Institutionen för systemteknik
Department of Electrical Engineering

Examensarbete

Coordinated model based throttle and turbo control

Examensarbete utfört i Fordonssystem
vid Tekniska högskolan vid Linköpings universitet
av

Petter Carlsson

LiTH-ISY-EX--13/4665--SE

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Titel
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Koordinerad modelbaserad gasspjäll och turbo reglering
Coordinated model based throttle and turbo control

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Sammanfattning
Abstract

Downsizing and turbocharging is one way to meet the high demands on fuel consumption and performance on todays engines. The air-path system in a turbocharged spark ignited engine is a complex system and because the intake manifold pressure is tightly connected with the engine torque a consistent and robust control is needed. The control strategy utilizes two control loops, one wastegate actuator to control the intercooler pressure and one throttle actuator to control the intake manifold pressure. These pressures are coupled, making both actuators affect both pressures. Because of the time delay and the dynamics in the actuators and the system dynamics between the wastegate and the intercooler pressure the controller overreacts causing a pressure overshoot and sometimes oscillations. The oscillatory behavior is caused by both actuators trying to minimize their respective control error, affecting the others pressure. The delay in the system dynamics causes the two controllers to enter a state where they counteract each other. A compensation strategy is suggested, which estimates the intercooler pressure derivative and uses that to predict the future intercooler pressure. The compensation strategy shows good performance in simulations, reducing the overshoots and eliminating the oscillations.

Nyckelord
Keywords

MVEM, Turbocharge, Oscillation, Compensation control

Abstract

Downsizing and turbocharging is one way to meet the high demands on fuel consumption and performance on todays engines. The air-path system in a turbocharged spark ignited engine is a complex system and because the intake manifold pressure is tightly connected with the engine torque a consistent and robust control is needed. The control strategy utilizes two control loops, one wastegate actuator to control the intercooler pressure and one throttle actuator to control the intake manifold pressure. These pressures are coupled, making both actuators affect both pressures. Because of the time delay and the dynamics in the actuators and the system dynamics between the wastegate and the intercooler pressure the controller overreacts causing a pressure overshoot and sometimes oscillations. The oscillatory behavior is caused by both actuators trying to minimize their respective control error, affecting the others pressure. The delay in the system dynamics causes the two controllers to enter a state where they counteract each other. A compensation strategy is suggested, which estimates the intercooler pressure derivative and uses that to predict the future intercooler pressure. The compensation strategy shows good performance in simulations, reducing the overshoots and eliminating the oscillations.

Sammanfattning

Nedskaling och överladdning är en sätt att möta dagens höga krav på motorer. Luftvägen i en överladdad motor är ett komplext system och eftersom insugstrycket är direkt kopplat till motorns utmoment krävs en konsekvent och robust reglering. Reglerstrategin som används har två reglerloopar, en wastegate aktuator som styr trycket i laddluftskylaren och ett gasspjäll som styr insugstrycket. Trycken är sammankopplade vilket gör att båda aktuatorerna påverkar båda trycken. Fördröjningar i dynamiken i aktuatorerna och systemet mellan wastegaten och trycket i laddluftskylaren gör att regulatorn överreagerar, vilket resulterar i en översläng och självsvängningar. Det självsvängande beteendet orsakas av att båda reglerlooparna försöker minimera deras respektive reglerfel och då påverkar båda trycken. Fördröjningarna i systemet gör att regulatorerna hamnar i otakt och motverkar varandra. En kompenserings strategi förslås, vilken skattar laddluftskylartryckets derivata och använder derivatan för att förutspå laddluftskylartryckets framtida värde. Simuleringar visar att kompenserings strategin reducerar överslängarna och eliminerar självsvängningarna helt och hållet.

Acknowledgments

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Petter Carlsson

Contents

Chapter 1

Introduction

1.1 Background

Engines have today high demands on driveability, fuel economy and emissions. One way to meet those demands is to downsize and turbocharge the engine. For spark ignited (SI) engines, engine torque is tightly connected to the air-mass flow, which is controlled by the throttle and the turbocharger. Dynamic modeling of the air path through a turbocharged engine is challenging, but to have consistent control behavior in all ambient conditions a model-based control is desirable. For the throttle, which is modeled as a flow through an orifice, the pressure drop over the throttle has a nonlinear behavior, which results in more complex control strategies than for a linear system. For the turbocharger the flow capabilities of the compressor and turbine are characterized by the performance at different operating points. Typically the performances are represented as mapped data provided by the manufacturer during steady state conditions, where the turbocharger speed is maintained fixed during a series of mass-flow measurements. After the measurements a new turbocharger speed is set and the procedure repeats itself until the entire operating area is mapped. Because the data is mapped during steady state conditions there may be difficulties in applying a dynamic model. Since both the throttle and the turbocharger affects the air-mass flow, they need to be coordinated in order to obtain the right amount of air-mass into the cylinders. This thesis will focus on two problems that arises due to the interactions between throttle and turbocharger.

- Throttle operation at low pressure drops. Throttle/turbo interactions occur when the throttle operates to increase intake manifold pressure. The increased throttle flow will initially decrease boost pressure which the boost control will compensate for, but with a delayed response due to actuator and system dynamics. When the boost pressure increases, the intake manifold pressure already has increased to its correct level due to throttle control. This results in an overshoot in manifold pressure and the system may get in to a state of self-oscillation that transfer to oscillations in vehicle torque.

The current control deals with these pressure drops and achieves stability. One of the objectives of this thesis is to propose a model-based approach to solve this problem.

- Throttle and boost control during gear-shift. During gear-shifts the engine reduces torque before changing gears and then increase torque after the shift is completed. There are strict requirements on how fast and accurate the torque control should be and the torque directly corresponds to the intake manifold pressure. The boost control should provide adequate boost during gear-shift and ensure a fast pressure build up during the end of the shift. The throttle typically induce a boost pressure overshoot when reducing torque, which the boost controller will react on and reduce boost pressure. The problem is similar to the throttle operating at low pressure drops but occurs during more transient operation conditions. The output torque is directly coupled to the intake manifold pressure which makes the control important for the driveability. Because there are two systems, throttle and turbocharger, they have to be coordinated to achieve good control performance.

1.2 Problem formulation

There is a self-oscillating-behavior in the intake manifold pressure that arises from the interaction between the throttle and the turbocharger. Listed below are a few hypotheses as what causes these self-oscillation. The thesis will investigate the hypotheses to determine the origin to the self-oscillation.

- Poorly tuned boost controller
- Poorly tuned throttle controller
- Time delay in turbo-actuator
- Time delay in sensors
- Other dynamics in the intake air system.

1.3 Purpose and Goals

The purpose with this work is to investigate the interactions between throttle and boost control. With an understanding of the underlying physics, a model-based approach can be used to characterize the dynamic couplings in the intake air system. This will be used to investigate the self-oscillation-behavior in the closed-loop control system. The main objectives are:

- Derive representative simulations that describes the self-oscillation-behavior in the engine measurements.

- Use model simulations to either confirm or falsify the hypotheses listed in 1.2.
- Propose and evaluate control algorithms that would improve control robustness.
- Test findings in a vehicle.

1.4 Related Research

The most common way to model an engine is a component based mean value engine model (MVEM), where the mean value of one or more cycles is modeled. The MVEM equations is, for example, described in Modeling and control of engines and drivelines by L.Eriksson and L.Nielsen [9]. A model of the turbocharger is further described in [7] where the turbine and the compressor is modeled as components in the MVEM-framework. In addition to the turbine and compressor model components in the MVEM-library, a master thesis was performed by E.Linden and D.Elofsson [14] to develop a wastegate model for the existing library and they also proposed a wastegate control strategy. The MVEM for the air path is well formulated in [2], which aims to estimate the amount of air charged in the cylinder. The models described in [9],[14] and [2] will be the cornerstones in the models used in this thesis.

To control the air-mass flow to the cylinder, several methods and ideas have been tested and evaluated. One example is the controller described in a paper by P.Moulin and J.Chauvin, [15], which is based on a motion-planning strategy first forumlated in a paper by T.Leroy [18]. The strategy proposed is to consider the throttle and wastegate as two independent systems that are active simultaneous. The throttle control strategy is based on model-based motion-planing where an air-mass trajectory is computed and then translated into a reference intake manifold pressure. To deal with model uncertainties in the volumetric-efficiency model, an observer is utilized to estimate and compensate for the bias error. The target manifold pressure is translated into an throttle angle with the use of dynamic inversion as a feedforward control law and fine tuned with a PI-controller as feedback control law. The wastegate controller in Moulins paper [15] is based on feedback linearization and constrained motion planning. The principle is the same as for the throttle controller, where a control law for the feedforward term is obtained through motion planning which takes the constraints in consideration. Because of the constraints, an integrator anti-windup is implemented. The controller also uses a feedback strategy in order to improve robustness and it’s implemented as a PI-controller where the P-part is given by linearization through dynamic inversion and the I-part is to guarantee convergence. The result of this approach shows good dynamic performances with a limited calibration effort.

Another control approach is studied by G.Colin et al. [10] in a paper about neural control for nonlinear systems. The paper suggests separated but coordinated controls for the throttle and the turbocharger. For the throttle controller an Internal Model Controller (IMC) is suggested. The controller is then evaluated against a classical feedforward control plus a PID. The results of the IMC is clearly better than the classical controller, but the main advantages with the IMC is stated to be the easy synthesis and tuning. To control the wastegate, ie the turbocharger a Nonlinear Model Predictive Control (NMPC) is suggested. To deal with the fact that the NMPC is computationally demanding and the solution isn't always the global minimum, a linearization is performed and a neural black-box model to estimate the supercharged pressure is implemented. The neural model is used to replace the physical model which is to complex to be implemented in the MPC framework. The control concept demonstrates good performance. Instead of the MPC approach it is possible to utilize the IMC approach for wastegate control as well as for the throttle, this is demonstrated in [12] where an IMC wastegate control is described.

For this thesis a slightly more interesting control approach is the coordinated throttle and wastegate control described in [17]. There the control problem is divided in three regions. A low region for when the ambient pressure is sufficient as boost pressure. Then the wastegate will be wide open and the throttle will control the air-mass flow. A mid region where the throttle and wastegate is used simultaneously, the throttle is maintained at a certain set point at steady state. The set point makes sure that the throttle can react fast when more air-mass flow is needed. A high region for high loads, where the throttle will be wide open and the wastegate will control the air supply. This controller shows fast torque responses and fairly high efficiency, the mid region is shown to lose 2-4 % in pumping losses in comparison with having the throttle wide open. The controller also demonstrated an oscilliative behavior while going from high region to mid region. This was solved by slowing down the throttle movement in the transition in exchange for a slower torque response.

Another interesting control approach is suggested in [3], where an “exact” air charge controller is described. They suggest a multiple input multiple output (MIMO) system that use a feedforward control. The control inputs are the opening of the wastegate and the throttle plate angle. The key idea is to use a nonlinear tenth order MVEM and instead of reduce the order they design a multi-variable feed forward control. The controller is computational demanding but also very accurate.

The MIMO approach is also proposed in [16], but as a future work to their multiple model control. The controller handles noise and model uncertainties as well as nonlinearities easier than other existing approaches. A MIMO approach is implemented in [4] for a spark ignited engine without turbocharger. The controller uses the MPC-framework and simplified models to reduce the computational effort. The controller gives a faster torque response and handles transients in lambda better compared to a conventional controller. But it also gives a small overshoot in the torque response which the conventional controller doesn't.

A Decentralized Two Input Two Output (TITO) controller of the throttle and wastegate is proposed in [11], where the throttle is used to control the intake manifold pressure while the wastegate is used to control the boost pressure. They propose a PI-controller with integrator antiwind-up for both control loops as well as a feedforward component in the wastegate controller. The wastegateloop is tuned to have a slow bandwidth and the throttleloop is tuned to have a fast bandwidth to be able to handle the non-minimum phase for the wastegate. The proposed TITO-system shows a tradeoff between actuator response and the boost pressure settling time. In an effort to improve this an output feedback controller is added in the control structure. The added controller improved the throttle response as well as the boost pressure settling time but the wastegate response still wasn't satisfactory. As a future work, a nonlinear controller is suggested to improve the wastegate response.

Another approach commonly researched for the engine air path is the fuzzy logic control, see for example [13], [1]. Where [13] states that the main advantage with this approach is the systematic way to deal with a large class of nonlinear systems.

1.5 Expected Results

The expected results in this thesis is to determine the reason why these self-oscillations mentioned in section 1.1 occur and to develop a solution for it. The solution will be in form of a model-based controller that should be implementable in a real ECU and be fairly easy to calibrate. The controller shall reduce the self-oscillatory behavior during low pressure drops and gear shifts. The main objectives for the thesis is listed below.

- To characterize and find the self-oscillation behavior for when the throttle operates at low pressure drops in simulations.
- To characterize the overshoot that initializes the self-oscillation behavior and find the reason why it occur in a simulation environment.
- To propose a possible solution for the above mentioned problems with support from the simulation environment.

1.6 Method

The method in this thesis is first to model the entire engine, after which the recreation of the self-oscillation and overshoot begins. When the problem is recreated some model components will be unlinked and its dynamics will be removed or modified to find out how a certain components dynamics will effect the air system. The control strategy utilized is two controllers, the throttle controller to control the intake manifold pressure and the wastegate controller to control the intercooler pressure. The throttle controller utilizes a model based feedforward and a PI-controller to fine tune. The wastegate controller utilizes a static feedforward and a PI-controller.

Chapter 2

Approach/Modeling

This chapter describes the modeling part of the thesis. The base to the engine model is a component based mean value engine model (MVEM). The majority of the components is a part of a MVEM-library called MVEM-lib, created by Lars Eriksson [6]. The MVEM equations is well described in for example [2] and [9]. To parameterize the model a set of measurements has been provided by Division of Vehicular Systems at Linköping University. In Figure 2.1 an overview of the entire model is shown including ECU, air path model, driver gas pedal interpretation, effective area calculations and blocks for manual inputs.

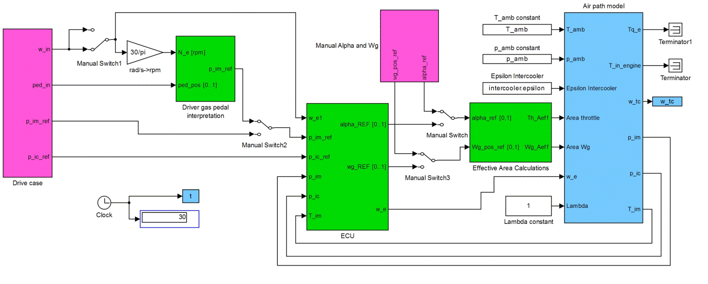

Figure 2.1. An overview of the entire model. The magenta colored blocks are for manual input, the green blocks are from the left; Driver gas pedal interpretation, ECU and effective area calculations. The blue block is the air path model.

2.1 MVEM-lib

This section will give a short description of Lars Erikssons MVEM-lib, a more extensive description is given in [8]. To model the airflow through the engine, the MVEM-lib is structured in a number of components.

- Receiver or control volume.
- Incompressible flow restriction.
- Compressible flow restriction.
- Compressor torque.
- Compressor temperature.
- Intercooler temperature
- Engine flow.
- Engine torque.
- Engine out temperature.
- Exhaust temperature drop.
- Turbine torque.
- Turbine temperature.
- Inertia with friction.
- Adiabatic mixer.

The basic idea behind the MVEM-lib is to put restrictions between control volumes or (receivers) and describe the air path in terms of control volumes and restrictions. The restrictions calculates the air-mass flow through the restrictions by the given pressure and temperature before and after the restriction. The control volumes handles the gas dynamics and have states for temperature and pressure. Some of the components in the air path model uses the standard MVEM-lib components while some of the components have to be customized. The components are described in section 2.3.

