Modeling, Control and Optimization of the Transient Torque Response in Downsized Turbocharged Spark Ignited Engines

OSCAR FLÄRDH

Doctoral Thesis in Automatic Control
Stockholm, Sweden 2012
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Abstract

Increasing demands for lower carbon dioxide emissions and fuel consumption drive technological developments for car manufacturers. One trend that has shown success for reducing fuel consumption in spark ignited engines is downsizing, where the engine size is reduced to save fuel and a turbocharger is added to maintain the power output. A drawback of this concept is the slower torque response of a turbocharged engine. Recent hardware improvements have facilitated the use of variable geometry turbochargers (VGT) for spark ignited engines, which can improve the transient torque response. This thesis addresses the transient torque response through three papers.

Paper 1 presents the optimal control of the valve timing and VGT for a fast torque response. Optimal open-loop control signals are found by maximizing the torque integral for a 1-D simulation model. From the optimization it is found that keeping the ratio between exhaust and intake pressure at a constant level gives a fast torque response. This can be achieved by feedback control using VGT actuation. The optimal valve timing differs very little from a fuel consumption optimal control that uses large overlap. Evaluation on an engine test bench shows improved torque response over the whole low engine speed range.

In Paper 2, model based, nonlinear feedback controllers for the exhaust pressure are presented. First, the dynamic relation between requested VGT position and exhaust pressure is modeled. This model contains an estimation of the on-engine turbine flow map. Using this model, a controller based on inverting the input-output relation is designed. Simulations and measurements on the engine show that the controller handles the strong nonlinear characteristic of the system, maintaining both stability and performance over the engine's operating range.

Paper 3 considers the dependence of the valve timing for the cylinder gas exchange process and presents a torque model. A data-based modeling approach is used to find regressors, based on valve timing and pressures, that can describe the volumetric efficiency for several engine speeds. Utilizing both 1-D simulations and measurements, a model describing scavenging is found. These two models combine to give an accurate estimation of the in-cylinder lambda, which is shown to improve the torque estimation. The models are validated on torque transients, showing good agreement with the measurements.
One does not simply walk into Mordor. Nor does one simply write a doctoral thesis. Without the support from a helpful fellowship, this thesis would never have been written.

First of all, I would like to thank my advisor Håkan Hjalmarsson for giving me the opportunity to have a second chance as a doctoral student. I am very grateful for your guidance and help that made me accomplish this thesis. In any situation, you always know how to make an improvement. My co-advisor Jonas Mårtensson has in an excellent way handled the transition from co-worker and officemate to supervisor. Your support has been invaluable.

My first supervisors; Kalle Johansson, Mikael Johansson and Carlo Fischione, also deserves thanks for teaching me the first steps in becoming a researcher.

A lot of this work has been carried out in collaboration with the Internal Combustion Engine department at KTH together with Gustav Ericsson and Fredrik Westin. It has been a lot of fun to work with you, and I’ve learned a lot. Also the mechanics at the combustion engine department have been very helpful in keeping the engine up and running or replacing throttles with a smile. Thanks also to Master’s thesis student Erik Klingborg for the help with simulations and experiments.

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To my friends, thanks for helping me relax and clear my mind, wether it be playing rackets ports, having a nice dinner, hiking or just having interesting discussions about life.

Last but not least, I would like to thank my family; Peter, Sylvia, Jacob, Joanna and Edvin for your incessant love and support.

Oscar Flärdbh
Stockholm, September 2012

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<td>BMEP</td>
<td>Brake mean effective pressure</td>
</tr>
<tr>
<td>BSFC</td>
<td>Brake specific fuel consumption</td>
</tr>
<tr>
<td>CAD</td>
<td>Crank angle degrees</td>
</tr>
<tr>
<td>DI</td>
<td>Direct injection</td>
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<tr>
<td>DISI</td>
<td>Direct injection spark ignition</td>
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<tr>
<td>EGR</td>
<td>Exhaust gas recirculation</td>
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<td>EVC</td>
<td>Exhaust valve closing</td>
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<td>FMEP</td>
<td>Friction mean effective pressure</td>
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<td>FL</td>
<td>Feedback linearization</td>
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<td>FTP</td>
<td>Federal test procedure</td>
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<td>IMEP</td>
<td>Indicated mean effective pressure</td>
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<td>IOMI</td>
<td>Input-output model inversion</td>
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<tr>
<td>IVO</td>
<td>Intake valve opening</td>
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<td>LQ</td>
<td>Linear Quadratic</td>
</tr>
<tr>
<td>PFI</td>
<td>Port fuel injection</td>
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<tr>
<td>PID</td>
<td>Proportional integral derivative</td>
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<tr>
<td>PMEP</td>
<td>Pump mean effective pressure</td>
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<td>MIMO</td>
<td>Multiple input multiple output</td>
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<td>MPC</td>
<td>Model predictive control</td>
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<td>MSE</td>
<td>Mean square error</td>
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<tr>
<td>NEDC</td>
<td>New european driving cycle</td>
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<td>SI</td>
<td>Spark ignited</td>
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<td>SISO</td>
<td>Single input single output</td>
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<td>VGT</td>
<td>Variable geometry turbine</td>
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<td>VNT</td>
<td>Variable nozzle turbine</td>
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<td>Variable valve timing</td>
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Part I

Introduction
Over the last century, the automobile has come to play a more and more important role in the society. The flexibility and freedom of road traveling is attractive for both private as well as for business transportation. In Sweden, 64% of the kilometers traveled are done by car and 53% of the population in ages 6–84 make at least one trip with a car an average day [SIKA 2007]. On the business side, 86% of the national freight and 60% of the total freight (national, import and export) was transported on the road in 2010 [Trafikanalys 2012].

A majority of today’s cars and trucks are powered by internal combustion engines. The only alternative that so far has reached the market is electric motors in hybrid or electric vehicles. However, combustion engines are likely to play an important role yet some years. One important reason for that is the much longer range due to higher energy density of its energy source. Gasoline and diesel have an energy density of 41–44 MJ/kg [Heywood 1988], while the batteries of modern hybrid and electric vehicle have 0.30–0.48 MJ/kg [Matthe et al. 2011; Tesla Motors 2012].

However, there is also impact on the environment from the use of combustion engines. Haagen-Smit (1952) showed environmental effects of the hydrocarbon and nitrogen oxide emissions, coming largely from automobiles. Such discoveries eventually led to legislation restricting the emission levels, which pushed the technological development. To meet these requirements, engines were equipped with, e.g., fuel injection systems and electronic throttle, as well as after-treatment systems like the three-way-catalyst. To utilize this new hardware and more precise actuators, good control systems are needed.

Lately, the discovery of the connection between CO$_2$ emissions and global warming (see, e.g., Weart 2003) have given rise to a similar development. The quest to reduce greenhouse gases like CO$_2$, together with the fact that the oil reserves are limited, is a large incentive to reduce the fuel consumption. In the European Union, a legislation on the average CO$_2$ emissions of 130 g/km is being phased in between 2012 and 2015. See Figure 1.1 for the development of and future goals on the fuel consumption.
CHAPTER 1. BACKGROUND

One technology that has been very popular for improving fuel consumption in spark ignited engines over the last decade is downsizing. A downsized engine has a smaller displacement volume, but is equipped with a turbocharger and possibly also other, more advanced hardware, like variable valve timing and direct injection to maintain the same power output. The smaller volume of the engine, however, reduces the fuel consumption. See Section 2.2 for a more detailed description of and references to downsizing.

1.1 Motivation

Even though many downsized engines matches or triumphs the larger engines in terms of power and fuel consumption, they still can not match a larger natural aspirated engine in the transient torque response. This problem is illustrated in Figure 1.2 where the measured torque response of a turbocharged engine is shown. The engine very quickly reaches around 190 Nm of torque, while the rest of the transient is much slower. The fist part, called the natural aspirated part, of the transient is limited by the actuator dynamics of the throttle and the gas dynamics in the intake manifold. The second, boosted, part of the transient is limited by the turbocharger dynamics. After the natural aspirated part, the pressure in the intake manifold is almost atmospheric. To further increase the torque, the turbine must have high enough rotational speed to allow the compressor to increase the pressure.

The maximum torque for this engine is over 300 Nm, and hence the time to reach that level is dominated by the response time of the boosting system. This is referred
1.2 Problem Formulation

The main problem studied in this thesis is how to control the variable geometry turbocharger and the valve timing to achieve a fast torque response. The goal is to find a generic control strategy that gives good transient performance for different load steps without affecting steady state fuel consumption. In this process, modeling and control of the exhaust pressure using the variable geometry turbocharger is investigated. Moreover, an analysis of the influence of variable valve timing on the gas exchange process and torque generation is performed.

1.3 Outline

The first part of this thesis is organized as follows. Chapter 2 describes the basic principles of turbocharged spark ignition engine operation, as well as the concept of downsizing and a brief description of the types of engine models used in this work. This chapter is directed to readers with little knowledge of combustion engines. It is followed by a presentation of related research results in Chapter 3. A description of the main contributions and a summary of the appended papers follows in Chapter 4 before Chapter 5 concludes the first part.

Figure 1.2: Measured torque response for a turbocharged SI engine at 1750 rpm.

Figure 1.2: Measured torque response for a turbocharged SI engine at 1750 rpm.
An internal combustion engine mixes air and fuel and generates work and emissions through a combustion process. This thesis focuses on a four stroke spark ignited engine with direct injection, variable valve timing and variable geometry turbine. This chapter describes the basics of such an engine, with focus on the concepts important for this thesis. For a thorough description of internal combustion engines, see e.g. Heywood (1988).

2.1 Air Path

In Figure 2.1 a sketch of the engine setup is shown. The air enters through an air filter, and is then compressed by the compressor. An intercooler is used to cool down the compressed air, and a throttle controls the air flow. The air in the intake manifold then passes through the valves into the cylinder, where fuel is injected. This happens during the intake stroke, the first of the four strokes. The air and fuel mixture is then compressed during the compression stroke, after which it is ignited using a spark plug. The ignition starts the combustion process and the released energy is transferred to mechanical work as torque on the rotating crankshaft. Finally, the burned gases are expelled from the cylinder into the exhaust manifold during the exhaust stroke. The completion of the four strokes corresponds to one engine cycle. The energy in the exhaust gases is utilized by the turbine to drive the compressor. Finally, the burned gas mixture passes the catalytic converter.

