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Online modelling and control of a turbo charging system using Simulink

Master's Thesis in Automotive Engineering

JACOB HEIMERSSON

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Department of Applied Mechanics
Division of Combustion
CHALMERS UNIVERSITY OF TECHNOLOGY
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Cover:
Example of turbocharger design, see Section 2.1.1 for more information about turbochargers. Figure reference Autoparts (2012).

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ABSTRACT

Combustion system development for future engine applications can be performed on a single cylinder combustion research engine and then with a highly controlled environment. To be able to operate such an engine at relevant conditions comparable to a multi cylinder engine, it is desirable to model and control the boundary conditions, e.g. trough simulation of turbo charging. This work was done as a master thesis during the spring of 2012 at Chalmers University of Technology and the aim was to develop an online simulation of a single turbo system for a single cylinder combustion research engine.

The simulation was developed in Matlab/Simulink and integrated in an AVL PUMA test bed system using AVL ARTE.Lab. The turbo performance was predicted by using measured data from the turbo developer, measured according to SAE standard. The model strategy was to predict what charge pressure needed in order for the engine to run at a certain lambda value with a given amount of fuel injected and then calculate whether or not the turbo system would be able to deliver such a charge pressure under the prevailing conditions.

The model has been validated both by reference test runs with a real engine as well as comparison with simulations carried out in the software GT-Power. These validations showed that the prediction of charge pressure works well and that the prediction of the turbocharger behavior shows the characteristic of the turbo. Although improvements in this section may be needed for a better accuracy of the prediction as an offset in the values can be seen.

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Preface

In this study an online simulation of charge pressure has been developed, during the spring of 2012. The project is a master thesis work done by Adam Johannesson and Jacob Heimersson. Examiner has been Associate Professor Sven Andersson at the Division of Combustion at the Department of Applied Mechanics at Chalmers University of Technology, who we would like to thank for his valuable advices and thoughts in different situations.

All tests have been done at Volvo Car Corporation (VCC) Torslanda site in Göteborg. Supervisor at VCC has been Dr Håkan Persson at the Diesel Concepts and Attributes group at the Engine Engineering department, whom without this thesis could not have been accomplished.

We would also like to thank the entire Diesel Concepts and Attributes group at VCC for welcoming us to their group and for the support in different issues during our work. Special thanks should go to Per Gustavsson, whom has taught us what cannot be read in books and encouraged us every day, and David Willermark at the Engine Simulation group, who has supported us with engine simulations and turbocharger data management.

Göteborg, June 2012

Adam Johannesson and Jacob Heimersson

Notations

Roman letters

Eff_{comp}	Compressor efficiency
ER_{turb}	Turbine expansion ratio
$H_{actual_{rise}}$	Actual enthalpy rise
$H_{isentropic_{drop}}$	Isentropic enthalpy drop across turbine stage
$H_{isentropic_{rise}}$	Isentropic enthalpy rice across compressor stage
N_{comp}	Compressor speed
$N_{comp_{corr}}$	Corrected compressor speed
N_{turb}	Turbine speed
$\dot{m}_{corrected}$	Corrected mass flow rate of a map
$\dot{m}_{max-corrected}$	Maximum corrected mass flow rate of a map
$\dot{m}_{air}$	Air mass flow
$\dot{m}_{fuel}$	Fuel mass flow
Mf_{comp}	Air mass flow trough compressor
$Mf_{comp_{corr}}$	Corrected air mass flow trough compressor
Mf_{turb}	Exhaust mass flow trough turbine
P_{amb}	Ambient pressure
$P_{comp_{in}}$	Compressor inlet pressure
$P_{comp_{out}}$	Compressor outlet pressure
P_{ref}	Reference pressure for corrected input data
$P_{turb_{in}}$	Turbine inlet pressure
$P_{turb_{out}}$	Turbine outlet pressure
PR	Actual pressure ratio of a map
PR_{comp}	Compressor pressure ratio
$PR_{normalized}$	Normalized pressure ratio
PR_{min}	Minimum pressure ratio of a map
PR_{max}	Turbine maximum pressure ratio
PR_{stall}	Compressor maximum pressure ratio
R_{ref}	Reference gas constant for a map
$RPM_{max-corrected}$	Maximum corrected speed of a map
$RPM_{max-reduced}$	Maximum reduced speed of a map
S_v	Swept volume
T_{amb}	Ambient temperature
$T_{comp_{in}}$	Compressor inlet temperature
T_{int}	Temperature in intake tank
T_{ref}	Reference temperature for corrected input data
$T_{turb_{in}}$	Turbine inlet temperature
TFP	Turbine Flow Parameter
TSP	Turbine Speed Parameter
Vf_{comp}	Volume flow through compressor
$\dot{V}_{max-reduced}$	Maximum reduced volumetric flow rate of a map
$\dot{V}_{max-corrected}$	Maximum corrected volumetric flow rate of a map
$\dot{V}_{normalized}$	Normalized volumetric flow rate of a map
$\dot{V}_{reduced}$	Reduced volumetric flow rate of a map

Greek letters

λ	Lambda value
ρ_{req}	Required density
ρ_{amb}	Ambient density

List of Abbreviations

AVL	Anstalt für Verbrennungskraftmaschinen List (Austrian based automotive engineering consulting firm)
CAC	Charge Air Cooler
CAD	Crank Angle Degree
CMP	Compressor (file format for extrapolated compressor map)
EGR	Exhaust Gas Recycling
MPE	Model Parameter Editor
SAE	Society of Automotive Engineers
TFP	Turbine Flow Parameter
TRB	Turbine (file format for extrapolated turbine map)
VCC	Volvo Car Corporation

1 Introduction

1.1 Background

Combustion system development for future engine applications can be performed on a single cylinder research engine and then with a highly controlled environment. To be able to operate such an engine at relevant conditions comparable to a multi cylinder engine, it is desirable to model and control the boundary conditions, e.g. trough simulation of a charging system.

Volvo Car Corporation (VCC) today has single cylinder research engines with the equipment for controlling these boundary conditions. The problem is to determine the target values for these systems for different operating conditions of the engine to imitate the conditions that can be found in a multi cylinder engine.

1.2 Purpose

The purpose of this master thesis is to design tailored models using Simulink in order to determine the boundary conditions for the turbo system of a single cylinder engine and to feed set point values to the engine control. The model should also be able to evaluate whether or not a specific turbo system is suitable for the engine at different driving conditions.

Further, the models shall be implemented in VCC's new single cylinder automation system, verifying the Simulink model against real data. The end result will be a model running on-line, updating boundary conditions and feeding set point values to the engine control.

1.3 Scope

This thesis work covers the simulation of a single turbo system regulated by a wastegate. It does not include other types of charging systems, such as multiple turbo systems, variable geometry charging systems, compressor charging systems or any combination of these. The reason for the limitation of a single turbo system is due to the time frame in a master thesis.

1.4 Methodology

The work procedure in this master thesis has been an iterative process, as illustrated in Figure 1. The work was carried out in steps and if any of the steps failed during the process one or several steps backwards were taken to search for a solution for the problem. Below follows an explanation to each of the steps in the process and what purpose the step has.

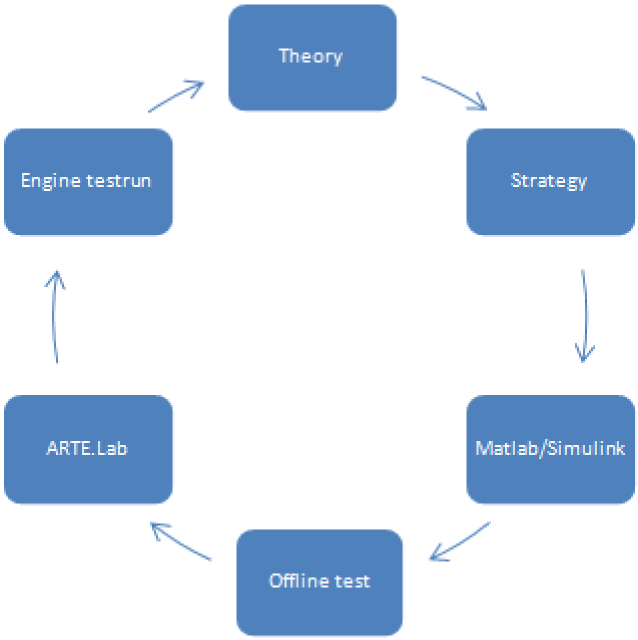

Figure 1 *Work procedure.*

Theory – In this stage theory relevant for the area that is needed to be modeled was examined to ensure a solid theory background for the models.

Strategy – Based on the theory found in the theory step, strategies were built up of how to approach the modelling problem. This step could be said to be the link between the theory and the actual model, explaining how the models work in principle and how they are supported by theory, without going too deep into the model and the software part of modelling.

Matlab/Simulink – In this step models are built up according to the strategies by programming code and graphical interface, using the software Matlab/Simulink. Input signals are programmed to be loaded from the engine test rig and update continuously to enable real time simulation, where the output signals are feed continuously to the engine test rig.

Offline test – In this stage an offline verification of the models is carried out. An offline test is a test of the model in the Matlab/Simulink environment which means that the basic function of the model is tested. Signals that in online tests are loaded from the engine test rig are changed to constants in the offline test since there is no connection to the engine test rig.

ARTE.Lab – To enable the feature of loading input signals from the engine rig the Matlab/Simulink models is compiled with a software called ARTE.Lab. This software compiles the models into code that can be read by the engine test rig system PUMA (see Section 2.4.2 for more information about PUMA). It is not until the code is integrated in the PUMA system that the loading of input signals and feeding of output signals to the engine will work, which explains why the step *offline test* is needed to verify the models at an earlier stage. More information about ARTE.Lab can be found in Section 2.4.3.

