Institutionen för systemteknik
Department of Electrical Engineering

Examensarbete

Enthalpy Based Boost Pressure Control

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

Emil Hilding

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Entalpibaserad laddtrycksstyrning
Enthalpy Based Boost Pressure Control

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Author

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Sammanfattning
Abstract

A turbo system is driven by the excess energy in the exhaust gases. As a result, variation in exhaust temperature cause variations in boost pressure. By using the information about the available exhaust energy in the turbo controller directly through a feedforward controller, an unexpected variation in turbo boost can be avoided. A model based controller is developed that calculates the desired turbine power from the boost pressure reference and then, by observing the available exhaust energy, controls the generated turbine power to match the desired power. A Mean Value Engine Model has been used to make simulation with the developed controller implemented. Steps between different boost pressure references are used to evaluate controller performance. Tests in a car have also been made to make sure the simulation results are consistent in a real environment.

Nyckelord
Keywords

Turbo, Controller, Model based, MVEM, Wastegate, Variable geometry turbine

Abstract

A turbo system is driven by the excess energy in the exhaust gases. As a result, variation in exhaust temperature cause variations in boost pressure. By using the information about the available exhaust energy in the turbo controller directly through a feedforward controller, an unexpected variation in turbo boost can be avoided. A model based controller is developed that calculates the desired turbine power from the boost pressure reference and then, by observing the available exhaust energy, controls the generated turbine power to match the desired power. A Mean Value Engine Model has been used to make simulation with the developed controller implemented. Steps between different boost pressure references are used to evaluate controller performance. Tests in a car have also been made to make sure the simulation results are consistent in a real environment.

Sammanfattning

Turbosystem drivs av överskottsenergin i motorns avgaser. Detta innebär att temperaturvariationer i avgaserna orsakar variationer i genererad turbineffekt och därmed ökat laddtryck från turbosystemet. Används informationen om den tillgängliga energin i avgaserna när man styr turbinen så kan man motverka oväntade laddtrycksförändringar. I denna rapport har en modellbaserad turboregulator med en framkoppling som beräknar en önskad turbineffekt från givet referenstryck utvecklats. Sedan tas en styrsignal fram till turbinen som, genom att använda informationen om den observerade energin i avgaserna, matchar den önskade turbineffekten. En model av en medelvärdesmotor har används för att validera prestandan i regulatorn via stegsvar mellan olika referenstryck. Det har även utförts tester i bil för att avgöra om resultatet blir detsamma under verkliga förhållanden.

Acknowledgments

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Emil Hilding

Contents

Chapter 1

Introduction

Minimizing fuel consumption is important in cars. One way to save fuel is to use smaller engines. To reach the same performance as the larger engines, the smaller engines have to be able to swallow the same amount of air as larger engines, i.e. a higher air density is needed. To achieve this a turbo can be fitted to the engine that extracts excess energy from the exhaust gases using a turbine. The turbine is connected to a compressor that increases the engine air-supply by increasing the air-intake pressure, so-called boost pressure.

Due to more strict emission legislation, future engines must get more sophisticated exhaust treatment systems. Some of these systems cause variations in the exhaust temperature, which affect the boost pressure. It would therefore be beneficial to develop a boost pressure controller, from here on called turbo controller, that uses information of the exhaust temperature to further refine the turbo behavior.

1.1 Problem description

The turbo system is driven by the excess energy in the exhaust gases. That means that the variations in exhaust temperature cause variations in the boost pressure. One exhaust treatment system that affect the exhaust temperature is the diesel engine particle filter. It needs several combustion settings, for example when it has to be cleaned the fuel injection is delayed (ineffective combustion) to get higher exhaust temperature and burn the particles. That temperature change increases the exhaust energy and if the turbo controller does not take those variations into account, it will lead to an unexpected turbo boost. This means that the turbo controller has to be calibrated for each combustion setting of the exhaust treatment system. This makes it beneficial to use information about the available exhaust energy in the turbo controller directly.

1.2 Resources

A Mean Value Engine Model (MVEM) implemented in MATLAB SIMULINK has been used for various simulations. The MVEM is provided by Vehicular System at Linköpings University and is extended with the developed turbo controller. Measurement data from a real car and the possibility to validate the result from the simulations has been provided by Saab Automobile Powertrain AB.

1.3 Related work

In this thesis the MVEM is used for simulation and controller development. The MVEM equations are well documented in [2] and [7]. A lot of work has been done on turbo control. In several papers a turbo control problem formulation is stated from efficiency models of the compressor and turbine but with different control methods. An approach to the control problem with state space representation and full state feedback with state linearization is documented in [8]. Calculations of compressor power demand using a torque control objective with inverted mapping of IMEP (Indicated Mean Effective Pressure), and trough that set a wastegate actuator signal, is documented in [9]. A turbo controller tuning method is proposed in [11] based on a wastegate actuator modeling, both static and dynamic, to capture the boost pressure response. Turbo system modeling with an energy balance approach are documented in [10].

In this thesis the demanded power from the compressor and the available energy in the exhaust gas is studied. The turbo system model equations will be stated with the energy balance approach as in [10]. Most of the previous work studies the control of the wastegate with pressure models for the compressor. As the exhaust energy runs the turbo system, a controller that observes the available energy will be developed in this thesis, to control the wastegate so that the turbine can deliver the specific power demand from the compressor.

To calculate the energy in the exhaust gases it may be important to accurately estimate the exhaust temperature. Mean value models for exhaust temperatures are documented in [4].

Chapter 2

Approach

A modelbased turbo controller has been developed to control the air intake pressure and mass flow into the engine to get a specific response in torque from the driver pressing the accelerator pedal. The idea is to decide what amount of power that should be generated by the turbine, and set the actuator signal for the turbine to a desired value. The actuators differ depending on what turbo system that are applied to the engine, but in general it controls the mass flow through the turbine and thereby the generated power. For more information about turbo systems see Section 3.1. To calculate the desired turbine power the approach is to convert the desired boost pressure through model equations of the system components between the intake manifold and the compressor then to the turbine, see Figure 2.1. When a desired value for the turbine power is given, a specific control signal for the turbine can be determined according to that value.

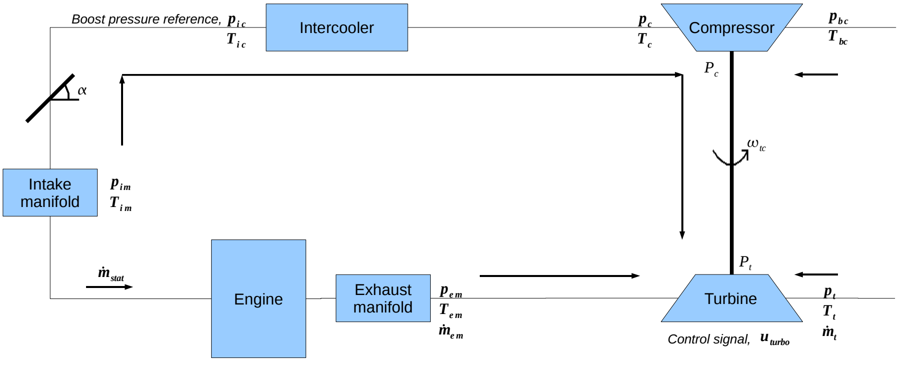


Figure 2.1. *An overview of the components in the turbo charged engine. The arrows show the calculation route in the system for how to reach the desired turbine power, and through that a control signal for the turbine system, from a demanded boost pressure reference.*

2.1 Mean value engine model

A Mean Value Engine Model, MVEM, is provided by Vehicular Systems at Linköping University. It is a complete model of a turbo charged engine with a wastegate (wg) turbine implemented, for more information about different turbo systems see Section 3.1. The MVEM is implemented in MATLAB SIMULINK which makes it possible to run simulations on different drive cycles or scenarios and evaluate step responses and at the same time observe the system behavior. An overview of the implementation of the MVEM is shown in Figure 2.2. The model equations used in the engine model are collected in [6]. The input signals required to run the MVEM are shown in Table 2.1.

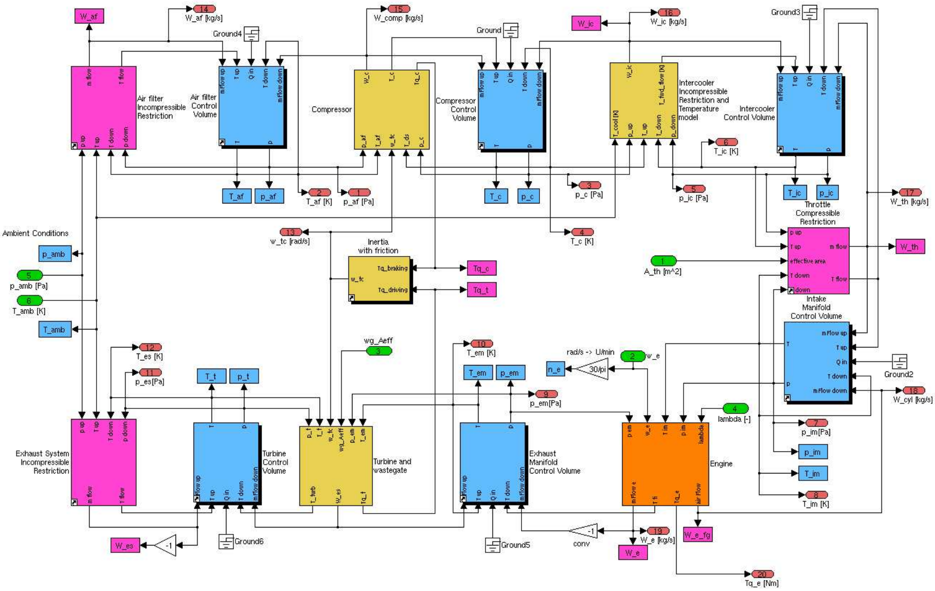


Figure 2.2. *The MVEM implemented in* MATLAB SIMULINK. *To control the turbo the wastegate is set to a specific effective area.*

Table 2.1. *Input signals needed in the MVEM for simulation.*

MVEM input signals	
Air-to-fuel equivalence ratio	λ
Ambient pressure	p_{amb}
Ambient temperature	T_{amb}
Effective area of the throttle	A_{th}
Engine speed	N_e
Wastegate control signal	$wg_{A_{eff}}$

Chapter 3

Turbo controller

A turbo charger is a compressor and a turbine linked together mechanically. The energy in the exhaust gas pass through the turbine that powers the compressor. The compressor raises the pressure on the air going into the engine, so-called boost pressure. To get the right amount of boost pressure from the turbo system, a turbo boost pressure controller is needed to control the mass flow through the turbine. Figure 3.1 shows an overview of the controller developed in this thesis. The main effort is put in the static feed forward link and the PID-controller which is tweaked manually and used to evaluate the performance of the controller.