The work, $W$, generated on the crankshaft is generally divided into three parts.

$$ W = W_g + W_p + W_f $$

where $W_g$ is the gross work (during the compression and power strokes), $W_p$ is the pumping work (generated during the exhaust and intake strokes) and $W_f$ is the friction work. The gross work is given by

$$ W_g = \int p_{cyl} \, dV $$

(2.1)
where the integral is taken over the compression and power strokes. The pumping work is given by the same integral but over the exhaust and intake strokes. The pumping work is mostly negative, but for supercharged engines it is possible to have positive pumping work at some operating points. The friction work is always negative, and a simple approximation is that it grows linearly with engine size and quadratically with engine speed. It is common to normalize the work with the engine size, given by the displacement volume $V_d$, to compare the performance of engines with different sizes. Since the $W/V_d$ is a pressure quantity, it is referred to as mean effective pressure. The work on the crankshaft is usually denoted brake mean effective pressure, BMEP, and IMEP, PMEP and FMEP are used for the indicated gross, pump and friction mean effective pressures giving the relationship

$$\text{BMEP} = \text{IMEP} + \text{PMEP} + \text{FMEP}$$

from \[2.1\].

In the catalytic converter, the emissions are converted into nitrogen, oxygen, carbon dioxide and water. The reduction of emissions is only efficient if the ratio between air and fuel mass is stoichiometric, i.e. the chemical reactions are bal-
2.1. AIR PATH

A normalized version of this ratio is denoted $\lambda$, and the air/fuel ratio is stoichiometric when $\lambda=1$. To comply with the emission legislations, $\lambda$ thus needs to be accurately controlled to one. This is an important control task, and usually requires both feedforward and feedback control to achieve good enough performance in transient as well as steady state conditions. Feedforward control of the injected fuel to match the amount of air in the cylinders requires, in turn, precise knowledge of the air mass. Since the air mass entering the cylinders can not be measured accurately, models for this are required.

2.1.1 Volumetric Efficiency

A common way to model the air flow into the cylinders is through the volumetric efficiency, $\eta_{vol}$. It is defined as the volume flow of air into the cylinders divided by the rate of volume displaced by the cylinders

$$\eta_{vol} = \frac{2 \cdot 60 \dot{m}_{\text{air}}}{\rho_{\text{air}} V_d N_{\text{eng}}}$$

(2.4)

where $\dot{m}_{\text{air}}/\rho_{\text{air}}$ is the volume flow of air, $V_d$ is the displacement volume and $N_{\text{eng}}$ is the engine speed in revolutions per minute (rpm). Since the engine speed and the temperature and pressure in the intake manifold are measured, the mass flow of air, $\dot{m}_{\text{air}}$, can be calculated given an accurate model of $\eta_{vol}$. The volumetric efficiency typically depends on the pressures and temperatures in the intake and exhaust manifold, the engine speed and the valve openings.

2.1.2 Variable Valve Timing

The opening and closing of the valves are governed by a rotating camshaft that pushes the valves open. The camshaft is mechanically connected to the engine’s crankshaft which guarantees that the valve timing is always correctly in phase with the engine cycle. With variable valve timing this relation can be phase shifted, meaning the valve lift curves in Figure 2.2 are shifted with respect to the crank angle. This has the advantage that the valve timing can be adapted to different conditions on the engine. As can be seen in Figure 2.2, it is possible for both the intake and exhaust valves to be opened at the same time. The number of crank angles between intake valve opening, $\\phi_{IVO}$, and exhaust valve closing, $\\phi_{EVC}$ is called overlap and denoted $\\phi_{OL}$.

At low loads, i.e. low intake pressure, the exhaust gases remaining in the cylinder after the combustion enters the intake manifold when the intake vales open. As the piston moves down, these gases enter the cylinder again together with the air. If the overlap is large and the pressure difference between intake and exhaust manifold is large enough, there might even be gases from the exhaust manifold entering the intake manifold through the cylinders. This significantly reduces the volumetric efficiency, which however can have positive effect. To achieve the same amount of air with lower volumetric efficiency, the pressure in the intake manifold needs to be
increased which reduces the pumping losses. On the other hand, too much exhaust gases is bad for the combustion.

2.1.3 Turbocharging

A turbocharger consists of a turbine and a compressor connected with a shaft. The turbine converts the energy in the exhaust gases to rotational energy on the turbine shaft. This energy is, in turn, used by the compressor to compress the air going into the cylinders. Utilizing the energy in the exhaust gases is one of the main advantages of a turbocharger compared to, e.g., a mechanical supercharger where energy is taken from the crankshaft to drive the compressor. The power, $P_{\text{trb}}$, generated by the turbine can be described by

$$P_{\text{trb}} = \eta_{\text{trb}} \dot{m}_{\text{trb}} c_p T_{\text{exh}} \left( 1 - \left( \frac{p_{\text{es}}}{p_{\text{exh}}} \right)^{\frac{\gamma-1}{\gamma}} \right)$$

(2.5)

where $\eta_{\text{trb}}$ is the turbine efficiency, $\dot{m}_{\text{trb}}$ is the mass flow thorough the turbine, $T_{\text{exh}}$ and $p_{\text{exh}}$ are the temperature and pressure in the exhaust manifold, $p_{\text{es}}$ is the pressure after the turbine, $c_p$ is the specific heat capacity at constant pressure and $\gamma$ is the ratio of specific heat capacities. Thus the turbine power increases with higher mass flow, higher exhaust temperature and higher exhaust pressure. For more details on turbochargers, see [Watson and Janota (1982)]

2.1.4 Variable Geometry Turbines

Turbochargers have a fixed relation between mass flow and pressure ratio. This might result in a too low boost at low mass flows or a too high at high mass flows. This is usually addressed by designing a small turbine to have enough boost at low loads and then use a wastegate at high mass flows to direct the excess exhaust gases past the turbine. Thus not all the energy in the exhaust is utilized.

Another way to address this is to use a variable geometry turbine (VGT). There are several types of VGTs, but they all have in common that they have an actuator
2.1. AIR PATH

Figure 2.3: Turbine flow map for the VGT used in this thesis. There is clearly a big difference in flow characteristics between fully open (100%) and fully closed (0%) VGT position. The flow also depends on the turbine speed, and the two lines for each actuator position shows the maximum and minimum flow over the whole speed range.

that can alter the relation between mass flow and pressure. An example of this is seen in Figure 2.3, where the flow characteristics for the two end positions of the VGT controller is shown. Since the flow characteristics can be changed using the VGT, there is generally no need for a wastegate and all the energy in the exhaust gases can be utilized. Altering the VGT actuator position also changes the turbine efficiency, i.e. how much of the energy in the exhaust gases that are converted to torque on the turbine shaft. With this improved freedom of using a VGT, it is possible to get higher boost pressure over a larger operating range, as well as faster boost pressure increase in transients.

The tradeoff when controlling the VGT during the transient is to have the right amount of exhaust pressure. Closing the VGT gives higher $p_{exh}$ which gives more turbine power and thus more boost pressure. However, too high $p_{exh}$ has negative effects and there are mainly three limiting factors. Firstly, high $p_{exh}$ increases the pumping losses. Secondly, a high $p_{exh}$ prevents good gas exchange. It means that there will be low volumetric efficiency and more exhaust gases in the cylinder which lowers the torque. The low volumetric efficiency also gives lower mass flow and thus less turbine power. Thirdly, a high $p_{exh}$ lowers the turbine efficiency and thereby the turbine torque.

A common type of VGT uses variable nozzles in the turbine house. By changing the position of these nozzles, the open area is changed and thus the relation between
pressure and flow. These types of turbines are called variable nozzle turbine (VNT), and is the type used in this thesis.

VGTSs are more sensitive to high temperatures, and are thus mainly used in diesel engines which have lower exhaust temperatures. However, hardware improvements have made them more durable making them interesting also for SI engine applications. There is one car with SI engine and VGTT on the market today, produced by Porsche ([Porsche AG] 2012).

2.2 Downsizing

Downsizing can improve the fuel economy by reducing the size of the engine. This reduces the pumping losses, both by the reduced volume but also since it forces the engine to work at a higher load to produce the same power. Also the friction losses are reduced. However, reducing the engine size also reduces the engine’s maximum torque. This can be compensated for by utilizing a turbocharger, and can be understood through the ideal gas law. The amount of air in the cylinder is given by

$$m = \frac{pV_d}{RT}$$

(2.6)

where $p$ is the pressure, $T$ is the temperature and $R$ is the gas constant. A reduction in $V_d$ can be compensated for by an increase in $p$ to give the same amount of air available for the combustion.

Utilizing turbochargers for the purpose of downsizing was presented by General Motors in 1978 ([Wallace] 1978), but has not gained significant attention until the last decade (see, e.g., [Eichhorn et al.] 2012, [Han et al.] 2007, [Lecointe and Monnier] 2003, [Lumsden et al.] 2009, [Schernus et al.] 2011). One important reason for this is the introduction of direct injection (DI) and variable valve timing. Figure 2.4 illustrates the benefits of utilizing variable valve timing on a direct injected spark ignited (DISI) downsized engine. To the left, the maximum torque as function of engine speed is shown, together with the brake specific fuel consumption (BSFC) as function of both engine speed and load. The BSFC measures the amount of fuel that is needed to generate one unit of energy. The maximum torques are represented by the thick, solid lines and it shows that downsized, turbocharged engines have higher torque at low engine speeds than larger, natural aspirated engines. It also shows that direct injection and variable valve timing (red) improves the torque for lower speeds compared to a turbocharged engine with port fuel injection (PFI) and fixed valve timing (black). The thin, dashed contour lines show the BSFC as function of torque and engine speed. The natural aspirated engine (blue) has slightly higher maximum efficiency, but the downsized engine has higher efficiency over almost the whole operating range. Especially at low engine speeds and low to medium load, which are the most common conditions during normal driving, the downsized engine is always more efficient than the larger, natural aspirated engine. The figure to the right shows the transient torque response for the three engines. Utilizing direct
2.2. DOWNSIZING

Figure 2.4: Measured BSFC (left) and transient response (right) from a 2.6 liter natural aspirated engine and a 1.8 liter turbocharged DISI engine. From Figure 1 and 4 in Kleeberg et al. (2006). Reprinted with permission from SAE Paper No. 2006-01-0046 © 2006 SAE International.

injection and variable valve timing significantly improves the torque response, but there is still a large difference compared to a natural aspirated engine as shown by the hatched area.

Two reasons for direct injection and VVT to be efficient on downsized engines are the increased volumetric efficiency and the possibility of scavenging. The increase in volumetric efficiency comes partly from the fact that no fuel needs to be inducted. Moreover, the cooling of the air due to vaporization of the injected fuel increases the density and thereby the volumetric efficiency. In total, these effects gives around 9% increase in volumetric efficiency in experiments. Wyszynski et al. (2002) present both theoretical and experimental results on the effect on volumetric efficiency of direct injection. The stronger cooling also increases the resistance against knock, which allows for a higher compression ratio and thus increased efficiency of the torque generation. Direct injection also enables scavenging, where large valve overlap allows air to pass through the cylinder to the exhaust manifold. For a PFI engine, scavenging would mean that unburned fuel passes directly through to the exhaust manifold, which is harmful for the catalytic converter. On DI engines, however, the fuel is injected after the exhaust valve is closed and no fuel can pass directly to the exhaust manifold. With scavenging the will be less exhaust gases in the cylinder, and hence more torque can be produced. The lower amount of exhaust gases also lowers the temperature and thus reduces the knock sensitivity. Finally, scavenging gives higher mass flow, which helps the turbine to generate more power and speed up faster.
2.3 Engine Models

In a development process, models are useful. They can serve as a basis for model based controllers, and they can also be used to simulate and analyze the properties of the system. In this thesis, two types of models are utilized. The first is control-oriented models, the second is more advanced simulation models.