Engine test run – To verify that the model works as intended it was tested in the environment where it is supposed to work. The model is installed in the engine test rig and runs at the same time as the engine exchanging values. This step controls that the reading of signals from the engine works and that the feeding of output signals from the model works. Also the output values from the model is verified by running the engine in reference points and comparing the output from the model with data that is logged from the turbo system in a multi cylinder engine running at the same point.

2 Theory

This chapter explains the basic theory relevant for this master thesis. Also the physical and software environment in which the project is carried out in is introduced.

2.1 Charging Systems

The society of today are demanding a decrease in the emissions from combustion engines and a popular approach to this from the car manufacturers side is downsizing, which means decreasing the swept volume of the engines, with smaller and less number of cylinders. But in contradiction to this the customers are demanding more powerful engines and better acceleration. Since the amount of air in the combustion is a major limiting factor for the power output from the engines, different charging systems to force more air into the cylinder, have become more and more important within engine development.

There are two major charging systems that are used today, exhaust driven turbochargers and mechanical driven compressors, or superchargers. This thesis concentrates on the simulation of turbochargers and hence those will be explained more in dept in the following section.

2.1.1 Turbochargers

The definition of a turbocharger from SAE Turbocharger Nomenclature and Terminology, SAE International (2011) is, "*Turbocharger – A device used for increasing the pressure and density of the fluid entering an internal combustion engine using a compressor driven by a turbine which extracts energy from the exhaust gas*". From the last part of this definition the difference between a supercharger and a turbocharger can be distinguished. In a turbocharger the energy needed to compress the fluid entering the engine is taken from the exhaust gases, and in a supercharger the energy is extracted with a mechanical connection to the engine.

The following explanations of turbochargers and superchargers are compiled from Heywood (1988), Bosch (2004), Borg Warner (2012), SAE International (1995) and SAE International (2011).

The turbine and compressor are connected by a shaft to transfer power in between. The turbine extracts energy from the exhaust gases and hence the power available from the turbine depends on the energy available in the exhaust gases in terms of mass flow, temperature, volume flow etc. The power extracted by the turbine is transferred by a shaft to a compressor, which uses the energy to compress the air induced in the engine.

A turbocharger consists of four basic components, a compressor, a turbine, a boost control system and a bearing system. These will be further described below, and especially the first three, of which the understanding is of special importance for this thesis work.

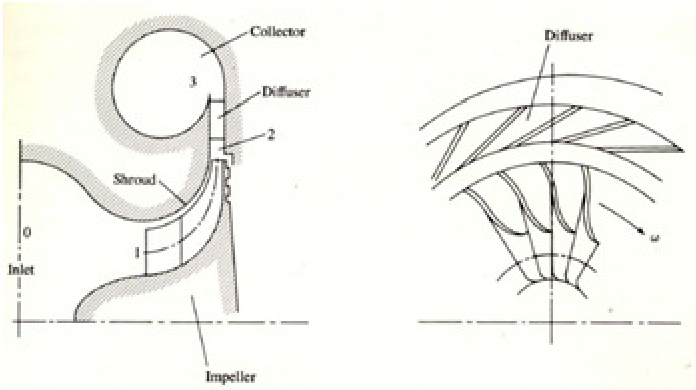

Figure 2 Centrifugal compressor, Heywood (1988).

2.1.1.1 Compressor

In most cases the compressor in a turbocharger is a centrifugal compressor such as in Figure 2. For this type of compressor to deliver a suitable pressure ratio for an internal combustion engine (up to about 3.5) quite high rotational speeds are needed, which can be delivered by an exhaust driven turbine.

The compressor consists of an impeller and housing. The impeller is rotating which makes the impeller blades, which can be seen in the right part of Figure 2, to suck air in. The air is sucked in axially, accelerated in the impeller and then sent out in a radial direction towards the diffusers (see left of Figure 2). The diffusers slow the air down, more or less without losses, which causes both the pressure and temperature to raise Borg Warner (2012). The geometry of the diffusers affects this change of the fluid and hence the geometry of the diffusers is important for the compressor characteristics. After the diffusers the air is collected in a collector which leads the air into pipes to the intake manifold of the engine.

The performance of compressor can be evaluated from a compressor map, as seen in Figure 3. Below the different parts of the map will be explained.

Volume flow rate: On the x-axis the volume flow rate of the fluid can be seen. The volume flow rate is normalized against a reference temperature so that the map can be used under different conditions by,

$$Volume\ flow\ rate = Vf_{\text{comp}} \cdot \sqrt{\frac{T_{\text{ref}}}{T_{\text{comp}_{\text{in}}}}} \tag{1}$$

Pressure ratio: On the y-axis the pressure ratio over the compressor can be seen. The pressure ratio is defined as:

$$PR_{comp} = \frac{Pressure\ before\ compressor}{Pressure\ after\ compressor} \tag{2}$$

Surge line: When the flow through the turbine is decreased and the compressor is kept at a constant speed there will come a point when the flow is too low to give a certain pressure ratio. This phenomenon will occur when the air no longer can adhere to the suction side of the impeller blades, causing a backflow through the compressor until stability is reached again with positive volume flow. Once stability is reached the pressure will build up again and the same cycle repeats. The regions to the left of the surge line an unstable region where the compressor should not be operated in.

Choke line: On the right side the map is limited by the choke line. Choking occurs when the speed of the fluid entering the impeller inlet reaches sonic velocity and hence the mass flow cannot increase further unless the speed is increased. At a certain point the narrowest section of the inlet of the compressor will be choked which gives that a further increase in speed will not any longer result in a higher mass flow.

Constant speed lines: The lines in vertical direction on the map shows how the pressure ratio is varied with volume flow rate when the speed is kept constant. The speed in the map is corrected (normalized) by,

$$N_{comp_{corr}} = N_{comp}\sqrt{\frac{T_{\mathrm{ref}}}{T_{\mathrm{comp_{in}}}}} \tag{3}$$

Constant efficiency lines: The oval shaped lines in the map marks the efficiency of the compressor.

2.1.1.2 Turbine

The most common type of turbine for passenger cars is a radial flow turbine, which operates in a similar way as the centrifugal compressor explained in Section 2.1.1.1, but with a major difference. Since the exhaust gas is expanded over the turbine the gas flows into the turbine in a radial direction and flows out in an axial direction, in other word, in the opposite direction of the flow in the centrifugal compressor.

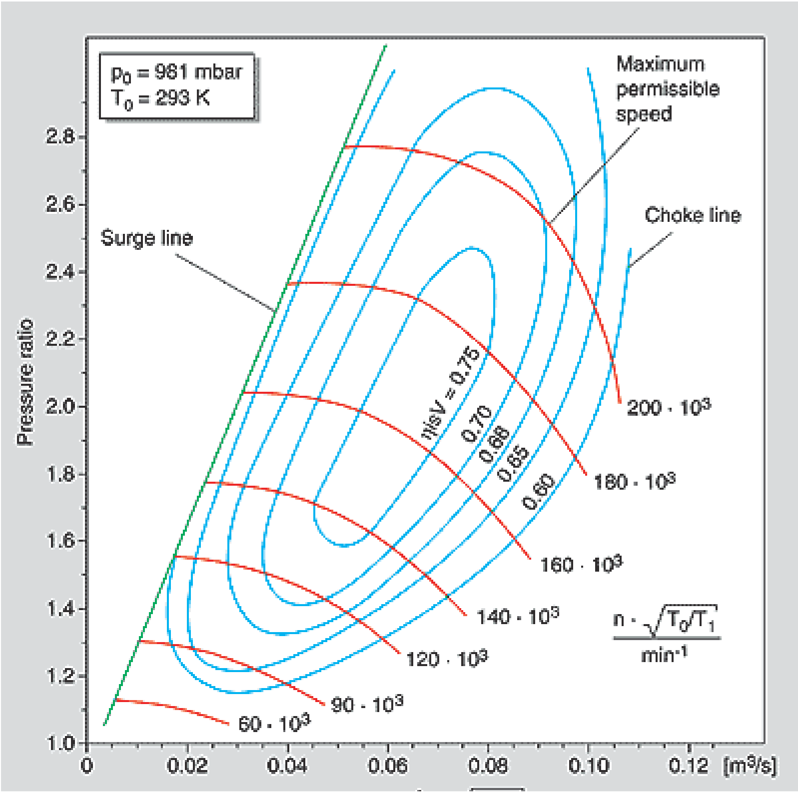

Figure 3 Example of a compressor map, Borg Warner (2012).

The radial flow turbine consists of a turbine wheel (impeller, rotor) and turbine housing (see Figure 4). The turbine converts the energy in the exhaust gases into mechanical energy in the rotating turbine wheel from the pressure drop over the turbine. Hence the performance of the turbine increases as the pressure drop over the turbine increases. The impeller is designed to enable a pressure drop almost down to atmospheric pressure which gives that the pressure before the turbine is the major factor determining the performance of the turbine. The performance of a turbine can be evaluated from a turbine map of which an example can be seen in Figure 5.