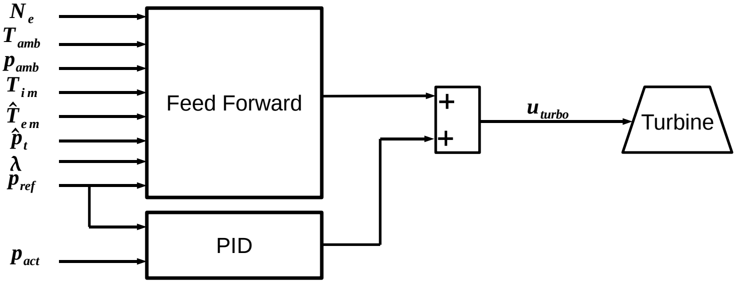


Figure 3.1. *Overview of the turbo controller that controls the mass flow through the turbine to extract a certain power to match the boost pressure reference signal p_{ref}. u_{turbo} is the control signal, N_e the engine speed, T_{amb} and p_{amb} the ambient temperature and pressure respectively, T_{im} the intake manifold temperature, $\hat{T}_{em}$ the estimated or measured exhaust manifold temperature, $\hat{p}_t$ the estimated or measured turbine pressure, λ the air-to-fuel ratio and p_{act} the measured boost pressure.*

3.1 Different sets of turbo systems

Two types of turbo systems are discussed in this thesis. The first system contains a turbine with a wastegate (wg) valve parallel to the turbine. The wg can be closed or opened in order to lead the exhaust gases through or past the turbine. The power generated by the turbine can then be controlled by the *wg*. The second system contains a variable geometry turbine (*vgt*). All the exhaust gases goes through the turbine but the power generated by the turbine can be controlled by changing the turbine geometry. There are different methods to change the geometry, but the car studied in this thesis has a turbine where the angle of the nozzles are controlled. Figure 3.2 show the difference between the two systems. The equations for the control signal are also stated and are adjusted for a gasoline engine with a *wg* and a diesel engine with *vgt*.

Figure 3.2. *Turbine systems with wastegate for a gasoline engine to the left and a variable geometry turbine applied to a diesel engine to the right.*

3.2 Model equations in the turbo controller

The turbo controller receives a reference value for the pressure before the throttle and convert it to a demanded turbine power and, depending on the type of turbine, to a pressure ratio over the turbine and finally to a *wg* or a *vgt* control signal. The control signal set to either the *wg* or the *vgt* will make the turbine generate a specific power to the compressor which gives a pressure boost corresponding to the boost pressure reference.

To convert the reference to a demanded turbine power, models for the intercooler, compressor, turbine speed and the air pressure drop from the environment to before the compressor are used. These models are needed because most of the signals needed from the system are not measured. The model equations and an overview of the system components are shown in Figure 3.3. How to convert the demanded turbine power to a pressure ratio and a control signal to the turbine is presented in Section 3.2.2 and 3.2.3.

The equations from the air intake through the compressor and to the engine are similar for gasoline and diesel engines, except that some diesel engines does not have a throttle. However they require different exhaust systems due to different combustion and therefore the equations after the engine will differ. Due to a more complex exhaust system, the diesel engine has a pressure sensor after the turbine while a model for that pressure will be required for the gasoline engine.

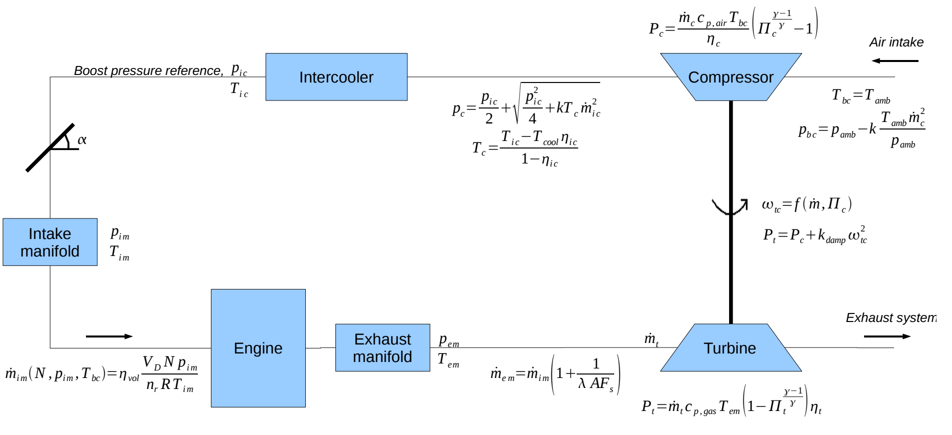


Figure 3.3. *System overview with equations used in the turbo controller to convert a given boost pressure reference to a demanded turbine power and, depending on the type of turbine, to a control signal for the wg or the vgt. These models are needed since most of the signals needed are not measured and each model are discussed further in this Section.*

3.2.1 Boost pressure reference to desired turbine power

To be able to control the turbine and extract the correct power from the exhaust gases a control signal has to be set. The control signal is calculated from a desired pressure ratio that matches the desired turbine power, see Section 3.2.3. As a first step it is sufficient to know the relation between the turbine power and the boost pressure. This section explains the set of model equations used to estimate the system parameters that are needed to achieve an expression for the desired turbine power.

The controller models that are used to calculate the demanded turbine power only depend on components in the air intake side of the engine, i.e. components before the engine. Therefore the same model equations are used for both the diesel and gasoline engine, independent of which turbo system that are used, since the turbine is placed in the exhaust side, i.e. after the engine.

Pressure and temperature before the throttle

The pressure before the throttle p_{ic} (after the intercooler) is the pressure that is given as the reference pressure, i.e. the boost pressure reference. In the MVEM with *wg* the reference pressure is given as an input for the simulations. For the car measurements with *vgt*, it is sent from the car Engine Management System.

$$p_{ic} = p_{ref}, \text{ boost pressure reference}$$

The temperature before the throttle is approximated with the temperature after the throttle in the intake manifold, which has a sensor for temperature measurement.

$$T_{ic} = T_{im}$$

Stationary air mass flow

To get the compressor power later on, the mass flow through the compressor needs to be calculated. The controller calculates the stationary feed forward response and at a stationary state the mass flow will be constant from the air intake to the engine.

$$\dot{m}_c = \dot{m}_{im} = \dot{m}_{stat}$$

The stationary mass flow is therefore calculated using a volumetric efficiency model.

$$\dot{m}_{stat} = \frac{V_D}{n_r R} \frac{N p_{im}}{T_{im}} \eta_{vol} \tag{3.1}$$

The volumetric efficiency η_{vol} in (3.1) can be modeled as a constant value, $\eta_{vol} = C_{\eta_{vol}}$, but at high pressure ratios over the engine the mass flow differs due to residual gases that affect the volumetric efficiency. Residual gases are the gases that remain in the combustion chamber after the exhaust valve has been closed and some of that expands into the intake manifold during the air intake. The residual gases have higher temperature than the intake air and it will affect the mass flow, see page 81 in Eriksson and Nielsen (2009) [7]. To take the residual gases into account a factor is added that depends on the compression ratio, $r_c = \frac{\text{maximum cylinder volume}}{\text{minimum cylinder volume}}$ and the pressure ratio over the engine, $\frac{p_{em}}{p_{im}}$.

$$\eta_{vol} = C_{\eta_{vol}} \frac{r_c - \left(\frac{p_{em}}{p_{im}}\right)^{\frac{1}{\gamma}}}{r_c - 1} \tag{3.2}$$

One drawback with using this volumetric model is that the reference pressure is the pressure before the throttle. Due to the pressure drop over the throttle it will be incorrect to just use $p_{im} = p_{ic}$. Though in this thesis, all the test and simulations are made with a fully open throttle, which minimizes the pressure drop. A throttle model between the intercooler and the intake manifold that describes the drop in pressure should be implemented, but due to insufficient time it has been suggested as future work.

Pressure and temperature after compressor

The desired pressure before the throttle is sent as a boost pressure reference input to the controller. To follow the calculation route in Figure 2.1 to finally reach a control signal, the first step is to calculate the pressure drop over the intercooler. From the boost pressure reference a desired pressure after the compressor p_c can be estimated from

$$p_c = \frac{p_{ic}}{2} + \sqrt{\frac{p_{ic}^2}{4} + k T_c \dot{m}_{ic}^2} \tag{3.3}$$

where the constant parameter k is determined using the least square method with measured data, from the engine modeled in the MVEM. But due to insufficiant sensors in the car with *vgt* the parameter is set to the same value in both controller models. The model is validated against another set of data and is shown in

Figure 3.4. The maximum relative error is 1.5% and occurs at a massflow around 0.035 kg/s.

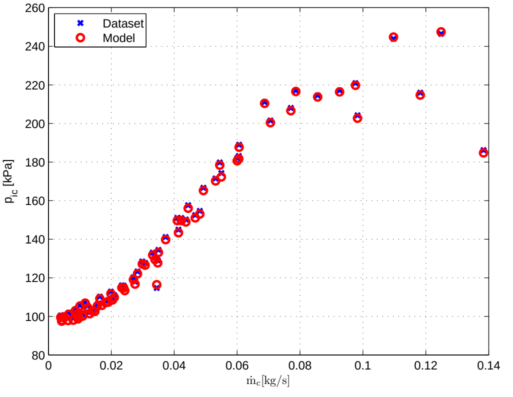


Figure 3.4. *Figure shows the model* (3.3), *with parameter k estimated from a data set, validated on another data set from the same engine. The model shows very good fit to data. The maximum relative error is 1.5% and occurs at a massflow around 0.035 [kg/s]*

The temperature after the compressor T_c can be estimated using

$$T_c = \frac{T_{ic} - T_{cool}\eta_{ic}}{1 - \eta_{ic}} \tag{3.4}$$

where $T_{cool} = T_{amb}$, the temperature of the cooling air. The intercooler efficiency, η_{ic}, is estimated in the same way as k in (3.3), and the validation is shown in Figure 3.5. The model show a very good fit at mass flows below 0.1 kg/s, but deviates up to 100 K at higher mass flows. Sensitivity analysis on the compressor temperature for simulations done on the MVEM, show that error at ±100 degrees gives an deviation in boost pressure to 1-2 kPa.