Control-oriented models are in general simpler, with only a few states and low computational complexity but still capturing the main dynamics of the system. Such models, as well as model based controllers, are presented in, e.g., Guzzella and Onder (2004); Kiencke and Nielsen (2005). Mean value engine models (MVEM) typically falls into this category. They describe the engine’s behavior as a continuous process rather than as series of discrete (combustion) events that occur every cycle. The signals in an MVEM are thus seen as mean values over one or several engine cycles. This thesis will use the MVEM concept for modeling and control of parts of the engine. MVEM for the whole engine with all components are presented in, e.g., Andersson (2005); Eriksson et al. (2012); Hendricks and Sorenson (1990); Müller et al. (1998).

For detailed simulation and analysis of the engine performance, MVEMs are not accurate enough. Instead, more advanced models capturing the discrete events like valve openings are used. This means that phenomena that occur during an engine cycle, like the pulsating pressure and flow in the manifolds, are captured. These models generally consider the length, diameter and curvature of the pipes and then solve partial differential equations for the intake and exhaust system. There is a distinction between 1-D and 3-D models, where the first ones considers the flow to be equal over the cross-sectional area and thus no turbulence is modeled. In this thesis, the 1-D simulation software GT-POWER (Gamma Technologies 2012) is used. It is based on the Navier-Stokes equations for the fluid dynamics in the pipes in the engine.
Chapter 3

Related Work

There is a vast literature on engines and turbochargers, where the books already mentioned on combustion engines by Heywood (1988) and turbocharging by Watson and Janota (1982) cover the fundamentals in a very thorough way. For more recent developments and applications, however, other sources exist. This chapter presents research results related to the papers included in the thesis.

3.1 Variable Geometry Turbine and Transient Optimization

As variable geometry turbines are mainly used on diesel engines, most of the literature is found in that area. An overview on the control problems on diesel engine is given by Guzzella and Amstutz (1998), for later references see, e.g., Wahlström (2009). A major difference is that on a diesel engine, the \( v_{GT} \) is needed to produce high enough exhaust pressure to push back the exhaust gases into the intake manifold through the exhaust gas recirculation (EGR) system. Some results, however, have relations to the work in this thesis and will be presented here. This section also presents results on investigations and optimization of turbocharged SI engines. Alternative ways to improve the transient response of supercharged engines by using, e.g., mechanical superchargers and/or electrically assisted turbochargers are not covered here. A good overview of such results is given in Westin (2005), and later references are found in Eriksson et al. (2012).

The control strategy for the \( v_{GT} \) on the SI engine in production by Porsche is not presented in detail, but it is briefly mentioned in Knirsch (2007). The \( v_{GT} \) is more closed at low loads and controlled depending on the mass flow and desired boost pressure at higher loads.

In Ericsson et al. (2010), optimization of the \( v_{GT} \) and \( v_{VT} \) for fast transient torque response on the same engine as the one used in this project is studied. The problem is addressed by a simulation study in a 1-D simulation environment. An extra bypass valve after the compressor is added to the model to bleed of the extra energy that accelerates the turbine. This makes it possible to simulate transient conditions at steady state. The transient is divided into four time segments, and
the best transient out of an initial screening of different settings of vgt position and valve overlap gives a reference case. For the optimization, the states of the model at the start of each segment are saved. These states are then simulated in steady state, and the effects of different vgt and cam settings are investigated. It is concluded that settings that maximize the product of volumetric efficiency and exhaust pressure in steady state, give good transient performance. The simulation study is done for one transient and is not evaluated on the engine. The optimized torque response compared to the reference case is shown in Figure 3.1.

In Kihar et al. (2007) a controller for boost pressure control using the wastegate is presented. The throttle controls the intake pressure when the requested pressure is below atmospheric, and in this case the wastegate is fully open. When the intake pressure request is higher than atmospheric, the throttle is fully open and the wastegate controls the pressure. The wastegate is controlled using a linear PD controller with a static nonlinearity. There is also a switching strategy for the reference to activate the waste gate controller at boosting conditions. This control scheme is evaluated on a 0.6 l turbocharged engine from the Smart car.

In Lezhnev et al. (2002), a control scheme for throttle and vgt on a direct injected si engine is presented. The control objective is to follow reference values on internal variables, e.g. \( p_{\text{int}} \) and \( p_{\text{exh}} \), that are given from a steady state calibration. Several control strategies are presented and tuned for transient response, all of them contain one siso loop each for vgt and throttle. The controllers are optimized for
3.1. VGT AND TRANSIENT OPTIMIZATION

Figure 3.2: Simulated transient response of a turbocharged SI engine at 2000 rpm. From Figures 16 and 17 in Lezhnev et al. (2002). Reprinted with permission from SAE Paper No. 2002-01-0709 © 2002 SAE International.

Fast boost pressure increase with no overshoot, and the Pareto optimal front for this tradeoff is presented. It is concluded that gain scheduled PI controllers with feedforward perform best. The whole study is carried out on a mean value engine model described in Buckland et al. (2000). Simulated step responses are shown in Figure 3.2. It is also concluded that having the VGT closed before the transient is beneficial for fast torque response.

Bozza et al. (2007) develop a model of a downsized DISI engine and evaluates the effect on transient performance of different turbochargers. Increasing the size of the inlet turbine housing gives slower boost pressure transient. A larger turbocharger with higher flow capacity also gives slower transient response with respect to boost pressure increase.

Lefebvre and Guilain (2005) develop a one-dimensional GT-POWER model of a turbocharged engine and compare the transient torque response to engine measurements. First, the authors evaluated the measurements on the engine test bed and concluded that measuring 10 transients per experiment and then using the time average of that was a good compromise between low variance and short experiment time. Moreover, they used 10 minutes between each transient for settling of all states and argued that the exhaust temperature was the most crucial state. Finally, they show that deviations in the equivalence ratio λ has a significant effect on the torque response. They also present ways to improve the model’s ability to predict the transient torque response. Two ways to significantly reduce the model errors are presented. The first one is to let the combustion modeling parameters of the Wiebe function, $c_{A50}$ and burn duration, to be dependent of engine speed and load. The second improvement is made by developing an in-cylinder heat transfer model that is dependent on engine speed and load.
Kleeberg et al. (2006) compare VVT control strategies for fast transient response of a turbocharged, DI engine. Two strategies are compared, both with constant cam settings during the whole transient. The first strategy has large overlap, while the second has a smaller overlap. Figure 3.3 shows the transient torque response for these two cases, compared to an engine with fixed cam timing. The larger overlap gives lower torque at the start of the transient, but has a faster torque increase later in the transient and reaches maximum torque earlier. This is explained by the fact that early in the transient, the pressure ratio over the engine is not favorable for scavenging and thus there will be a significant amount of residual gases in the cylinder. This in turn requires late spark ignition which gives lower produced torque but higher energy in the exhaust and thus higher turbine speed. Furthermore, they show that the intake temperature has a significant effect on the transient response.

In Ushida (2006), a one dimensional model is used to evaluate transient performance of a turbocharged SI engine. The engine is equipped with both VGT and variable inlet guide vanes (VIGV) for the compressor. An open-loop strategy is presented, where the VIGVs are kept constant during the first 0.6 s of the transient and then ramped down to zero at 2 s. How the VGT is controlled is not mentioned. This control gives a faster boost pressure buildup, with about 0.15 bar higher boost pressure after 2 s.

Simulation studies of utilizing optimal control techniques (LQ and MPC) for tran-
sient responses on a turbocharged engine with wastegate are done in both Kristoffersson (2006) and Bloisi and Argolini (2007). The result is that it is beneficial in some operating points and load changes to have the wastegate opened during transients. The benefits of opening the wastegate were only present at low loads and/or for improved fuel consumption. Both studies were carried out on an MVEM presented in Andersson (2005).

In Eriksson et al. (2002), fuel optimal control is compared to driveability optimal control for an SI engine with a wastegate turbocharger. The fuel optimal controller keeps the wastegate open as much as possible to reduce pumping losses, achieving a fuel saving of about 2-4%. The driveability optimal controller keeps the wastegate closed as much as possible and only opens the wastegate to prevent the boost pressure from exceeding its maximum level. This gives higher exhaust pressure and thereby higher turbine speed during normal driving conditions. Simulations show that the response time, defined as the time to reach 90% of maximum torque, can be reduced by about 0.5 s for normal driving conditions.

This concept is extended to an SI engine with VGT in Eriksson et al. (2012). Therein, a component based mean value model is developed to evaluate different hardware configurations for boosting SI engines. In a simulation study, a transient optimal strategy with the VGT always fully closed is compared with a fuel optimal strategy with the VGT always fully open. This gives 16% higher fuel consumption in the NEDC, but also a reduced response time of 1 s. The response time for the fuel optimal controller with electric power assist to the turbine is reduced with about 0.3 s. The simulated transient responses are shown in Figure 3.4.

Navrátil et al. (2004) develop a one-dimensional simulation model in GT-POWER...
Figure 3.5: To the left, the control scheme for the VGT is shown. The right plot shows the improved boost pressure increase by the control scheme compared to a fully open VGT. From Figures 8 and 11 in Ito et al. (2007). Reprinted with permission from SAE Paper No. 2007-01-0263 © 2007 SAE International.

of an SI engine with turbocharger and wastegate. The wastegate is controlled by feed-forward from the pedal position, keeping the wastegate fully open for pedal position up to 50%. The wastegate position is then ramped down to fully closed for 100% pedal position. There is also a mechanism that opens the wastegate if the boost pressure reaches its upper limit. This engine is then compared with a 33% larger, naturally aspirated engine in both acceleration and fuel consumption. The downsized, turbocharged engine has 1.1 s faster 0-100 km/h acceleration and 24% lower fuel consumption in the New European Driving Cycle (NEDC).