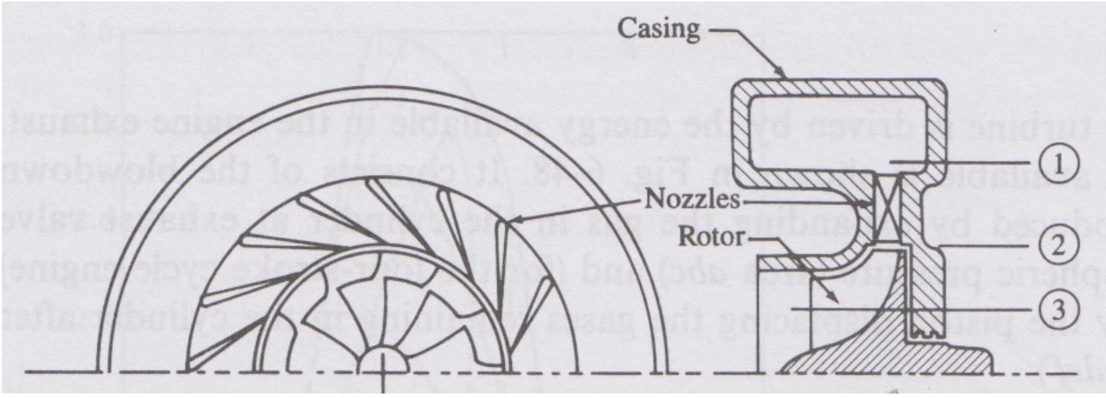

Figure 4   Radial flow turbine, Heywood (1988).

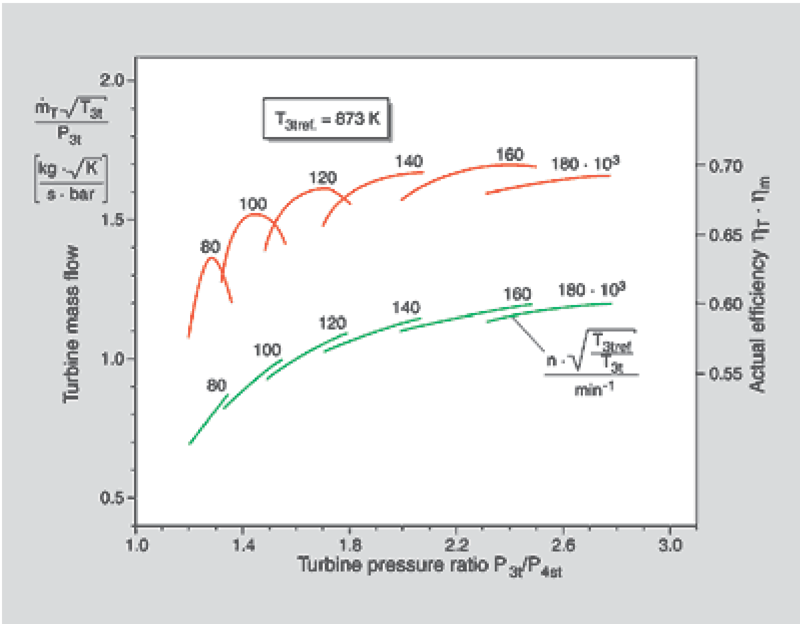

Figure 5   Example of a turbine map, Borg Warner (2012).

2.1.1.3 Boost control system

Since a passenger car engine operates over a wide rpm span the turbo system must be able to deliver charge pressure under different conditions. As the engines rpm is increasing the demanded air mass into the engine, as well as the exhaust mass output is increasing. For an engine to have a good driveability the turbo system must be designed to deliver at quite low engine rpm, otherwise the driver will experience what is most commonly known as turbo lag. When a system is designed like this, the exhaust flow just below the engine rpm where maximum torque is achieved from the engine will be enough to deliver the required boost pressure. After this point the remaining exhaust flow will contribute to a higher exhaust pressure, resulting in a less effective engine. To avoid this phenomenon a boost control system is fitted to the turbo system. There are three common ways to design a boost control.

Bypass valve: The simplest way to control charge pressure, and the one studied in this thesis, is to fit a bypass valve, as seen in

Figure 6, which allows the excess of exhaust flow to be routed around the turbine to the exhaust system downstream the turbine.

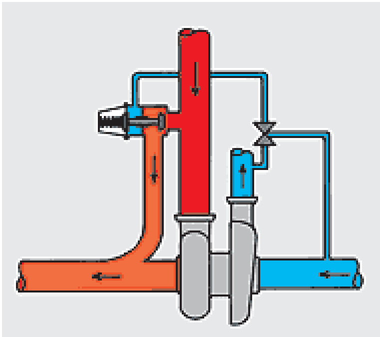

Figure 6 Bypass valve layout, Borg Warner (2012).

Traditionally the position (open/closed) of the bypass valve has been controlled by a spring loaded diaphragm dependent on the boost pressure on the intake side. Although this gives a drawback of that the bypass valve is set to open at a certain pressure ratio over the compressor, protecting against too high charge pressure. To further develop the use of the bypass valve the valve control are made electrically, making it possible to regulate the charge pressure with a signal from the engine control unit.

2.1.1.4 Bearing system

The bearings of turbochargers are not of interest for this report, but they will still be briefly discussed to give the reader an understanding of their importance for the turbocharger system.

Since the shaft for the compressor and turbine wheels rotates at speeds up to 300 000 rpm the demands on the bearings of the shaft is high. The most common type of bearings is sleeve bearings, both in axial and radial direction. This type of bearings meets the high demands at reasonable cost.

2.2 Charge Air Cooler

A charge air cooler, also called intercooler, is a cooling device which decreases the temperature of the intake air. As the name intercooler implies the air is cooled between the charging device and the cylinder. The reason for adding a intercooler is that the intake air is heated as a by-product of the charging and in order to keep the combustion temperatures down, to keep emissions down and to keep the density of the intake air high to better utilize the volume flow into the engine a charge air cooler is used.

There are different designs of intercoolers divided into two main categories, Air-cooled intercoolers and Coolant-cooled intercoolers. Air-cooled intercoolers have an air-to-air heat exchange and they are the most common type on passenger cars and commercial vehicles. The drawback of this system is that it needs to be placed in such a way that air will flow through it (in front of the car, with scoops etc). Coolant-cooled intercoolers has a better flexibility in placement, but it has the drawback of that the coolant needs a low temperature in to the heat exchanger and therefore the regular coolant for the engine is not optimal to use, since its temperature is usually around 90°C.

For this thesis the interest lies in the pressure and temperature drop that occurs over the charge air cooler.

2.3 Exhaust Gas Recycling (EGR)

Exhaust gas recycling means that some of the exhaust gases are recycled into the intake manifold and there mixed with the fresh fuel and air mixture. Since the new mixture with EGR has lower oxygen content than a pure fuel and air mixture the peak combustion temperature is lowered. The effect of this is that the produced nitrogen oxides (NO_X) are reduced since NO_X is strongly temperature dependent.

There are two types of EGR, internal and external. Internal EGR occurs in every engine and depends on the overlap of the intake and exhaust valves. External EGR is when the exhaust gases are forced back into the intake manifold and it is this type of EGR that this chapter describes.

The exhaust gases are routed back into the intake manifold via an EGR valve and an EGR cooler, see

Figure 7.

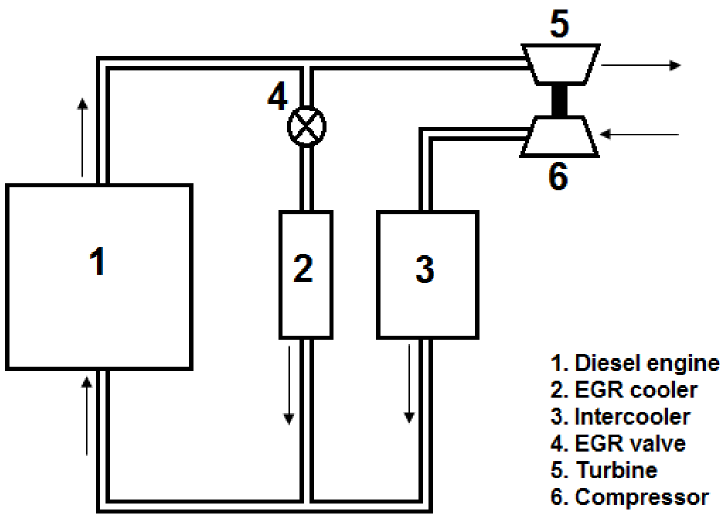

Figure 7 Schematic view of the EGR system.

To optimize the effects of the EGR an EGR cooler is used to lower the temperature of the exhaust gases and therefore also decrease the peak combustion temperature and pressure even more.

One way of categorize different types of EGR is by the route. It is called short route EGR when the exhaust are lead back to the intake manifold directly after the cylinders (see

Figure 7). If instead the exhausts are lead to the intake manifold after the turbo turbine it is called long route EGR.

2.4 Environment

In this section the environment that the resulting model is developed in and for will be described. Both the physical environment and the software's will be described.

2.4.1 Engine Test Cell

The engine test cell is a one cylinder research diesel engine at VCC in Torslanda. The test cell and engine is controlled via AVL PUMA Open test bed system, which is explained in Section 2.4.2. In

Figure 8 below, the most important parts of the test cell is shown and will be explained below. The dotted lines represent intake flows and the unbroken lines represent exhaust flows.

Dyno – A dynamometer that brakes the engine to a desired speed.

1-cyl engine – A 1-cylinder combustion research engine.

Intake tank – A large volume tank where the intake air is mixed with EGR. The tank helps evening out pressure pulses in the system.

Exhaust tank – A large volume tank where pressure pulses from the engine is evened out. In this tank EGR gas is extracted and lead to the intake tank and emissions are measured.

Charge pressure valve – A valve that controls the pressure in the intake. Upstream the valve is highly pressurized air so that the pressure drop over the valve decides the pressure in the intake tank.

Backpressure valve – A valve that controls the backpressure in the exhaust system to simulate pressure drops such as turbines, mufflers etc. The backpressure is needed to drive the EGR from the exhaust system to the intake system.

