to tune the closed-loop system so it guarantees stability, it also implies that

the simulation time has to be reasonable. A prediction horizon of $H_p = 20$ sample intervals was chosen. The length of the prediction- and control-horizon need to satisfy this $H_p \gg H_c$ relation. Small control horizon length promotes stability and it was set to $H_c = 4$ sample intervals.

For the controller to react, a set point y_{ref} was created to suit the planned gear change, see Equation 3.9. There is no extra compensating input error and other disturbances, but there was a mismatch that had to be solved. It turned out that it was the internal disturbances added by MPC-toolbox, that had to be cancelled out.

Constraints on the manipulated variable $(u(k))$ and the output variable $(y(k))$ are hard boundaries and they are set to $15 \leqslant Torque \leqslant 150$ Nm and $0 \leqslant Engine\,speed \leqslant 250$ rad/s.

To complete this controller it is time to add some weights. From previous section penalising cost was added to the manipulated variable $W_u = 10$ and to the output variable $W_y = 100$ but there also is the manipulated variable rate $(\Delta u_{\max})$ that needed some tightening $W_{\Delta u_{\max}} = 200$, otherwise the torque would have increased too fast.

4.4.2 MPC with fuel consumption penalty

To be able to implement a penalty for the fuel cost an MPC optimizer called CustomCostFcn where used. This feature is enabled in the MPC control by setting $Optimizer.CustomCostFcn = true$. CustomCostFcn uses inputs like e.g. $y(k)$, y_{ref}, $u(k)$, u_{ref} and $\Delta u_{\max}$ and then returns among other variables a specified arbitrary cost function (J^*). This function replaces the standard quadratic cost function. Instead is $fmincon$ used from Optimization Toolbox to solve this non-linear problem. Previously chosen parameters, such as weights, are not in use, only the constraints with hard boundaries are still active while using this template. The cost function is defined like this,

$$J^* = \mathbf{Q} \cdot \{y - y_{ref}\}^2 + R \cdot \{\Delta u\}^2 + \text{Ł} \cdot f_\alpha(y, u) \qquad (4.10)$$

where $\mathbf{Q}$, R and Ł are weights and have been tuned to level out the contribution from the different parts in this equation. H_p is the length of the prediction horizon, y, y_{ref} and Δu is plant model output speed (ω_{ce}), reference speed (ω_{ref}) and manipulated variable rates of change $(\frac{T(i)-T(i-1)}{\Delta time})$. The fuel cost Equation 4.9 had to be modified to fit the cost function, where all α_i are the same as before. $\mathbf{Q}$ is created by multiplying the weight $Q = 3$ with an identity matrix with the same size as the prediction horizon,

$$\boldsymbol{Q} = \begin{bmatrix} Q(H_w) & 0 & \ldots & 0 \\ 0 & Q(H_w + 1) & \ldots & 0 \\ \vdots & \vdots & \ddots & \vdots \\ 0 & 0 & \ldots & Q(H_p) \end{bmatrix}$$

$H_w = 1$ is the 'window' parameter. The remaining weights have been set to $R = 9$ and $Ł = 1 \cdot 10^9$. Besides tempering with the weights values, prediction horizon and control horizon have been changed and it is shown in the results section what the impact these values had.

5

Results

The results are first displayed in Table 5.1 and then more detailed with figures in order for the reader to get a better visual review.

Table 5.1: Collected outcome from simulations*

| Controller Type	Baseline	Basic MPC	MPC with Fuel Cost
Shift Time [ms]	370 + 300**	380	450
Calculation Time [s]	-	3.36	32.25
Δu_{Max} [Nm/s]	269	1036	904
Receding Horizon	-	20	70
Fuel Consumption [mg]	376.53	288.10	284.69
Change from Baseline [%]	-	-23.5	-24.4