2.2 Model inputs

The model has both mandatory and optional inputs. The mandatory inputs have to be supplied for the model to run, while the optional inputs are utilized to create specific simulation cases. The mandatory inputs are a target intake manifold pressure, engine speed and the ambient conditions. The target intake manifold pressure can either be given by the acceleration pedal through the driver gas pedal interpretation, see section 2.7, or by bypassing the driver gas pedal interpretation block and choose the target intake manifold pressure directly. There is also possible to control the target intercooler pressure as well as the throttle angle and the wastegate positon manually. The possibilities is shown i table 2.1.

Model inputs	Mandatory/Optional
Ambient pressure	Mandatory
Ambient temperature	Mandatory
Engine speed	Mandatory
Acceleration pedal position	Mandatory/Optional
Target intake manifold pressure	Mandatory/Optional
Target intercooler pressure	Optional
Throttle angle	Optional
Wastegate position	Optional

Table 2.1. Model inputs. The mandatory inputs has to be supplied for the model to run. One of the two inputs labeled mandatory/optional have to be supplied while the other is optional. The inputs labeled optional are utilized to create specific simulation cases.

2.3 Air path model

The main part of the thesis is the model over the air flow through the engine. In this section the components of the air path model is described component by component. In Figure 2.2 an overview of the air path model is shown. The magenta colored subsystems represent restrictions, blue colored subsystems represent control volumes, the red colored subsystems represent temperature models and the two yellow subsystem is the adiabatic mixer and the rotation inertia model of the turbocharger. The last component is the grey subsystem, which is the model of the combustion. Later in this section a short description of the subsystems is presented.

Airfilter

The airfilter consists of one incompressible flow restriction connected to one control volume, which is the pipe between the airfilter and the compressor. Both the restriction and the control volume uses the standard MVEM-lib components.

Compressor

The compressor is in the MVEM-lib modeled as a restriction, but the compressor consist of a set of sub models; a torque model, a temperature model, an air-mass flow model and an efficiency model, more on that in section 2.4. The compressor is connected to a compressor receiver, which is the control volumes that is the pipe between the compressor and the intercooler.

Inertia with friction

An inertia with friction is implemented to calculate the speed of the turbocharger.

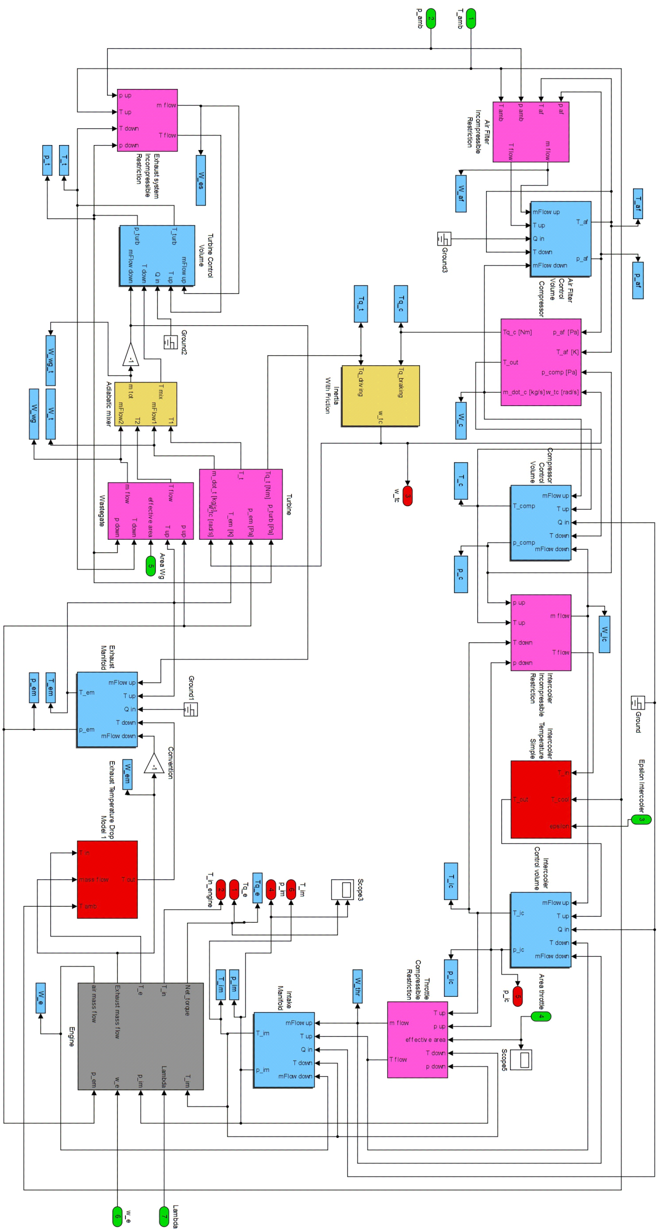

Figure 2.2. An overview of the air path model. The magenta blocks represents restrictions, the blue blocks control volumes, the red blocks represents temperature models and the two yellow subsystem is the adiabatic mixer and the rotation inertia model of the turbocharger. The grey subsystem is the combustion model. The air enter at the top-right block and go through the other components in a semi-circle and finally exit at the top-left corner.

Intercooler

The intercooler has three components from the MVEM-lib. First there is the incompressible flow restriction and secondly to model the temperature drop, a simple temperature model is implemented. Finally the intercooler flow restriction is connected to the intercooler receiver, which is the pipe between the intercooler and the throttle.

Throttle

The throttle consists of a compressible flow restriction. The flow through the throttle is controlled by controlling the effective open area. The effective area calculations is shown in section 2.6.

Intake Manifold

The intake manifold consists of the control volume that connects the throttle with the cylinders. This component uses the standard MVEM-lib component.

SI engine

The engine is modeled as a customized restriction, but it is modeled with unmodified blocks from the MVEM-lib. The engine consists of models for engine flow, engine torque and engine out temperature.

Exhaust Manifold

The exhaust manifold consists of a temperature drop model and a receiver from the MVEM-lib. The receiver connects the cylinders with the turbine and wastegate.

Wastegate

The wastegate consists of a compressible flow restriction which is controlled by controlling the effective open area. More about the effective area in section 2.6. The Wastegate works parallel with the turbine and indirect controls the turbine by affecting the pressures connected to the turbine.

Turbine

The turbine submodel consists of a temperature and torque model from the MVEM-lib as well as a control volume. In addition a model for the turbine air-mass flow and the efficiency is implemented, see section 2.4.

Adiabatic Mixer

An adiabatic mixer is implemented after the turbine and wastegate to mix the flows from the turbine and the wastegate. This is a standard MVEM-lib component.

Exhaust system

The exhaust systems exists of an incompressible flow restriction which is a standard MVEM-lib component.

2.4 Turbocharger modeling

This section describes the parts of the turbocharger model which isn't a standard MVEM-lib component. First the compressor and its massflow and efficiency model is described followed by a description of the turbine and its massflow and efficiency model.

2.4.1 Compressor

The compressor model is divided into two submodels, one massflow model and one efficiency model. The model equations is taken from [2] and can be seen below. Table 2.2 gives an explanation for the variable symbols in the equations.

Variable	Symbol
Blade tip speed	U
Diameter	D
Pressure ratio	Π
Temperature	T
Air-mass flow	$\dot{m}$
Specific heat capacity	c_p
Ratio of specific heats	γ
Efficiency	η

Table 2.2. Table of variables for the compressor model.

Massflow model

$$U_c = \omega_{TC} \frac{D_c}{2} \tag{2.1}$$

$$\Pi_c = \frac{p_{af}}{p_c} \tag{2.2}$$

$$\Pi_{c,max} = \left(\frac{U_c^2 \Psi_{max}}{2 c_p T_{af}} + 1 \right)^{\frac{\gamma}{\gamma-1}} \tag{2.3}$$

$$\dot{m}_{c,corr} = \dot{m}_{c,corr,max} \sqrt{1 - \left(\frac{\Pi_c}{\Pi_{c,max}} \right)^2} \tag{2.4}$$

$$\dot{m}_c = \dot{m}_{c,corr} \frac{p_{af}/p_{ref}}{\sqrt{T_{af}/T_{ref}}} \tag{2.5}$$

Ψ_{max} and $\dot{m}_{c,corr,max}$ are model parameter and is estimated with the MATLAB-function *lsqcurvefit*.

Efficiency model

$$\eta_c = \eta_{c,max} - \left[\frac{\dot{m}_{c,corr} - \dot{m}_{c,corr@\eta_{c,max}}}{\sqrt{\Pi_c - 1} - \left(\Pi_{c@\eta_{c,max}} - 1\right)}\right]^T \begin{bmatrix} Q_{11} & Q_{12} \\ Q_{12} & Q_{22} \end{bmatrix} \left[\frac{\dot{m}_{c,corr} - \dot{m}_{c,corr@\eta_{c,max}}}{\sqrt{\Pi_c - 1} - \left(\Pi_{c@\eta_{c,max}} - 1\right)}\right] \tag{2.6}$$

$\Pi_{c@\eta_{c,max}}$, $\eta_{c,max}$, Q_{11}, Q_{12}, Q_{22} and $\dot{m}_{c,corr@\eta_{c,max}}$ are model parameter and is estimated with the MATLAB-function *lsqcurvefit*.

Validation

To validate the models, they are plotted against the measured data. As seen in the Figure 2.3 the model gives a good match against the measured data. A closer look at the left figure shows that the model is less accurate at higher pressure ratios.

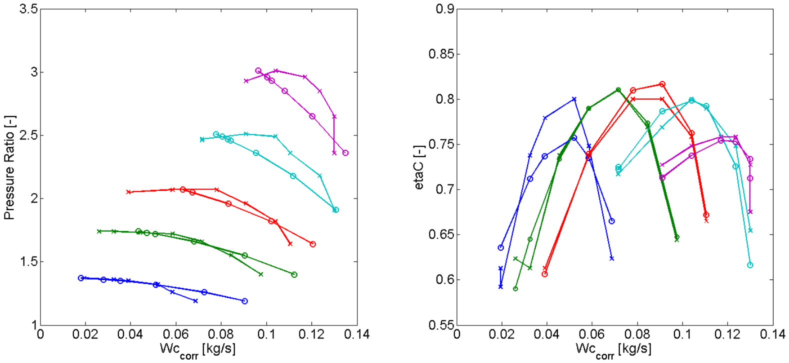

Figure 2.3. Validation of compressor model, where x is measured and o is modeled. The different colors represent different turbocharger speeds. Left figure: Validation of the mass flow model shows the pressure ratio plotted against the corrected mass flow. The model is a good match for the lower pressure ratios but gets less accurate for higher pressure ratios. Right figure: Validation of the efficiency model, the efficiency plotted against the corrected mass flow. The figure shows a good match for all speed lines except the blue one, which is the lowest turbocharger speed.

2.4.2 Turbine

As the compressor model, the turbine model is divided into two submodels, one massflow model and one efficiency model. The submodels used in this thesis is taken from [2]. The model equations is shown below. Table 2.3 gives an explanation for the model variables.

Variable	Symbol
Pressure ratio	Π
Temperature	T
Air-mass flow	$\dot{m}$
Turbine flow parameter	TFP
Pressure	p
Efficiency	η
Diameter	D
Specific heat capacity	c_p
Ratio of specific heats	γ
Blade speed ratio	BSR
Angular speed	ω

Table 2.3. Table of variables for the turbine model.

Massflow model

$$\Pi_t = \frac{p_t}{p_{em}} \tag{2.7}$$

$$TFPmod = \begin{cases} TFP_{max}\sqrt{1 - \Pi_t^{TFP_{exp}}} \ , \Pi_t^{TFP_{exp}} \leq 1 \\ 0, \ \text{otherwise} \end{cases} \tag{2.8}$$

$$TFPmod = \dot{m}_t \frac{\sqrt{T_{em}}}{p_{em}} \tag{2.9}$$

TFP_{max} and TFP_{exp} are model parameters and are estimated with the MATLAB-function *lsqcurvefit*.

Efficiency model

$$BSR = \frac{D_t}{2} \cdot \frac{\omega_{TC}}{\sqrt{2c_{p_{eg}}T_{em}\left(1 - (\frac{1}{\Pi_t})^{\frac{\gamma_{eg}-1}{\gamma_{eg}}}\right)}} \tag{2.10}$$

$$\eta_t = \eta_{t_{max}}\left(1 - \left(\frac{BSR - BSR_{\eta_{t_{max}}}}{BSR_{\eta_{t_{max}}}}\right)^2\right) \tag{2.11}$$

$BSR_{\eta_{t_{max}}}$ and $\eta_{t_{max}}$ are model parameters and are estimated with the MATLAB-function *lsqcurvefit*.

Validation

To validate the models, they are plotted against the measured data. As seen in Figure 2.4 the model gives a good match against the measured data.

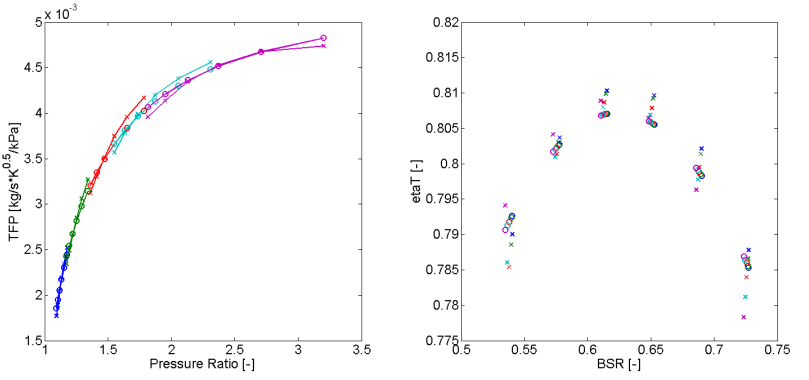

Figure 2.4. Validation of turbine model, where x is measured and o is modeled. The different colors represent different turbocharger speeds. Left figure: Validation of the mass flow model, where the TFP is plotted against the pressure ratio. The model approximates the mass flow with good accuracy. Right figure: Validation of the efficiency model, where the efficiency is plotted against the BSR. The model gives a good approximation to the measured efficiency where all measured and modeled points is of the same magnitude.

2.5 Actuator dynamics modeling

There are two actuators, one to actuate the throttle and one to actuate the wastegate. Both actuators are modeled with at first-order system

$$\text{pos} = \frac{1}{1+\tau s}\text{ref} \tag{2.12}$$

The throttle actuator first-order system time constant is, $\tau = 30ms$. In addition to the first-order system the throttle also have a time delay of 20 ms. The wastegate actuator first-order system has a time constant of $\tau = 100ms$.

2.6 Effective Area Calculations

The effective area is needed for the compressible flow restrictions in the throttle and wastegate. The equations for the compressible flow is described in [2]. The effective area is the area times the discharge coefficient, C_d. A variable description is shown in table 2.4 below followed by the model equations.