EGR valve – A valve that controls the EGR flow from the exhaust system to the intake system based on that the pressure in the exhaust system is higher than that in the intake system.

In addition to this the engine is equipped with multiple sensors which measures for example temperature, pressure and flow at different locations.

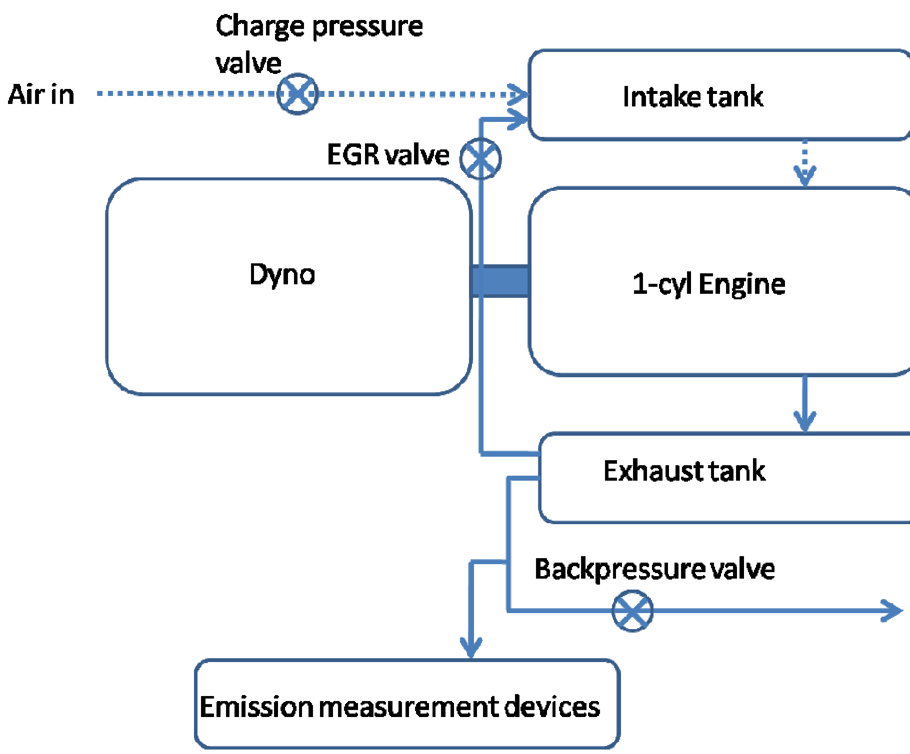

Figure 8   Engine test cell overview.

2.4.2 AVL Puma Open

AVL PUMA Open is a test bed automation system used to control the test bed attributes, such as dynamometer, valves, pumps, coolers etc. PUMA also handles measuring sensors in and around the engine to display temperatures, pressures, engine speeds, emission equipment etc. In the test cell used for this thesis PUMA is connected to INCA-PC which enables PUMA to also control the ECU, with parameters such as injection timing, pilot injection control etc.

2.4.3 AVL ARTE.lab

ARTE.lab is a AVL software for integration of real-time applications into AVL test bed systems, such as AVL PUMA Open. After creating a model in Simulink the model can be compiled with ARTE.Lab and installed in the PUMA test bed system.

When building the model, a Model Parameter Editor (MPE) is created with parameters that should be displayed to the user of the model in the PUMA test bed can be defined, with explanations, min and max value for each value etc. An example of how this window would look like for the user in PUMA test bed is shown in Figure 9 below.

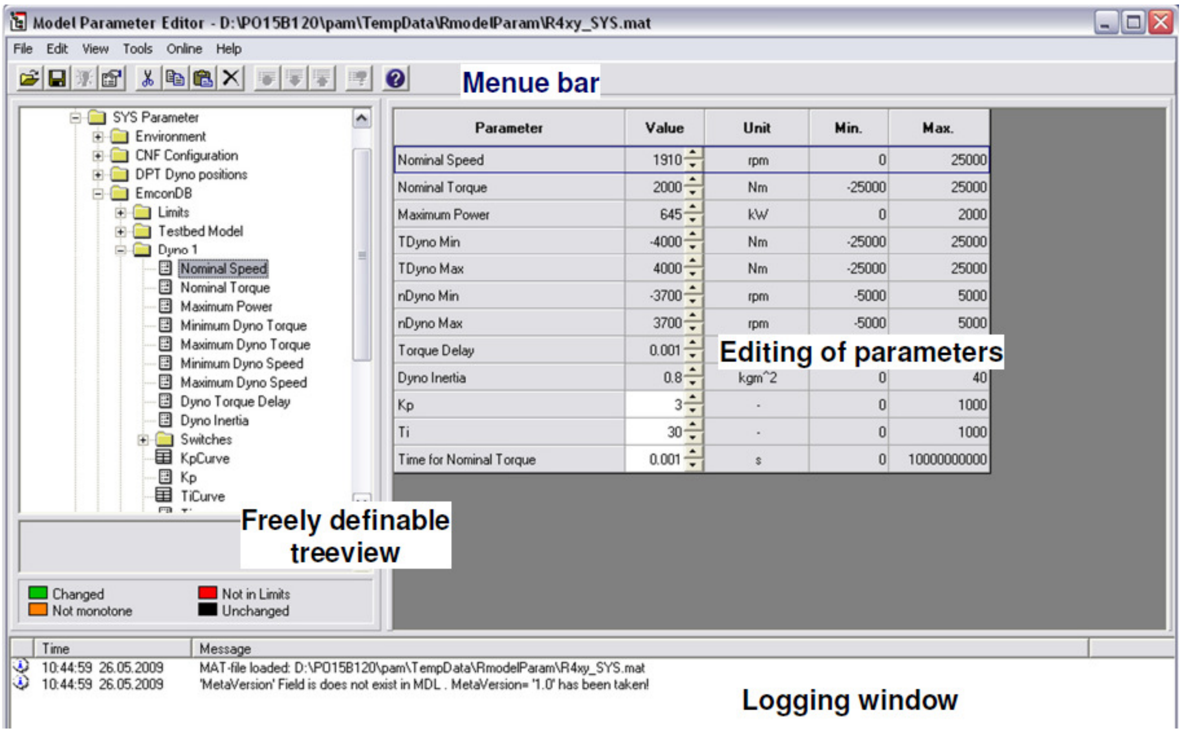

Figure 9 Model Parameter Editor window example.

3 Model Development

3.1 User Demands

The user should be able to define what lambda he/she wants to run on and what fuel amount that should be injected. Given this and input data measured from the running engine a charge pressure should be calculated that would satisfy the given conditions. Furthermore the program should be able to predict if the conditions that the user wants the engine to operate in is suitable for the turbocharger setup of choice.

The data regarding the turbochargers should be the same data as is used for the gas exchange simulation in engine simulation software such as GT-Power, which gives the advantage that no new data is needed to be measured and the data is already available at VCC.

3.2 Modulation Strategy

To identify which parts that needed to be taken into account in the model a typical turbo system installation was studied. Since the simulation runs online and receiving variables from a running engine in real-time the combustion process in the cylinder does not need to be modelled and neither does the valve train and its components. The areas that this thesis focuses on modelling are the turbine, compressor, charge air cooler and the EGR-circuit. The theory behind these areas is explained in Chapter 0 and how they affect the modelling will be described below. As mentioned in previous section, Section 3.1, also the required charge pressure needs to be calculated to determine whether or not a specific turbo system is suitable.

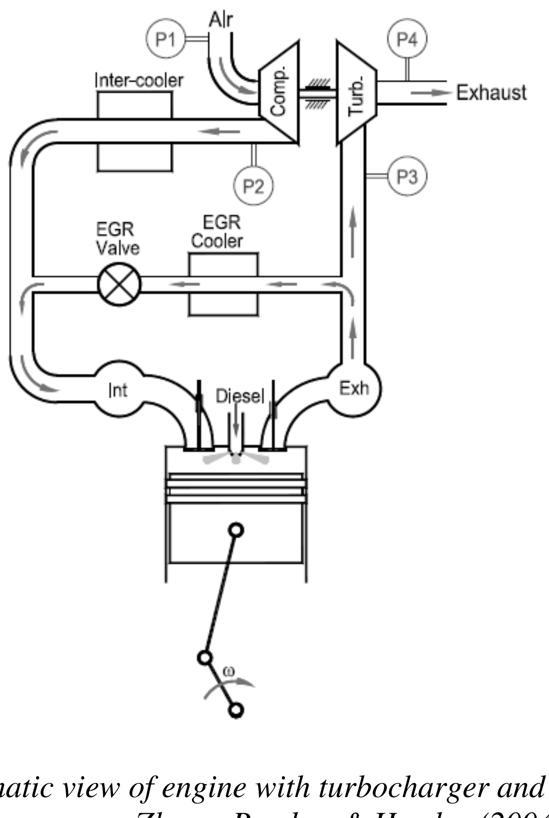

Figure 10 Schematic view of engine with turbocharger and charge air cooler, Zheng, Reader, & Hawley (2004).

3.2.1 EGR Content in Intake Tank

To predict the volume in the cylinder available for the charge of fresh air at each cycle the amount of EGR-gas in the intake air has to be known. In the PUMA system the measurement for the EGR-gas content in the intake tank is displayed by the CO_2-content in the intake tank. To calculate the content of EGR-gas in the intake tank, the measurement of the CO_2-content in the exhaust tank is used. Assuming that the CO_2-molecules are evenly spread in the exhaust tank, each molecule of CO_2 in the intake tank should correspond to a certain amount of EGR-gases, as shown in the relationship in Equation (4):

$$\frac{EGR\ gas_{intake}}{Exhaust\ gas_{exhaust}} = \frac{CO_2 intake - 0.04}{CO_2 exhaust} \qquad [-] \qquad (4)$$

Due to that the CO_2-content in air is 0.04 per cent.