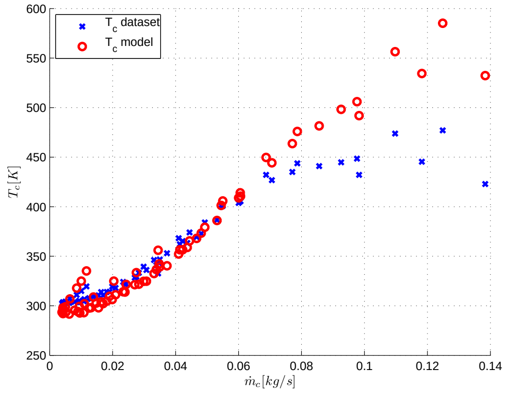


Figure 3.5. *Figure shows the model* (3.4)*, with parameter* η_{ic} *estimated from a data set, validated on another data set from the same engine. The model shows very good fit to data at mass flows below 0.08 kg/s but deviates up to 100 degrees at higher mass flows. Simulations done on the MVEM, show that error at ±100 degrees gives an deviation in boost pressure to 1-2 kPa*

Pressure and temperature before compressor

The pressure before the compressor p_{bc} can be calculated from the ambient pressure p_{amb} through the pressure drop over the air filter.

$$p_{bc} = p_{amb} - k\frac{T_{amb}\dot{m}_c^2}{p_{amb}} \tag{3.5}$$

The constant parameter k is determined using the method of least square. Figure 3.6 show a validation of the model, where the maximum relative error is 2.6% and occurs at low mass flows.

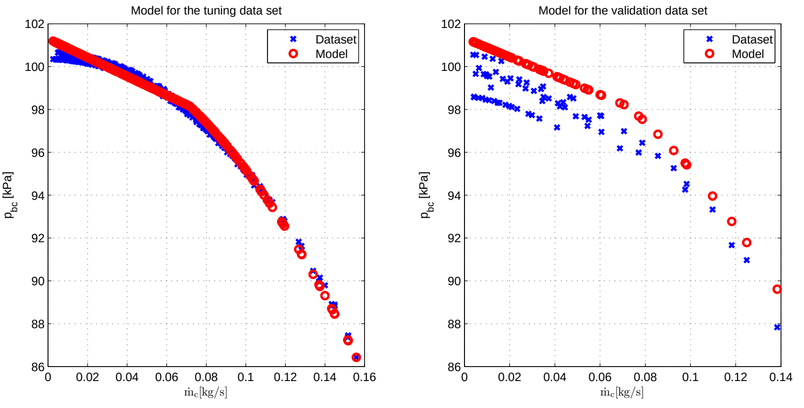


Figure 3.6. *The model* (3.5), *with parameter k estimated from a data set, validated on another data set from the same engine. The model shows very good fit for the tuning data set but some deviation in the validation. The maximum relative error is 2.6% for the validation data set*

Compressor efficiency

The efficiency of the compressor is defined as

$$\eta_c = \frac{P_{c,ideal}}{P_c} = \frac{\left(\frac{p_{ds}}{p_{us}}\right)^{\frac{\gamma-1}{\gamma}} - 1}{\frac{T_{ds}}{T_{us}} - 1} \tag{3.6}$$

where $P_{c,ideal}$ is the power needed for the ideal process and P_c is the actual consumed compressor power. The subscript *us* and *ds* means upstream (before) and downstream (after). The compressor power is the sought parameter since it is needed to calculate the desired turbine power. Therefore a model for the compressor efficiency is sufficient. To model the compressor efficiency a dimensionless flow parameter is used.

$$\Phi = \frac{\dot{m}_c T_{bc} R}{N_{tc} D_c^3 p_{bc}} \tag{3.7}$$

In [5] it is described how dimensional analysis are used to reduce cost of measurements needed for turbocharger performance determination and how it can be applied in engine modeling. By using (3.7), η_c is modeled by a fitted quadratic formula of Φ according to

$$\eta_c = \frac{\Phi(2\Phi_{max} - \Phi)}{\Phi_{max}^2} \eta_{c,max} \tag{3.8}$$

where $\eta_{c,max}$ is the maximum efficiency of the compressor and Φ_{max} is the value of Φ at $\eta_{c,max}$. Those parameters can be estimated using the least square method. This model for the compressor efficiency has been evaluated against data from a compressor map and the result is shown in Figure 3.7. Table 3.1 show the maximum relative error between model and data set for different turbo speeds. The model is more accurate at lower turbo speeds.

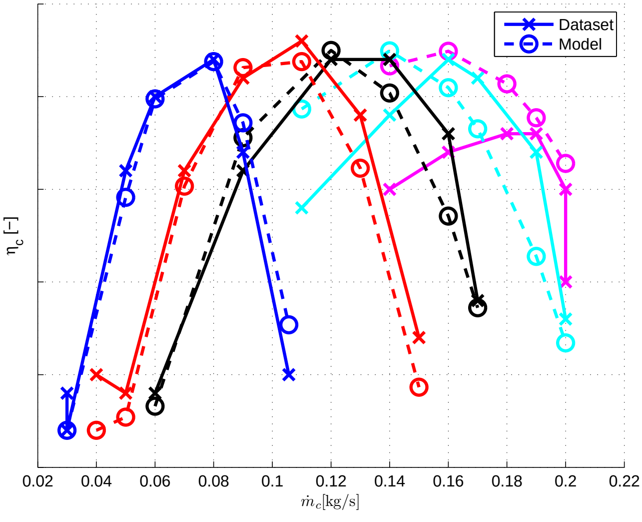


Figure 3.7. *Validation of the compressor efficiency model* (3.8)*, where the solid line is data from a compressor map and the dashed line is the values calculated with the efficiency model. The data is collected at different turbo shaft speeds, in this case at 80000, 110000, 130000, 150000 and 170000 rpm. The model shows very good fit for every operating point under 150000 rpm.*

Table 3.1. *Compressor efficiency model validation, showing the maximum relative errors at different turbo speeds*

Turbo shaft speed [rpm]	Relative error [%]
80000	4.4
110000	5.0
130000	6.1
150000	7.8
170000	9.8

Compressor power

The static power consumed by the compressor is

$$P_c = \frac{\dot{m}_{stat} c_{p,air} T_{bc}}{\eta_c} \left(\Pi_c^{\frac{\gamma-1}{\gamma}} - 1 \right) \tag{3.9}$$

where $\Pi_c = \frac{p_c}{p_{bc}}$ is the pressure ratio for the compressor, $c_{p,air}$ is the specific heat at constant pressure. A constant $c_{p,air}$ is used and the other parameters have been previously determined.

Turbine power

Since the compressor is mechanically linked to the turbine the turbine speed is equal to the compressor speed. The power generated from the turbine will ideally be the same as the power transferred to the compressor, but including friction losses the turbine power is

$$P_{t,des} = P_c + k_{frik}\omega_{tc}^2 \tag{3.10}$$

which is the desired turbine power calculated from the boost pressure reference. The dynamic behavior has been neglected since a stationary value is sought in the feed forward link. Now the desired power from the turbine has been calculated.

3.2.2 Turbine power to turbine pressure ratio

The desired turbine power is now known from the demanded boost pressure reference. The actual turbine power is calculated through

$$P_t = \dot{m}_t c_{p,exh} T_{em} \left(1 - \Pi_t^{\frac{\gamma-1}{\gamma}}\right) \eta_t \tag{3.11}$$

where $\Pi_t = \frac{p_t}{p_{em}}$ is the pressure ratio for the turbine and the parameter $c_{p,exh}$ is modeled as a constant. T_{em} is modeled with the engine mass flow for the *wg* and measured from the car for the *vgt*. The turbine efficiency η_t is modeled with the blade speed ratio, BSR, discussed on the next page. That leaves two unknowns, $\dot{m}_t$ and Π_t. Starting with using models for the mass flow that depends on the pressure ratio, a pressure ratio can be calculated to match the desired turbine power so that $P_t = P_{t,des}$.

Stationary exhaust mass flow

The stationary exhaust mass flow is modeled as the sum of the air mass flow and the fuel mass flow.

$$\dot{m}_{em} = \dot{m}_{stat} + \dot{m}_f = \dot{m}_{stat}\left(1 + \frac{1}{\lambda AF_S}\right) \tag{3.12}$$

Turbine mass flow

The calculation of the pressure ratio differs depending on whether a wastegate or a *vgt* is used. In the turbine with a wastegate the mass flow through the turbine is $\dot{m}_t = \dot{m}_{em} - \dot{m}_{wg}$, where $\dot{m}_{wg}$ is the mass flow controlled through the actuator by setting the effective area in the *wg* valve, see (3.25). For the *vgt* the turbine mass flow is known from the stationary mass flow, because all flow goes through the turbine, $\dot{m}_t = \dot{m}_{em}$. For the *wg* a mass flow model is used, according to (3.13) and (3.14a). The corrected mass flow stated in (3.14a) is used to represent turbine data in maps. More information about corrected massflow is given at page 140 in [7].

$$\dot{m}_t = \frac{p_{em}}{\sqrt{T_{em}}}\dot{m}_{t,corr} \tag{3.13}$$

$$\dot{m}_{t,corr} = c_0\sqrt{1-\Pi_t^{c_1}} \tag{3.14a}$$

There is a similar model for the corrected massflow for the *vgt* as well but then the constant c_0 and c_1 is depending on the *vgt* blade angle.

$$\dot{m}_{t,corr} = c_0(\theta_{vgt})\sqrt{1-\Pi_t^{c_1(\theta_{vgt})}} \tag{3.14b}$$

The equations for the corrected mass flow will be used when calculating the control signal for the *wg* and the *vgt* position, but that will be discussed further in Section 3.2.3, where an estimation of the parameters in (3.14) is estimated. The mass flow models are also validated against data and are shown in Figure 3.9 and 3.10

Temperature before the turbine

The temperature before the turbine is modeled from the engine out temperature and the temperature drop in the exhaust manifold, as proposed in [6].

$$T_{em} = T_e - T_{drop,em}$$

The temperature drop is given by the heat transfer from the engine exhaust gas to the exhaust manifold and therefore depends on the construction. The engine out temperature can be modeled as in [1].