Andersen et al. (2006) present an experimental evaluation of six different VGT turbochargers on a two liter SI engine. Five of them uses variable nozzles, i.e. are of the VNT type. The sixth has two scrolls connected by fixed vanes. A flap can direct the flow to the smaller, inner scroll only or open up and also allow flow through the outer scroll. A matching check for turbine flow capacity and compressor efficiency is carried out to evaluate the six turbochargers’ on-engine performance. This matching then acts as a reference when comparing the results from the evaluation. The VGTs are compared with a fixed turbine in terms of low speed performance, transient response, maximum power and cold start emissions. The transient evaluation was made with fixed positioning of the VGT. They conclude that a VGT can increase the torque with around 10% at both low and high engine speeds, having equal transient performance. However, the increased temperature drop over the VGTs caused longer time for the catalytic converter to reach its efficient working range. Out of the tested VGTs, the one with two scrolls and a flap
Table 3.1: Table of measured time to reach different boost pressure levels during full acceleration starting from different engine speeds in a production car. Data from Moody (1986).

<table>
<thead>
<tr>
<th>Test scenario</th>
<th>Time to reach boost pressure [s]</th>
<th>Production turbo</th>
<th>VGT turbo</th>
<th>Reduction (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Boost (psi)</td>
<td>Speed (mph)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>7.5</td>
<td>0</td>
<td>1.62</td>
<td>1.22</td>
<td>25</td>
</tr>
<tr>
<td>7.5</td>
<td>10</td>
<td>1.13</td>
<td>0.60</td>
<td>47</td>
</tr>
<tr>
<td>7.5</td>
<td>20</td>
<td>1.06</td>
<td>0.60</td>
<td>43</td>
</tr>
<tr>
<td>7.5</td>
<td>30</td>
<td>0.98</td>
<td>0.88</td>
<td>10</td>
</tr>
<tr>
<td>9.0</td>
<td>0</td>
<td>1.76</td>
<td>1.36</td>
<td>23</td>
</tr>
<tr>
<td>9.0</td>
<td>10</td>
<td>1.37</td>
<td>0.71</td>
<td>48</td>
</tr>
<tr>
<td>9.0</td>
<td>20</td>
<td>1.22</td>
<td>0.72</td>
<td>41</td>
</tr>
<tr>
<td>9.0</td>
<td>30</td>
<td>1.11</td>
<td>0.96</td>
<td>14</td>
</tr>
<tr>
<td>12.0</td>
<td>0</td>
<td>-</td>
<td>1.61</td>
<td></td>
</tr>
<tr>
<td>12.0</td>
<td>10</td>
<td>-</td>
<td>0.92</td>
<td></td>
</tr>
<tr>
<td>12.0</td>
<td>20</td>
<td>-</td>
<td>0.88</td>
<td></td>
</tr>
<tr>
<td>12.0</td>
<td>30</td>
<td>-</td>
<td>1.04</td>
<td></td>
</tr>
</tbody>
</table>

to control the flow was the most suited for SI application.

Ito et al. (2007) present a control scheme for an SI engine with VGT. This particular VGT has fixed nozzle vanes, but a valve that can direct the flow in an inner scroll (inside the vanes) directly, or to both an outer and inner scroll. The valve is controlled in three zones in the engine speed and $p_{int}$ plane, see Figure 3.5. For low speeds and low pressures, the VGT is closed and it is opened up for higher speeds and loads. Transient evaluation of this control scheme is performed in a prototype vehicle by comparing with keeping the VGT fully open throughout the transient. For a first gear WOT acceleration, the time to reach maximum boost pressure was reduced from around 3.1 to 1.5 s. The activated VGT controller also gives higher vehicle acceleration between 1 and 2.5 s. They also modify the torque converter to allow a faster engine speed increase at the start of the transient. This increases the mass flow and helps reduce the turbo lag.

Moody (1986) replaces the original turbocharger of a 3.8 liter V6 production engine with a VGT. The trade off between building boost pressure and not having too high exhaust pressure is presented, and it is concluded that it is not beneficial to close the VGT completely. The nonlinear relation between vane position and boost pressure is identified as a major difficulty, and is handled by gain scheduling the control gain according to the size of the boost control error. A steady state full load test in an engine test bed shows that the VGT can improve the torque by around 10% over the whole engine speed range. The transient response is evaluated through acceleration tests in a vehicle. The car is accelerated from constant speeds of 0, 10, 20 and 30 mph and evaluation is performed by measuring the time to build boost pressure of 7.5 psi ($\approx 0.52$ bar) and 9 psi ($\approx 0.62$ bar). With the VGT, the
Figure 3.6: Simulated transient responses from Xu et al. (2011). The y-axis shows torque, the x-axis is time. ① is a 1.5 liter natural aspirated engine, the others are from a 1.0 liter supercharged engine. ② is a standard turbocharger, ③ is standard turbocharger with 40 CAD valve overlap, ④ uses an additional mechanical supercharger in series with the standard turbocharger and ⑤ a VGT turbocharger that is closed to 40% at the start of the transient. © 2011 IEEE.

time to build boost is reduced by 10-48%. The VGT was operating in open-loop and kept constant at a position near fully closed during the transient. The data from the experiment is presented in Table 3.1.

Xu et al. (2011) perform a simulation study in GT-POWER of the effect of valve overlap and boosting technologies on the transient torque response. The evaluation is conducted on a 1.0 liter turbocharged engine. Firstly, the effect of constant valve overlap is investigated by simulating transients with 0, 10, 20, 30 and 40 CAD overlap. The transient torque response is faster with increasing overlap. Moreover, the effect of using a VGT turbocharger is studied. Three simulations are performed, where the VGT is fully open (100%) at the start of the transient and then kept open and closed to 60% respectively 40%. Closing to 40% gives the fastest torque response. A mechanical supercharger in series with the standard turbocharger is also simulated, and this gave the fastest transient response. A comparison of the boosting technologies is shown in Figure 3.6.

3.1.1 Related work on Diesel Engines

Control of the VGT with model inversion on a diesel engine is presented in Wahlström and Eriksson (2010). This is done in a MIMO setup, where both the turbine flow and the EGR flow models are inverted to form static nonlinear compensators to be used together with two linear PID controllers.

Another diesel application is found in Buratti et al. (1997). There, SISO boost
control design using VGT actuation on a passenger car diesel engine is developed. At steady state, a gain scheduled PI controller is used for boost pressure control. It is only the total gain of the controller that changes, the ratio between the integral and proportional gains is kept constant. When a transient is detected, through a threshold on the first order derivative on the boost pressure, the control is switched to a PD controller. Validation is performed in an experiment where the load is too low to achieve the reference boost pressure. A step in load large enough to achieve reference boost pressure is then analyzed, and the switching controller performs much better than the PI. See Figure 3.7.

Alberer and del Re (2009) perform optimization of the transient response on a diesel engine. The objective is to minimize deviations in torque, $\text{NO}_x$ and PM from reference values using the controlled inputs fuel, VGT position and EGR valve. One specific load step, from 4 to 9 bar BMEP at 1650 rpm engine speed is considered. The optimization problem is solved numerically, and measurements at the test bench are used for each function evaluation. The objective function is computed using measured emissions and torque, averaged over a few transients. Hence few function evaluations is desirable to reduce experiment time and assure similar conditions throughout the optimization. Hence the open-loop trajectories are parameterized in steps from 6000 values initially down to three. Through the optimization, the emission peaks in the transient is eliminated while the torque response is preserved. Since it is not possible to have the emissions as the control outputs, intermediate control targets are suggested. These control targets are the in-cylinder oxygen concentrations before and after combustion, respectively. Based on oxygen measurements in the exhaust manifold and a volumetric efficiency model these are also found through the optimization. In Alberer (2009), the oxygen concentration trajectories are used as references in a MPC implementation on the engine. Hence a
feedback implementation is found, however, whether these oxygen trajectories are optimal for other transients is not evaluated.

### 3.2 Gas Exchange Modeling

[Smith et al. (1999)] use a 1-D simulation model to show that the steady state volumetric efficiency models are valid also during transients in engine speed, and thereby engine flow. [Öberg and Eriksson (2007)] investigate the performance of residual gas models during cam phasing transients using 1-D simulations. The standard deviation of the model error is increased with 3–17% during the change of cam phasing compared to steady state conditions. The usefulness of black box models for volumetric efficiency is discussed in [Nicolao et al. (1996)].

[Turin et al. (2008)] develop a volumetric efficiency model based on energy conservation during the period when the intake valve is open. The model considers both variable valve lift and variable valve timing. Effects of the exhaust pressure are not included; it is assumed to be atmospheric. The model has twelve parameters and is calibrated and evaluated using data from a simulation model. Around 70% of the validation points have less than 5% error, and 90–98% of the points (depending on valve lift profile) have less than 10% error.

[Stefanopoulou et al. (1998)] present an MVEM with a polynomial volumetric efficiency model for a natural aspirated engine with constant overlap but varying overlap center. The model is a polynomial, third order model in overlap center offset, intake pressure and engine speed. The model has 21 parameters and covers overlap center offset from 0°–35° after top dead center (TDC) and engine speed up to 2000 rpm. The volumetric efficiency model is not validated separately, but only the whole MVEM where the airpath time constants agree well with measurement data. An evaluation of the effect of the cam timing on air mass flow and thereby torque is also performed. It is concluded that, within the studied range, overlap center at TDC gives highest mass flow.

[Yi et al. (2004)] investigate the effect of intake cam phasing on the trapped mass in the cylinder of a DISI engine. A 3-D computational fluid dynamics simulation model is used for the investigation. The study is performed for four values of intake valve opening (IVO): -30°, -20°, 0° and 20° offset from TDC and constant intake pressure of 0.62 bar. Earlier IVO gives monotonically increased trapped mass in the cylinder. The trapped fresh air, however, has a maximum between -20° and 0° IVO. The authors explain the reason for this as the cause of two counteracting effects. First, a later IVO means that there is less residual gases being pushed out of the cylinder early in the induction process. Thus more fresh air can enter the cylinder later in the induction process. Second, a late IVO means that fresh air will be pushed out of the cylinder during the end of the induction phase, just before the intake valve closes.

[Colin et al. (2009)] develop two models for the gas exchange process. The first model describes scavenging and residual gases, while the second describes the vol-
3.2. GAS EXCHANGE MODELING

Figure 3.8: Volumetric efficiency evaluation. Left is the scavenged air (negative values) or residual gases (positive values) in mg as function of valve timings. This is for $N_{\text{eng}} = 2000$ rpm and $p_{\text{int}} = 1.4$ bar. To the right, the air mass and the estimated error is shown. Reprinted from Figures 3 and 7 in Colin et al. (2009). © (2008), with permission from Elsevier.