Variable	Symbol
Air-mass flow	$\dot{m}_{air}$
Gas constant	R
Temperature	T
Pressure	p
Pressure ratio	Π
Ratio of specific heats	γ
Area	A
Throttle angle	α
Discharge coefficient	C_d

Table 2.4. Variable description

Effective area equations

$$\dot{m}_{air} = \frac{p_{before}}{\sqrt{R \cdot T_{before}}} \Psi(\Pi) C_d A(\alpha) \tag{2.13}$$

$$\Pi = min(\frac{p_{after}}{p_{before}}, 1) \tag{2.14}$$

$$\Psi^*(\Pi) = \frac{\sqrt{\frac{2\gamma}{\gamma-1}(\Pi^{\frac{2}{\gamma}} - \Pi^{\frac{\gamma+1}{\gamma}})}}{\sqrt{\frac{2\gamma}{\gamma-1}((\frac{2}{\gamma+1})^{\frac{2}{\gamma}} - (\frac{2}{\gamma+1})^{\frac{\gamma+1}{\gamma-1}}))}} \tag{2.15}$$

$$\Psi(\Pi) = \begin{cases} 1 \text{ , if } 0 < \Pi \leq (\frac{2}{\gamma+1})^{\frac{\gamma}{\gamma+1}} \\ \Psi^*(\Pi) \text{ , otherwise} \end{cases} \tag{2.16}$$

Rewriting 2.13 to solve for the effective area gives:

$$C_d A(\alpha) = \dot{m}_{air} \frac{\sqrt{R \cdot T_{before}}}{p_{before} \cdot \Psi(\Pi)} \tag{2.17}$$

2.6.1 Throttle effective area

Two different effective area models are used, one to translate the throttle angle to an effective area and one simpler model for the feedforward part in the throttle controller.

Effective Area Model

In [2] an effective area model is suggested, where the model equation is shown below.

$$C_d A(\alpha) = A_1(1 - \cos(a_2\alpha^2 + a_1\alpha + a_0)) + A_0 \tag{2.18}$$

Using the MATLAB-function *lsqcurvefit* to determine the model parameters. The calculated effective area is then plotted against the modeled effective area in Fig-

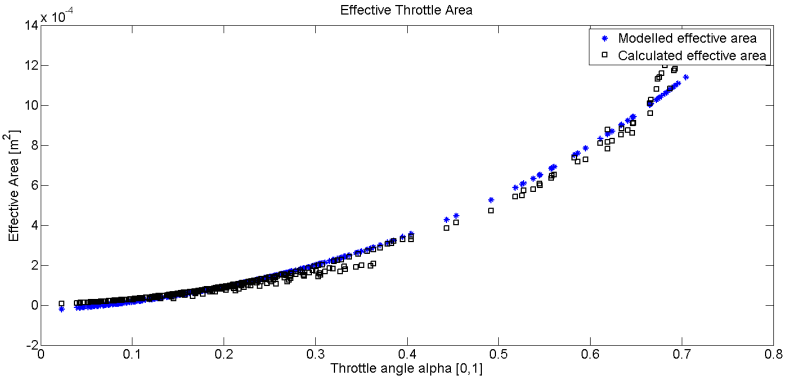

Figure 2.5. Validation of the throttle effective area model. Blue marks the model and black the calculated effective area. The model shows a good approximation for $0 < \alpha < 0.65$ and but loses in accuracy for $\alpha > 0.65$. The measured data only supplied measurements for throttle angle up to 0.7

ure 2.5. The figure shows that the model gives a good approximation of the effective area.

To implement the effective area model in simulink the cosine is approximated with a Maclaurin-series with 3 terms. Inserting the Maclaurin-series in (2.18) gives:

$$\begin{aligned} C_dA(\alpha) &= A_1(1-(1-\frac{(a_2\alpha^2+a_1\alpha+a_0)^2}{2!}+\frac{(a_2\alpha^2+a_1\alpha+a_0)^4}{4!}))+A_0 \\ &= A_1(\frac{(a_2\alpha^2+a_1\alpha+a_0)^2}{2!}-\frac{(a_2\alpha^2+a_1\alpha+a_0)^4}{4!}))+A_0 \qquad (2.19) \end{aligned}$$

Simple Effective Area Model for feedforward

A simpler effective area model is implemented in the feedforward part of the throttle controller, the reason to utilize a simpler version of the effective area model is because the feedforward part calculates the effective area and then translate it to an angle using the inverse of the effective area model. The simple effective area model is suggested in [9].

$$C_dA(\alpha) = A_0 + A_1\alpha + A_2\alpha^2 \qquad (2.20)$$

As seen in Figure 2.6 the simple model is a little more off than the more complex effective area model, but it still gives a good approximation.

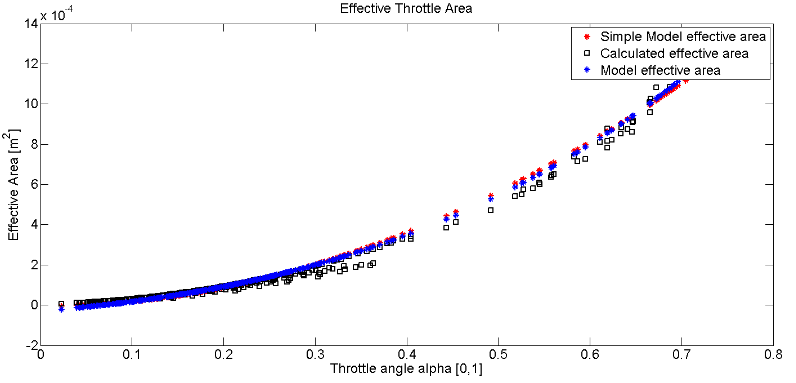

Figure 2.6. Validation of the simple effective area model. Red marks the simple model, blue the more complex model and black the calculated effective area. The simple model gives a less accurate approximation of the calculated effective area than the more complex effective area model. But it still gives a good approximation.

2.6.2 Wastegate effective area

The wastegate is more difficult to model, due to the fact that the mass-flow isn't measured. This thesis uses the same model as in [2].

$$A_{eff} = C_d A_{wg,max} wg_{pos} \tag{2.21}$$

Because the lack off measurements there is no validation of this specific component.

2.7 Driver gas pedal interpretation

The driver gas pedal interpretation model structure is made by the Division of Vehicular Systems at Linköping University. The model translates a given acceleration pedal position to a request in torque and then calculate the required intake manifold pressure to produce the requested torque.

Driver torque request

The translation from acceleration pedal position to requested torque is done by two maps. One map containing the maximum available torque for a few engine speeds, the other map contains the minimum available torque for the same engine speeds. The maps are then linearized to represent every engine speed and every acceleration pedal position with a given driver torque request.

Target intake manifold pressure

To calculate the target intake manifold pressure from the driver torque request a model of the brake mean effective pressure (BMEP) is needed. A model for BMEP is presented in [9]. A description to the model variables is shown in table 2.5 and the model equations is shown in (2.22)-(2.23).

Variable	Symbol
Torque	T_q
Brake mean effective pressure	BMEP
Displacement volume	V_D
Intake manifold pressure	p_{im}
Number of crank revolutions in a complete power generation cycle	n_r

Table 2.5. Variable description BMEP model

$$T_q = \frac{BMEP(p_{im})V_D}{n_r 2\pi} \tag{2.22}$$

$$BMEP(p_{im}) = -C_1 + C_2 p_{im} \tag{2.23}$$

By calculating BMEP from the measured data and estimate C_1 and C_2 with the method of least squares. In Figure 2.7 the BMEP model is validated. The figure shows that the model is a good approximate to the measured data.

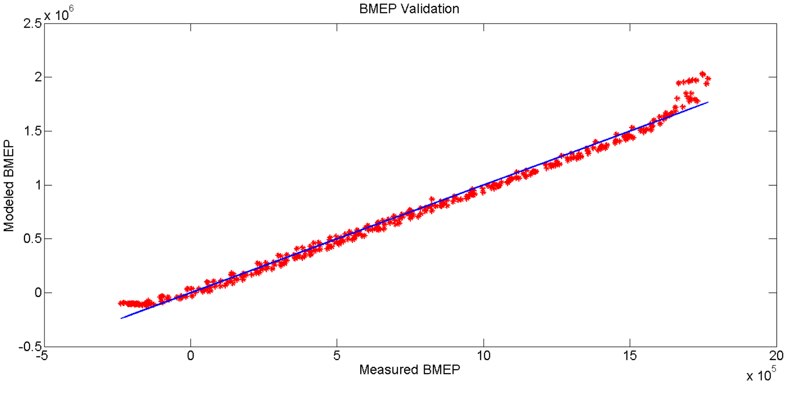

Figure 2.7. Validation of the BMEP model. The blue line is the ideal model, while the red stars represent measured BMEP plotted against the modeled BMEP. The figure shows a good agreement, the red dots is place around the ideal blue line.

2.8 Volumetric efficiency, η_{vol}

In the feedforward part of the throttle controller a volumetric efficiency model is used to estimate $\dot{m}_{air}$. The volumetric efficency η_{vol} is described in [9] as well as the suggested model that was developed by Hendricks and Sorenson (1990).

Variable	Symbol
Volumetric efficiency	η_{vol}
Air-mass flow	$\dot{m}_a$
Displacement volume	V_D
Intake manifold pressure	p_{im}
Number of crank revolutions in a complete power generation cycle	n_r
Engine speed	N

Table 2.6. Variable description volumetric efficiency model

A variable description is shown in table 2.6 and the model equations is shown in (2.24)-(2.25).

$$\eta_{vol} = \frac{\dot{m}_a n_r}{p_{im} V_d n_{cyl} N} = \frac{\dot{m}_a n_r}{p_{im} V_D N} \tag{2.24}$$

$$\eta_{vol} = c_0 + c_1 N + c_2 N^2 + c_3 p_i m \tag{2.25}$$

By calculate η_{vol} with measured data and then use the MATLAB-function *lsqcurve-fit* to estimate the constants $c_0 - c_3$. Figure 2.8 shows that the model is a good approximation for η_{vol} bigger than 0.65.

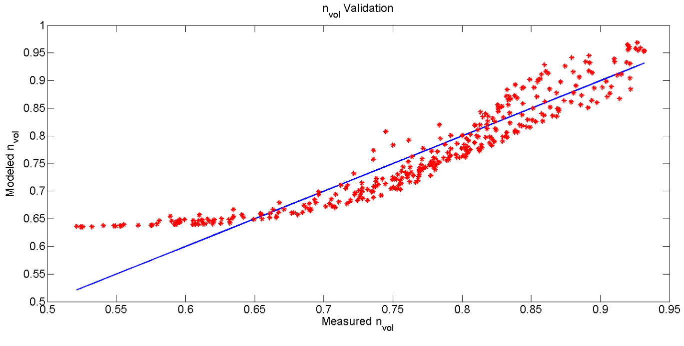

Figure 2.8. Validation of η_{vol}-model. The blue line represent a perfect model and the red stars is the result of plotting the measured η_{vol} against the modeled η_{vol}. The model gives a decent approximation for η_{vol} over 0.65 seen as the red dot is centered around the blue ideal line.

2.9 ECU

In this section the controllers are described. First the throttle controller and then the wastegate controller. Both controller has been discretized with a 10 ms sample time.

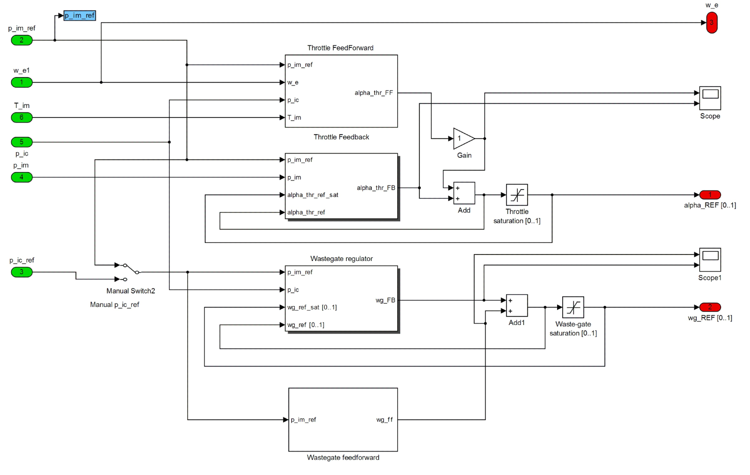

Figure 2.9. An overview of the ECU. From the top down, throttle feedforward, throttle PI-controller, wastegate PI-controller and wastegate feedforward.

2.9.1 Throttle Controller

The throttle controller consists of two parts, one feedforward part and one feedback part. The feedforward part estimates a throttle angle, α, and the feedback part fine tune the angle to get the correct intake manifold pressure. The feedback part has a tracking functionality to prevent integrator wind-up. The feedforward and the feedback contributions are then added to a final throttle angle which are saturated between 0 and 1.

Feedforward

The feedforward part has its core in the effective area equations, (2.13)-(2.16) and the η_{vol} equations, (2.24)-(2.25). The idea is to use the η_{vol} model with the target intake manifold pressure to estimate the required air-mass flow, $\dot{m}_{air}$. Given the estimated air-mass flow an effective area can be estimated using the effective area equations. The effective area is then translated into an throttle angle α using the inverted simple effective area model, (2.20).

$$\alpha_{ff} = f(\eta_{vol}, N, p_{im,ref}, V_D, p_{im}, p_{ic}, T_{im}) \tag{2.26}$$

Feedback

The feedback part consists of a PI-controller with integrator anti wind-up.

$$e_n = p_{im,ref} - p_{im} \tag{2.27}$$

$$I_{thr,n} = I_{thr,n-1} + K_{p,thr}\frac{T_s}{T_{i,thr}}e_n + \frac{T_s}{T_{i,thr}}(\alpha_{sat,n} - \alpha_n) \tag{2.28}$$

$$\alpha_{fb} = K_{p,thr}e_n + I_{thr,n} \tag{2.29}$$

$$\alpha_n = \alpha_{ff,n} + \alpha_{fb,n} \tag{2.30}$$

$$\alpha_{sat,n} = \begin{cases} 1 \ , \ \alpha_n > 1 \\ \alpha_n \ , \ 0 \leq \alpha_n \leq 1 \\ 0 \ , \ \alpha_n < 0 \end{cases} \tag{2.31}$$

2.9.2 Wastegate Controller

The wastegate controller also consists of one feedback part with a PI-controller with tracking to prevent integrator wind-up and one feedforward part. The feedforward part is a static feedforward which has been mapped for a few different target intercooler pressures. The PI-controller is then used to fine tune the wastegate position. The throttle setpoint, $\Delta p_{thr,ref}$, is the desired pressure drop over the throttle.

$$wg_{pos,ff} = f(p_{ic,ref}) \tag{2.32}$$

$$p_{ic,ref} = p_{im,ref} + \Delta p_{thr,ref} \tag{2.33}$$

$$e_n = p_{ic,ref} - p_{ic} \tag{2.34}$$

$$I_{wg,n} = I_{wg,n-1} + K_{p,wg}\frac{T_s}{T_{i,wg}}e_n + \frac{T_s}{T_{i,wg}}(wg_{pos,sat,n} - wg_{pos,n}) \tag{2.35}$$

$$wg_{pos,fb} = K_{p,wg}e_n + I_{wg,n} \tag{2.36}$$

$$wg_{pos,n} = wg_{pos,ff,n} + wg_{pos,fb,n} \tag{2.37}$$

$$wg_{pos,sat,n} = \begin{cases} 1 \ , \ wg_{pos,n} > 1 \\ wg_{pos,n} \ , \ 0 \leq wg_{pos,n} \leq 1 \\ 0 \ , \ wg_{pos,n} < 0 \end{cases} \tag{2.38}$$

2.10 Measurements

This section described the measurements provided by the Division of Vehicular systems and Volvo Car Corporation.

Division of Vehicular systems

Division of Vehicular systems provided a map consisting of a steady state measurements. These measurements are used to parameterize the MVEM-model. A map containing turbine and compressor measurements was also provided to parameterize the turbocharger.

Volvo Car Corporation

Volvo Car Corporation provided a set of transient measurements made in an engine rig. There are six measurements. The first four are made with a constant engine speed 3800 rpm and the target intercooler pressure equal to the target intake manifold pressure. The last two measurements are made with a constant engine speed at 4600 rpm and the intercooler target pressure is set to be 10% higher than the intake manifold pressure.