$$T_e = T_0 + c\cdot\dot{m}_e \tag{3.15}$$

when the engine operates close to when the air-to-fuel ratio $\lambda = 1$, why it is only suited for gasoline engines, since diesel engines operates at much higher λ. There is no simple model for the temperature in a diesel engine. Therefore the temperature before the turbine in the *vgt* turbo controller uses the measured value from the car sensor when doing the performance tests. The temperature drop is modeled as

$$T_{em} = T_{amb} + (T_e - T_{amb})\cdot e^{-\frac{h_{tot}A_i}{\dot{m}_e c_{p_{exh}}}} \tag{3.16a}$$

where A_i is the inner area of the exhaust pipe, T_{amb} is the coolant temperature and h_{tot} is the total heat transfer coefficient, which depends on the external heat transfer coefficient h_e and the internal heat transfer coefficient $h_{cv,i}$ as

$$\frac{1}{h_{tot}} = \frac{1}{h_{cv,i}} + \frac{1}{h_{ext}} \tag{3.16b}$$

No validation has been done for this temperature model due to lack of temperature sensor in the exhaust manifold, in the available data sets. For the implementation, the coefficient $h_{cv,i}$ has been calculated through

$$Nu = c_0 Re^{c_1} Pr^{c_2} \left(\frac{\mu}{\mu_s}\right)^{c3} \tag{3.17a}$$

$$Nu = h_{cv,i} Nu_{const} \tag{3.17b}$$

used in [3]. Re is the Reynolds number, Pr the Prandtl number, μ the viscosity and $Nu_{const} = \frac{l_{pipe}}{\lambda}$. The constants c_i are correlation constants, and are set by Meisner-Sorenson as $c_i = (0.27,\ 0.603,\ 0,\ 0)$ [3]. This leads to

$$h_{cv,i} = \frac{0.27 Re^{0.603}}{Nu_{const}} \tag{3.18}$$

and h_{ext} has been estimated with the method of least square from a linear model depending on the mass flow [1].Read more about the gas temperature at the exhaust valve in [7] and [1].

Turbine efficiency

The turbine efficiency is modeled as a function of the blade speed ratio (BSR). BSR is the rotor tip velocity divided with the velocity with isentropic gas expansion through the turbine

$$BSR = \frac{r_t \omega_{tc}}{\sqrt{2 c_{p,exh} T_{em} \left(1 - \Pi_t^{\frac{\gamma-1}{\gamma}}\right)}} \tag{3.19}$$

see [12] for more details about the BSR. The turbine efficiency peaks at a $BSR_{\eta_{t,max}}$ for BSR around 0.7 why it is modeled as a parabolic function of the BSR

$$\eta_t = \eta_{t,max} \left(1 - \left(\frac{BSR - BSR_{\eta_{t,max}}}{BSR_{\eta_{t,max}}}\right)^2\right) \tag{3.20}$$

with $BSR_{\eta_{t,max}} = 0.7$. Validation of turbine efficiency model (3.20) at different vgt positions, from 10% closed to fully closed, are shown in Figure 3.8. Fully closed vgt does not mean that the mass flow is cut of, but that the turbine is set so that it can generate as much power as possible. The model shows good fit to data and the maximum relative error, MRE, and average relative error, ARE, are shown in the plot titles.

The purpose is to calculate the pressure ratio over the turbine and for that we need the turbine efficiency. Since the *BSR* depends on that pressure ratio, according to (3.19), and the turbine efficiency depends on the *BSR* problem arises. According to page 165 in [12], the turbine efficiency plotted against *BSR* is insensitive to the pressure ratio. Therefore the problem is solved by a fix point iteration. Starting with a initial value for the pressure ratio Π_t^0 for the turbine efficiency, gives a Π_t^1 in (3.22) and is used in the next step for η_t

$$\begin{gathered}
\eta(\Pi_t^0) \to \Pi_t^1 \\
\eta(\Pi_t^1) \to \Pi_t^2 \\
\vdots \\
\eta(\Pi_t^n) \to \Pi_t^{n+1}
\end{gathered}$$

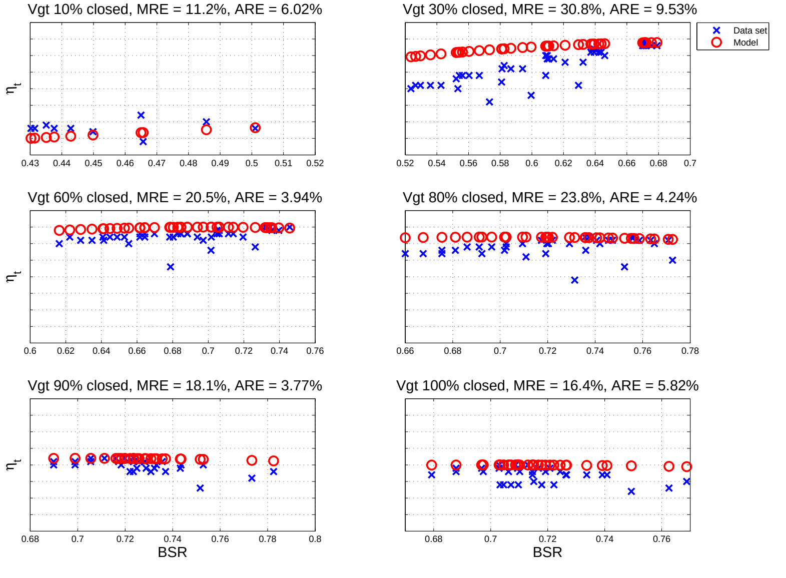


Figure 3.8. *Validation of turbine efficiency model* (3.20) *at different vgt positions, from 10% closed to fully closed. Fully closed vgt does not mean that the mass flow is cut of, but that the turbine is set so that it can generate as much power as possible. The model shows good fit at every preference but with deviation in some points. The figure show the maximum relative error, MRE, in the title of each plot, and also the average relative error, ARE. The ARE are probably a more interesting quantitative measure since it accounts for all measurements. In the third plot with vgt 60% closed for example, the ARE is a much more reliable value since the MRE will apply to the one point that deviates.*

Pressure after the turbine

The pressure after the turbine differs between a gasoline engine and a diesel engine due to the different exhaust treatment systems. The diesel exhaust treatment system is more complex to model and therefore a pressure sensor is used in the diesel engine. The model applied for the gasoline engine is dependent of the mass flow out from the engine $\dot{m}_{em}$, the temperature after the turbine T_t and the backpressure from the environment p_{amb}.

$$p_t = \frac{p_{amb}}{2} + \sqrt{\frac{p_{amb}^2}{4} + kT_t\dot{m}_{em}^2} \tag{3.21a}$$

where k is set by a least square estimation from measured data from the vehicle and

$$T_t = T_{em}\left(1 - \eta_t\left(1 - \Pi_t^{\frac{\gamma-1}{\gamma}}\right)\right) \tag{3.21b}$$

Turbine pressure ratio

From the turbine power in (3.11), the mass flow equations (3.13) and (3.14) and by using that $p_{em} = \frac{p_t}{\Pi_t}$ the equation for the pressure ratio can be written as

$$\frac{P_t}{c_{p,exh}\sqrt{T_{em}}p_t\eta_t} = \frac{1}{\Pi_t}c_0\sqrt{1 - \Pi_t^{c_1}}\left(1 - \Pi_t^{\frac{\gamma-1}{\gamma}}\right) \tag{3.22a}$$

for the *wg*, and

$$\frac{P_t}{c_{p,exh}T_{em}\eta_t\dot{m}_t} = 1 - \Pi_t^{\frac{\gamma-1}{\gamma}} \tag{3.22b}$$

for the *vgt*, which becomes simplier since the mass flow is known for the *vgt*. The right hand side of both equations are strictly decreasing functions of Π_t and thus have a unique solution for each value of the right hand side, as long as $\Pi_t > 0$. The pressure ratio over the turbine is further used for calculating the control signal that will actuate the turbine *wg* or the *vgt* blade positions.

3.2.3 Pressure ratio to turbine control signal

The desired pressure ratio is calculated to match the desired power from the turbine and by evaluating the available energy in the exhaust gas. Now the mass flow through the turbine can be evaluated by using (3.13) and (3.14), and finally a control signal to get the desired boost pressure is obtained.

Wastegate valve turbine control signal

The mass flow in the exhaust manifold is given by (3.12) and the *wg* mass flow is given by

$$\dot{m}_{wg} = \dot{m}_{em} - \dot{m}_t \tag{3.23}$$

The mass flow through the wastegate valve depends on how open the wastegate is. The model used in this thesis is a compressible flow restriction defined as

$$\dot{m}_{wg} = \frac{p_{us}}{\sqrt{RT_{us}}} A_{wg}(u_{wg}) C_{D,wg}(u_{wg}) \Psi(\frac{p_{ds}}{p_{us}}) \tag{3.24}$$

where the mass flow depends on how much the *wg* is open. It is the actuator signal u_{wg} that can be controlled and by modeling both the area A_{wg} and the discharge coefficient $C_{D,wg}$ together it can be stated as

$$A_{eff}(u_{wg}) = \frac{\dot{m}_{wg}\sqrt{RT_{em}}}{p_{em}\Psi} = \frac{(\dot{m}_{em} - \dot{m}_t)\sqrt{RT_{em}}}{p_{em}\Psi} \tag{3.25a}$$

where $A_{eff}(u_{wg}) = C_D(u_{wg})A_{wg}(u_{wg})$ is a function of the actuator signal u_{wg} [2]. The pressure in the exhaust manifold, $p_{em} = \frac{p_t}{\Pi_t}$ and $\Psi = \Psi(\Pi_t)$ is a function of the pressure ratio as

$$\Psi(\Pi_t) = \begin{cases} \sqrt{\frac{2\gamma}{\gamma-1}\left(\Pi_t^{\frac{2}{\gamma}} - \Pi_t^{\frac{\gamma+1}{\gamma}}\right)} & \text{for } \Pi_t > \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}} \\ \sqrt{\frac{2\gamma}{\gamma-1}\left(\left(\frac{2}{\gamma+1}\right)^{\frac{2}{\gamma-1}} - \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{\gamma-1}}\right)} & \text{otherwise} \end{cases} \tag{3.25b}$$

Ψ determines the fluid velocity which saturates by the sonic velocity which occurs at a critical pressure ratio $\Pi_{t,crit} = \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}}$ [7]. Since the effective area depends on the *wg* actuator signal, an actuator model for the *wg* is needed to convert the effective area to the actuator signal u_{vgt}, which is a pulse-width modulation signal (*pwm*) that controls the actuator. In this thesis however the simulations with the wastegate model is done on the MVEM, see Section 2.1, which takes A_{eff} as input directly. Therefore no effort has been put into development of a *wg* actuator model, but more can be read about it in [11].