Since the amount of exhaust gas in the exhaust from obvious reasons are 100 per cent, the amount of EGR in the intake system, in per cent, can be calculated as in Equation (5):

$$EGRcontent_{intake} = \frac{CO_2 intake - 0.04}{CO_2 exhaust} * 100 \qquad [\%] \qquad (5)$$

3.2.2 Required Charge Pressure

If the user want to run the engine at a given lambda with a certain fuel amount a certain mass of air will be required, given from the formula, in Equation (6):

$$\lambda = \frac{\left(\frac{\dot{m}_{air}}{\dot{m}_{fuel}}\right)_{actual}}{\left(\frac{\dot{m}_{air}}{\dot{m}_{fuel}}\right)_{stoichiometric}} \qquad [-] \qquad (6)$$

To determine the volume of fresh air induced into the cylinder each stroke the content in the cylinder has to be sorted out. Figure 11 shows the content in the cylinder after a finished intake process. Some of the volume will be taken up by residual gases left in the cylinder from last combustion cycle and from the intake there has been induced a mixture of fresh air and EGR-gases. To determine how much volume that is occupied by each of the parts the following calculations are done.

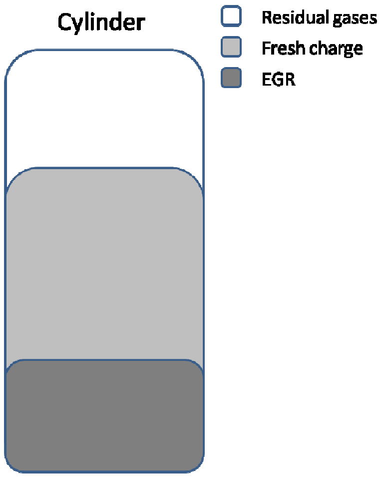

Figure 11   Content in cylinder after intake process.

The theoretic air consumption of the engine, which means the highest possible air consumption that could be achieved by the engine, is defined by the swept volume of the engine by Equation (7):

$$Airconsump_{theoretical} = S_v \cdot \frac{N}{2 \cdot 60} \quad \left[\frac{m^3}{s}\right] \tag{7}$$

The theoretic air consumption is reduced with a volume, residual gases, which are not exchanged each cycle. In this thesis it has been assumed that 85 per cent of the volume in the cylinder is exchanged each cycle, giving that the total induced volume is, calculated by Equation (8):

$$vol_{induced,tot} = 0.85 \cdot Airconsump_{theoretical} \quad \left[\frac{m^3}{s}\right] \tag{8}$$

Since the total induced volume consists of both air and EGR-gases the induced air volume can be calculated from Equation (9):

$$vol_{induced,air} = vol_{induced,tot} \frac{(100 - EGRcontent_{intake})}{100} \quad \left[\frac{m^3}{s}\right] \tag{9}$$

By determining the volume of fresh air induced into the cylinder each stroke, the density of the air required to deliver the mass of air can be calculated from Equation (10):

$$\rho_{req} = \frac{m_{air}}{vol_{induced,air}} \quad \left[\frac{kg}{m^3}\right] \tag{10}$$

The increase in density can be achieved by regulating the air pressure. This will then give the required charge pressure by applying the ideal gas law, presented in Equation (11):

$$pV = nRT \tag{11}$$

where p is the pressure, V is the volume, n is the number of moles, R is the ideal gas constant and T is the temperature of the gas. This equation applies to the air in the intake tank as well as air at atmospheric conditions. Based on this, the pressure raise that is needed to raise the density to a certain value can be calculated as follows.

Since R is the ideal gas and hence the same both before and after the compression. Also the mass is constant and hence the number of moles is constant. These statements, Equation (10) and (11) gives an expression for the charge pressure presented in Equation (12):

$$p_{charge} = \frac{p_{amb}\rho_{req}T_{int}}{\rho_{amb}T_{amb}} \qquad [kPa] \tag{12}$$

This is also what happens in a throttle in a CI-engine, as the pressure drops over the partly closed throttle, but in that case the pressure is decreased below atmospheric pressure to decrease the air density.

3.2.3 Compressor

When a compressor is created the measurement follows SAE standard, SAE International (1995) and SAE International (2011), which means that all compressor maps are measured in a standardized way.

An example of a compressor map can be seen in Figure 3 and this figure is supplied from the turbocharger manufacturer together with the measured data.

The drawback with the SAE format is that the standardized corrections are different on the turbine side compared to the compressor side and therefore this is explained further in this and the next chapter.

The available SAE data in a compressor map are corrected speed, corrected mass flow, pressure ratio and efficiency which are all defined below in Equations (13) to (16).

$$Mf_{comp_{corr}} = \frac{Mf_{comp}\sqrt{\frac{\mathrm{T_{comp_{in}}}}{298}}}{p_{comp_{in}}/100} \qquad \left[\frac{kg}{s}\right] \tag{13}$$

Equation (13) shows how the compressor mass flow is dimensionless corrected to the inlet conditions of the compressor. This is done to be able to use the same compressor map in varying compressor inlet conditions. As can be seen in equation (13) the turbocharger manufacturer has measured the compressor flow at an inlet temperature of 298 K and an inlet pressure of 100 kPa according to SAE standard.

$$PR_{comp} = \frac{P_{comp_{out}}}{P_{comp_{in}}} \qquad [-] \qquad (14)$$

Equation (14) simply describes the pressure ratio, corresponding to the pressure difference created by the compressor.

$$N_{comp_{corr}} = \frac{N_{comp}}{\sqrt{\frac{T_{comp_{in}}}{298}}} \qquad [rpm] \qquad (15)$$

The compressor speed calculated in equation (15) is corrected in the same way as compressor mass flow to be able to use one compressor map at varying compressor inlet conditions.

$$Eff_{comp} = \frac{H_{isentropic_{rise}}}{H_{actual}} \qquad [-] \qquad (16)$$

The efficiency of the compressor is calculated, as in Equation (16), by the isentropic enthalpy rise over the compressor stage divided by the actual enthalpy rise over the compressor stage. The definition of an isentropic efficiency of a compressor is, "*the ratio of the work input required to raise the pressure of a gas to a specified value in an isentropic manner to the actual work input*" Cengel & Boles (2007).

Since the SAE data provided from a turbocharger manufacturer could be represented in different ways, but still with the same content, this master thesis uses the extrapolated data generated from a GT-Power simulation instead. The extrapolated data is based on the SAE data but it is always a text file with 60 extrapolated compressor speeds and 60 extrapolated compressor ratios and this means that there will be a 60 by 60 matrix of compressor mass flow and compressor efficiency. The big advantage with having the data presented in the same format each time is that it makes it easier for the user to switch between turbocharger maps. The format of the extrapolated files is explained further in Section 3.2.5.

3.2.4 Turbine

In the same way as a compressor the turbine is measured in a standardized way following SAE International (1995) and SAE International (2011) an example of a turbine map can be seen in Figure 5. The figure is supplied from the turbocharger manufacturer together with the measured data.

The available SAE data in a turbine map are expansion ratio, turbine flow parameter, turbine speed parameter and efficiency and they are all defined below in Equations (17) to (20).

$$ER_{turb} = \frac{P_{turb_{in}}}{P_{turb_{out}}} \qquad [-] \qquad (17)$$

Equation (17) shows how the expansion ratio of the turbine is calculated.

$$TFP = \frac{Mf_{turb}\sqrt{T_{turb_{in}}}}{P_{turb_{in}}} \qquad \left[\frac{\left(\frac{kg}{s}\sqrt{K}\right)}{kPa}\right] \qquad (18)$$

The turbine flow parameter is calculated with equation (18). Compared to the compressor mass flow the TFP is corrected in a different way which results in a non dimensionless correction. The TFP is corrected to the inlet temperature and pressure of the turbine, which means the temperature and pressure of the exhausts.

$$TSP = \frac{\mathrm{N}_{turb}}{\sqrt{T_{turb_{in}}}} \qquad \left[\frac{rpm}{\sqrt{K}}\right] \qquad (19)$$

The turbine speed parameter showed in equation (19) is also corrected with a non dimensionless correction in the same way as the TFP. The correction of both TFP and TSP is done to be able to use the same turbine map with different inlet temperature and pressure of the turbine.

$$Eff_{turb} = \frac{H_{actual}}{H_{isentropic_{drop}}} \qquad [-] \qquad (20)$$

The efficiency calculated with equation (20) is a combined efficiency with both the efficiency of the turbine and the mechanically efficiency included. It is calculated by the actual enthalpy rise over the turbine stage divided by the isentropic enthalpy drop over the turbine stage. According to Cengel & Boles (2007) the definition of isentropic efficiency is, "*the ratio of actual work output of the turbine to the work output that would be achieved if the process between the inlet state and the exit pressure where isentropic*"

With the same reason as explained in Section 3.2.3 the turbine data used in the model are the extrapolated data generated from GT-Power.

3.2.5 The extrapolated data format

The simulated turbocharger is, as said before, based on the SAE data but the data used in the Simulink is the extrapolated SAE data generated from GT-power. This chapter describes these files and the syntax of them in more detail.

The extrapolated data generated from GT-Power is a text file with the file extension cmp if it is a compressor map and trb if it is a turbine map and an example of a cmp file can be seen in Appendix A. Since the cmp-file only is a text file it is possible to load it into Matlab and a Matlab script was developed to be able to convert the text string into useable data in the Simulink model.