Volumetric efficiency. The first model considers the two cases where either residual gases are trapped or there is scavenging. It is a neural network model with one hidden layer, twelve neurons and four regressors: Intake pressure, engine speed, intake and exhaust cam timing. This gives 73 parameters in total. As the scavenging and trapped residual gases cannot be measured in the engine, a 1-D model from Berr et al. (2006) is used for calibration and evaluation. The model is shown to the left in Figure 3.8 and has a mean absolute error of 9.6% on validation data. The volumetric efficiency model is also a neural network model with one hidden layer, six neurons and four regressors: Intake pressure, engine speed, intake and exhaust cam timing. This gives 37 parameters in total, and validation with measured data gives a mean absolute error of 2.3% and a maximum absolute error of 12%. The evaluation, compared both to measured data and data from the simulation model, is shown to the right in Figure 3.8. Leroy et al. (2008) present a mean value model for the air flow through the intake valves for a turbocharged engine with VVT. The model is based on three components. These are the cylinder filling, remaining gases from previous combustion and a term capturing scavenging or backflow of residual gases. The model is validated using measurements, see Figure 3.9.
CHAPTER 3. RELATED WORK

3.3 Summary

For control of the VGT on SI engines, almost all proposed schemes use only feedforward. There are many results showing that proper use of both VGT and VVT can improve the transient response, however, most of the them focus on fast boost pressure increase and not the torque response. Moreover, few papers use optimization to achieve a fast response; it is more common to compare open-loop controllers. For diesel engines, a controller activated only during the transient is presented, as well as an optimization of the transient response with respect to both torque and emissions.

Models of volumetric efficiency, residual gas fraction and scavenging for SI engines covering the effect of VVT generally has mean errors in the range 2 to 5%, utilizing twelve to around 70 parameters. Moreover, the scavenging phenomena is shown to be much more sensitive in the overlap direction, than in the overlap center direction. The steady state volumetric efficiency might differ from the volumetric efficiency during cam phasing transients.
In this chapter, the contributions of the thesis is summarized. First, a list of the papers appended are presented together with previously published papers on related topics. Second, a brief discussion relating the contributions to the published results presented in Chapter 3 are given. Finally, the last three sections present a summary of each of the three appended papers.

4.1 List of Papers

The first paper concerns optimal transient torque response.


Paper 2 presents models and feedback controllers for the exhaust pressure.


Initial results in this direction are available in the following conference paper.


Paper 3 presents models for the gas exchange process and its application for torque prediction.

Parts of those results are presented in the conference publication


The following paper on gas flow modeling is also published, but is not included in the thesis.


### 4.2 Relation to Previous Work

Many of the papers addressing the transient torque response on SI engines focus on fast boost pressure response. However, higher boost pressure does not always give higher torque as shown in Figure 4.1. There, two vgt control strategies are compared. One that has the vgt almost fully closed (dashed) and one that only closes to about 60% (solid). Clearly, a more closed vgt gives higher boost pressure but also significantly lower torque. This is due to the high exhaust pressure, which lowers the volumetric efficiency. Hence, to achieve a fast torque response a good tradeoff between obtaining boost pressure increase and not having to high exhaust pressure is needed. Moreover, only a few papers use optimization techniques to achieve good transient performance, and almost all utilize open-loop trajectories. To the authors knowledge, there are no contributions where feedback control based on optimization is considered.

### 4.3 Paper 1 - Transient Optimization

Paper 1 addresses the problem with the slow transient torque response for turbocharged, downsized engines. The focus is on air path control, injection and ignition is not considered.

#### 4.3.1 Main Contributions

There are three main contributions in this paper. The first is solving the optimization problem of maximizing the torque response. A model based approach is pursued to find the optimal control of the variable valve timing (VVT) and the vgt during a torque transient. The goal of the optimization is to maximize the torque integral. This optimization problem is solved for three different engine speeds.

The second key contribution is to design a feedback controller that resembles the optimal control for any transient. Since several optimization problems are solved,
Figure 4.1: Measured torque responses for two VGT control strategies. Even though a more closed VGT (dashed lines) gives higher boost pressure, the torque response is lower due to the higher exhaust pressure.

generic properties of a fast torque response are identified. By only activating the feedback controller during the transient, the steady state settings are not affected and hence the steady state fuel consumption is not affected. Instead, the best way to control the engine between two states, from low load to full load, is sought for.

The third key contribution is that the optimal and feedback controllers are evaluated on an engine mounted in a test bench. While the model based optimal inputs do not always give good performance due to model errors, the feedback controller achieves fast transient response for all three engine speeds.

4.3.2 Experimental Setup and Modeling

The base engine is a GM L850/Ecotec SI engine with a cylinder head from the LNF engine series. It has direct injection (DI), variable valve timing (VVT) and the original twin scroll turbocharger is replaced with a Mitsubishi VGT turbocharger (MHI TD04HL-VG turbine and a TD04H-15TK31 compressor). This VGT has variable nozzles, i.e., it is of the VNT type.
The engine is controlled by an open source engine control system, which is fully described in Backman (2011). It is specifically designed for rapid prototyping of new control algorithms and has complete control of the engine, i.e. there is no bypass. The engine control system has been calibrated for lowest brake specific fuel consumption in steady state. There are six degrees of freedom: vgt-position, intake cam position, exhaust cam position, fuel pressure, start of injection and ignition timing that have to be optimized for each load point. Ignition timing is determined by cylinder pressure feedback to control CA50 to 10 CAD after TDC, together with a spark retard functionality to prevent knocking. For the other five parameters, feedforward maps depending on engine speed and requested air flow are calibrated. The air flow control signals for the cams and vgt are optimized first, with the fuel pressure and start of injection optimized in a second step. The air path parameters are then optimized again, continuing in an iterative manner until the change in control signal is small enough. The maps are also smoothed to not vary too much between neighboring operating points.

For this project a one-dimensional engine model of the engine in GT-POWER is used. It is calibrated using steady state measurements in the range 3 bar imep to wide open throttle (WOT) and 1500 to 3000 rpm engine speed, as well as load transients with different settings of the vgt and vvt.

4.3.3 Problem Formulation and Optimization

The problem that studied is how to control the valve timing and the vgt to achieve fast transient torque response at fixed engine speed. To reduce the complexity, the cam timing is only varied in the overlap direction, \( \phi \), and the center of the overlap is always kept at TDC. The fuel amount is set to keep \( \lambda = 1 \) throughout the transient and the injection and ignition timings are determined by the base calibration.

The considered torque is IMEP + PMEP, which here is denoted IMEP720 since it corresponds to the indicated torque during one cycle. The objective function in the optimization is the torque integral

\[
\max_{\text{vgt}, \phi} \int_0^{t_{\text{opt}}} T(\text{vgt}(t), \phi(t)) \, dt
\]  

(4.1)

for the first \( t_{\text{opt}} \) s of a torque reference step from three bar IMEP720 to full load. Since the engine speed is kept constant during the transient this integral corresponds to the produced work during the first \( t_{\text{opt}} \) seconds of the transient. The optimization horizon \( t_{\text{opt}} \) is chosen to 1.5 s, which is considered as a good tradeoff between short term torque gains using fast pressure dynamics and long term torque gain by the slower turbo dynamics.

To reduce the number of optimization parameters, the input trajectories are optimized at specific time instances and linear interpolation is used for determining the input signal between these samples. The selected time instances are \( t = [0 \ 0.1 \ 0.2 \ 0.4 \ 0.6 \ 0.8 \ 1.0 \ 1.2 \ 1.4] \) seconds after the requested load step.
4.3. PAPER 1 - TRANSIENT OPTIMIZATION

The optimization problem (4.1) can not be solved analytically, due to the complexity of the model relating the input to the torque integral. Thus an iterative, gradient-free approach using the Matlab function fmincon is performed. The optimization problem (4.1) converges very slowly, why the following optimization problem is solved instead

$$\min_{v_{gt}(t_i), \varphi_{ol}(t_i)} \int_0^{1.5} (T_{\text{ref}} - T(v_{gt}(t), \varphi_{ol}(t)))^2 \, dt$$

s.t. $0 \leq v_{gt}(t_i) \leq 95$, $\forall i$

$0 \leq \varphi_{ol}(t_i) \leq 78$, $\forall i$

(4.2)

with the torque reference being 22 bar IMEP720. This optimization gives almost identical results as (4.1).

Optimization Results

The resulting torque responses for the three engine speeds are shown in Figure 4.2. The torque gains become significant just before 1 s and then increase up until 1.5 s. There is also a small bump in the torque just before 1.5 s, after which the torque flattens out. This is most likely an effect of that the optimization is only performed up until 1.5 s, and a quick gain from reduced pumping losses is favorable over increasing the boost pressure. This is achieved by opening the VGT in the end of the transient, see Figure 4.3. Figure 4.4 shows the overlap during the transients, which is kept at maximum almost throughout the whole transient. The objective function functions values as well as the torque integral are shown in Table 4.1.
Figure 4.3: Initial and optimal VGT trajectories. Optimized values are solid, the base calibration is dotted.

Figure 4.4: Initial and optimal overlap trajectories. Optimized values are solid, the base calibration is dotted.
4.3. **PAPER 1 - TRANSIENT OPTIMIZATION**

<table>
<thead>
<tr>
<th></th>
<th>1500 rpm</th>
<th>1750 rpm</th>
<th>2000 rpm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Initial Objective $J_2$</td>
<td>186</td>
<td>176</td>
<td>156</td>
</tr>
<tr>
<td>Optimized Objective $J_2$</td>
<td>175</td>
<td>1621</td>
<td>141</td>
</tr>
<tr>
<td>Initial torque integral $J$</td>
<td>17.1</td>
<td>17.6</td>
<td>18.7</td>
</tr>
<tr>
<td>Optimized torque integral $J$</td>
<td>17.8</td>
<td>18.7</td>
<td>20.1</td>
</tr>
<tr>
<td>Iterations</td>
<td>23</td>
<td>29</td>
<td>30</td>
</tr>
<tr>
<td>Function evaluations</td>
<td>486</td>
<td>585</td>
<td>612</td>
</tr>
</tbody>
</table>

**Table 4.1:** Initial and optimized objective values, torque integral and number of iterations and function evaluations for the three different transients when solving (4.2) on the GT-POWER model.

![Figure 4.5](image-url)

**Figure 4.5:** The ratio $p_{exh}/p_{int}$ during the simulated transients. Optimized values are solid, the base calibrations are dotted.

### 4.3.4 Feedback Control Design

By analyzing the results from the optimization, a feedback strategy is found. In Figure 4.5, the ratio between exhaust and intake pressure, $p_{exh}/p_{int}$, for the simulated optimal (solid) and default trajectories (dotted) are shown. For the optimal transients, this ratio is almost constant at about 1.2 while it for the base calibration drops to around 1 at the end of the transient. This gives that the pressure ratio $p_{exh}/p_{int}$ should be kept at a constant value $K$.

The overlap during the optimized transients do not differ that much from the base calibration, and the difference when considering the actuator dynamics on the engine are even smaller. Hence the base calibration is used for the cam timing control. The overlap center for the base calibration is 15 CAD before TDC at the
start and 3 CAD after TDC during the maximum overlap part of the transient.