Variable geometry turbine control signal

With a *vgt* the mass flow through the turbine is controlled by changing the geometry of the turbine. The car in this thesis, that the measurements has been made on, is controlled by changing the angle θ_{vgt} of the nozzles. The turbine mass flow is known from the stationary mass flow after the engine and through (3.13) the corrected mass flow can be calculated. With that given mass flow and the pressure ratio from (3.22b) a *vgt* blade angle is to be set corresponding to those values. To do so (3.14b) has been used and the parameters $c_0(\theta_{vgt})$ and $c_1(\theta_{vgt})$ has been estimated from the flow characteristics for the *vgt* settings used in the controller performance validation in Chapter 4.

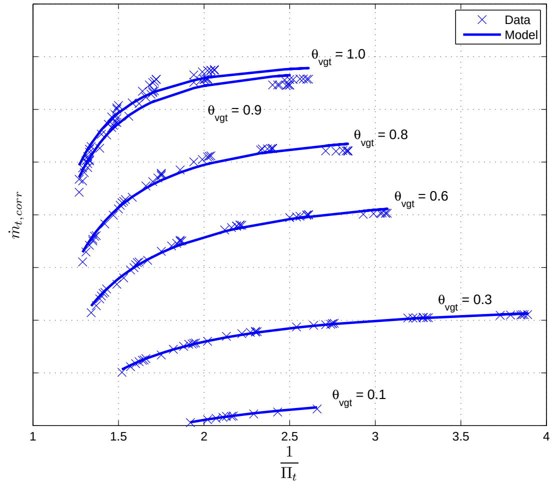


Figure 3.9. *Flow characteristics for the vgt turbine. The mass flow model,* (3.14b), *estimated for six different nozzle angles, θ_{vgt}, and validated against turbine data. The x marks the measured data and the solid lines the estimated model for the corrected mass flow that neglect the turbine speed dependency. The model shows a good fit for each nozzle angle.*

First $c_0(\theta_{vgt})$ and $c_1(\theta_{vgt})$ were estimated from data using the least square method, with fixed θ_{vgt}. In reality there is a turbine speed dependency when deciding a specific mass flow from a given pressure ratio. But the mass flow model, (3.14b), that is neglecting the dependency provides a good fit to the data for every fix nozzle angle. This is shown in Figure 3.9 where The mass flow model is marked with solid lines and the data is marked with x.

The problem with estimating the parameters for these six fix angles θ_{vgt} is that six different values for $c_0(\theta_{vgt})$ and $c_1(\theta_{vgt})$ is achieved. That means that six models for the mass flow is at hand, and each suitable for only one value of the nozzle angle. To assemble these models into on model for an arbitrary θ_{vgt} it has been assumed that the relation between the parameters $c_0(\theta_{vgt})$ and $c_1(\theta_{vgt})$ and the *vgt* nozzle angle is quadratic

$$c_{0,par}(\theta_{vgt}) = r_1\theta_{vgt}^2 + r_2\theta_{vgt} + r_3 \tag{3.26}$$

$$c_{1,par}(\theta_{vgt}) = s_1\theta_{vgt}^2 + s_2\theta_{vgt} + s_3 \tag{3.27}$$

The values of $c_{0,par}(\theta_{vgt})$ and $c_{1,par}(\theta_{vgt})$ for the six models provides the parameters r_i and s_i, using the method of least square. By using the parametrized parameters from (3.26) and (3.27) in the mass flow model, makes it suitable for all angles at the cost of some deviation from the data. The result is shown in Figure 3.10, it has almost as good fit as the models with fixes angles.

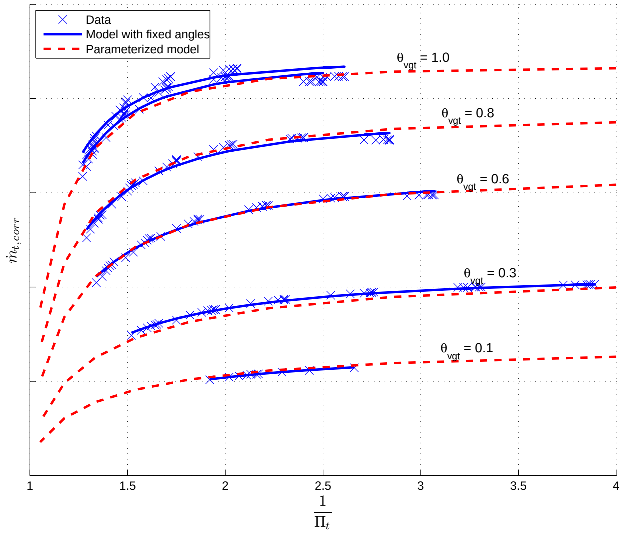


Figure 3.10. *This figure is a complement to Figure 3.9. The mass flow model* (3.28)*, with the parametrized parameters, marked with a dashed line, has been plotted for some angles. Then it has been compared to the models for a fix angle, marked with solid lines. The new mass flow model shows almost as good fit to the data, marked with x.*

The final step to calculate a *vgt* control signal is to solve the new corrected mass flow equation with *vgt* parametrized parameters, with θ_{vgt} as the only unknown variable.

$$\dot{m}_{t,corr} = c_{0,par}(\theta_{vgt})\sqrt{1 - \Pi_t^{c_{1,par}(\theta_{vgt})}} \tag{3.28}$$

Since this equation can not be solved analytically for θ_{vgt}, first the quadratic equation $c_{0,par}$ will be solved for a constant $c_{1,par}$ and a fix point iteration will be applied to $c_{1,par}$. Solving (3.28) for $c_{0,par}(\theta_{vgt})$ gives

$$c_{0,par}(\theta_{vgt}) = \frac{\dot{m}_{t,corr}}{\sqrt{1 - \Pi_t^{c_{1,par}(\theta_{vgt})}}} \tag{3.29}$$

and that combine with (3.26) gives a solution for θ_{vgt} as

$$\theta_{vgt} = -\frac{r_2}{2r_1} \overset{(+)}{-} \sqrt{\left(\frac{r_2}{2r_1}\right)^2 - \frac{1}{r_1}\left(r_3 - \frac{\dot{m}_{t,corr}}{\sqrt{1 - \Pi_t^{c_{1,par}(\theta_{vgt})}}}\right)} \tag{3.30}$$

Which solution to use has been chosen by simulation and comparison with turbine data. The estimation of parameter r_1 is negative which explains the minus choice.

Since $c_{1,par}(\theta_{vgt})$ in (3.30) depends on θ_{vgt} it is solved by fix point iteration for that variable.

$$
\begin{aligned}
\theta_{vgt}^{(1)} &= \theta_{vgt}(c_{1,par}(\theta_{vgt}^{(0)})) \\
\theta_{vgt}^{(2)} &= \theta_{vgt}(c_{1,par}(\theta_{vgt}^{(1)})) \\
&\vdots \\
\theta_{vgt}^{(n)} &= \theta_{vgt}(c_{1,par}(\theta_{vgt}^{(n-1)}))
\end{aligned}
$$

with $\theta_{vgt}^{(0)} = 0.5$ as an initially value. In simulations, the calculations of θ_{vgt} has been insensitive to the initial value. It has been set to 50 % because it felt natural, but it only takes 2-3 iterations to settle to the correct value.

With these sets of equations and a given pressure ratio and mass flow from previous calculations, a control signal to actuate the turbine for a demanded response can be set.

3.3 Controller implementation

The model based controller developed from the system equations is implemented in Matlab Simulink. Model blocks containing the equations that calculates, a turbine actuator signal that can be followed in Section 3.2. Controller parameters that are needed for the simulation are taken or estimated from the car data or MVEM data and saved in workspace. The input signals to the controller are those the controller will need while running and are connected from the engine control system (ECU) or the MVEM observers.

3.3.1 Wastegate controller in the MVEM

The controller is fitted and connected to the MVEM with a wastegate turbine. The Matlab Simulink environment where to the MVEM is connected with the *wg* controller is shown in Figure 3.11. The big model block in the middle contains the MVEM. The turbine in the MVEM are controlled by a *wg* effective area, which is sent from the controller. The inputs to simulate the MVEM is given in Table 3.2. The outputs used by the controller is the measured boost pressure, used in the feed back PID-controller, and the temperature in the intake manifold. The other outputs gives the possibility to observe variables after simulation.

The block to the right contains the developed controller. The controller need the input signals listed in Table 3.3. The boost pressure reference is taken from Matlab workspace, and is constructed manually by the user. It could, for example, be a driving scenario or a step. The output is the control signal, *wg* effective area, $wg_{A_{eff}}$. The control signal is sent to MVEM.

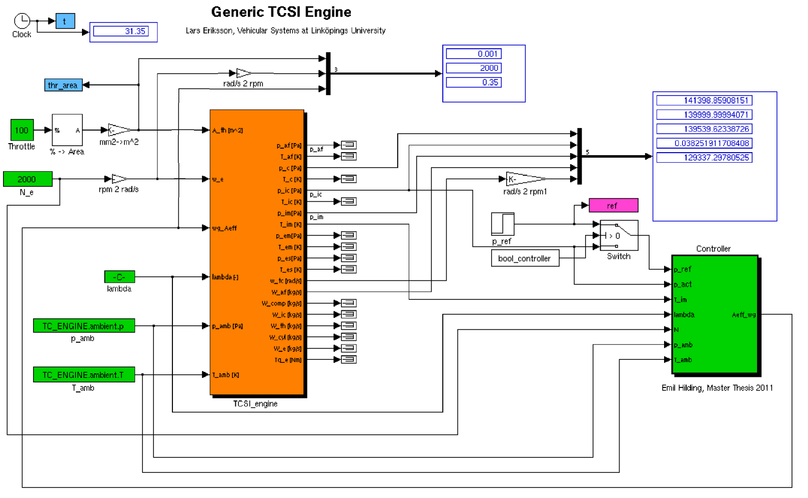


Figure 3.11. *Overview of the* MATLAB SIMULINK *environment where the MVEM is connected to the wg controller. The controller takes the inputs and send a control signal for the wg effective area in the engine model. The switch can be changed to run the controller with a step response in pressure reference or to run parallel to the engine model to verify the controller states.*

Table 3.2. *Input signals needed in the MVEM for simulation.*

MVEM input signals	
Air-to-fuel equivalence ratio	λ
Ambient pressure	p_{amb}
Ambient temperature	T_{amb}
Effective area of the throttle	A_{th}
Engine speed	N_e
Wg control signal	$wg_{A_{eff}}$

The controller model block calculates the wg control signal, according to the reference signal and the equations in Section 3.2. An overview of the model based controller implementation is showed in Figure 3.12. The model blocks contains equations for each sub model. Except for the PID-controller down to the right in the figure, all blocks are associated with the feed forward link, that calculates the stationary control signal for a given boost pressure reference. The PID-controller then contributes to the control signal to compensate for the stationary error between the boost pressure reference and the actual pressure.