The extrapolated data is always normalized against the maximum value, to be able to extrapolate between 0 and 1, which means that the cmp file also includes maximum speed of turbocharger, maximum mass flow and maximum pressure ratio.

The first lines of the text file include essential data about the specific map, such as the maximum values, and the ones used in the developed Simulink model are explained below. The parameters that are not explained but can be seen in Appendix A are options that can be selected in GT-power when the cmp-file is created but since they are not used in this master thesis they are not explained any further.

The maximum speed of the compressor is presented in the cmp file as maximum reduced speed (MXRSP, variable name presented in brackets are the variable name in the cmp file). But to be able to use the SAE standard compressor format this is recalculated to maximum corrected speed which can be seen in equation (21).

$$RPM_{max-corrected} = RPM_{max-reduced} \cdot \sqrt{T_{ref}} \qquad [rpm] \qquad (21)$$

Since the extrapolated cmp file is normalized equation (22) is used to calculate the actual corrected compressor speed.

$$RPM_{corrected} = RPM_{normalized} \cdot RPM_{max-corrected} \qquad [rpm] \qquad (22)$$

The maximum mass flow rate of the compressor is presented in the cmp file as maximum reduced volumetric flow rate (MXRVOL) which is recalculated in equation (23) to fit the SAE standard format for compressors, similar to the maximum speed of the compressor showed in equation (21).

$$\dot{m}_{max-corrected} = \frac{\dot{V}_{max-reduced} \cdot P_{ref}}{\sqrt{T_{ref}} \cdot R_{ref}} \qquad \left[\frac{kg}{s}\right] \qquad (23)$$

Also the mass flow in the cmp file is normalized and equation (24) is used to calculate the actual compressor mass flow.

$$\dot{m}_{corrected} = \dot{V}_{normalized} \cdot \dot{m}_{max-corrected} \qquad \left[\frac{kg}{s}\right] \qquad (24)$$

The maximum compressor pressure ratio (PMAX) and the minimum compressor pressure ratio (PMIN) in the cmp file is used to convert the normalized pressure ratio, see equation (25), to actual compressor pressure ratio.

$$PR = PR_{normalized} \cdot (PR_{stall} - PR_{min}) + PR_{min} \qquad [-] \qquad (25)$$

The text file syntax between a cmp and a trb file is almost the same. There are some minor differences in the first lines with some different options to choose in GT-Power in a trb file. But since this is not used in this master thesis it will not be explained any further.

The differences worth describing are how the format in a trb file is converted to fit the SAE format of turbine maps.

The maximum reduced speed in a trb file, see equation (21), is the same as turbine speed parameter as it is called in the SAE format. This means that there is no need to convert this as in the case of a compressor map.

The maximum reduced volume flow rate needs to be recalculated to fit the SAE format with turbine flow parameter and this is shown in equation (26).

$$TFP = \frac{\dot{V}_{max-reduced}}{R_{ref}} = \frac{\dot{m}_{max-corrected} \cdot \sqrt{T_{ref}}}{P_{ref}} \qquad \left[\frac{kg}{s}\right] \tag{26}$$

3.2.6 Charge Air Cooler Pressure Drop

To predict the pressure drop over a Charge Air Cooler (CAC) in the system, data have been measured on a multi-cylinder engine. The pressure was measured before and after the CAC at different engine speeds and charge pressures. Figure 12 below shows a plot of the different pressure drops. Each star in the figure represents a pressure drop and at each speed the pressure drop increases with higher charge pressure.

How this data will be used is explained in Section 3.3.5.

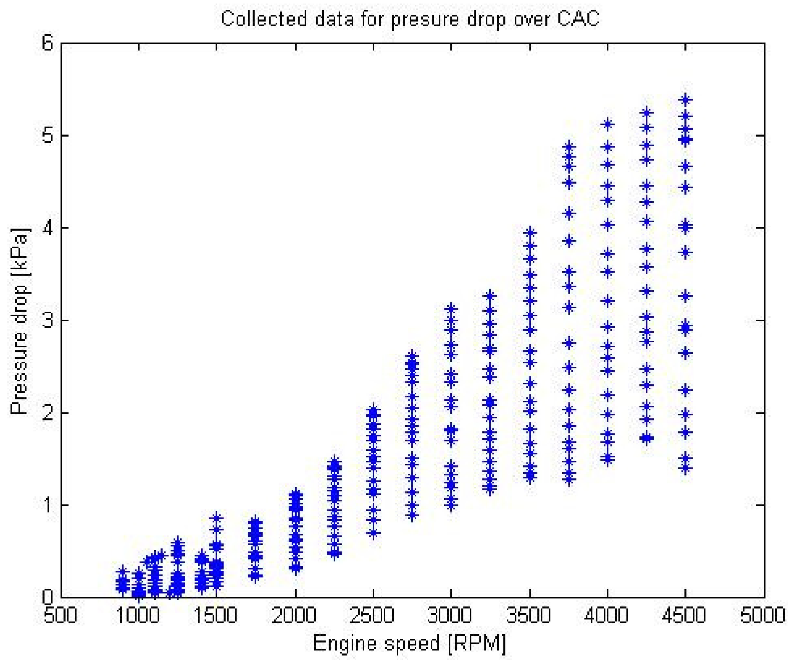

Figure 12 Pressure drop over CAC as a function of engine speed.

3.3 Simulation Models

This section describes how the model strategies from Section 3.2, Modulation Strategy, are incorporated in Matlab/Simulink.

3.3.1 EGR-content Model

The EGR-content in the intake tank is calculated with the model in Figure 13, using equation (5).

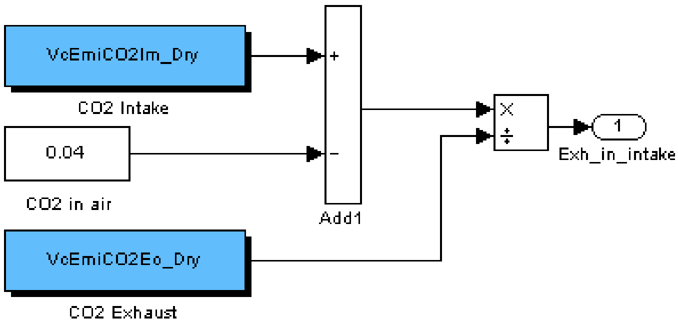

Figure 13 EGR Simulink sub model.

3.3.2 Required Charge Pressure Model

In Figure 14 the Simulink model used to calculate the required charge pressure is presented (a larger version is shown in Appendix B). This model is built by the strategy and equations presented in Section 3.2.2 resulting in an output of a required charge pressure.

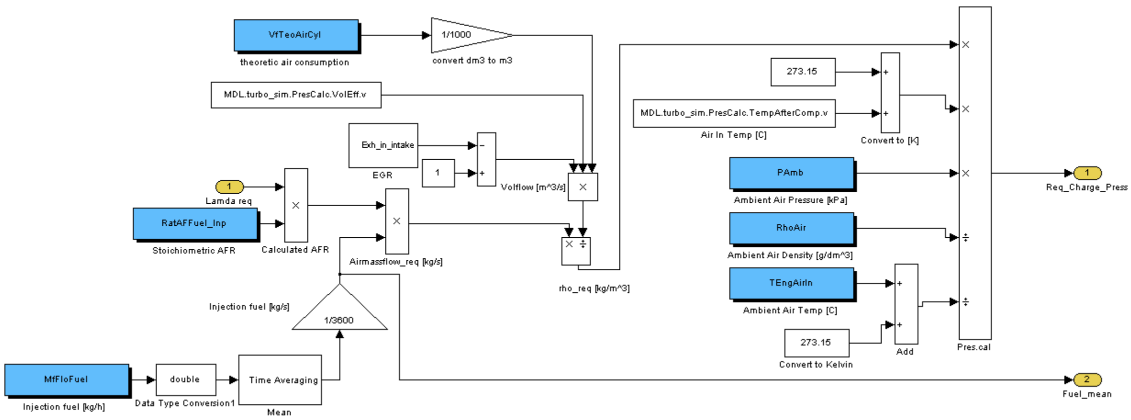

Figure 14 Required charge pressure model.

3.3.3 Compressor Model

In Figure 15 the Simulink model used to handle the data described in Section 3.2.3 is presented. The input variables EFFI_comp (matrix), Mass_comp (matrix), PR_comp (vector) and RPM_comp (vector) are loaded from the cmp-file explained in Section 3.2.5. The variable PR_search is a pressure ratio calculated from the required charge pressure model (see previous section) and atmospheric pressure. The variable mass_search is the mass flow through the compressor. This variable is based on the measurement of the mass of air flowing into the cylinder, which then is recalculated to a corrected air mass flow as described in Equation (13).