The opening of the VGT and reduction of overlap seen from the optimization is not included in the feedback. This is due to the fact that they occur to gain extra short term torque, and are hence only important when the reference torque is obtained. This is verified by performing another optimization with horizon $t_{opt} = 1.7$ s, resulting in these control actions being delayed with 0.2 s.

4.3.5 Evaluation

When evaluating the feedback control strategy on the engine, the control objective, keep $p_{exh} / p_{int} = K$, is reformulated as letting $Kp_{int}$ be reference for $p_{exh}$. This reference can then be tracked using the input-output model inversion controller from Paper 2.

<table>
<thead>
<tr>
<th></th>
<th>1500 rpm</th>
<th>1750 rpm</th>
<th>2000 rpm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Base calibration</td>
<td>19.6</td>
<td>19.5</td>
<td>20.0</td>
</tr>
<tr>
<td>Model based optimal</td>
<td>18.7</td>
<td>19.7</td>
<td>21.2</td>
</tr>
<tr>
<td>Feedback</td>
<td>20.0</td>
<td>20.3</td>
<td>21.1</td>
</tr>
</tbody>
</table>

**Table 4.2:** Measured torque integral on the engine for the base calibration, the trajectories from the model based optimization and the feedback trajectories.

The torque responses for the base calibration, the model based optimal and the feedback controllers are all compared at 1500 rpm, 1750 rpm and 2000 rpm respectively. The exhaust pressure $p_{exh}$ was controlled to $1.15 p_{int}$ for all three engine speeds. Note that the best value of $K$ is close to 1.15 for all engine speeds, see Figure 4.6. The torque integral for the transients evaluated on the engine are presented in Table 4.2. The feedback controller gives good results and is only beaten by the model based optimal input at 2000 rpm.

The torque response at 1500 rpm is shown in Figure 4.7. The model based optimal controller gives a huge dip in the torque after 0.8 s when the VGT is closed (c.f. Figure 4.3). The reason for this is that $p_{exh}$ increases much more on the engine than on the model when closing the VGT in this case, see Figure 4.8 and compare with Figure 4.5. It is clear that the feedback controller manages to keep the pressure ratio close to the desired 1.15 during the whole transient, which gives a fast and steady torque increase.

Figure 4.9 shows the torque response at 2000 rpm. Both the model based and the feedback controllers are significantly better than the base calibration. Also note that, just before 1.5 s, the model based controller gets a little extra torque from opening the VGT (c.f. Figure 4.3). At 1750 rpm, the transient responses are similar to the ones at 2000 rpm. The major difference is that the model based controller has a torque dip at around 0.8 s, corresponding to the dip at 1500 rpm but smaller.
Figure 4.6: IMEP720 integral as function of pressure ratio reference $K$ for different engine speeds. The best transient response for all engine speeds is given by $K \approx 1.15$.

Figure 4.7: Torque response for the base calibration, the model based optimal and the feedback at 1500 rpm.
Figure 4.8: The pressure ratio $p_{\text{exh}}/p_{\text{int}}$ for the base calibration, the model based optimal and the feedback at 1500 rpm.

Figure 4.9: Torque response for the base calibration, the model based optimal and the feedback at 2000 rpm.
To summarize, the feedback controller performs well in all three transients and is clearly better than the base calibration. The model based optimal feedforward controller performs bad on 1500 rpm, better on 1750 rpm and is the best one at 2000 rpm. Model discrepancies prevents the model based optimal controller to perform better at all engine speeds. Note also that the model based optimal feedforward controller opens the VGT just before 1.5 s to gain extra short term torque, neither the feedback nor the base calibration does that.

### 4.4 Paper 2 - Exhaust Pressure Modeling and Control

The work in this paper is motivated mainly by two purposes. First, the transient optimization in Paper 1 shows that it is important with accurate feedback control of the exhaust pressure during load transients. Second, Paper 3 shows the importance of the exhaust pressure for the engine breathing process. Hence, a model for the exhaust pressure is beneficial for cylinder air charge estimation.

#### 4.4.1 Main Contributions

The main contributions of this paper are twofold. First, a model describing the turbine flow on the engine using only four parameters is proposed. Second, a model based nonlinear control structure is presented which can be tuned using the two parameters of a PI controller.

The model for the turbine flow is a nonlinear function of VGT position and exhaust pressure. It is shown to capture the behavior over the engine operating range as well as the range of the VGT. Together with first order linear dynamics, this describes the exhaust pressure. The model can be identified using stationary measurements as well as dynamic data and is utilized in the controller.

The basic idea of the controller, presented in Flärdh and Mårtensson (2010), is to first find the mapping from states and input to output, and then invert the dependence between the input and the output. Feedback is then added to compensate for model errors. The controller is tuned with two parameters, corresponding to the proportional and integral gain of a PI controller. A feedback linearization controller for the same problem is also presented and evaluated, as well as the relation between the IOMI controller and a feedback linearization controller.

Both the model and the controller are illustrated and validated with good results using simulations and experiments on an engine mounted in a test bench.

#### 4.4.2 Modeling

Two models for the exhaust pressure, $p$, are presented, both based on the relation

\[
\begin{align*}
p[k+1] &= p[k] + \beta (w_e[k] - w_t[k]) \\
y[k] &= p[k]
\end{align*}
\]  

(4.3)
where $k$ is the time index, $w_e$ is the mass flow through the exhaust valves, $w_t$ describes the turbine flow and $\beta$ is a parameter depending on the gas constant, the volume of the exhaust manifold, the exhaust temperature and the sampling time.

The turbine flow is modeled as

$$w_t = \alpha_1 (p - 1)^{\alpha_2} \ln(\alpha_3 u + \alpha_4)$$

(4.4)

where $\alpha_i$, $i = 1, \ldots, 4$ are parameters and $u$ is the position of the vanes in the VGT. This model structure is validated in Figure 4.10. In the left plot, the exhaust pressure is plotted as a function of VGT position for ten operating points ranging from 1750 rpm and 1 bar BMEP to 3000 rpm and 11 bar BMEP. It is clear that the static relation between pressure and VGT position varies with the operating point. The plot to the right, however, shows the scaling that follows from the structure (4.4). Except for three operating points with the lowest mass flow, the fit is very good. Hence this is a way of parameterizing the on-engine turbine flow for a VGT turbine using only four parameters.

Utilizing the model (4.4) for the turbine flow, the first order exhaust pressure model becomes

$$p[k + 1] = p[k] + \beta(w_e[k] - \alpha_1 (p[k] - 1)^{\alpha_2} \ln(\alpha_3 u[k] + \alpha_4))$$

$$y[k] = p[k]$$

(4.5)

This model assumes that the VGT actuator is ideal. By assuming first order linear
actuator dynamics, the following second order system is obtained

\[
\begin{align*}
x_a[k+1] &= \zeta x_a[k] + (1 - \zeta) u[k] \\
p[k+1] &= p[k] + \beta (w_e[k] - \alpha_1 (p[k] - 1)^{\alpha_2} \ln(\alpha_3 x_a[k] + \alpha_4)) \\
y[k] &= p[k]
\end{align*}
\]

(4.6)

where \(x_a\) is the actual VGT position and \(\zeta\) is the time constant of the actuator.

Both the presented models are estimated and validated using dynamic data. A part of the validation data is shown in Figure 4.11 which displays the measured and simulated pressure for both models. As seen in the figure, both models captures the dynamic behavior well and also captures the absolute level. The models are compared by the fit value which is defined as

\[
1 - \frac{\|y - \hat{y}\|_2}{\|y - \text{mean}(y)\|_2}
\]

(4.7)

where \(y\) is the measured output and \(\hat{y}\) is the modeled output. The fit for the models are 55% for the first order model and 64% for the second order model.

The residual of these models are well described by 5th order ARX models, giving almost white residuals. Including these models increases the fit to 64% and 67% for the first and second order model respectively, and hence the main dynamics is described by the presented nonlinear models.
4.4.3 Control Design

The IOMI control design is based on the idea of inverting the input-to-output mapping. It is designed for a SISO nonlinear system

\[ x[k + 1] = f(x[k], u[k]) \]
\[ y[k] = g(x[k]) \]  

(4.8)

with states \( x \), input \( u \) and output \( y \). The goal is to track an output reference \( r \). For a system with relative degree \( N \geq 1 \), a mapping \( Y \) from states and input to output is defined as

\[ y[k + N] = Y(x[k], u[k]) \]  

(4.9)

If this mapping is invertible with respect to \( u \) for all \( x \) in the domain of interest, a controller can be formed as

\[ u[k] = \hat{Y}_u^{-1}(\hat{x}[k], z[k]) \]  

(4.10)

where the hats denote the controllers model and state estimate of the true system. With \( z = r \), perfect tracking is achieved if there are no model errors. However, by using

\[ z[k] = \hat{r}[k] - (y[k] - \hat{Y}(\hat{x}[k], u[k - 1])) \]  

(4.11)

with

\[ \hat{r}[k] = y[k] + K_i(r[k] - y[k]) + K_p(r[k] - y[k] - (r[k - 1] - y[k - 1])) \]  

(4.12)

both compensation for model errors and tuning is introduced.

For a specific system structure which is affine in the input

\[ x[k + 1] = Ax[k] + B\gamma(x[k])u[k] \]
\[ y[k] = Cx[k] \]  

(4.13)

the IOMI controller becomes a PI controller. For a system on the form (4.13) with relative degree \( N \), it holds that the first \( N - 1 \) impulse response coefficients are zero: \( CA^jB = 0, j = 0 \ldots N - 2 \). This gives the controller

\[ u[k] = u[k - 1] + \frac{1}{\hat{CA}^{N-1}\hat{B}\hat{\gamma}(\hat{x}[k])} (\hat{r}[k] - y[k]) \]
\[ = u[k - 1] + \frac{1}{\hat{CA}^{N-1}\hat{B}\hat{\gamma}(\hat{x}[k])} (K_i e[k] + K_p (e[k] - e[k - 1])) \]  

(4.14)

with \( e[k] = r[k] - y[k] \). Hence this control scheme gives a PI controller whose gain varies with \( 1/\hat{\gamma}(\hat{x}[k]) \).

Moreover, this control structure has several implementation advantages. It has anti-windup and bumpless transfer at mode switches, as well as parameter changes during steady state.
4.4. PAPER 2 - EXHAUST PRESSURE MODELING AND CONTROL

Figure 4.12: Simulations of steps in exhaust pressure reference with the iomi controller using one state model. The parameters are $K_p = 0.2, K_i = 0.01$ (solid black), $K_p = 0.4, K_i = 0.01$ (solid grey) and $K_p = 0.2, K_i = 0.025$ (dotted black). The system response corresponds to the proportional and integral gains of a PI controller.