Table 3.3. *Input signals needed to the wg controller for simulation.*

Controller input signals	
Air-to-fuel equivalence ratio	λ
Ambient pressure	p_{amb}
Ambient temperature	T_{amb}
Boost pressure reference	p_{ref}
Intake manifold temperature	T_{im}
Measured boost pressure	p_{act}

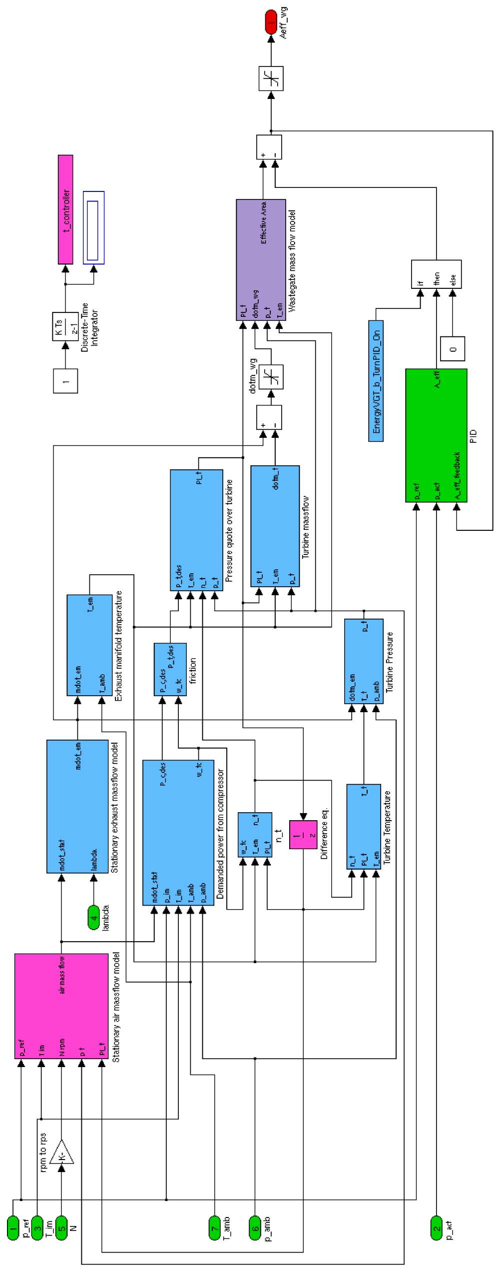


Figure 3.12. *An overview of the controller implemented in* MATLAB SIMULINK. *The reference signal is the pressure before the throttle p_{ic}. The measured input signals are intake manifold pressure p_{im}, engine speed N, intake manifold temperature T_{im}, air-to-fuel ratio λ, ambient temperature and pressure T_{amb} and p_{amb}. The signal flow can be followed from boost pressure reference to the output control signal. The block down to the right is the PID-controller that corrects the control signal.*

3.3.2 Variable geometry turbine in a car

The controller was fitted and connected to the car with a variable geometry turbine sending a specific angle to set the turbine nozzles in right position to match the reference value for the intake pressure given by the car. The controller is connected to the car control system by using an Intecrio control box. Through the control box the model can be downloaded to the car and the control signal overrides the original control signal given by the car's ECU. Figure 3.13 shows how the control box is implemented, where the big box contains the developed controller. The signals entering is the input signals needed by the controller and some additional signals for hardware protection, for example turbo speed limitation. The input signal needed for running the *vgt* controller in the car is shown in Table 3.4.

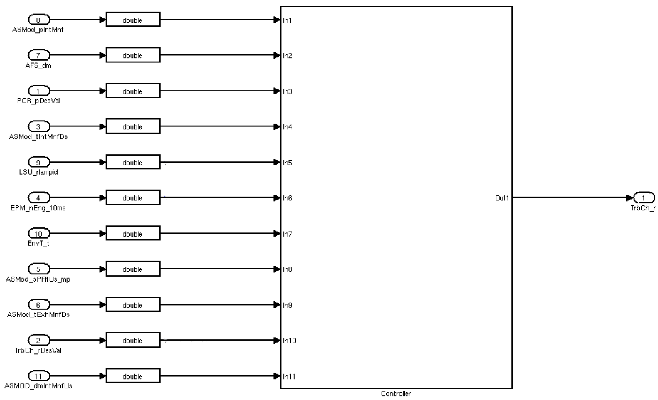


Figure 3.13. *The signal representation that takes input from the car signals and send a vgt blade angle in percentage open from 0-100 %. Most of the signals represent the input signals to the controller and some to hardware protection models implemented, for example to ensure that the turbine speed does not exceed its maximum.*

The MATLAB SIMULINK implementation for the controller with *vgt* is very similar to the wastegate controller, see Figure 3.14. The mass flow model separating the flow between the turbine and wastegate is not needed since all mass flow in the *vgt* passes the turbine and the final mass flow models used in the final step before returning the control signal differ according to Section 3.2.3. Since the model for the pressure after the turbine is more complex for the *vgt* the measured value from the car sensor is used directly.

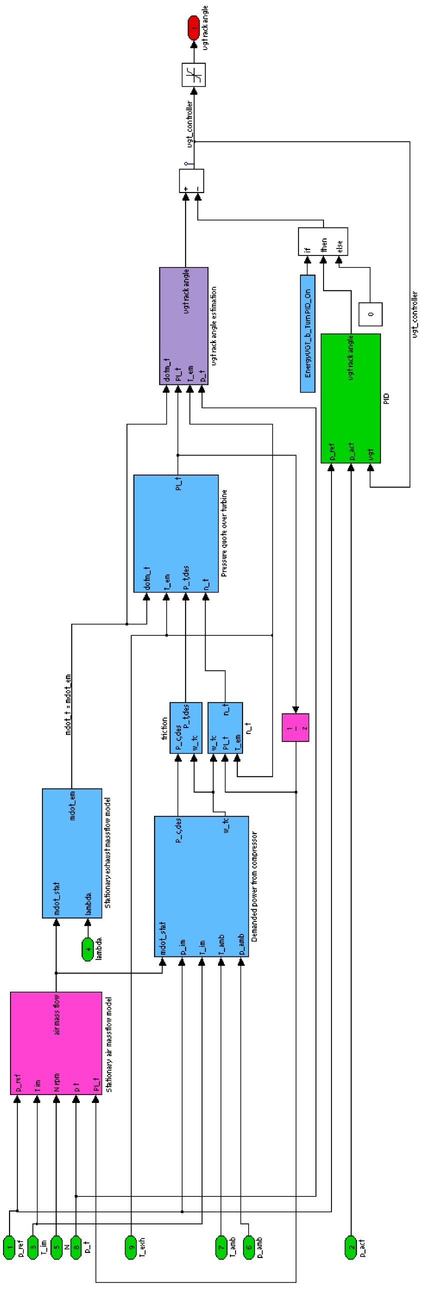


Figure 3.14. *An overview of the controller implemented in* MATLAB SIMULINK. *The reference signal is the pressure before the throttle p_{ic}. The measured input signals are intake manifold pressure p_{im}, engine speed N, intake manifold temperature T_{im}, air-to-fuel ratio λ, ambient temperature and pressure T_{amb} and p_{amb}*

Table 3.4. *Input signals needed to the vgt controller for running in car.*

Controller input signals	
Air-to-fuel equivalence ratio	λ
Ambient pressure	p_{amb}
Ambient temperature	T_{amb}
Boost pressure reference	p_{ref}
Intake manifold temperature	T_{im}
Measured boost pressure	p_{act}
Pressure after the turbine	p_t

3.3.3 PID-controller

The turbo controller developed in this thesis is for feed forward control, but to see how the system reacts within a closed system, a feed back link with a PID-controller is implemented. The PID-controller corrects the *wg* effective area or the *vgt* blade angle to minimize the error between the boost pressure reference and the actual pressure. Not much effort has been put into the PID-controller optimization and therefore the PID-parameters are just tweaked manually in the simulation environment of MATLAB SIMULINK to see that the controller does what it should and the stationary error is taken care off.

Chapter 4

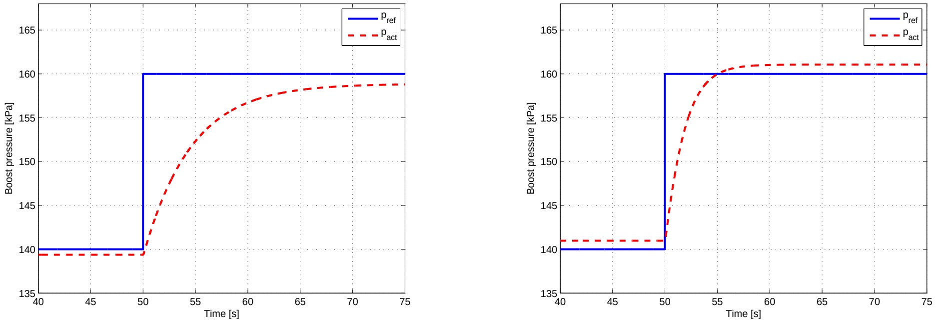
Results

The performance of the developed turbo controller will be evaluated in this chapter. First the controller with the wastegate turbine model is connected to the MVEM and simulated test results will be analyzed in Section 4.1. In Section 4.2 the controller with variable geometry turbine model will be implemented and evaluated in a car.

4.1 Turbo controller simulations with the MVEM

The MVEM, described in Section 2.1, have been used to evaluate the turbo controller with the *wg* turbine. Through various simulations the controller sub-models could be monitored and a verification of the behavior on each sub-model could be made. Step responses in boost pressure reference has been simulated with the turbo controller implemented in the MVEM. The presented step responses in Figure 4.1 have been made from 140 kPa to 160 kPa at the engine speeds 2000 rpm and 4000 rpm. The feed forward controller does not raise the boost pressure very fast since it calculates the final value for the wastegate or variable geometry turbine at the specified pressure reference and it takes time to build up the pressure. To lower the rise time for the boost pressure a PID-controller is implemented that gives an addition to the control signal when the boost pressure differ from the reference, which means that it closes the *wg* to build pressure until it reaches the desired value from the feed forward controller.