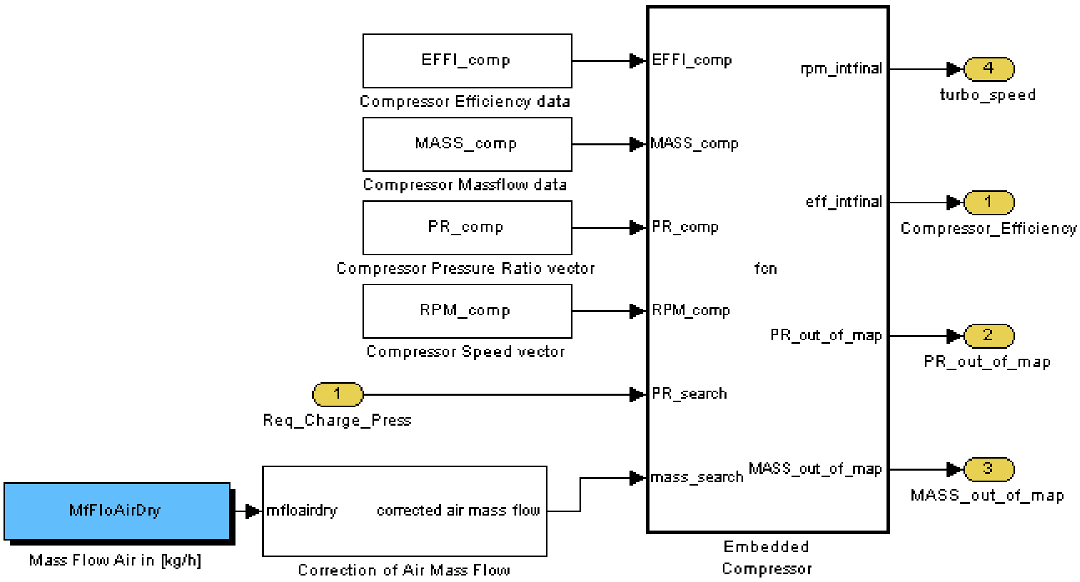

Figure 15 Compressor model

The embedded compressor function contains a Matlab script that based on the requested pressure ratio, PR_search, finds the two pressures closest to the requested pressure ratio in the PR_comp vector.

Figure 16 shows the principle of how the Matlab script works. The two values (rectangles) closest to PR_search(circle) are found. Each of these pressure ratios corresponds to a number of mass flows, depending on the RPM. To find mass_search an interpolation is carried out between the closest values on each pressure ratio, after which two points are found (the crosses) which each correspond to a speed. By interpolating these speeds the final speed for the compressor is found called turbo_speed as output variable.

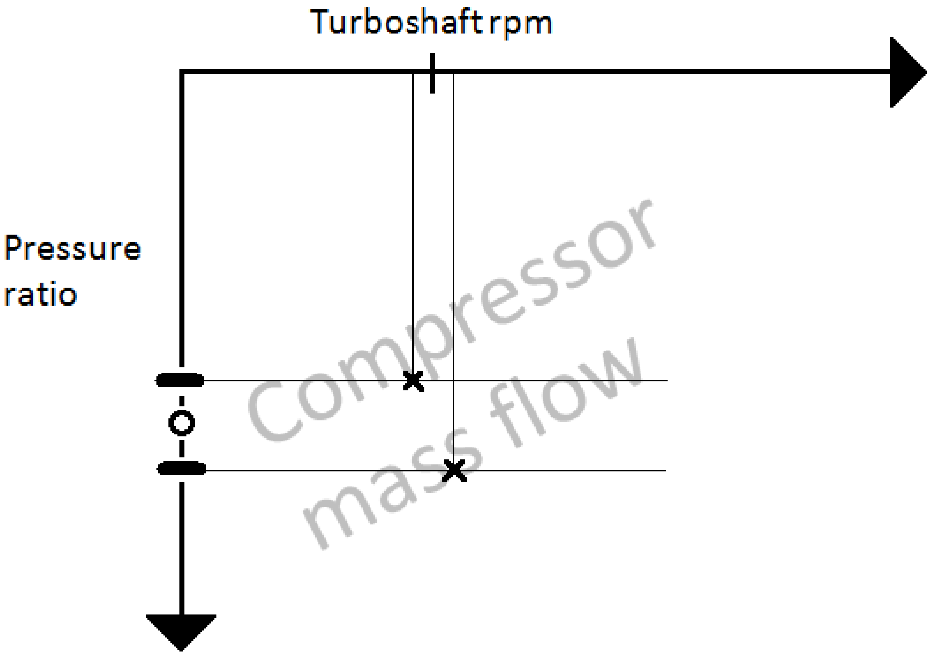

Figure 16 Principle behind embedded compressor function script.

In a similar way the compressor efficiency is found but instead of compressor mass flow the matrix is changed to compressor efficiency and the output variable is now called Compressor_Efficiency.

The two output variables PR_out_of_map and MASS_out_of_map are used to display if the input values on pressure ratio and mass flow are too large to be found in the matrices. If this is the case the compressor of choice is not suitable for the current driving condition.

3.3.4 Turbine Model

In Figure 17 the Simulink model used to handle the data described in Section 3.2.4 is presented. The input variables RPM_turb (vector), MASS_turb_TFP (matrix), EFFI_turb (matrix) and PR_turb (vector) are loaded from the trb-file explained in Section 3. The input variable rpm_intfinal is calculated from the output variable turbo_speed from the Compressor model in Section 3.2.3 through equation (15). P_exh is the backpressure in the exhaust system and is a user defined variable. T_exh is the temperature of the exhaust gases, measured in the engine rig and an assumption has been made that this temperature is roughly the same temperature as upstream the tubine. Mf_exh is the exhaust mass flow.

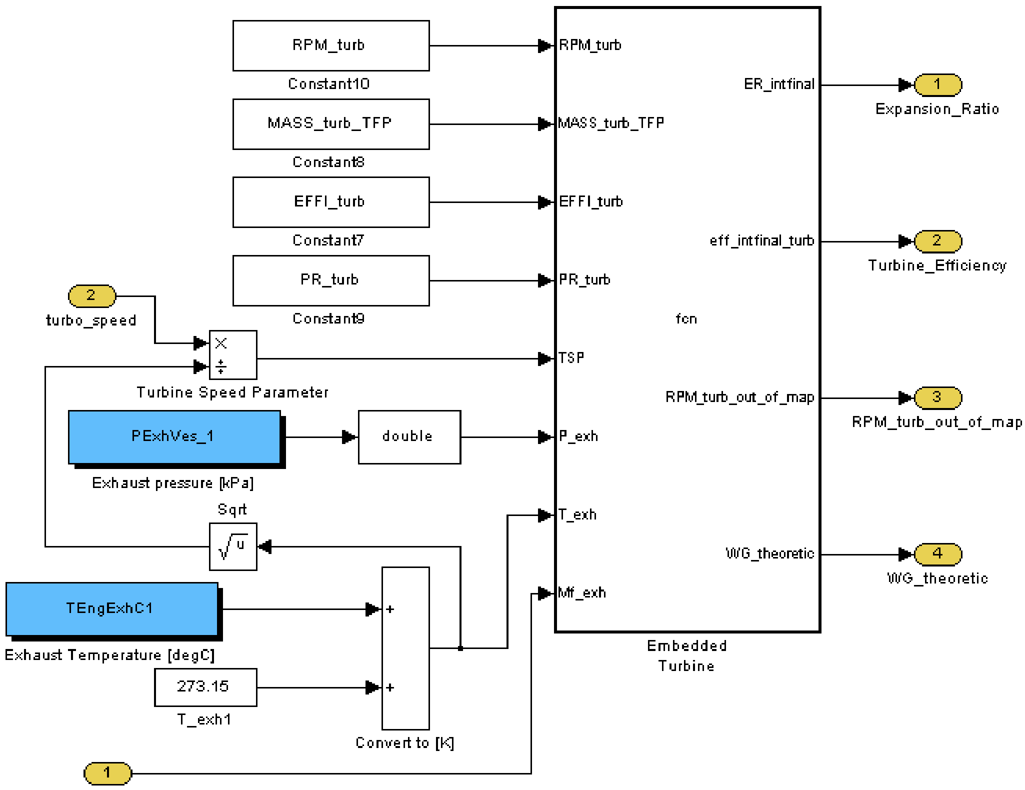

Figure 17 Turbine model.

The embedded turbine function contains a Matlab script that in a similar way as in the embedded compressor function interpolates to find the expansion ratio over the turbine and the turbine efficiency.

The output variable RPM_turb_out_of_map displays if the input variable rpm_intfinal has a too high value to be found in the RPM_turb vector. If so is the case the driving conditions is not suitable for the turbine of choice. If the exhaust flow is too high for the turbine some of the flow will be routed around the turbine by the wastegate. If this occurs the output variable WG_theoretic will display how much flow that needs to be routed around the turbine.

3.3.5 Charge Air Cooler Model

The sample data in Figure 12 is loaded in the initiation function of the Simulink model to be able to use in the Simulink model.

To estimate the pressure drop at a certain driving condition the current RPM and charge pressure of the engine is measured. These values are then fed into an embedded Matlab function, which can be seen in Figure 18, together with the sampled data from the initiation file. The pressure drop corresponding to the measured values of RPM and charge pressure can now be interpolated from the sampled data by finding the RPM and charge pressure sampled points closest to the current driving condition.

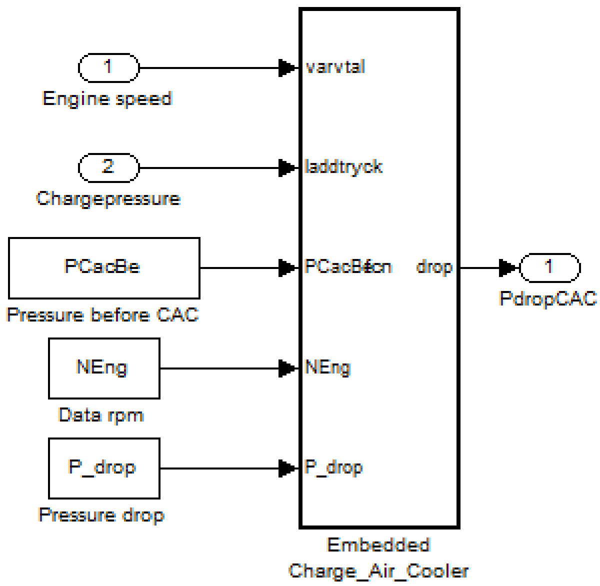

Figure 18 Charge Air Cooler Simulink sub model.