- The first set is two step responses from steps in acceleration pedal position 20-70% at a constant engine speed at 3800 rpm.
- The second set is the same setup as the first set, but there are 4 steps, where the first two set are identically with the first set and the later two steps are made with the wastegate position fixed.
- The third set is made with a fixed wastegate position during constant engine speed at 3800 rpm and several steps in throttle angle.
- The fourth set is made with the same engine speed and throttle steps as in the third set, but the acceleration pedal position is fixed at 80% to get a fixed target intercooler pressure for the wastegate controller.
- The fifth set is two step responses from steps in acceleration pedal position 20-75% at a constant engine speed at 4600 rpm.
- The sixth set is measured for an engine speed fixed at 4600 rpm, a constant acceleration pedal position at 65% and several steps in throttle angle. During the first half of the set the wastegate controller is on and during the second half of the measurement the wastegate position will be fixed.

In addition to these engine measurements made in a rig, several transient measurements have been done on the road in a test vehicle.

2.11 Model Simplifications and Limitations

There is a few things that have been excluded from the model. A surge valve model hasn't been implemented, which means that the model can't dump boost pressure. The reason for not implementing the surge valve is because of the focus of this thesis is the pressure build up and the phenomenon that arises during the pressure increase and not when the pressure decreases. Another aspect in limitation is the wastegate implementation. The thesis model makes it possible to control the effective area directly and letting a certain wastegate position correspond to a certain effective area. In reality the wastegate PWM control signal controls a solenoid valve which is connected to two different pressures, which means that a certain wastegate position control signal will give a position varying with the to the solenoid connected pressures. There is also no λ-controller to control the amount of injected fuel, λ is assumed to be constant 1.

Another aspect when comparing simulated with measured results. The model is parameterized with data from another engine so there are differences. But the phenomenon occur in both simulations and real engine measurements. The model results shows a good match with the actual measurements and therefore the model approach seems adequate to handle the throttle/turbo effects.

Chapter 3

Simulation Study

This chapter describes the experimental setup and the given result. The idea is to remove/change the dynamic of a certain component and then evaluate and compare the simulated results to find the cause for the overshoot and oscillation problem. The chapter starts with a characterization of the overshoot and oscillation, followed by experiments to find the origin why these phenomenon occur.

3.1 Characterization of the overshoot and oscillations

To recreate the phenomenon described in the problem section the entire model is simulated with a constant speed and a step in acceleration pedal position. The model inputs is shown in Figure 3.1. The inputs are the same as for the measurements made in engine rig. In this experiment, the target intercooler pressure is set to be the same as the target intake manifold pressure.

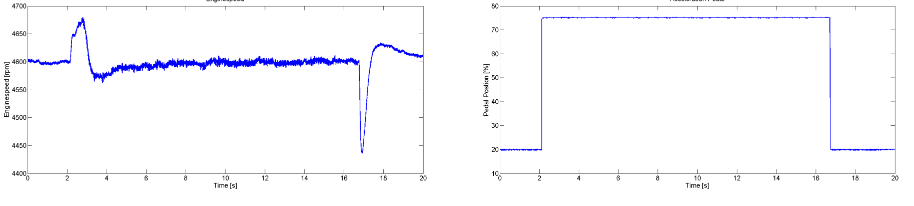

Figure 3.1. Engine speed to the left and step in acceleration pedal position to the right. Both these inputs is taken from the measurements. The engine speed is aimed to be constant 4600 rpm and a step in acceleration pedal position from 20-75%.

The pressure response to the simulation is shown in Figure 3.2. The intercooler pressure (blue) as well as the intake manifold pressure (black) overshoots. Because the throttle is faster than the wastegate to control the pressure, the intake manifold pressure overshoot is compensated for faster. When the intake manifold pressure has reached its target pressure, the intercooler pressure has dropped to much, resulting in an undershoot for both the intercooler and intake manifold pressure. Then they will follow each other to the target pressure. The throttle angle and wastegate position during the simulation can be seen in Figure 3.3. As seen in the figure, the throttle cuts the flow at around 3 seconds by going from the value 1 to 0.4 and back to 1 again. The figure also shows how the wastegate position overcompensates for the overshoot, which is seen in the overshoot before it reaches its target value.

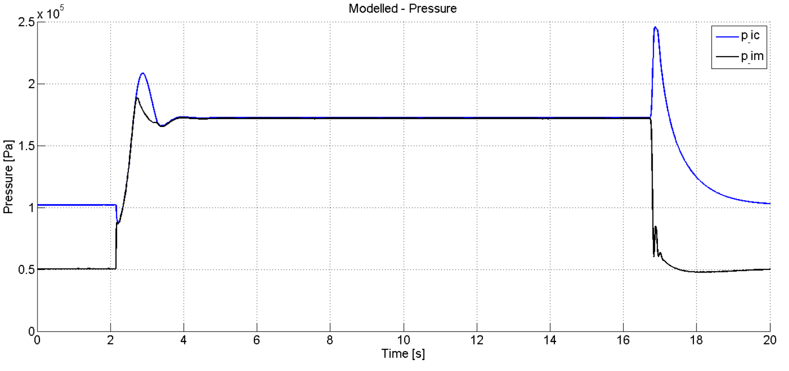

Figure 3.2. Pressure responses. Intercooler pressure, p_{ic}, (blue) and intake manifold pressure, p_{im}, (black). In the beginning both pressures are constant, p_{ic} at 101kPa and p_{im} at 50kPa. When the step comes, p_{im} rapidly increases while p_{ic} decreases. When the two pressures meet, they both increase as the turbocharger spins up. The pressure reaches its target value and the throttle cuts the flow with a delay resulting in an overshoot in p_{im}. p_{ic} also overshoots and the controller overcompensate the overshoot resulting in an undershoot which takes p_{im} with it below the target value. The controllers then slowly takes both p_{ic} and p_{im} to its target value.

Another interesting figure is Figure 3.4, where the compressor torque, turbine torque and the turbocharger rotational speed is shown. The figures shows an overshoot as well as an oscillatory-behavior in all three subplots.

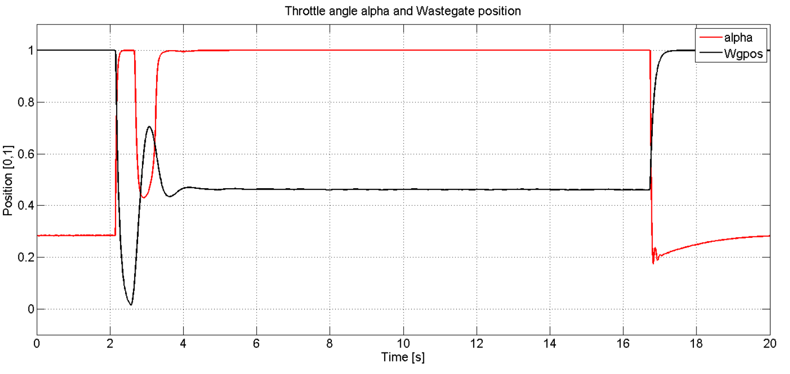

Figure 3.3. Throttle angle alpha, α_{thr} , (red) and Wastegate position, wg_{pos} , (black) during the simulation. First both the wg_{pos} and α_{thr} are constant, when the step comes, the α_{thr} gets fully open and the wg_{pos} goes from 1 against 0. At about 3 seconds, the throttle cuts the flow, seen by the α_{thr} goes from 1 to about 0.45 and back to 1 again. During the same time wg_{pos} reacts to the p_{ic} overshoot in Figure 3.2. wg_{pos} overcompensate the p_{ic} overshoot by opening the wastegate to 70%. Then the wg_{pos} slowly decreases and finally reaches its final value.

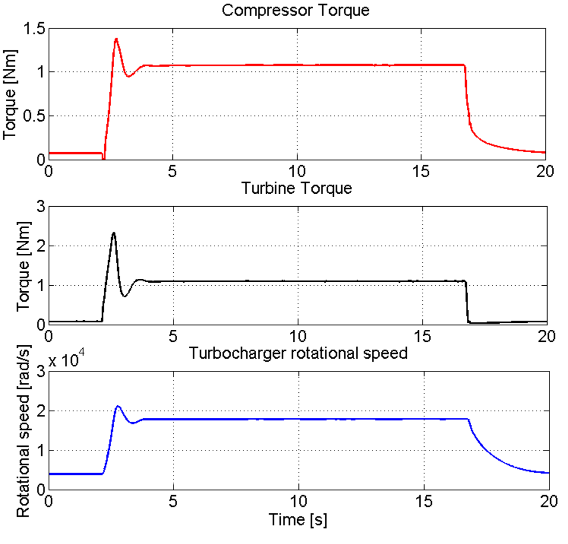

Figure 3.4. Top: Compressor braking torque. Middle: Turbine driving torque. Bottom: The resulting turbocharger rotational speed. All three subplots are linked together, the turbine driving torque, which accelerates the turbocharger rotational speed and the compressor braking torque which decelerates the turbocharger rotational speed. The three subplots shows that the oscillations in present in the turbocharger system as well as the pressure system.

In some cases it is desired to have a pressure drop over the throttle, which enable the throttle to quickly react on an increased torque demand. In that case, the intercooler pressure will be above the desired intake manifold pressure, typically about 10% higher. The same experimental setup as before, with the model input set to be the same input as in Figure 3.1 but with a 10% pressure drop over the throttle, in other words the target intercooler pressure is set to be 1.1 times the target intake manifold pressure. The resulting pressure is shown in Figure 3.5 and the throttle angle and the wastegate position is shown in Figure 3.6. Comparing these figures with the corresponding figures, Figure 3.2 and Figure 3.3, a more oscillatory-behavior can be found in the set up with a pressure drop over the throttle. In Figure 3.6 its clear that the controllers counteracts each other which results in an oscillating pressure response.

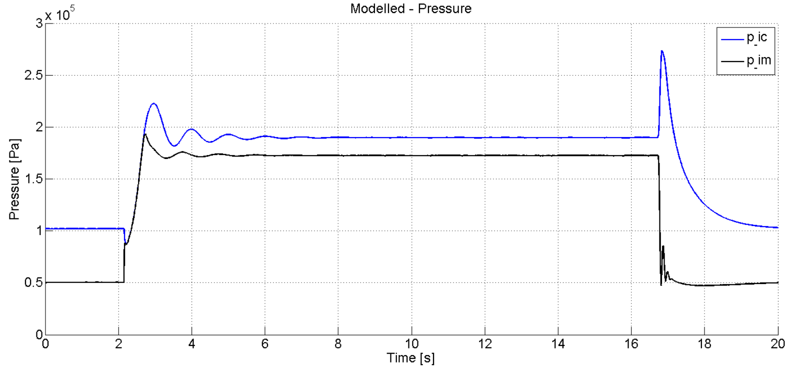

Figure 3.5. Pressure response with a 10% pressure drop over the throttle. Intercooler pressure, p_{ic}, (blue) and intake manifold pressure, p_{im}, (black). In the beginning both pressures are constant, p_{ic} at 101kPa and p_{im} at 50kPa. When the step comes, p_{im} rapidly increases while p_{ic} decreases. When the two pressures meet, they both increase as the turbocharger spins up. The pressure reaches its target value and the throttle cuts the flow with a delay resulting in an overshoot in p_{im}. p_{ic} also overshoots. Both actuators respond to the overshoots and because the pressures affects each other, it leads to an overcompensation resulting in an undershoot. The actuators overcompesates for the undershoot which leads to an overshoot in both pressures and so on. The magnitude of the over/undershoots is decreasing and they are damped out after a few seconds. Comparing with Figure 3.2 the pressure drop over the throttle results in a more oscillatory behavior due to the fact that both actuators is trying to minimize the control error without considering the other actuator.

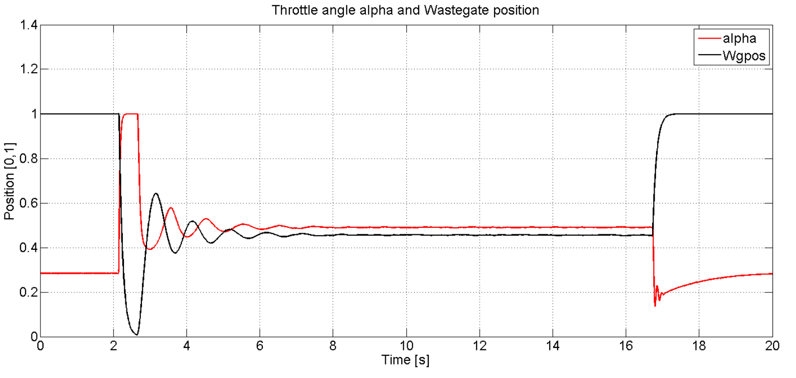

Figure 3.6. Throttle angle, α_{thr} , (red) and Wastegate position, wg_{pos} , (black) during the simulation. First both wg_{pos} and α_{thr} are constant, when the step comes, α_{thr} gets fully open and wg_{pos} goes from 1 against 0. At about 3 seconds, the throttle cuts the flow, α_{thr} goes from 1 to about 0.40. During the same time wg_{pos} reacts to the p_{ic} overshoot in Figure 3.5. wg_{pos} overcompensate the p_{ic} overshoot by opening the wastegate to 65%. Then both controllers try to minimize their control error but they counteract each other resulting in a oscillatory behavior which slowly is damped out.

3.1.1 Closed vs Open Loop

To investigate if the problems with overshoots and oscillations occurs in both open and closed loop, a series of simulations was performed.

Closed loop

The closed loop simulations is done in the section 3.1. In the closed loop simulation, both overshoot and oscillations in pressure are visible. See in Figure 3.2 and Figure 3.5.

Open loop

This simulation uses a fixed engine speed at 4600 rpm with both controllers off. There is a step from 1 to 0.46 in wastegate position and a step from 0.28 to 0.49 in throttle angle. The steps are taken from where the positions have settled in Figure 3.6. The resulting pressure and both the control signals, throttle angle and wastegate position, can be seen in Figure 3.7 and Figure 3.8 respectivly. In the open loop response, there are no overshoots or oscillations in pressure.

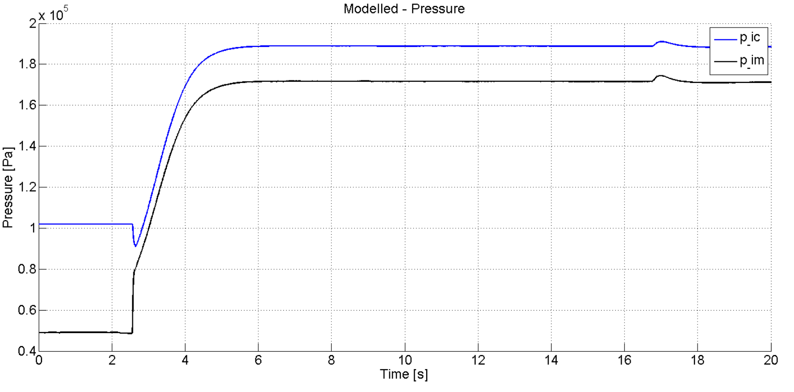

Figure 3.7. Pressure - Throttle controller off, wastegate controller off. There are no overshoots or oscillations in either p_{ic} or p_{im} . The bulb on the pressure lines at t=17s is caused by the dip in engine speed seen in Figure 3.1

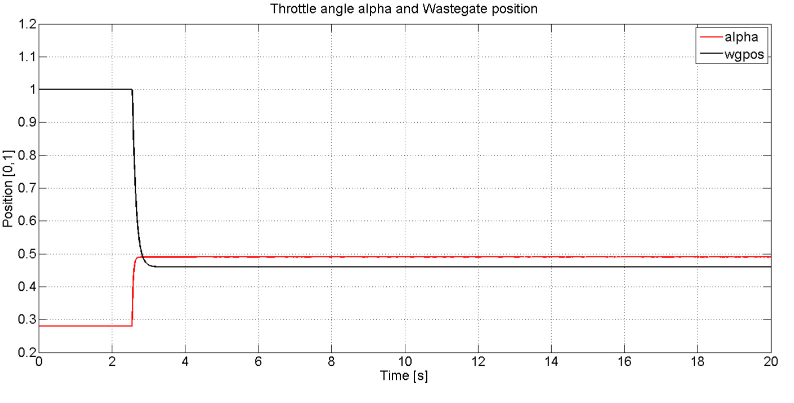

Figure 3.8. Wastegate and throttle position - Throttle controller off, wastegate controller off. Step from 1 to 0.46 in wastegate position and a step from 0.28 to 0.49 in throttle angle. The corresponding pressure response in shown in Figure 3.7.