Figure 4.2 shows the difference in the control signal during a step response, and how the rise time and stationary error differs when the PID-controller is implemented in Figure 4.3. The interesting thing to look at in the feed forward is the stationary error, which is only 1-2 kPa, and indicates that the feed forward controller comes close to the stationary points. Now the PID-controller only have to correct the small stationary error. The parameters in the implemented PID-controller has only been tweaked manually to show that it is possible to correct the stationary error and decrease the responce time. Preferable is to use some PID-tuning method, for both turbo types. One way to do PID-tuning to minimize the response time without oscillations is proposed in [11].



Simulation at engine speed 2000 rpm *Simulation at engine speed 4000 rpm*

Figure 4.1. *Step response in boost pressure reference from 140 kPa to 160 kPa at engine speeds 2000 rpm and 4000 rpm. The solid line shows the step reference and the dashed line shows the measured boost pressure from the MVEM with the feed forward controller. The stationary error is between 1-2 kPa which is corrected with a PID-controller, see Figure 4.3*

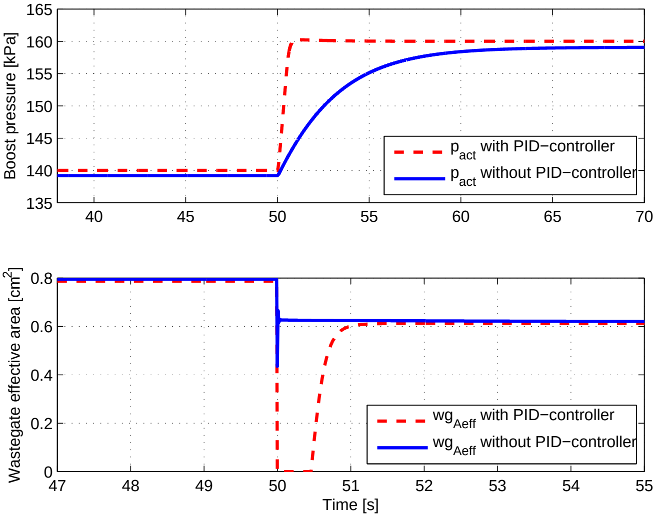
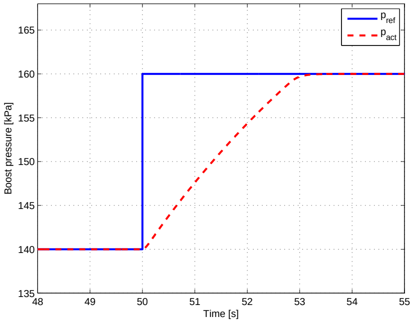
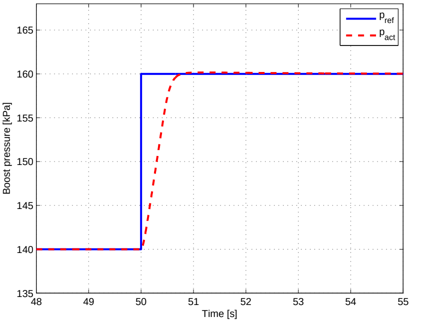


Figure 4.2. *The control signal for the MVEM wg turbine with and without the PID-controller implemented. The PID-controller makes the wg close to build up pressure during the step response at 50 seconds, and when the pressure is built up it sets at the same value as the controller without PID-controller, except for the compensation for the stationary error. The solid line representing the control signal without PID-controller is fat at time 50 s because the feedforward controller takes two iterations to settle.*



Simulation at engine speed 2000 rpm with PID-controller



Simulation at engine speed 4000 rpm with PID-controller

Figure 4.3. *Step response in boost pressure reference from 140 kPa to 160 kPa at engine speeds 2000 rpm and 4000 rpm. The solid line shows the step reference and the dashed line shows the measured boost pressure from the MVEM with the feed forward controller including a PID-controller. The stationary errors that was visible in Figure 4.1 has been corrected and the rise time improved.*

4.2 Turbo controller performance test on car

A car has been provided by Saab Automobile Powertrain AB for measuring and test of the turbo controller. The car has a diesel engine with *vgt* turbine. The turbo controller for the *vgt* has been implemented, overriding the car control signal to the turbine.

First a steady state measurement at different engine speeds are made with only the feed forward link in the controller. The measurement is made at 1500-4500 rpm and shows how accurate the feed forward controller is at different engine speeds and what the stationary error is. The measurement is shown in Figure 4.4 and the interesting points to observe, which is pointed out with arrows, is when the engine speed is kept as close to constant as possible and the actual boost pressure has saturated. The difference between the pressure reference and the actual pressure at the arrows, show the stationary error for the different engine speeds for the feed forward link in the controller. The areas between the constant boost pressure reference is when the engine speed accelerates and is not of important for the stationary behavior. The feed forward link is very accurate at engine speeds between 2000-3500 rpm but starting to deviate when approaching 4000 rpm. Looking at the point in the middle at about 3000 rpm and 140 kPa boost reference, the stationary error is about 2 kPa which show a great simularity to the simulation in Figure 4.1 before the step.

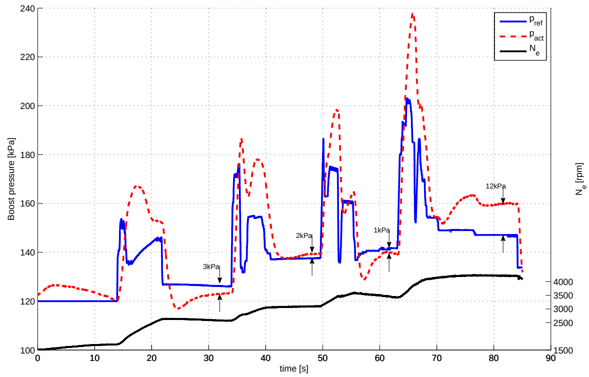
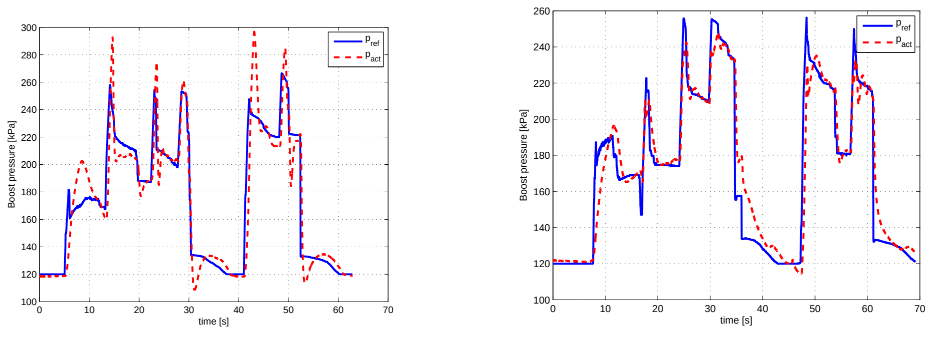


Figure 4.4. *Steady state measurement at different engine speeds with only the feed forward link in the controller. The arrows show the stationary error, the difference between boost pressure reference and actual boost pressure when the actual boost pressure has reach its end value. The feed forward link is very accurate at engine speeds between 2000-3500 rpm but starting to deviate when approaching 4000 rpm. The engine speed is held as constant as possible and the values shown on the right Y-axis, is the engine speeds that correspond to the steady state measurements.*



Measurement in car with developed controller implemented

Measurement in car with originally implemented controller

Figure 4.5. *The solid line is the boost pressure reference from the car and the dashed line is the measured pressure. The left plot with measured pressure from the developed controller shows some overshoot, probably caused by the manually tweaked PID-controller. Comparing to the originally controller in the right plot the developed controller is a bit faster with more overshoot, but by tweaking the PID-controller it could at least reach the same performance. Time spent on calibration for the developed controller is about one hour, which is considerable very little time.*

Figure 4.5 shows the fit between the boost pressure reference and the measured pressure in the car. The actual pressure does follow the pressure reference

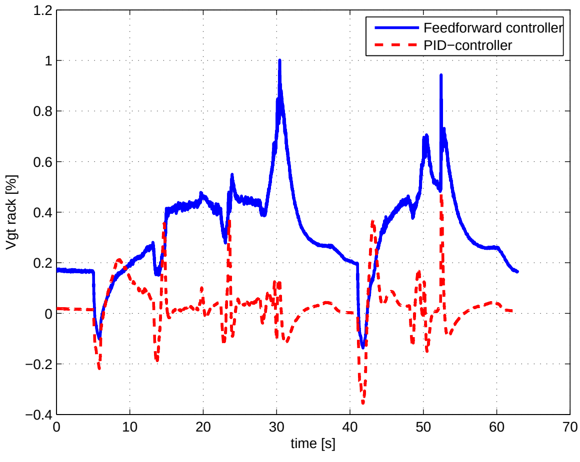


Figure 4.6. *Control signal distribution between the feed forward controller and the PID-controller, from the same car measurement as in the left plot in Figure 4.5. This plot is presented to show that the feed forward controller does the main control signal contribution and the PID-controller contribution is smaller and only compensate for the error between boost pressure reference and actual pressure.*

accurately but with some overshoot. The overshoots could be prevented if the PID-parameters were better tweaked. Compared to the originally controller in the car the performance of the developed controller is good. The overshoot is more significant in the developed controller but it holds close to the same performance elsewhere. One thing to keep in mind is that the time for calibration for the developed controller is about one hour, and the PID-tuning was made in the car manually while measuring. To show that the feed forward controller does the main contribution to the control signal, Figure 4.6 show the distribution to the control signal between the feed forward controller and the PID-controller.

4.3 Disturbance rejection test

The idea with this thesis was to use the information about the exhaust gas temperature in the turbo controller, to handle variations in that temperature and give the correct boost pressure. To see the difference in performance between a controller that do not take information about the temperature into account and one that does, a disturbance test was performed. The disturbance was implemented as a step as (4.1) in the MVEM, since similar tests on a car could not be performed.

$$T_{em} = \begin{cases} T_{em} & \text{for t } < 50s \\ T_{em} + 100 & \text{for t } > 50s \end{cases} \tag{4.1}$$

The results from the simulations, with only the feed forward controller and no PID-controller, is shown in Figure 4.7. The left plot shows the boost pressure with a controller that has no information of the temperature T_{em} and the right plot the boost pressure by using the measured temperature as input to the developed controller. The simulation with the measured temperature, in the right plot, shows that the controller reacts almost immediately, with almost no deviation except for a minor stationary error. The left plot with no knowledge about the temperature gets a big stationary error in the feed forward controller. This test shows that the controller can handle variations in temperature before the turbine very well.