**IOIMI Control Design**

The IOIMI control design is applied to the two presented exhaust pressure models. To have more compact expressions, the functions $\gamma(x) = \alpha_1 (x-1)^{\alpha_2}$ and $h(u) = \ln(\alpha_3 u + \alpha_4)$ from (4.4) that describes the turbine mass flow are introduced. The controller based on the first order model is given by

$$u[k] = Y_u^{-1}(x[k], \tilde{r}[k] - (y[k] - Y(x[k], u[k-1])))$$
$$= e^{\beta \alpha_1 (x[k]-1)^{\alpha_2}} u[k-1] + \frac{\alpha_4}{\alpha_3} \left( e^{\beta \alpha_1 (x[k]-1)^{\alpha_2}} - 1 \right)$$  \hspace{1cm} (4.15)

with $\tilde{r}[k]$ taken from (4.12).

The IOIMI controller for the second order model (4.6) considers the mass flow from the engine $w_e$ as a measurable disturbance. The control law is

$$u = e^{\beta \gamma(x_e - \beta \gamma(x_e)h(x_a) + \beta w)} u[k-1] + \frac{\zeta \alpha_4 x_a}{(1 - \zeta) \alpha_3} \left( e^{\beta \gamma(x_e - \beta \gamma(x_e)h(x_a) + \beta w)} - 1 \right)$$  \hspace{1cm} (4.16)

where the time index $[k]$ has been omitted for all variables except for $u$ on the right hand side.

Even though these systems are not affine in $u$ as in (4.13), the PI interpretation is still valid. This can be seen from the simulations shown in Figure 4.12. Altering the parameters $K_p$ and $K_i$ changes the system response in the same way as corresponding parameters of a PI controller would.
Feedback Linearization Control Design

A feedback linearization controller is also derived for the second order model. This is done mainly to benchmark the performance of the iomi controller, but in the paper general similarities between feedback linearization and input-output inversion is also presented. To design a feedback linearization controller, the system (4.6) is rewritten to the standard form for feedback linearization. This is done through the state transformation

\[
\begin{pmatrix}
\xi_1 \\
\xi_2
\end{pmatrix} = \begin{pmatrix}
p \\
-\beta \gamma(p)h(x_a)
\end{pmatrix}
\]

and the input transformation \( \bar{u} = h(\zeta x_a + (1 - \zeta)u) \). Then system can be written as

\[
\begin{pmatrix}
\xi_1[k+1] \\
\xi_2[k+1]
\end{pmatrix} = \begin{pmatrix}
1 & 1 & 0 \\
0 & 0 & 0
\end{pmatrix} \begin{pmatrix}
\xi_1[k] \\
\xi_2[k]
\end{pmatrix} + \begin{pmatrix}
0 \\
-\beta
\end{pmatrix} \gamma(\xi_1[k] + \xi_2[k] + \beta w[k]) \bar{u}[k] + \begin{pmatrix}
\beta \\
0
\end{pmatrix} w[k]
\]

Introducing \( \xi_3 \) as the integrated control error

\[
\xi_3[k+1] = \xi_3[k] + T_s(r[k] - y[k]) = \xi_3[k] + T_s(r[k] - \xi_1[k])
\]

and using \( \bar{u}[k] = \frac{-1}{\beta \gamma(\xi_1[k] + \xi_2[k] + \beta w[k])} v[k] \) gives

\[
\xi[k+1] = \begin{pmatrix}
1 & 1 & 0 \\
0 & 0 & 0
\end{pmatrix} \xi[k] + \begin{pmatrix}
0 \\
1
\end{pmatrix} v[k] + \begin{pmatrix}
\beta \\
0
\end{pmatrix} w[k] + \begin{pmatrix}
0 \\
T_s
\end{pmatrix} r[k]
\]

\[
y[k] = \begin{pmatrix}
1 & 0 & 0
\end{pmatrix} \xi[k]
\]

which is a linear system from \( v \) to \( y \). Since \((A_\xi, B_\xi)\) is controllable, the feedback \( v[k] = K_\xi \xi[k] \) can place the poles arbitrarily. Since the function \( h \) is monotone, the control \( u[k] \) can be found from \( \bar{u}[k] \) which in turn is given by \( v[k] \).

4.4.4 Evaluation

The second order model (4.6) is simulated with the IOMI controller (4.16) based on the second order model, the feedback linearization controller (FL) and a linear PI controller. The controllers have the correct model structure, but parametric model errors are used. The step responses are shown in Figure 4.13, where all controllers are tuned to give the same performance for the first reference step. The PI controller has a significantly larger overshoot already for the second step and becomes very poorly damped for the third step. The FL controller performs very well and has similar step responses for all cases. The IOMI also performs well.
4.4. PAPER 2 - EXHAUST PRESSURE MODELING AND CONTROL

Figure 4.13: Simulations showing reference steps for the IOMI compared with a feedback linearization controller and a linear PI controller.

Figure 4.14: Measured step responses for the linear PI controller for four different levels of the exhaust pressure. Neither performance nor stability can be maintained over the operating range.
but the overshoot becomes slightly larger for each step. This clearly illustrates the significant nonlinearity in the system. Steps in the mass flow $w_e$ are also performed, showing the same tendencies.

Evaluation is also performed in two experiments on the engine, comparing the IOMI controller based on the one state model with a linear PI controller. The controllers are again tuned for similar performance for a specific step response at one operating point of the engine ($\text{BMEP} = 8$ bar, $N_{\text{eng}} = 1750$ rpm). Subsequent reference steps are then evaluated, as well as reference steps at $8$ bar BMEP and $N_{\text{eng}} = 2500$ rpm with the same control parameters. The reference steps at $2500$ rpm are shown in Figure 4.14 and Figure 4.15. Clearly, the PI controller can not maintain stability while the IOMI controller maintains both stability and performance.

4.5 Paper 3 - Gas Exchange Modeling

Paper 3 concerns issues of the gas exchange process of direct injected engines with variable valve timing.

4.5.1 Main Contributions

The main contributions of this paper are four. The first is a model of the volumetric efficiency as a function of valve overlap and the pressures in the intake and exhaust manifolds. The models is derived for different engine speeds. Second, a trapping model which is a function of the volumetric efficiency is presented. Third, a torque model based on fuel injection and the air/fuel ratio inside the cylinder is derived.
4.5. PAPER 3 - GAS EXCHANGE MODELING

The engine considered in this paper is equipped with direct injection, VGT and dual independent variable cam phasing. Through the variable cam phasing, the valve timing can be shifted according to Figure 4.16. When the exhaust valve closing ($\phi_{evc}$) is late and/or the intake valve opening ($\phi_{ivo}$) is early (solid lines), then there is a period when both the intake and exhaust valves are open at the same time. The length of this interval, measured in crank angle degrees, is called the valve overlap and denoted $\phi_{ol}$. Hence $\phi_{ol} = \phi_{evc} - \phi_{ivo}$ and the overlap center is given by $\phi_{cen} = (\phi_{ivo} + \phi_{evc})/2$ as shown in Figure 4.16. $\phi_{ol}$ and $\phi_{cen}$ is an alternative representation of the valve timing, which is used in this paper. Moreover, $\phi_{cen}$ will always be at the top dead center ($\phi_{tdc}$) and thus the valve timing is uniquely determined by $\phi_{ol}$. The motivation for this is that the volumetric efficiency depends stronger on $\phi_{ol}$ than on $\phi_{cen}$.

When the overlap is large there might be direct flows through the cylinders, either from the intake to the exhaust manifold or vice versa. Direct flow from the intake to the exhaust is called scavenging, and flow from the exhaust to the intake is called internal exhaust gas recirculation. Which of these situations that occurs depends on the pressures in the intake and exhaust manifolds. Figure 4.17 shows GT-POWER simulations of the gas flows through the intake and exhaust valves.

Finally, these models are validated using engine measurements, both in steady-state and transient operation.
CHAPTER 4. CONTRIBUTIONS

Figure 4.17: Simulations of the mass flows through the intake and exhaust valves for different valve overlaps. a) No air is scavenged when the valve overlap is small. b) Fresh air is scavenged from the intake to the exhaust during the overlap period when the overlap is large and the intake pressure is high. c) Exhaust gas is recirculated to the intake manifold when the overlap is large and the intake pressure is low.

when there is a) no scavenging or gas recirculation b) scavenging and c) exhaust gas recirculation.

**Volumetric Efficiency**

The volumetric efficiency, here denoted $\eta_{vol}$, is defined as the ratio of the actual volume of air inducted to the cylinders and the displacement volume. The volumetric efficiency is then defined as

$$\eta_{vol} = \frac{2 \cdot 60 \cdot \dot{m}_{\text{air}}}{\rho_{\text{int}} V_d N_{\text{eng}}} \tag{4.21}$$

where $\dot{m}_{\text{air}}$ (kg/s) is the mass flow of air, $V_d$ is the displacement volume of the cylinders, and $N_{\text{eng}}$ (rpm) is the engine speed and $\rho_{\text{int}}$ is the density of the inducted air.

In a mean value engine model, the air flow through the cylinders is modeled as a volumetric pump. According to (4.21), the air mass flow is given by

$$\dot{m}_{\text{air}} = \eta_{vol} \frac{\rho_{\text{int}} V_d N_{\text{eng}}}{2 \cdot 60} \tag{4.22}$$
where now the volumetric efficiency can be modeled as a function of, e.g., engine speed, intake and exhaust manifold pressures, and intake and exhaust cam positions
\[ \eta_{\text{vol}} = \eta_{\text{vol}}(N_{\text{eng}}, p_{\text{int}}, p_{\text{exh}}, \varphi_{\text{OL}}, \varphi_{\text{CEN}}, \ldots) \] (4.23)

**Trapping Efficiency**

The volumetric efficiency describes the total amount of fresh air that passes through the cylinders during the complete engine cycle. In the event of scavenging, the amount of air that is trapped in the cylinder during the combustion phase is less than the total amount of air. This will be described by a trapping efficiency factor \( \eta_{\text{trap}} \), which is the ratio of the amount of air trapped in the cylinders and the amount of inducted air. In accordance with (4.22) the trapped air mass flow is given by
\[ \dot{m}_{\text{air,trap}} = \eta_{\text{trap}} \dot{m}_{\text{air}} \] (4.24)

### 4.5.3 Modeling of Volumetric Efficiency

Although the volumetric efficiency model is primarily intended for transient operating conditions, the model will be estimated from steady-state measurements. The reason for using steady-state data is that the flow meter is mounted ahead of the compressor, and during load transients the air flow ahead of the compressor will differ from the air-flow into the cylinders. During steady-state conditions the (cycle-average) air flow will be the same throughout the whole system.