Throttle controller on - Wastegate controller off

In this simulation a fixed engine speed at 3800 rpm and a step in acceleration pedal position are used as inputs. The wastegate is set to be fixed at 0.45. The result can be seen in Figure 3.9 and Figure 3.10 where also a small overshoot in intake manifold pressure (black) can be seen.

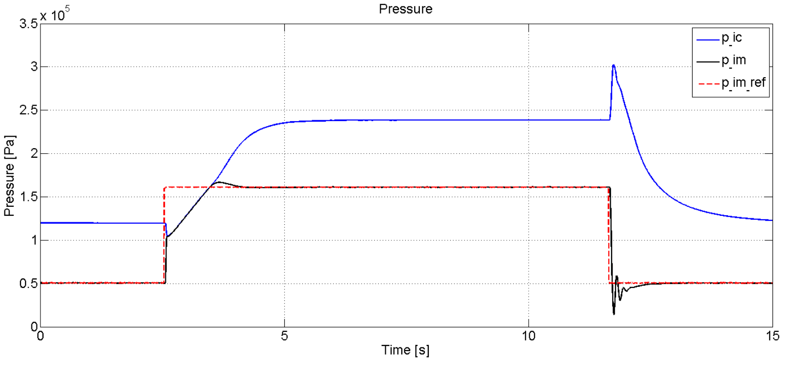

Figure 3.9. Pressure - Throttle controller on, wastegate controller off. There is a small overshoot in p_{im} but no oscillations. The fixed wastegate position give the p_{ic} response to be slow and that makes the p_{im} response slow.

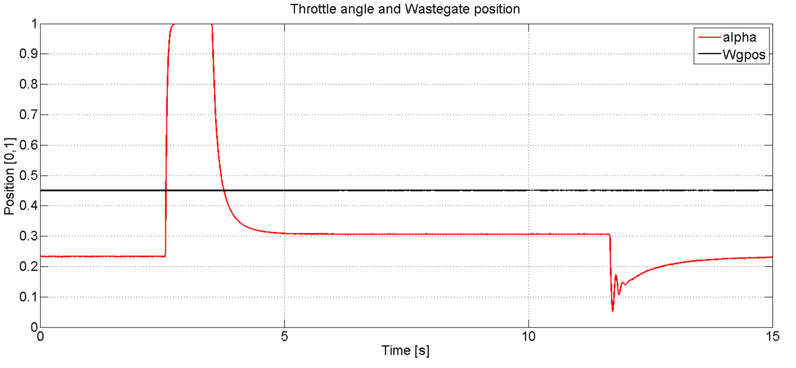

Figure 3.10. Wastegate (black) and throttle angle (red) - Throttle controller on, wastegate controller off. The wastegate position is held fixed at 0.45 at all times. The throttle starts at a constant value and when the step comes, the throttle gets fully open. When it has reached its target value, the throttle cuts the flow by closing and slowly decreasing and reaching a constant value.

Throttle controller off - Wastegate controller on

In this simulation a fixed engine speed at 3800 rpm and a constant acceleration pedal position at 80% are used as inputs. The throttle is set to perform a numerous of steps while the wastegate controller is turned on. The result is shown in Figure 3.11 and Figure 3.12 where the intake manifold pressure (black) shows an overshoot but no oscillations as a result of the throttle step and the intercooler pressure (blue) also shows an overshoot at the up-step of the throttle.

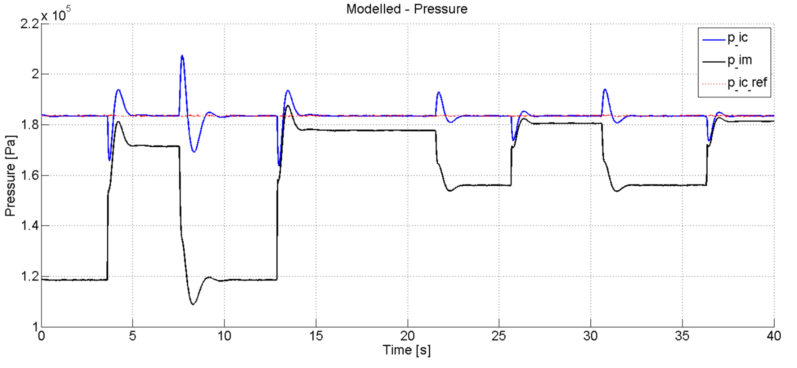

Figure 3.11. Pressure - Throttle controller off, wastegate controller on. p_{im} overshoots in every step, which is caused by the wastegate controller overshoots in its attempt to keep p_{ic} constant. p_{ic} shows an oscillatory behavior before it reaches its target value, caused by the pressure drop when the throttle opens and the resulting overcompensation that causes the overshoot.

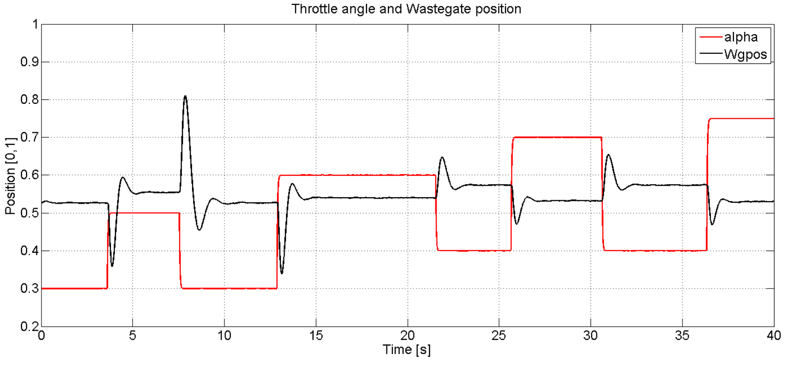

Figure 3.12. Wastegate and throttle position - Throttle controller off, wastegate controller on. The red line shows the numerous of step in throttle angle. The black line shows the controlled wastegate position. The wastegate controller reacts hard and overcompensate for the control error, causeing the overshoot and the oscillation in p_{ic} shown in Figure 3.11.

3.1.2 Summary of characterization

With both controllers off, there is no overshoot and no oscillations in pressure after a step in throttle angle and a step in wastegate position. With the throttle controller on and the wastegate position held fixed during a step in acceleration pedal position, the result is a very small overshoot in intake manifold pressure. The intercooler pressure shows neither oscillations or overshoots. With the wastegate controller on, with an aim to keep a constant intercooler pressure and the throttle doing a few steps in throttle angle. The intake manifold pressure shows an overshoot but no oscillations. With both controllers on, both overshoots and an oscillatory-behavior is present in the pressure response from the step in acceleration pedal position. To sum up, with the wastegate controller off, the overshoot is drastically reduced. But with the controller on, there is overshoot both with and without the throttle controller on. But the oscillatory behavior is only present when both controllers are on, and it is caused by the two controllers counteracting each other. The oscillations is most apparent in Figure 3.5 but it is also apparent in Figure 3.2. In Figure 3.2 when the intercooler pressure undershoots, it is forcing the intake manifold pressure to follow, causing it to undershoot. Then both controllers wants to increase pressure, but it is only the wastegate controller whom can affect the pressures.

3.2 Badly tuned controllers

This section investigates if the controllers tuning is causing the overshoots and oscillations. Because of the driveabilty aspect of the vehicle, the torque response from a step at acceleration pedal position needs to be fast. The torque is directly coupled with the intake manifold pressure, which means that the pressure response needs to be fast. Therefore a slow controller that will build up the pressure during a longer time to prevent overshoots and oscillations isn't an option. But to investigate the effect of different control parameters, a series of simulations with different control parameters is performed. The model inputs in this simulations is a constant engine speed at 4600 rpm with a step in acceleration pedal position 20-75%. The simulation utilizes a pressure drop over the throttle, set to be 10% of the target intake manifold pressure. This setup is to maximize the oscillation.

P-part of Wastegate Controller

This part tests four different settings on the proportional part of the PI-controller by varying the proportional gain $KpWg$. Because $KpWg$ also affects the I-part of the controller, the integration time is compensated with the same factor. First a simulation with the original $KpWg$ followed by simulations with $0.5KpWg$, $0.2KpWg$ and $2KpWg$. The result is shown below in Figure 3.13. A lower $KpWg$ gives a bigger overshoot as well as it takes longer time for the system to stabilize and a oscillatory behavior arises. A higher $KpWg$ gives a smaller overshoot but the controller always overcompensates the over/undershoots and therefore a very oscillatory behavior occur.

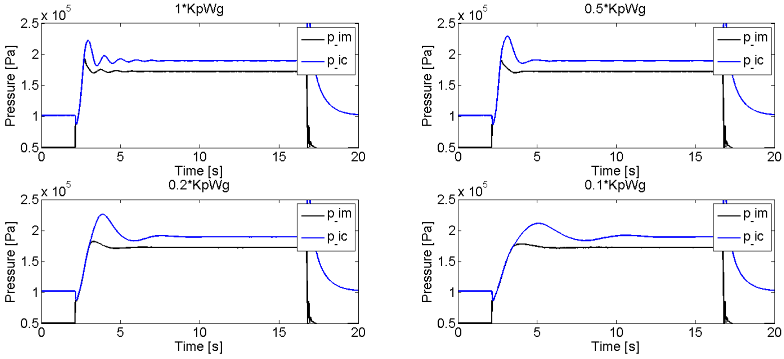

Figure 3.13. Pressure response with varying proportional gain at the wastegate controller, intercooler pressure, p_{ic} (blue) and intake manifold pressure, p_{im} (black). Top left is showing the pressure response from the original $KpWg$, showing both overshoots and oscillations and is used as a reference. Top right figure shows pressure response for $0.5KpWg$, showing a bigger overshoot and a more oscillatory behavior than the $KpWg$ response. Bottom left figure shows $0.2KpWg$ which has an even bigger overshoot and a more oscillatory-behavior. Bottom right is for $2KpWg$ and has a smaller overshoot than $KpWg$ but a very oscillatory-behavior.

I-part of Wastegate Controller

In this section same test as above is performed, but instead of changing the proportional gain the integration time, $TiWg$, is varying. The simulations is done with first the original $TiWg$ followed by $0.5TiWg$, $2TiWg$ and $4TiWg$. The Figure 3.14 clearly shows a more oscillatory behavior when lowering the integration time. All responses are equally fast and a longer integration time gives less oscillations. The downside with a small I-part, in other words a big $TiWg$, is that the elimination of steady-state error is much slower.

P-part of Throttle Controller

This part tests four different settings on the proportional part of the throttle PI-controller by varying the proportional gain $KpThr$. Because $KpThr$ also affects the I-part of the controller, the integration time is compensated with the same factor. First a simulation with the original $KpThr$ followed by simulations with $0.5KpThr$, $0.1KpThr$ and $2KpThr$. The result is shown below in Figure 3.15. The pressure responses for the different proportional gains are very alike, showing both oscillations and overshoots.

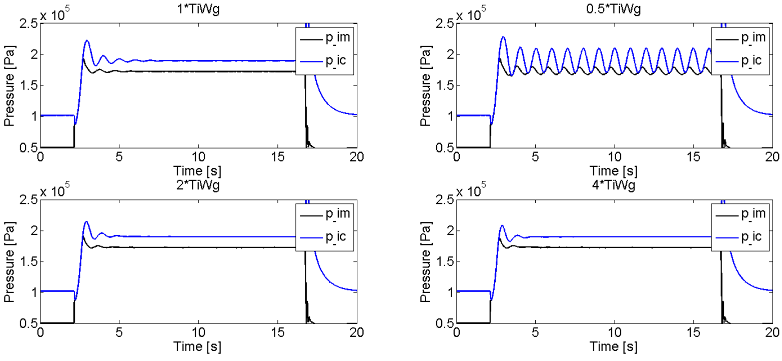

Figure 3.14. Pressure response with varying integration time at the wastegate controller, intercooler pressure, p_{ic} (blue) and intake manifold pressure, p_{im} (black). Top left is showing the pressure response from the original $TiWg$, showing both overshoots and oscillations and is used as a reference. Top right figure shows pressure response for $0.5TiWg$, showing a bigger overshoot and a much more oscillatory behavior than the $TiWg$ response. Bottom left figure shows $2TiWg$ which has a slightly smaller overshoot and a slightly less oscillatory-behavior comparing with $TiWg$. Bottom right is for $4TiWg$ and has a even smaller overshoot and a less oscillatory behavior than the $TiWg$ and $2TiWg$ simulations

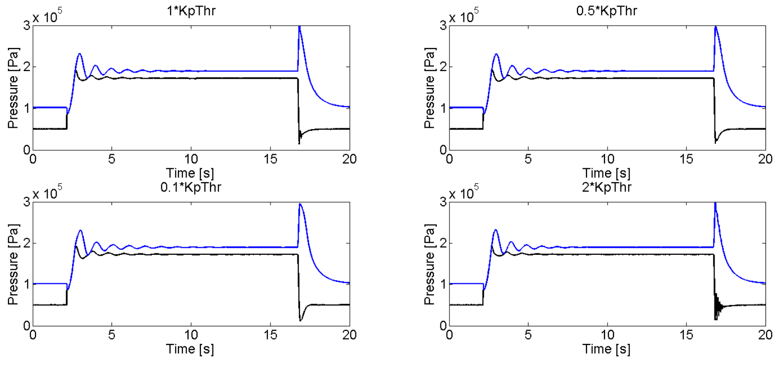

Figure 3.15. Pressure response with varying proportional gain at the throttle controller. The pressure response is pretty much the same for all four different $KpThr$, showing both overshoots and oscillations.

I-part of Throttle Controller

In this section same test as above is performed, but instead of changing the proportional gain the integration time, $TiThr$, is varying. The simulations is done with first the original $TiThr$ followed by $0.5TiThr$, $2TiThr$ and $4TiThr$. The pressure response is shown in Figure 3.16. A higher integration time reduces the oscillations, but the oscillations is still present. A higher integration time also increases the time to eliminate a steady-state error.

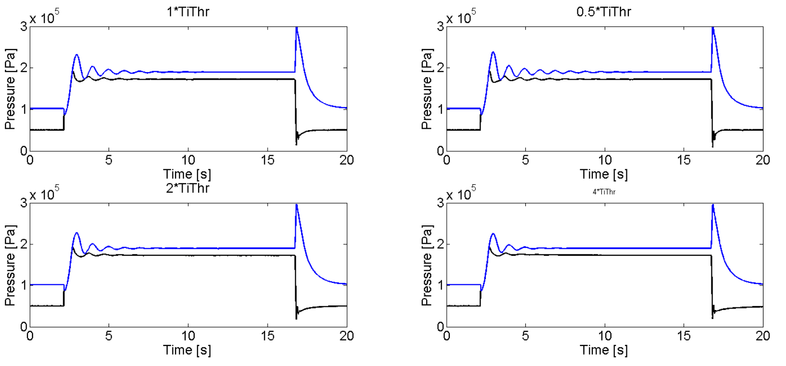

Figure 3.16. Pressure response with varying integration time at the throttle controller. The four responses are pretty much the same, but the oscillations damps out faster with a bigger integrational time $TiThr$.