(*) The computer used for simulation had an Intel^R Core^{TM} i5-6300U processor with 8 GB of RAM and ran with 64-bit MATLAB R2017b.
(**) Thr time between released Lock Up and the beginning of speed increase.
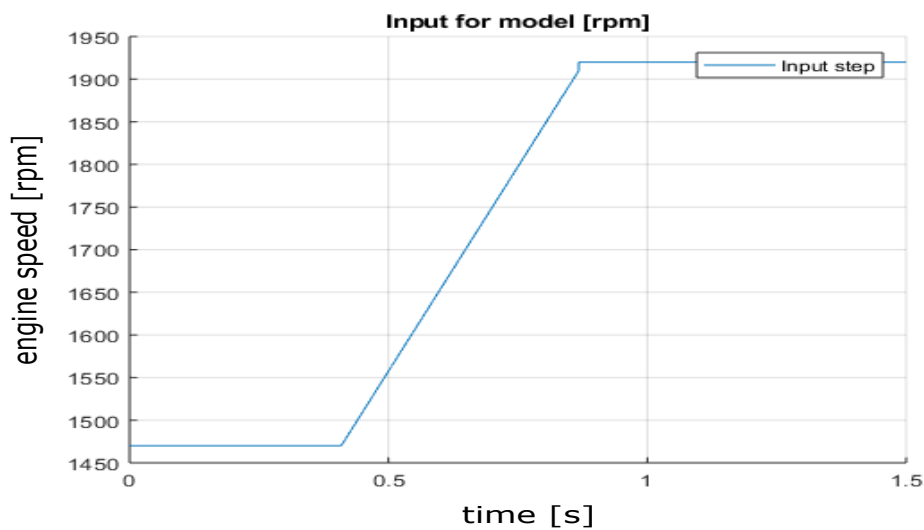
5.1 Engine model with PI controller

The simulations were performed in a Simulink setup shown in Figure 3.1.
An input signal was created to be able to recreate the measured signal in an accurate way. The slope of the new reference signal used in the PI controller was created by curve fit and can be seen together with the measured signal in Figure 5.1.
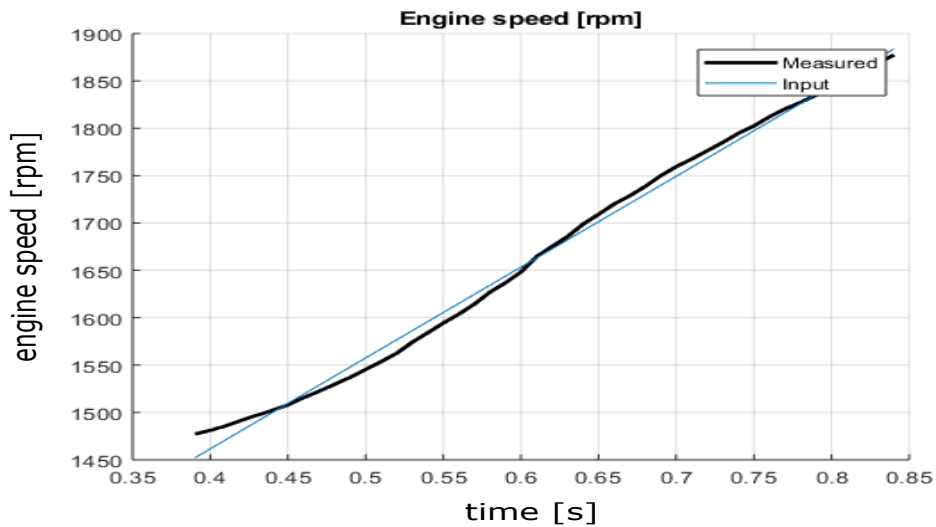
Figure 5.1: Creating an input step, by replicate the slope of the real gear change

To complete the new reference signal a constant start speed and finish speed were added to replicate the gear shift, which is presented in Figure 5.2.