The volumetric efficiency model is based on linear regression and one model is created for each engine speed. The paper presents models for 1500, 1750 and 2000 rpm, and the volumetric efficiency for intermediate engine speeds is found through linear interpolation. For each engine speed, the model depends only on the valve overlap \( \varphi_{\text{OL}} \) and the intake and exhaust manifold pressures \( p_{\text{int}} \) and \( p_{\text{exh}} \). The steady-state data points were generated by gridding the three variables \( \varphi_{\text{OL}}, p_{\text{int}} \) and \( p_{\text{exh}} \). By utilizing both the throttle and the VGT, the intake and exhaust pressures can be controlled independently within a certain range. During transient conditions, higher exhaust pressure is achieved than what can normally be achieved in steady state. Hence an additional valve in the exhaust system is used to obtain steady state measurements for these high exhaust pressures, see Figure 4.18.

**Regressor selection**

The volumetric efficiency is modeled as a linear regression model in terms of three variables \( \varphi_{\text{OL}}, p_{\text{int}} \) and \( p_{\text{exh}} \), as discussed above:
\[ \eta_{\text{vol}}(\varphi_{\text{OL}}, p_{\text{int}}, p_{\text{exh}}) = \sum_{k=1}^{n} \theta_k \tau_k(\varphi_{\text{OL}}, p_{\text{int}}, p_{\text{exh}}) \] (4.25)
is used where \( \{\theta_k\} \) are parameters to be estimated and \( \{r_k\} \) are regressor functions chosen among those displayed in Table 4.3. The regressor \( \eta_{\text{otto}} \) is given by

\[
\eta_{\text{otto}} = \frac{r_c - (p_{\text{exh}}/p_{\text{int}})^{1/\gamma}}{r_c - 1}
\]  

(4.26)

where \( r_c \) is the compression ratio and \( \gamma \) is the specific heat ratio and represents the volumetric efficiency of an ideal Otto cycle (Eriksson and Nielsen, 2006). The other candidate regressors are not chosen based on physics. An initial screening of the data, including many more combinations of different powers of pressures and the overlap, showed that these chosen regressors were potentially important for the volumetric efficiency. To evaluate how many and which parameters to use in the model, the parameter estimates for all possible combinations of the candidate regressors in Table 3.2 are calculated. In Figure 4.19 the mean square error (MSE) of the best model with exactly \( n \) regressors is plotted against \( n \) for 1750 rpm. The situation for the other engine speeds is similar. Even though the lowest MSE
Figure 4.19: For all possible combination of the 17 regressors, the corresponding parameter estimate and the mean square error over the validation data are calculated. Among all models with \( n \) regressors (from 1 to 17) the best model (in mean square error sense) is picked and the mean square error of that model is here plotted against \( n \).

is given by 12 parameters, using more than six parameters does not improve the model significantly. Hence a model with six regressors is sought.

The best model with six regressors is

\[
\eta_{\text{vol}} = b_1 + b_2 \varphi_{\text{ol}} + b_3 \varphi_{\text{ol}} p_{\text{int}} + b_4 \varphi_{\text{ol}}^2 p_{\text{int}} + b_5 \varphi_{\text{ol}} \sqrt{p_{\text{int}}} + b_6 \varphi_{\text{ol}}^2 \sqrt{p_{\text{exh}}} \tag{4.27}
\]

Table 4.4 shows the results from the identification for the three engine speeds.

<table>
<thead>
<tr>
<th>Engine speed</th>
<th>No. of parameters</th>
<th>MSE</th>
</tr>
</thead>
<tbody>
<tr>
<td>1500 rpm</td>
<td>6</td>
<td>2.0%</td>
</tr>
<tr>
<td>1750 rpm</td>
<td>6</td>
<td>1.7%</td>
</tr>
<tr>
<td>2000 rpm</td>
<td>7</td>
<td>1.5%</td>
</tr>
</tbody>
</table>

Table 4.4: Model size and mean square error on validation data for the volumetric efficiency models.

Even though the number of parameters are the same in two of the models, the selected regressors differ.
4.5.4 Modeling of Trapping Efficiency

It is not possible to measure how much air is actually trapped inside the cylinders. In Mårtensson and Flärdh (2010), simulations in GT-POWER were used to generate the trapping data. This showed that the trapping efficiency could be modeled as a function of volumetric efficiency with the following structure

\[ \eta_{\text{trap}} = \begin{cases} 
1, & \eta_{\text{vol}} \leq m \\
1 - k(\eta_{\text{vol}} - m), & \eta_{\text{vol}} > m 
\end{cases} \quad (4.28) \]

The parameters \( k \) and \( m \) are estimated from data using a torque model which is presented next.

4.5.5 Modeling of Torque Generation

The model of the torque \( M \) is based on three parts; the ideal torque generated from combustion of the fuel, a compensation for situations when \( \lambda < 1 \) and the trapping efficiency.

\[ M = (a + bM_{\text{otto}}) f_{\text{AFR}}(\eta_{\text{trap}} \cdot \lambda) \quad (4.29) \]

where \( M_{\text{otto}} \) is the torque produced by an ideal Otto cycle

\[ M_{\text{otto}} = \frac{\dot{m}_{\text{fuel}} \cdot q_{hv} \cdot \eta_{\text{otto}}}{\omega_{\text{eng}}} \quad (4.30) \]

here \( \dot{m}_{\text{fuel}} \) (kg/s) is the fuel mass flow, \( q_{hv} \) (J/kg) is the lower heating value of the fuel, \( \omega_{\text{eng}} \) (rad/s) is the engine speed, and \( \eta_{\text{otto}} = 1 - r_c^{1-\gamma} \) is the efficiency of an ideal Otto cycle. The function \( f_{\text{AFR}} \) models torque decrease when \( \lambda < 1 \) and is a quadratic function of \( \lambda \)

\[ f_{\text{AFR}}(\lambda) = 1 + (\lambda - 1)(c + d\lambda) \quad (4.31) \]

The factor \( \eta_{\text{trap}} \cdot \lambda \) models the local \( \lambda \) in the cylinder, with \( \eta_{\text{trap}} \) given by (4.28).

The calibration data points for the torque model are the same as those used for modeling the volumetric efficiency, see Section 4.5.3. The measured torque \( M \) is calculated from cylinder pressure measurements with crank angle resolution by calculating the integral \( \int p_{\text{cyl}} \, dV \) over the compression and power strokes of the engine cycle. The model is calibrated in three steps: First \( f_{\text{AFR}}(\cdot) = 1 \) is assumed and the torque is modeled as an affine function of \( M_{\text{otto}} \) for data points with \( \lambda = 1 \) and \( \eta_{\text{trap}} = 1 \). Second, points with \( \lambda < 1 \) is considered to calibrate (4.31). In the last step, points with \( \eta_{\text{trap}} < 1 \) is used for calibration of (4.28).

4.5.6 Evaluation

In Figure 4.20 the three model steps are shown. The upper left plot shows the first step, when \( \lambda = 1 \) and \( \eta_{\text{trap}} = 1 \) is assumed. The upper right plot of Figure 4.20
Figure 4.20: The measured torque is compared with the modeled torque. Clearly, the three modeling steps gives good agreement and captures the effects of scavenging.

shows the torque prediction using model step 2. The torque predictions using the final model on all data is shown in the lower plot of Figure 4.20. The relative root mean square errors on validation data for the three model steps are 4.5%, 2.9% and 0.92% respectively. The trapping model is shown in Figure 4.21 and the data fits well to the model structure.
Figure 4.21: The trapping efficiency $\eta_{\text{trap}}$ is modeled as a piecewise affine function of the volumetric efficiency.

**Transient Validation of the Models**

Neither the in-cylinder $\lambda$ nor the volumetric efficiency and trapping efficiency can be directly measured. To assess the accuracy of those models, the torque model (4.29) is compared with the torque measured with the cylinder pressure sensors. Figure 4.22 shows the measured torque and models with and without the trapping model. During the transient, the model estimates $\lambda$ using the volumetric efficiency model and the injected fuel amount. The torque predictions are very accurate when using the trapping model. In some cases, though, the torque prediction during part of the transient is not so good.
Figure 4.22: Examples of load transients. The trapping model improves the accuracy of the predicted torque also in transient conditions. Mostly the prediction is good (left plot), but in some cases the model deteriorates slightly (right).
Chapter 5

Conclusions and Future Work

The scope of this thesis is centered around the transient torque response on downsized, turbocharged DISI engines. This is studied on an engine equipped with variable valve timing and variable geometry turbine. Open-loop trajectories that maximize the torque response are found through model based optimization. It is found that these trajectories keep the ratio between the exhaust and intake pressures constant during the transient. A feedback strategy that uses the VGT to control this ratio at an optimal level in presence of the model errors is presented. For the camshaft timing, a fuel optimal strategy with large overlap is beneficial also for a fast torque response. Evaluation on the engine shows improved torque response regardless of operating point. This control scheme also has the advantage that it will not affect the steady state fuel consumption, since it is only activated during the transients. There are several directions of further research for this problem. Performing the same procedure on another engine and/or another turbocharger is needed to further validate the approach. For production implementation, the tradeoff between improved performance and higher hardware cost needs to be investigated. A way to detect transients and decide when the controller should be activated and deactivated also needs to be found.

To control the exhaust pressure accurately, a feedback controller that is able to track the exhaust pressure over the whole operating range is designed. This is achieved by utilizing a nonlinear model of the VGT’s effect on the exhaust pressure in the controller. The model, also presented in the thesis, contains a four parameter model for the turbine mass flow that is calibrated using measurements on the engine. Evaluations in simulations and experiments, as well as during the torque transients, show that both performance and stability is preserved over the operating range. An additional benefit of the controller is the simple tuning. Only two minutes of experiments are needed for model estimation and only two control parameters have to be tuned for one single step response. The control structure also has inherit anti-windup and bumpless transfer. Interesting extensions would be to investigate the control structure on a MIMO system, as well as including explicit handling of time delays. An example of such MIMO system could be joint control of the VGT.
and throttle, or even full airpath MIMO control including also the cam timings. The IOMI structure with bumpless transfer would then also be beneficial for switching between modes focusing on, e.g., fuel economy and transient response respectively.

The valve timing has a significant effect on the cylinder air flow. A volumetric efficiency model that includes the effect of the VVT is derived. For SI engines with direct injection and VVT, it is possible to utilize scavenging, and a model for how much of the air that is trapped in the cylinder is presented. This trapping model is calibrated and evaluated through a torque model that captures the effect of the in-cylinder lambda. The torque model and the trapping model shows good performance also in most transient conditions. A natural extension to this work is to consider the full flexibility of the independent valve timing, i.e., let overlap center not only be at TDC. Also, validation at engine speeds where there is no estimation data would be valuable. Moreover, this study indicates differences between transient and steady state volumetric efficiency and this phenomena requires further attention.