Summary of badly tuned controllers

To summarize the simulations with different tuning on the PI-controllers of both the wastegate and throttle controller shows that the original tuning is good and that the wastegate controller tuning have more effect on the overshoots and oscillations. A higher integration time in both controllers seems to reduce the oscillations, but at the cost of steady-state error elimination.

3.3 Actuator dynamics

In this section the actuator dynamics and thier impact on the overshoot and oscillations is investigated. The investigation consists of a series of simulations with varying time constants in the first-order system which the actuator dynamics is modeled.

Without actuator dynamics

First both actuator dynamic models are removed, i.e the target position and the actual position is the same at all times. Then a simulation with constant engine speed at 3800 rpm with a step in acceleration pedal position from 20-80% is performed. The result is shown in Figure 3.17, where the intercooler and intake manifold pressure is plotted. The figure shows a very small overshoot in intake manifold pressure and a bigger overshoot in intercooler pressure. In Figure 3.18 the throttle angle α and the wastegate position for the simulation is shown. In summary there is still an overshoot in pressure even though the wastegate position and throttle angle is controlled without delay and actuator dynamics.

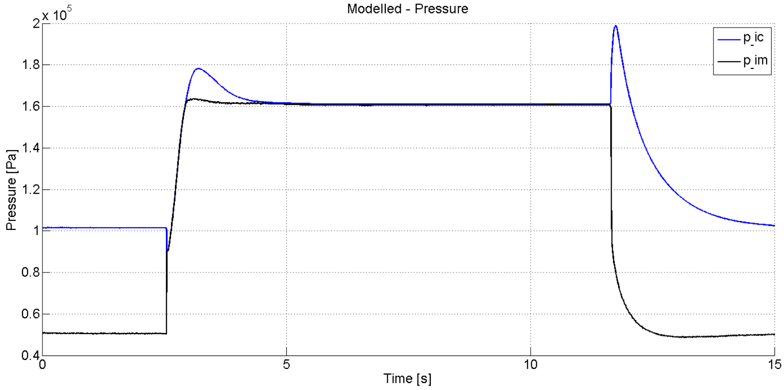

Figure 3.17. Step response in pressure without actuator dynamics. There is basically no overshoot in intake manifold pressure (blackline) but there is still a overshoot in intercooler pressure (blueline). At the end of the step, around $t = 12s$, the effects of a missing surge valve is seen, causing the p_{ic} to rapidly increase when the throttle is closed. With a surge valve, the pressure could have been dumped and the p_{ic} would have decreased faster.

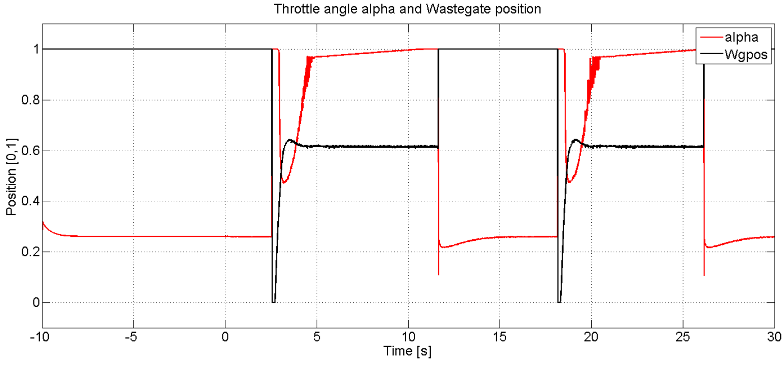

Figure 3.18. Wastegate and throttle position during step in pedal position. Throttle angle α_{thr} (red) and wastegate position wg_{pos} (black). In the beginning both positions is constant, when the step comes, both positions reacts, α_{thr} gets fully open and wg_{pos} goes from fully open to fully closed. When the pressures, see Figure 3.17, reaches its target value, both positions reacts and cuts the flow, the wg_{pos} gets a small overshoot before its settles at around 0.6. The α_{thr} reacts on the change in p_{ic} and keeps p_{im} at its target value by increasing and going towards 1.

Throttle actuator dynamics investigation

To investigate the throttle actuator dynamics impact of the overshoot and oscillations, simulations with constant speed and step in acceleration pedal position with varying time constant in the first-order system which approximate the actuator dynamics is performed. Figure 3.19 shows the intercooler and the intake manifold pressure as a result of a step in acceleration pedal position with constant engine speed of 3800 rpm. Figure 3.20 shows the same but with an engine speed at 4600 rpm. In both figures the throttle actuator time constant is varying. Time constant, from top to bottom, left to right; 0 s, 30 ms, 50 ms, 100 ms, 150 ms, 200 ms. The figures show that the intercooler pressure is pretty much the same for all the different time constants as well as the behavior of the intake manifold pressure, but the overshoot increases with increased time constant.

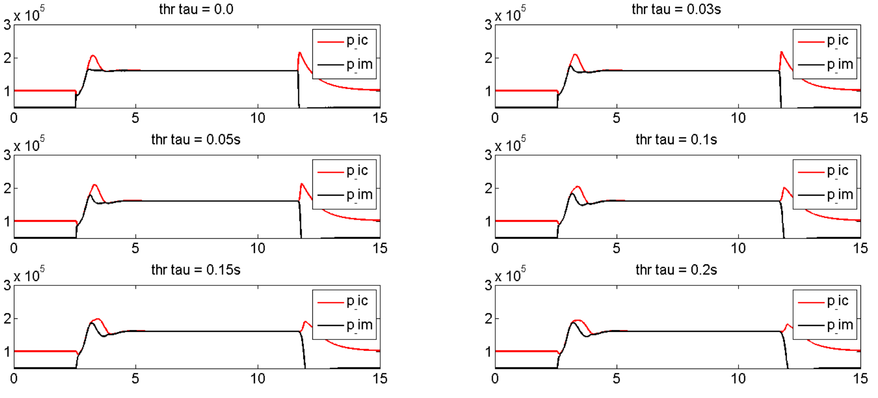

Figure 3.19. Throttle actuator dynamics simulations. p_{ic} (red) and p_{im} (black) as a result of a step in acceleration pedal position with constant engine speed at 3800 rpm. The time constant, τ, is set to 0 s, 30 ms, 50 ms, 100 ms, 150 ms ans 200 ms, with 0 s in the top left corner and 200 ms in the bottom right corner. The figure with a τ at 0s gives no overshoot or oscillations at p_{im} but still an overshoot in p_{ic} . For all other time constants, the behavior is the same. Both the p_{ic} and p_{im} overshoots, and the overshoot is overcompensated resulting in a small undershoot. A bigger τ gives a bigger p_{im} overshoot and makes the p_{ic} takes longer time to settle at its target value.

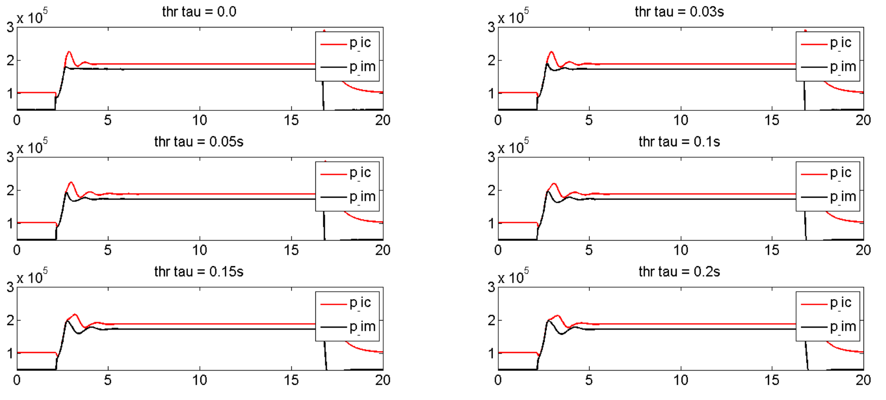

Figure 3.20. Throttle actuator dynamics simulations. p_{ic} (red) and p_{im} (black) as a result of a step in acceleration pedal position with constant engine speed at 4600 rpm. The time constant is set to 0 s, 30 ms, 50 ms, 100 ms, 150 ms and 200 ms, with 0 s in the top left corner and 200 ms in the bottom right corner. The figure with a τ at 0s gives no overshoot or oscillations at p_{im} but p_{ic} overshoots and oscillate. For all other time constants, the behavior is the same. Both the p_{ic} and p_{im} overshoots, the controller counteract each other resulting in an oscillatory behavior. A bigger τ gives a bigger p_{im} overshoot and makes the p_{ic} takes longer time to settle at its target value. The amount of oscillations is the same, but the magnitude increases with an increased τ.

Wastegate actuator dynamics investigation

As for the throttle actuator dynamics investigation the same simulations is done with varying time constants on the wastegate actuator dynamics. Figure 3.21 shows the intercooler and the intake manifold pressure as a result of a step in acceleration pedal position with a constant engine speed of 3800 rpm. Figure 3.22 shows the same but with the constant engine speed 4600 rpm. In both figures the wastegate actuator time constant, τ, is varying. Time constant, from top to bottom, left to right; 0 s, 50 ms, 100 ms, 150 ms, 200 ms, 300 ms. The figures show that the overshoot in both intake manifold pressure and intercooler pressure is increasing with increased time constants as well as that a self-oscillation behavior is getting more and more apparent with an increased time constants. Especially in the 4600 rpm simulation with wastegate actuator time constant set to 200-300 ms, Figure 3.22, the self-oscillations become very clear.

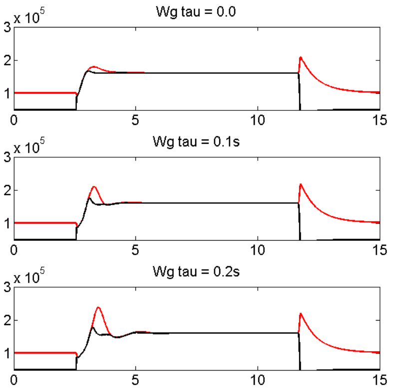

Figure 3.21. Wastegate actuator dynamics simulations. Intercooler(red) and intake manifold pressure(black) as a result of a step in acceleration pedal position with constant engine speed at 3800 rpm. Time constants, τ, from top to bottom left to right; 0 s, 50 ms, 100 ms, 150 ms, 200 ms, 300 ms. A higher τ gives a bigger overshoot in p_{ic} , and a bigger overshoot in p_{ic} gives a bigger overshoot in p_{im} . The bigger overshoots and the bigger delay in the actuator makes it harder control the pressures againt it target value, causing an oscillatory behavior.

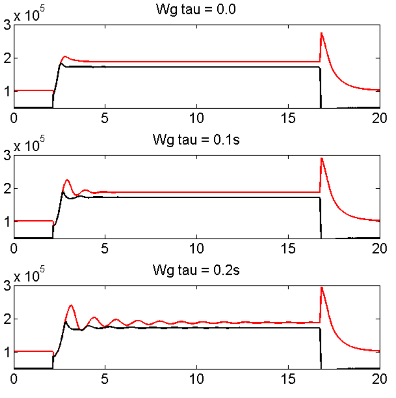

Figure 3.22. Wastegate actuator dynamics simulations. Intercooler(red) and intake manifold pressure(black) as a result of a step in acceleration pedal position with constant engine speed at 4600 rpm. Time constants, τ, from top to bottom left to right; 0 s, 50 ms, 100 ms, 150 ms, 200 ms, 300 ms. A higher τ gives a bigger overshoot in p_{ic} . The overshoot in p_{ic} causes an overshoot in p_{im} . A bigger τ makes it difficult for the pressures to settle at their target value cauing an oscillatory behavior. The oscillations is very apparant in simulations with $\tau = 200 - 300ms$.

Summary of Actuator dynamics

Without actuator dynamics for both the wastegate and the throttle there is a small overshoot in intake manifold pressure and intercooler pressure. Varying the time constant in the throttle actuator dynamic the intercooler pressure is pretty much the same for all constants, but the overshoot in intake manifold pressure is increasing with increasing time constants, that is due to that the throttle reacts slower and therefore cuts the flow later which gives a higher pressure. Varying the time constant in the wastegate actuator dynamic effects both the intercooler and intake manifold pressure. Both pressures overshoots increases with increased time constant and the additional time delay makes it difficult for the controller to get the pressures to settle at their target values, causing a self-oscillatory behavior. It could be noted that the controllers are tuned for a throttle actuator time constant at 30 ms and the wastegate actuator time constant at 100 ms.

3.4 Throttle cutting flow

Since the intercooler control volume is connected to the intake manifold control volume by the throttle, the pressure in both of these volumes are affected by the throttle. When the throttle restricts the flow between the intercooler and intake manifold there might be an increase of pressure in the intercooler pressure. To investigate this, two simulations is made. Both simulations use constant engine speed at 3800 rpm and a acceleration pedal step from 20-80%. In the first simulation, both controllers are on and the throttle cuts the flow when the intake manifold pressure reaches its target pressure. In the other simulation, the throttle initially have the same reaction on the step in acceleration pedal position, but the throttle cutting of the flow is ignored and instead the throttle stay fully open. Figure 3.23 shows the intercooler and intake manifold pressure responses and Figure 3.24 shows throttle angle and wastegate position from both simulations. When the throttle isn't cutting the flow, the intake manifold and intercooler pressure follow each other, given both an overshoot but no oscillations. With the throttle cutting active, the intake manifold pressure produces a smaller overshoot compared to when the throttle ain't cutting flow. But as a result of the throttle cutting the intercooler pressure will get an increased overshoot compared to when the throttle isn't cutting flow. The increased overshoot results in an overcompensation from the wastegate controller which results in an undershoot. As the intercooler pressure undershoots it takes the intake manifold pressure with it down below the target pressure, giving both pressures an oscillatory-behavior.

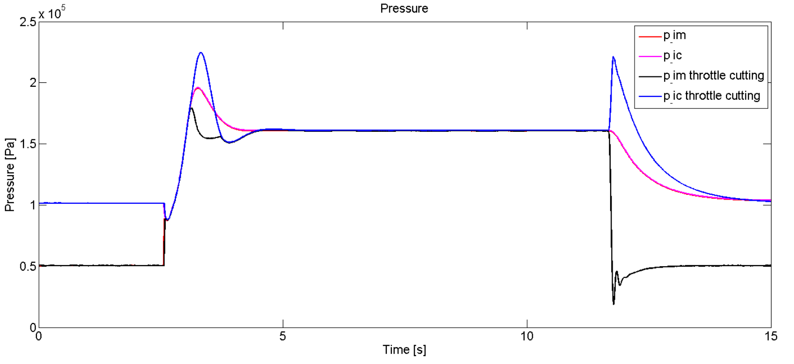

Figure 3.23. Intercooler pressure (p_{ic}, magenta and blue) and intake manifold pressure (p_{im}, red and black). Red and magenta shows the pressure when the throttle is set to not cut the flow, they are hard to separate as they follow each other. Black and blue shows pressure when the throttle cuts the flow. p_{ic} overshoots is both cases, but when the throttle ain't cutting the flow, the overshoot is smaller and it is controlled to its target value without undershoots. When the throttle cuts the flow, p_{ic} both over- and undershoots. p_{im} also overshoots in both cases, but with a smaller overshoot when the throttle cuts the flow, but it also follows the p_{ic} down in an undershoot before it reaches its target value.