Figure 5.2: Finished input step

By applying the PID example technique to the engine model, described in Section 3.2, to calculate suitable tuning values $K_D = 0$, $K_P = 4$ and $K_I = 1$. $K_D = 0$ means that the controller is actually only a PI-controller and the outcome will look like Figure 5.3.
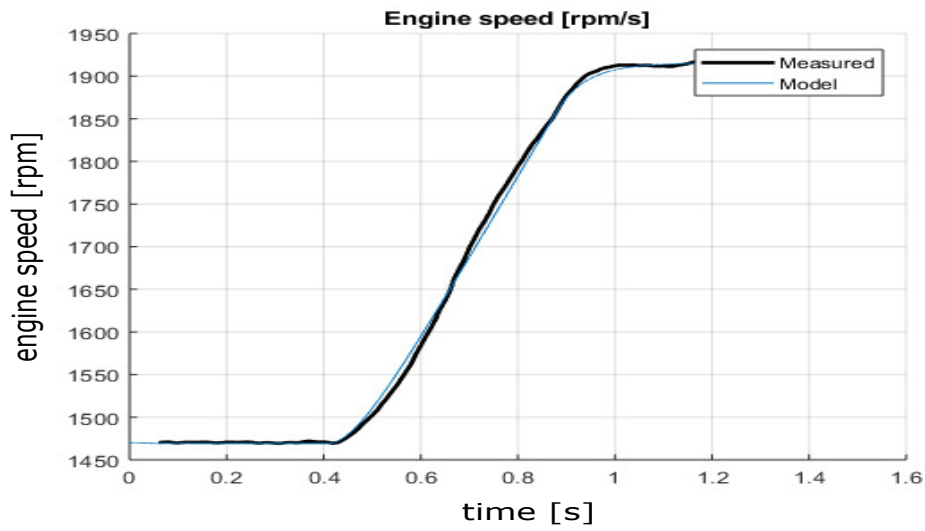
Figure 5.3: Gear change simulation with PI-controller

It became a good fit but with some extra tuning $K_P = 5$ was discovered to be even a better fit, shown in Figure 5.4. The latest version of PI-controller was then used as a baseline in the performance comparison. The figure also shows a simulated torque curve.
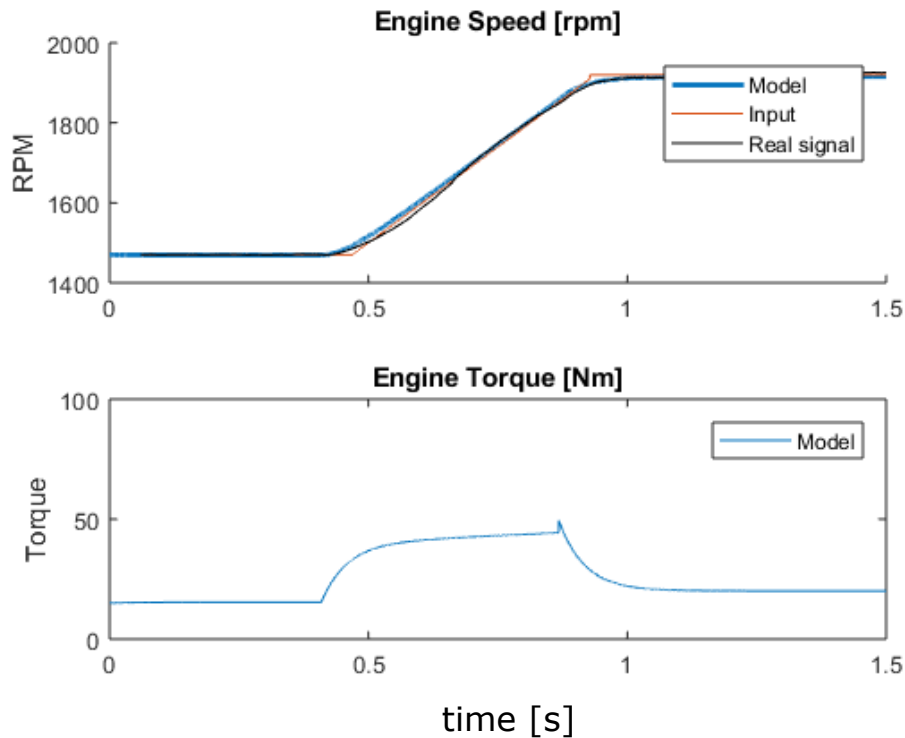
Figure 5.4: Gear change simulation with updated PI-controller

This simulation gave a fuel cost of 0.37653 grams for the increase of engine speed and with a maximal torque increase rate of $\Delta u = 269$ Nm/s.