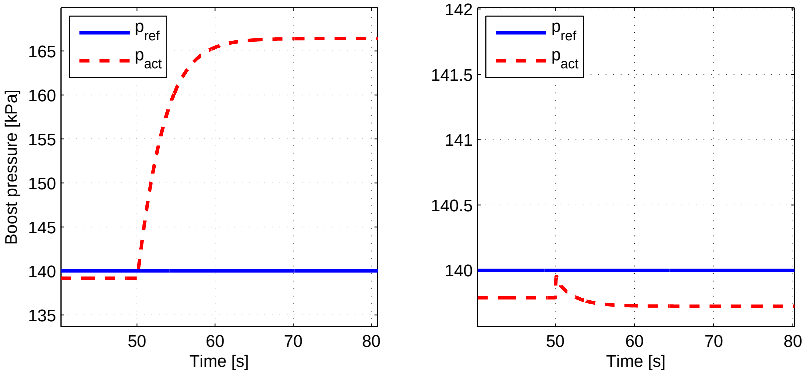


Figure 4.7. *Simulations with a disturbance added on the temperature before the turbine T_{em} according to (4.1), with no PID-controller active. The left plot shows the boost pressure using a controller without knowledge about the temperature before the turbine and the right plot the boost pressure by using the measured temperature as input to the controller. The right plot shows that the controller can handle variations in the temperature before the turbine well if the estimation of it is good. Note the different scaling on the Y-axis.*

With the PID-controller active, the disturbance's affect on the boost pressure can be compensated, even for the controller without knowledge of the temperature. However the improvement shows in the results in Figure 4.8, where the same disturbance test has been done with a PID-controller active. The dashed line shows the boost pressure using a controller without knowledge about the temperature before the turbine and the dash-dotted line the boost pressure by using the measured temperature as input to the controller. Both manage to compensate for the temperature disturbance, but the controller with knowledge of the temperature shows significantly better response.

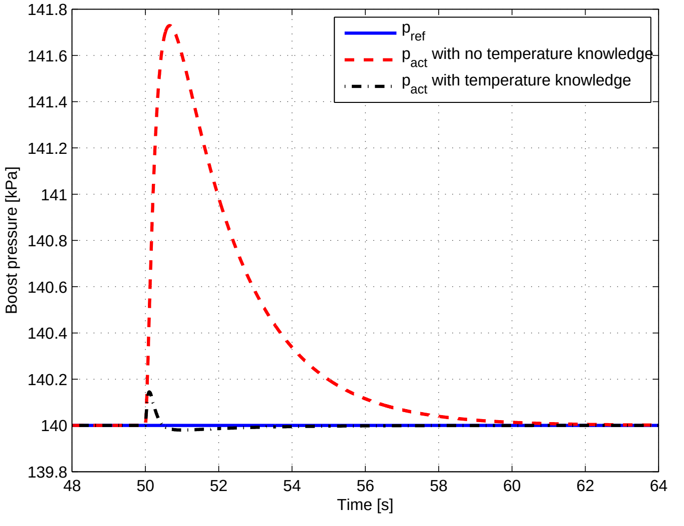


Figure 4.8. *Simulations with a disturbance added on the temperature before the turbine T_{em} according to (4.1), with a PID-controller active. The dashed line shows the boost pressure using a controller without knowledge about the temperature before the turbine and the dash-dotted line the boost pressure by using the measured temperature as input to the controller. Both manage to compensate for the temperature disturbance, but the controller with knowledge of the temperature shows significantly better response. The top value for the dashed line is 141.7 kPa and top value for the dash-dotted line is 140.1 kPa, where the solid line is the boost pressure reference at 140 kPa. The controller with temperature knowledge is getting some undershoot, which can be caused by change in stationary error after the disturbance step or poor PID-tuning, and it is less than 20 Pa.*

Chapter 5

Future work

There have been some areas in this thesis that should be interesting to study closer to improve the controller. They are presented here.

Include a throttle model for improved dynamic control

The boost pressure reference is the pressure before the throttle and the stationary mass flow is calculated from the volumetric mass flow model in the intake manifold depending on the pressure after the throttle. In this thesis the approximation $p_{ref} = p_{im}$ has been made and is acceptable since all the simulations and measurements are made with a fully open throttle. It would be interesting to implement a throttle model between the boost pressure reference and the intake manifold pressure as $p_{im} = p_{ref}(\theta_{throttle})$ to be able to use the throttle for faster dynamic control.

Improved model for temperature before after the turbine

A controller that takes information of the temperature before the turbine has been developed. To use that information a good estimation of the temperature before the turbine is needed.

The controller with *wg* applied on the MVEM, takes a modeled value for the temperature before the turbine. Since the MVEM models a turbocharged gasoline engine which generally do not have temperature sensor before the turbine. The model needs to handle temperature variations caused by ineffective combustion, for example. The currently model used, that calculates the temperature from the engine massflow and temperature drop in the exhaust manifold, might not be sufficient.

The controller with *vgt* applied on the diesel car lacks models for the temperature before the turbine, T_{em} and pressure after the turbine p_t. It takes measured values from the car as input instead. If suitable models were developed for those signals, the feedforward controller could get faster, since a model could calculate the end value instantly when the actual pressure and especially the temperature do not rise so fast.

Improve the feedback PID-controller

The feedback controller is a PID-controller that corrects the *wg* effective area or the *vgt* nozzle angle to minimize the error between the boost pressure reference and the actual pressure. The PID-controller parameters should be estimated and evaluated more accurately, to avoid overshoot. It would be interesting to investigate if it would be more beneficial to apply a feedback controller on some other variable. For example minimizing the error between estimated and measured mass flow, instead of the boost pressure.

Include wastegate actuator in the controller with a wastegate valve

To be able to implement the *wg* controller in a car to perform similar tests as for the *vgt* a wastegate actuator is needed to convert the demanded effective area to a control signal. This has not been done because the MVEM takes the effective area as input directly and no car with a wastegate turbo has been available.

Chapter 6

Summary and conclusions

6.1 Summary

In this thesis an enthalpy based boost pressure controller is developed. It takes a boost pressure reference and calculates a desired turbine power. Together with the available energy in the exhaust gas generate a control signal to the turbine system. The control signal is calculated as a static feedforward value, i.e. the value corresponds to the stationary end value that matches the boost pressure reference. A PID-controller is then implemented to correct the stationary error and to increase the pressure rise in the control signal based on the difference between the boost pressure reference and the measured boost pressure. The controller is module based and built in MATLAB SIMULINK and is applied to a mean value engine model for simulations and a car for measurements, to verify the performance.

6.2 Conclusions

6.2.1 Controller modeling

The developed controller contains model equations that convert a boost pressure reference to a control signal for the turbine. The model equations describes the system components in a turbocharged engine, from pressure and temperature drops over the intercooler and air filter, to efficiencies, generated power for turbine, turbo speed and pressure ratios. To put all these models together and get a working controller has demanded a lot of effort and validation. Finally the controller ended up working well with good results, both in simulation and in reality. This thesis show that working with Mean Value Modeling and simulations is an efficient way to start developing turbo controllers.

6.2.2 Performance test

The simulated values and measured data shows a good fit between the boost pressure reference and the actual pressure. The car measurements showed some overshoot, due to insufficient PID-controller calibration. The performance is almost as good as the originally implemented boost pressure controller in terms of difference between boost pressure reference and actual pressure. The calibration time is worth taking in consideration here, since the calibration has been done in about one hour before testing in the car and the PID-tuning was made in the car while doing the measurements. Given a little more time on the controller calibrations, the result would probably come much closer to the original controller performance. The advantage with the developed controller in this thesis is that it can handle variations in exhaust temperature since it uses the information on the exhaust side of the engine, which will lead to less turbine calibration.

Bibliography

[1] Filip Ainouz and Jonas Vedholm. Mean value model of the gas temperature at the exhaust valve, 2009.

[2] Per Andersson. *Air Charge Estimation in Turbocharged Spark Ignition Engines*. 2005. PhD thesis, Linköpings University.

[3] Christopher Depcik and Dennis N. Assanis. A universal heat transfer correlation for intake and exhaust flows in an spark-ignition internal combustion engine, 2002.

[4] Lars Eriksson. Mean value models for exhaust system temperatures. *SAE Transactions, Journal of Engines, 2002-01-0374*, 111(3), September 2003.

[5] Lars Eriksson. Modeling and control of turbocharged SI and DI engines. *Oil & Gas Science and Technology - Rev. IFP*, 62(4):523–538, 2007.

[6] Lars Eriksson. *Modeling of Turbocharged Engines with MVEM_lib*, 2008.

[7] Lars Eriksson and Lars Nielsen. *Modeling and Control of Engines and Drivelines*, 2009. Course material.

[8] A.Y. Karnik, J.H. Buckland, and J.S. Freudenberg. Electronic throttle and wastegate control for turbocharged gasoline engines. In *American Control Conference, 2005. Proceedings of the 2005*, pages 4434 – 4439 vol. 7, 2005.

[9] Philippe Moulin and Jonathan Chauvin. Modeling and control of the air system of a turbocharged gasoline engine. *Control Engineering Practice*, 19(3):287 – 297, 2011. Special Section: IFAC World Congress Application Paper Prize Papers.

[10] Martin Müller. Mean value models for exhaust system temperatures. *SAE Technical Paper Series, 2008 World Congress. Detroit, Michigan , 2008-01-1013*, 2008.

[11] Andreas Thomasson, Lars Eriksson, Oskar Leufven, and Per Andersson. Wastegate actuator modeling and model-based boost pressure control. In *IFAC Workshop on Engine and Powertrain Control, Simulation and Modeling*, Paris, France, 2009.

[12] Niel Watson and Marian S. Janota. *Turbocharging the internal combustion engine*. 1982.

Appendix A

Nomenclature

The variables and their subscripts that are used in this thesis are listed below.

Variable	Parameter	Unit
A	area	m^2
C_D	discharge coefficient	-
c_p	specific heat coefficient	-
γ	specific heat ratio	$\frac{J}{KgK}$
$\dot{m}$	mass flow	Kg/s
N	engine speed	rpm
p	pressure	Pa
P	power	W
Π	pressure ratio	-
T	temperature	K
ω_{tc}	turbo speed	rad/s

Subscript	Description
amb	ambient
bc	before compressor
c	after compressor
ds	downstream (after)
eff	effective
em	exhaust manifold
ic	intercooler
im	intake manifold
stat	stationary
t	after turbine
us	upstream (before)
vgt	variable geometry turbine
wg	wastegate