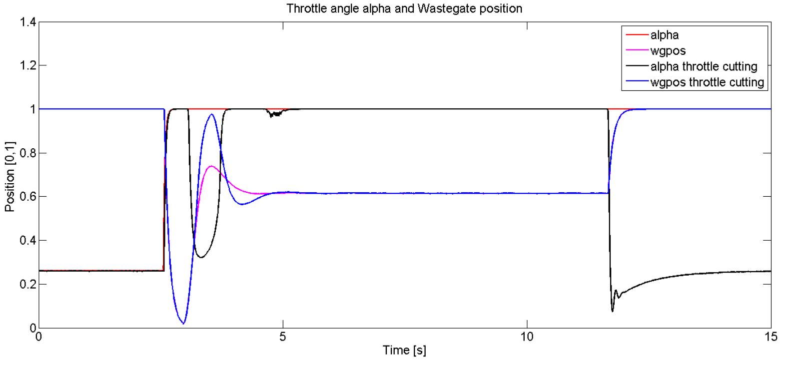

Figure 3.24. Throttle angle alpha (red and black) and wastegate position (magenta and blue). Red and magenta shows the position when the throttle is set to not cut the flow. Black and blue shows position when the throttle cuts the flow. When the throttle cuts the flow, the throttle angle goes from 1 to about 0.35 and back to 1 and the wastegate controller reacts harder as a result of the bigger overshoot in p_{ic} .

3.5 Sensor delay

This section investigates if there is a delay in the pressure sensors that causes this oscillations and overshoots. The simulations is done in two steps, first a simulation with no sensor delay and then a simulation with 50 ms delay on the pressure signals going to the ECU. The extra time delay causes a bigger overshoot and a more oscillatory behavior, see Figure 3.25. The corresponding control signals, throttle angle and wastegate position, can be seen in Figure 3.26 where the delay can be seen in the difference between when the controller reacts in the two cases. The delay causes the bigger overshoots which causes the wastegate to overreact and as a result giving the pressure a bigger undershoot as well.

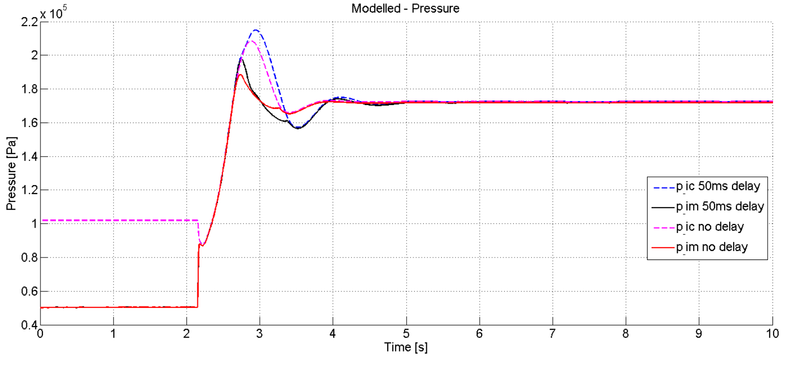

Figure 3.25. Pressure response with delay in pressure sensor compared to no delayed sensor. The figures shows both pressures giving a higher overshoot and more oscillations with delay.

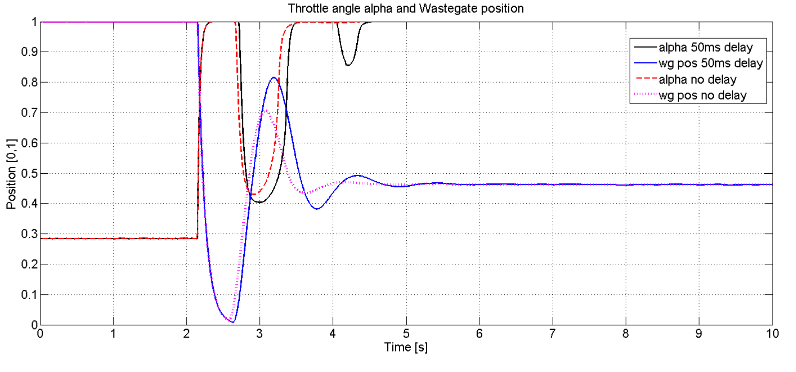

Figure 3.26. Control signals with delay in pressure sensors compared to no delayed sensors. The figures shows that the control signals reacted slower and more heavily.

3.6 Turbocharger dynamics

This section investigates if the pressure overshoot and oscillation have its core in the dynamics and delays in the turbocharger. A series of simulations with different configurations to unlink the turbochargers components from each other is performed, to capture which part that causes the pressure problems.

No delays and no dynamics

This setup is made by letting the wastegate control the rotational speed of the turbocharger directly. Mapping the turbocharger rotational speed in two points, one for the minimum speed and linking it to the wastegate effective area when the wastegate position is set to 1 and one for the maximum speed and linking it to the wastegate position 0. Linearization of the two points gives a corresponding turbocharger speed for every wastegate position. The wastegate PI-controller was also roughly tuned for the new situation. This configuration unlinks the turbine from the compressor. The simulation takes a constant engine speed at 4600 rpm and a step in acceleration pedal position from 20-80%. The results from the simulations is shown in Figure 3.27 and Figure 3.28.

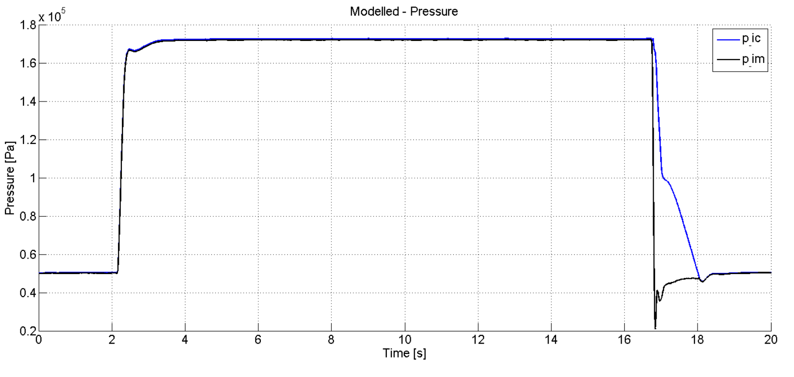

Figure 3.27. Pressure response with no delays in the turbocharger. Intercooler pressure, p_{ic} (blue) and intake manifold pressure p_{im} (black). The pressures follow each other and the step responses are fast without overshoots or oscillations. There is a small bulb on the pressure response caused by the controller, see Figure 3.28.

The figure 3.27 shows that the intercooler and intake manifold pressure follow each other without overshoot or oscillations. The little bulb on the pressure line, in the end of the pressure rise is caused by the controller which can be seen in the small oscillation before the wastegate position reaches it's final value in figure 3.28.

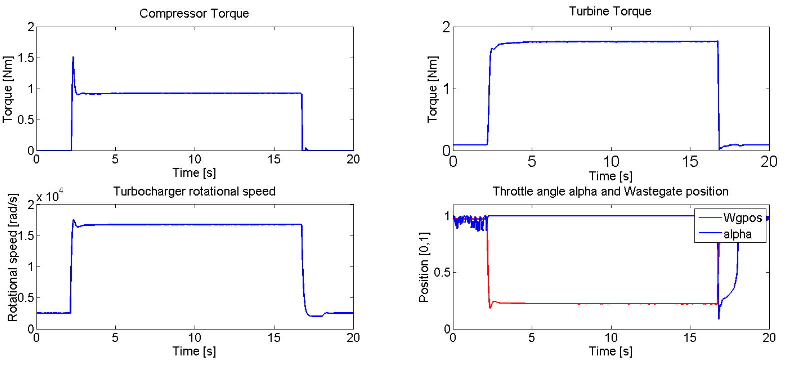

Figure 3.28. Top left: Compressor torque. Top right: Turbine torque. Bottom left: Turbocharger rotational speed. Bottom right: Throttle angle and wastegate position. The turbocharger rotational speed is controlled directly with the wastegate, giving it a very fast response. The fast response in rotational speed makes fast responses for both compressor and turbine torque. The control signals shows that the throttle angle is basically constant 1 at all times, and the wastegate starting at 1 and when reacting to the step, closes to about 0.2 with a little undershoot before settling at a constant value. That little overshoot causes the bulb mentioned in Figure 3.27.

Delay and dynamics with a first order system

Same setup as before, with the wastegate controlling the turbocharger speed directly, but adding a first-order system between wastegate controlled turbocharger speed and the actual turbocharger speed.

$$\omega_{TC} = \frac{\omega_{,TC,target}}{1 + s\tau} \tag{3.1}$$

Three simulations is performed with different time constants $\tau = 0.6s$, $\tau = 0.4s$ and $\tau = 0.2s$. The in parameters is constant engine speed at 4600 rpm and a step in acceleration pedal position 20-75%.

The result is seen in Figure 3.29 - Figure 3.31, which show the pressure response, control signals and turbocharger rotational speed. The result for the pressure for different τ is basically the same, but the response gets slower and slower and the intercooler pressure gets a bigger and bigger overshoot with a longer time constant. The alpha part of the control signals show that the throttle cuts the flow later for bigger τ and it also reacts harder as a result of the increased intercooler pressure overshoot. The turbocharger rotational speed also responds slower with a greater value on τ but the overshoot has the same height in all three simulations, but the time for the overshoot to reach the target value is longer with a bigger τ.

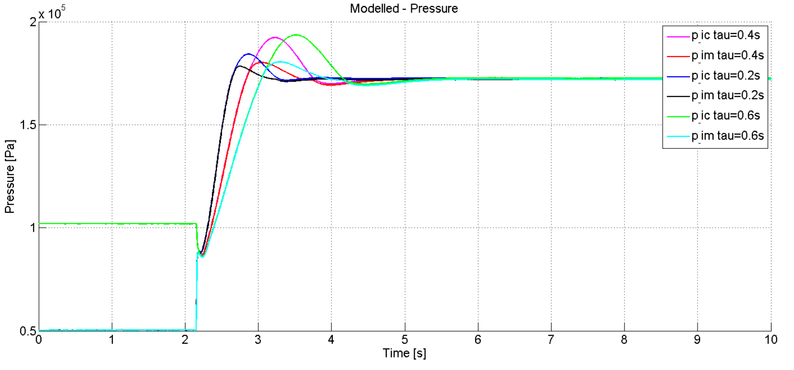

Figure 3.29. Pressure response with first-order system as delay and dynamics in the turbocharger. A bigger time constant, τ, causes the pressure response to get slower and slower, but it also causes the pressures to overshoot more, which is most apparent in p_{ic}, but p_{im} also gets a slightly bigger overshoot for a bigger τ. The settling time before the pressures have settled at the target value increases with increased τ.

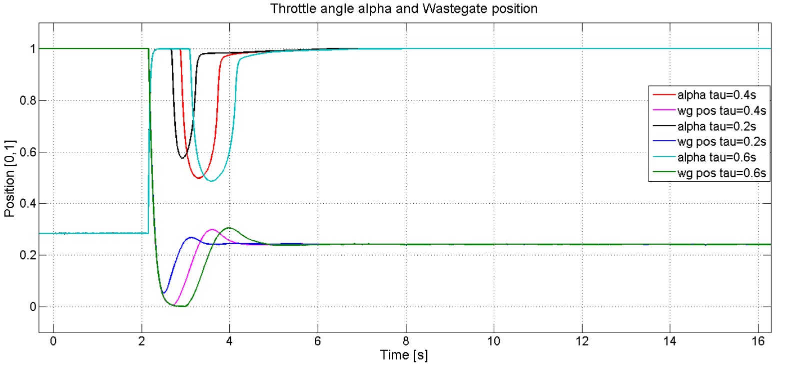

Figure 3.30. Control signals with first order system as delay and dynamics for turbocharger. The delays causes both control signals to act later and harder to compensate for the delayed response.

Controlling Turbine driving torque with wastegate

With this configuration, the turbine driving torque is controlled directly by the wastegate without any delays. This excludes the turbine dynamics and delays but keep the turbocharger shafts dynamics as well as the compressors dynamics. To let the wastegate control the turbine driving torque directly, a linearization is made, a wastegate position at zero correspond to 2.5 Nm and a wastegate position at one correspond to 0.0075 Nm. Wastegate position between 0 and 1 is then linearized between 2.5 and 0.075 Nm. The simulations takes a constant engine speed and a step in acceleration pedal position as inputs. The result is shown in Figure 3.32 and Figure 3.33. The configuration without the turbine dynamics and

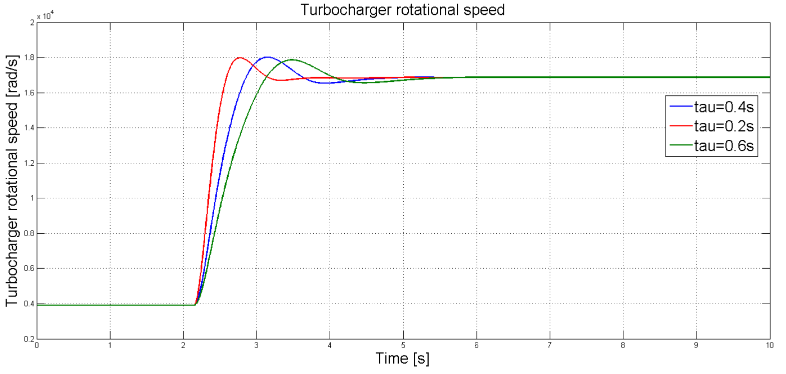

Figure 3.31. Turbocharger rotational speed with different time constants at the first-order system. A bigger delay gives a slower response but about the same in the overshoot height in all three simulations.

delay is a little bit faster than with the dynamics and delays of the turbine. But the overshoot and oscillations is still there.

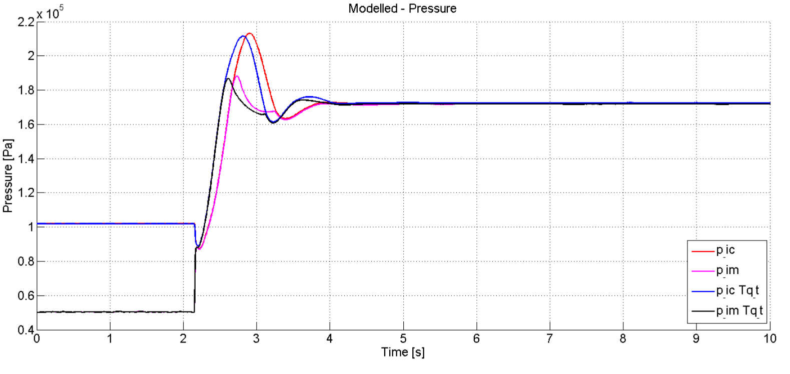

Figure 3.32. Pressure response. Red and magenta is with normal configuration. Blue and black is for the wastegate controlling turbine torque configuration. The wastegate controlling the turbine torque directly gives a faster response, but the characteristics of the pressure responses is the same as for the conventional configuration.

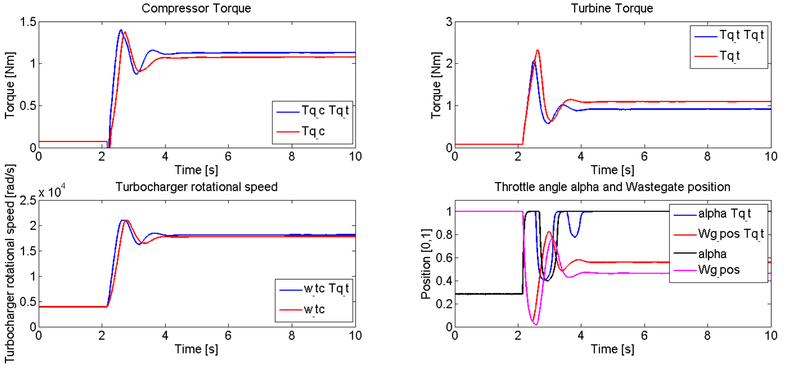

Figure 3.33. Top left: Compressor torque. Top right: Turbine torque. Bottom left: Turbocharger rotational speed. Bottom right: Throttle angle and wastegate position. The colors marked Tq_t is the simulations with wastegate controlling the turbine torque. The other is the normal configuration. When controlling the turbine torque directly, all responses are a little bit faster than the normal configuration, but the characteristics is the same for both cases.