5.2 Engine model with MPC controller

Algorithm 1 refers to subsection 4.4.1 (MPC without fuel consumption penalty) and Algorithm 2 refers to subsection 4.4.2 (MPC with fuel consumption penalty)

5.2.1 Algorithm 1

This solution is made by creating a traditional (implicit) MPC controller in Matlab with no regard to fuel cost. The purpose was to investigate Model Predictive Control ToolBox and to be able to compare a simple version of predictive control

against a more advanced algorithm which penalises high fuel cost behaviour. It is also of interest to be able to compare simulation time when a more advanced controller is in use.
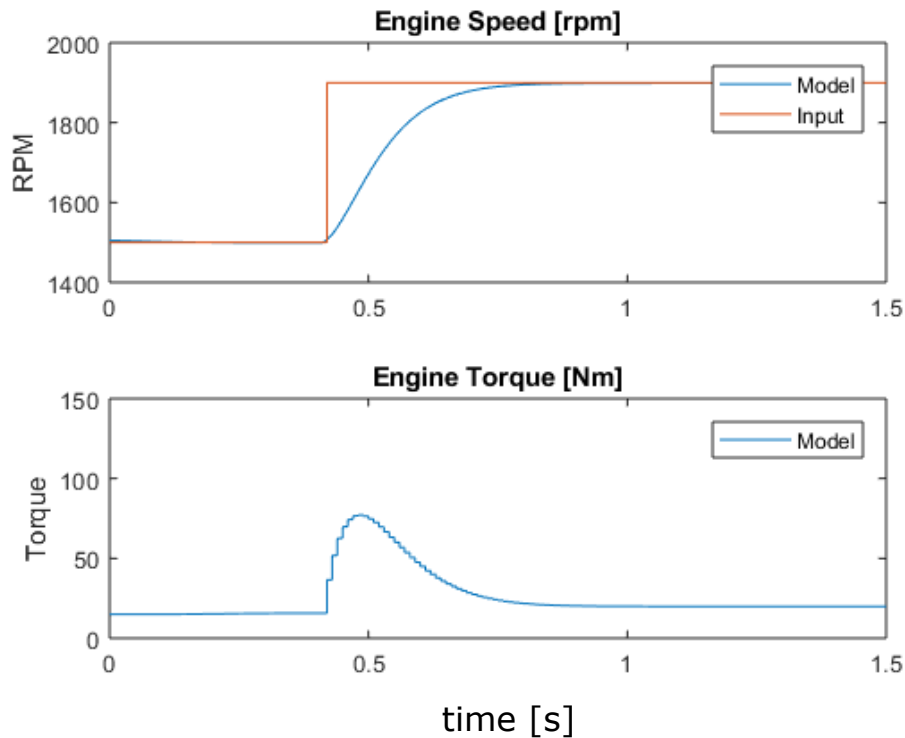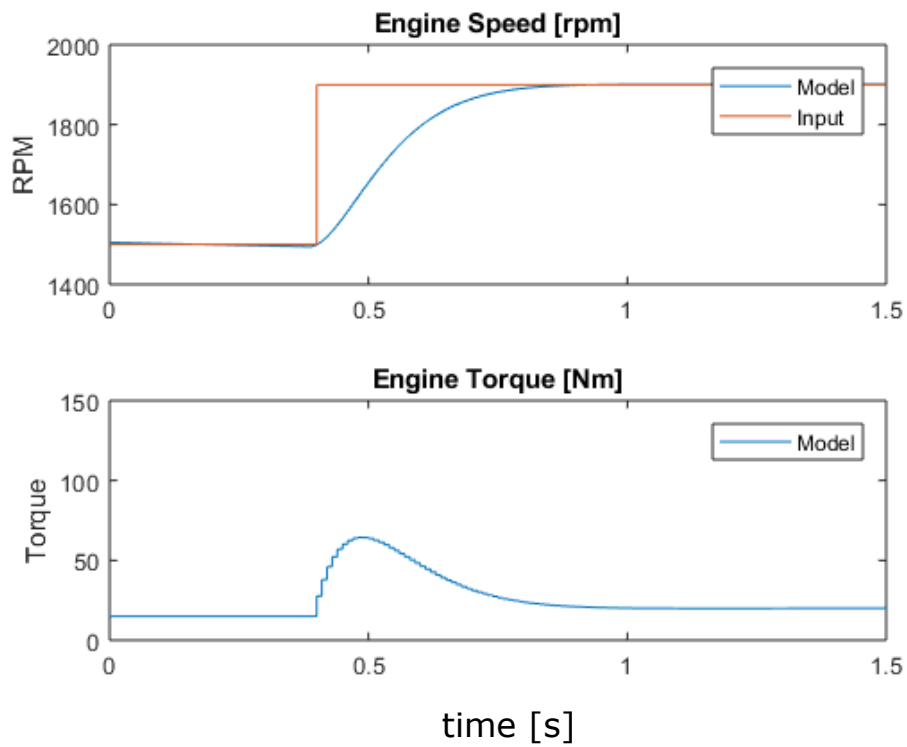

Figure 5.5: RPM and Torque plots from a regular MPC

This simulation gave a fuel cost of 0.28810 grams for the increase of engine speed and with a maximal torque increase rate of $\Delta u = 1036$ Nm/s.

5.2.2 Algorithm 2

This controller was made by using CustomCostFcn, which is described above, including sample time, constraints and weights. For this to turn out properly both prediction and control horizon had to be increased, $H_p = 70$ and $H_c = 7$. This resulted in an even smoother torque curve than in previous MPC controller without penalty on fuel consumption.

Figure 5.6: RPM and Torque plots from Custom MPC

This simulation gave a fuel cost of 0.28469 grams for the increase of engine speed and with a maximal torque increase rate of $\Delta u_{max} = 904$ Nm/s.

6

Conclusion

This thesis gives an overview of the most important subjects when designing an engine controller. Modern vehicles are complex and involve a great number of inputs together with small margins to operate within to follow government regulations. The thesis also describes how to develop a simple linear engine model to use in controller design.

The main achievement of the thesis has been the creation and analyse of two MPC controllers which have been executed in Matlab. The implementation has been done with an input signal from the engine model. Both controllers have shown improvement regarding fuel consumption. The second MPC controller, where a fuel map where acting as a weight, appeared to work best. The simulation time increased due to the additional length of the prediction horizon. It is probably possible to downsize this time when the controller becomes optimised and implemented in real time into the engine system control unit.

Regarding emissions, CO_2, which can be assumed proportional to the fuel consumption has been reduced, but nothing can be concluded about CO, HC, NO_x.

It clearly can be seen that fuel consumption reduction can be achieved by using an MPC controller during a gear shift.

6.1   Future work

To improve the modelling it would be interesting to include ignition timing, in a way that MPC controller could separate $T_{q,inst}$ and $T_{q,base}$ and use $T_{q,rsv}$ for instantaneous torque gain. Maybe even analyse if controlling camshafts and variable geometry turbo can be prepared to make the gear shift more efficient.

In order to future proof the MPC algorithm used in vehicles, it would be of interest besides fuel economy also to see emissions be implemented as constraints.

The algorithms in this thesis are only considering a small interval of the engine speed, it can be enlarged. Due to the transmission has its own control system, it can be explored to find more possibilities for control.

By developing an explicit MPC controller it is possible to trim the computations and save on memory usage, which usually is downsized to save cost at production.

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