Chapter 4

Controller compensation

This section describes and evaluates one compensation strategy for the wastegate controller to deal with the delays in the system dynamics and the throttles impact on the intercooler pressure. The structure have similarities with the smith-predictor, [5], using an inner loop to handle the delays in the system by predicting the future value. While the smith-predictor is used to compensate for pure time delays, this compensation is used to compensate for the dynamics. First the modified controller equations is described followed by a simulation study and an evaluation.

Controller equations

The strategy is to predict what the intercooler pressure will be X seconds in the future and use the predicted pressure to calculate the control error. This is done by estimating the derivative of the intercooler pressure using the ideal gas law (4.1) with the assumption of a constant temperature and mass-balance (4.2). The intercooler temperature is also consider to be equal to the intake manifold temperature. The derivative is assumed to be constant during the next X seconds. X is a control parameter which describes how far in the future the prediction is. The compensation is added in the feedback part of the wastegate controller, giving it the following equations.

$$\frac{dp}{dt} = \frac{\frac{dm}{dt} RT}{V} \tag{4.1}$$

$$\frac{dm}{dt} = \dot{m}_{ic} - \dot{m}_{thr} \tag{4.2}$$

$$\Delta p_{ic} = \frac{dp}{dt} \cdot X \tag{4.3}$$

$$wg_{pos,ff} = f(p_{ic,ref}) \tag{4.4}$$

$$p_{ic,ref} = p_{im,ref} + \Delta p_{thr,ref} \tag{4.5}$$

$$e_n = p_{ic,ref} - p_{ic} + \Delta p_{ic} \tag{4.6}$$

$$I_{wg,n} = I_{wg,n-1} + K_{p,wg}\frac{T_s}{T_{i,wg}}e_n + \frac{T_s}{T_{i,wg}}(wg_{pos,sat,n} - wg_{pos,n}) \tag{4.7}$$

$$wg_{pos,fb} = K_{p,wg}e_n + I_{wg,n} \tag{4.8}$$

$$wg_{pos,n} = wg_{pos,ff,n} + wg_{pos,fb,n} \tag{4.9}$$

$$wg_{pos,sat,n} = \begin{cases} 1 \ , \ wg_{pos,n} > 1 \\ wg_{pos,n} \ , \ 0 \leq wg_{pos,n} \leq 1 \\ 0 \ , \ wg_{pos,n} < 0 \end{cases} \tag{4.10}$$

Simulations with controller compensation

To test the modified controller a set of simulations is made with the assumption that both the intercooler mass flow, $\dot{m}_{ic}$, and the throttle mass flow $\dot{m}_{thr}$ is measured and given to the ECU. Usually non of these mass flows are measured. The first simulations is with constant engine speed of 4600 rpm with a step in acceleration pedal position 20-75%. The throttlesetpoint, $\Delta p_{thr,ref}$, is set to 0 and that gives the same target pressure for both pressures. Figure 4.1 shows the results from simulations with three different X, X=200ms, X=100ms and X=0ms. The last one with X=0ms gives the original controller. Figure 4.1 shows that the controller compensation is reducing the overshoots in both pressures as well as it removes the oscillatory behavior. The compensation makes the response slower, which is most apparent in the simulations with X=200ms. Figure 4.2 shows the control signals for the three simulations.

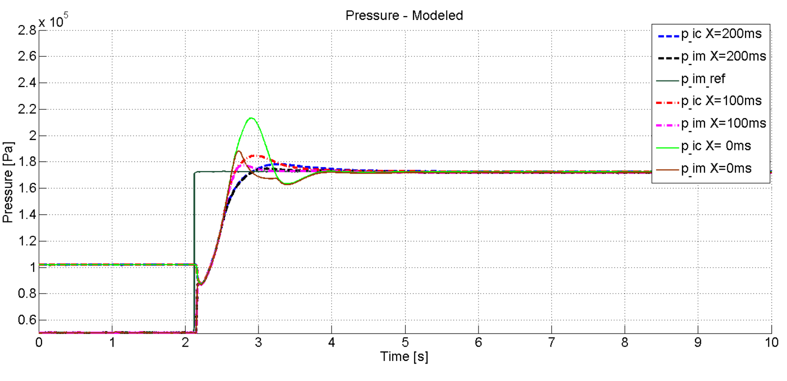

Figure 4.1. Pressure response with compensating controller. Without the compensation (p_{ic} green, p_{im} brown) the figure shows that both pressures overshoots and undershoots before it settles at the target value. For X=100ms the compensation reduces both the p_{ic} (red) and p_{im} (magenta) overshoots and removes the undershoots altogether. For X=200ms the p_{ic} (blue) and p_{im} (black) overshoots is even more reduced but the bigger X makes the response slower.

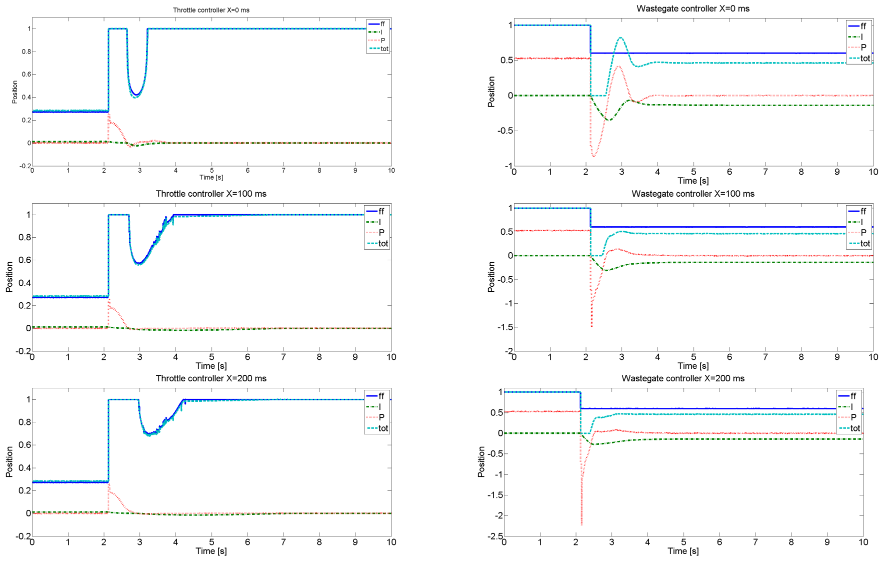

Figure 4.2. Throttle controller to the left and wastegate controller to the right. The blue is the feedforward part, red is the P-part and green is the I-part. The cyan is the total saturated control signal. The control signals, from the top row to the bottom row, X=0 ms, X=100 ms and X=200 ms.

The same experiments is performed with a throttlesetpoint, $\Delta p_{thr,ref}$, at 10%, giving the target intercooler pressure to be 10% higher than the target intake manifold pressure. The result is shown in Figure 4.3. Without the compensation, the simulation with X=0ms gives a overshoot in both pressures resulting in a oscillatory behavior. The simulations with the compensation, reduces both the overshoots and the oscillations. X=100ms gives an oscillatory behavior which is damped out quickly. X=200ms gives a small overshoot but no oscillations. Figure 4.4 shows the control signals for the three simulations.

Evaluation

The simulations performed in previous section shows that the controller with compensation handles the oscillation problem very well, totally eliminating them for a throttlesetpoint set to zero, shown in Figure 4.1 and drastically reducing them for X=100ms and eliminating them for X=200ms in the simulations with a 10% throttlesetpoint shown in Figure 4.3. The controller also handle the overshoot problem well, drastically reducing them. There might be possible to remove the p_{im} overshoot altogether at the cost of response time.

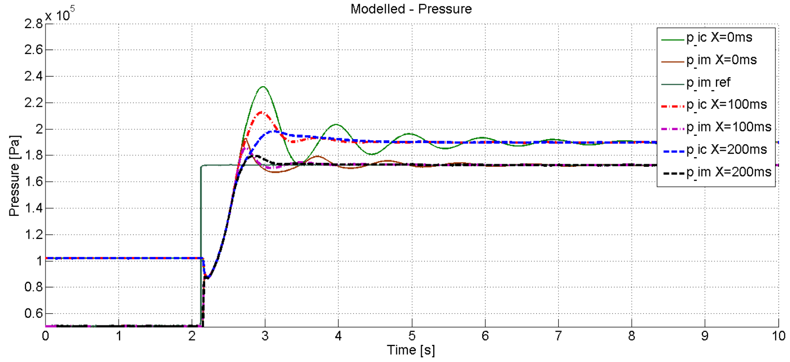

Figure 4.3. Pressure response with compensating controller. Without the compensation both pressures (p_{ic} green, p_{im} brown) overshoots and start to oscillate before it finally settles at the target value. For X=100ms the compensation reduces both the p_{ic} (red) and p_{im} (magenta) overshoots but not enough to remove the oscillations altogether, giving them to do an undershoot and another overshoot before they settle at their target values. For X=200ms the p_{ic} (blue) and p_{im} (black) overshoots are even more reduced and the oscillations are removed altogether.

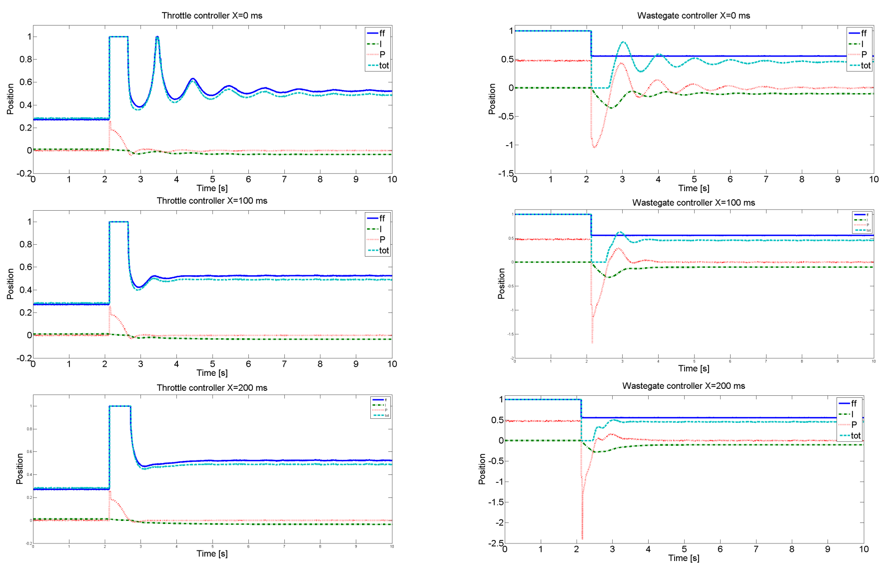

Figure 4.4. Throttle controller to the left and wastegate controller to the right. The blue is the feedforward part, red is the P-part and green is the I-part. The cyan is the total saturated control signal. The control signals, from the top row to the bottom row, X=0 ms, X=100 ms and X=200 ms.

Chapter 5

Conclusion

This section presents the conclusions that have been drawn from the simulation study.

Overshoot

The overshoot is most apparent in the intercooler pressure, but it is the intake manifold pressure that is most important to control due to the direct coupling with the engine torque. Starting with the intercooler overshoot. Even if the actuator dynamics is removed, there is still an overshoot in intercool pressure. That indicates that there are some other dynamics in the system that causes this overshoot. The actuator dynamics does affect the size of the overshoot, a bigger time constant in the first-order system approximating the wastegate actuator results in bigger overshoot. Experiments show that that parts of the overshoot is caused by the throttle cutting the flow when the intake manifold pressure reaches its target value. This is seen by an increase in intercooler pressure when the throttle is cutting flow compared to when the throttle isn't cutting flow, but there is an overshoot in both cases. That indicates that there are more dynamics affecting the pressure overshoot. There is no overshoots in the experiment with the turbocharger dynamics removed and with an addition of a first-order system to approximate the dynamics shows that a bigger time constant gives a bigger overshoot. But in the experiments with the fully open loop system, there is no overshoot even though the turbocharger dynamics is present. The overshoots are only present when the wastegate controller is used. The consensus is that the dynamics in the boost pressure system, ie the dynamics in the turbocharger and the wastegate actuator is making it difficult to control. The dynamics makes the wastegate controller react too late, causing an overshoot.

Continuing with the overshoot in intake manifold pressure. Intercooler pressure and intake manifold pressure is coupled, they are just separated by the throttle. If there is no overshoot in intercooler pressure, there won't be an overshoot in intake manifold pressure. The overshoots is mainly caused by the throttle actuator delay and dynamics. Given that without actuator dynamics the overshoot is really small, which might be caused by the uncompensated increase in intercooler pressure when the throttle cuts the flow.

Oscillations

The oscillations is induced by the overshoot, which causes the controllers to react to counteract the overshoot. The oscillatory-behavior is caused by the controllers, when the two controllers are counteracting each other. This statement is supported by the fact that there are no oscillations in open loop system, see section Figure 3.1. The oscillations is most apparent in Figure 3.5, where the the intercooler target pressure is set to be 10% higher than the target intake manifold pressure. The reason for why this setup causes most oscillations is due to when the intercooler pressure is controlled to have a certain pressure drop over the throttle, both the throttle and wastegate have the ability to affect both pressures, given that they counteract each other an oscillatory-behavior arises. A comparision with the corresponding simulation with the target pressures equal, Figure 3.2, the intercooler pressure takes the intake manifold pressure with it down below the target pressure. That causes the throttle to be unable to increase pressure and therefore it will only be the wastegate actuator that is active, and as a result the oscillations ceases when the intercooler pressure and intake manifold pressure reaches the target pressure. The reasons for why the controllers counteract each other and why the control problem is difficult is due to that both actuators affect both pressures and that there are dynamics mainly in the boost pressure system.

Controller

One way or another the controller needs to handle the dynamics and the delays in the boost pressure system. One way to solve this may be the suggested compensation in the wastegate controller described in chapter 4. The compensation reduces the overshoot and removes the oscillatory behavior. The compensation takes the dynamics in the boost system in consideration by predicting the pressure build up, but it also takes the throttles affection on the intercooler pressure in consideration by using the mass-balance equation. The controller is in need of more development and more evaluation, some of the work that needs to be done is mentioned in chapter 6.

Chapter 6

Future Work

In the future it would be interesting to implement a gearshift sequence. To get accurate simulation a set of measurements during gearshift is needed to find out how the vehicle reacts during gearshift in terms of torque reduction, engine speed etc. The measurements also need to capture the oscillatory behavior.

The controller compensation suggested needs more evaluation, more simulations to ensure that the controller can handle every operating condition. The robustness of the controller also needs to be tested and evaluated. A good model to estimate the intercooler mass flow is also needed to remove the need of an extra sensor for intercooler mass flow.

A more advanced control strategy, such as a MPC, would also be interesting to implement in a simulation environment. Starting with a complex model and then try to reduce it to meet the computational demands in the ECU.

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Appendix A

Nomenclature

Variable name and its symbols.

Variable	Symbol
Pressure	p
Temperature	T
Mass flow	$\dot{m}$ or W
Efficiency	η
Pressure ratio	Π
Angle	α
Volume	V
Diameter	D
Torque	Tq
Angular speed	ω
Rotational speed [rpm]	N

The subscripts indicates the location.

Location	Subscript
Ambient	amb
Air filter	af
Compressor	c
Intercooler	ic
Throttle	thr
Intake manifold	im
Engine	e
Exhaust manifold	em
Turbine	t
Wastegate	wg
Exhaust system	es
Turbocharger	TC
Cylinder	cyl
Exhaust gas	eg