4 Results

The result chapter is divided into three parts. The first part shows the final Simulink model of the turbocharger simulation, the second part shows the results from offline model verification and the third part shows the result from running the model online in the engine test cell.

4.1 The Simulink Model

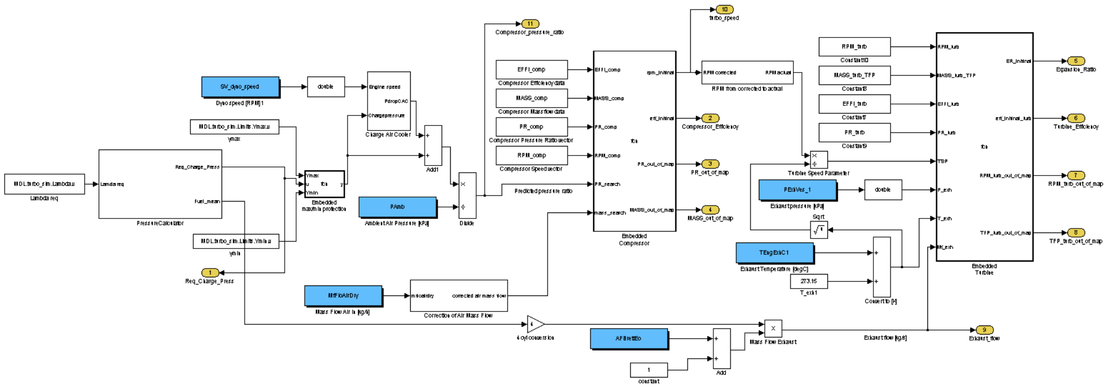

Figure 19 Top view of the complete Simulink model.

Combining the models described in Section 3.3 results in a model of the complete turbo charging system. An overview of this system can be seen in

Figure 19 below (a larger figure can be seen in Appendix C). The modulation flow of this model is illustrated in Figure 20. The figure is meant to give an overview of in which order the models described in Chapter 3 is used.

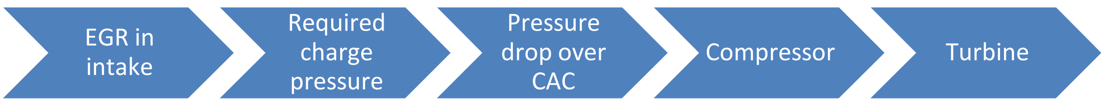

Figure 20 Modulation flow

4.2 Offline Model Verification

To be able to validate each part of the model without influence of possible errors from parts of the model each part was tested alone in offline simulations. The offline simulations were carried out by feeding the model with data from a simulation of a single turbo multi-cylinder engine in GT-Power. The reason to choose GT-Power simulated data as reference data is due to the fact that the physical engine test setup, with a single turbo system, was not available for testing during the verification of this model. Therefore the comparison against real engine data had to be made against logged data from previous testing, which means that the only parameters available are the ones that were logged during the tests. Since the tests that were carried out with the real engine were missing some of the parameters needed to verify the Simulink model, data from the corresponding engine simulated in GT-Power was used. Though, the GT-Power simulation of the engine has been validated against the testing of the real engine.

4.2.1 Required Charge Pressure

The charge pressure prediction was tested in Simulink by simulations of only this part of the model. This means that the sub model was fed with lambda and amount of fuel and then the charge pressure is estimated. Figure 21 below shows a comparison of GT-Power simulated charge pressure and Simulink model prediction of charge pressure. Y-axis is the normalized required charge pressure and x-axis is the test number. An increasing test number corresponds to an increasing engine speed at full load.

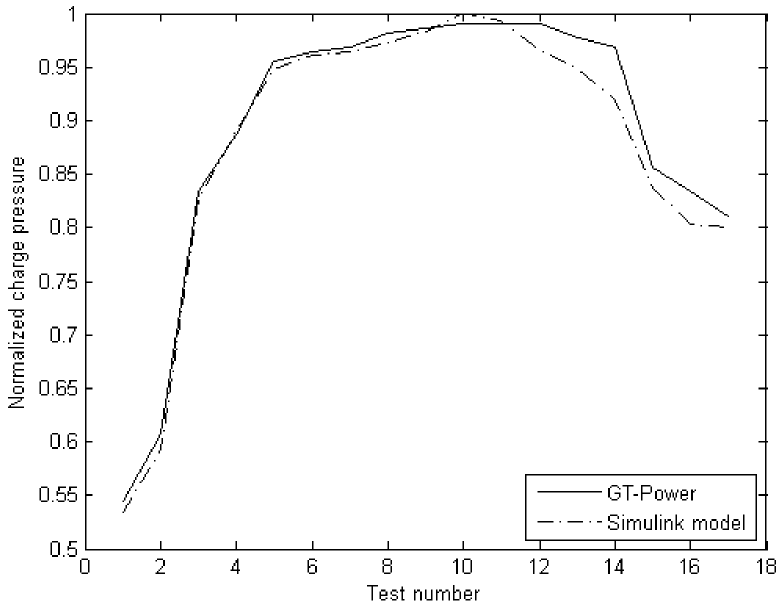

Figure 21 Comparison of charge pressure results.

The charge pressure prediction in Simulink conforms to the resulting charge pressure from the GT-Power model. There are some deviations at high test number, which corresponds to high engine speed, but they are small.

4.2.2 Compressor Model

The compressor model is verified individually to ensure that the sub model works as supposed. This was done by setting the pressure ratio and intake air mass flow as input signal to the model and the compressor (turbocharger) speed as the output. The input signal was the measured data from GT-Power simulations and not the output from the charge pressure prediction model. This was done to eliminate error sources when testing a sub model.

Figure 22 shows the result from this verification where the y-axis is the normalized compressor speed and the x-axis is the number of test.

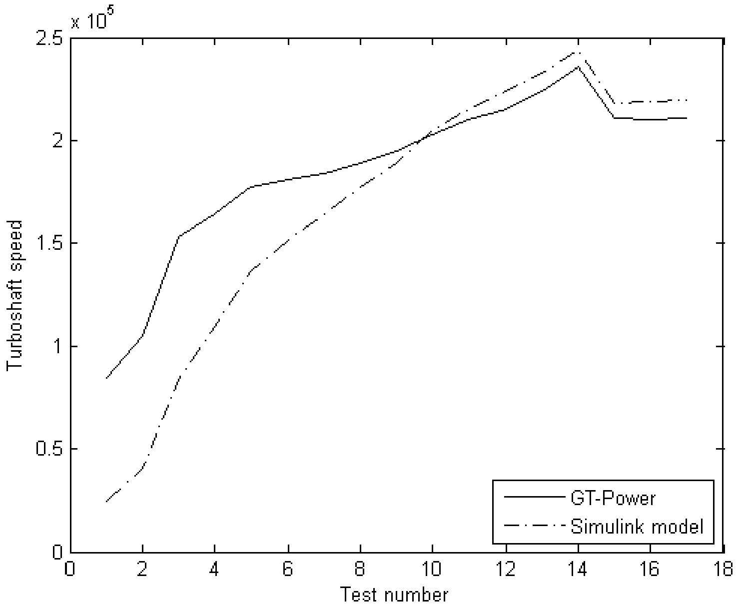

Figure 22 Normalized turbocharger speed results.

As can be seen in Figure 22 the Simulink results clearly follow the same behavior as the results from the GT-Power simulation, although there are some differences.

4.2.3 Turbine Model

The turbine model was validated in the same way as the compressor model but the turbine model was instead fed with the turbocharger speed and exhaust mass flow as input signals and the expansion ratio over the turbine as the resulting output signal. Also in this validation the simulated GT-Power results was used as reference data and the comparison between GT-Power and Simulink model results can be seen in Figure 23.

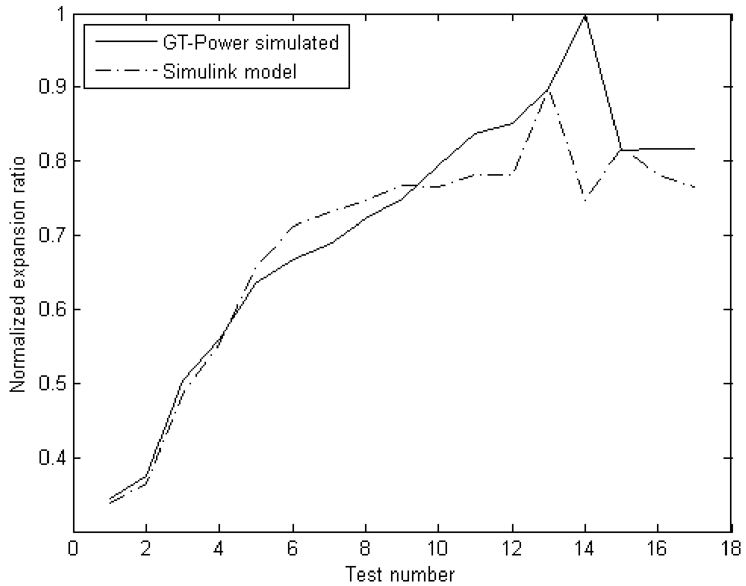

Figure 23 Comparison of turbine expansion ratio results.

The behavior of the Simulink model results follows the results from GT-Power in a similar way as the compressor (turbocharger) speed.

4.2.4 Complete Model

The complete Simulink model was also tested offline in the Matlab/Simulink environment with the GT-Power simulated data as input. Figure 24 shows the normalized charge pressure from the complete model simulation.

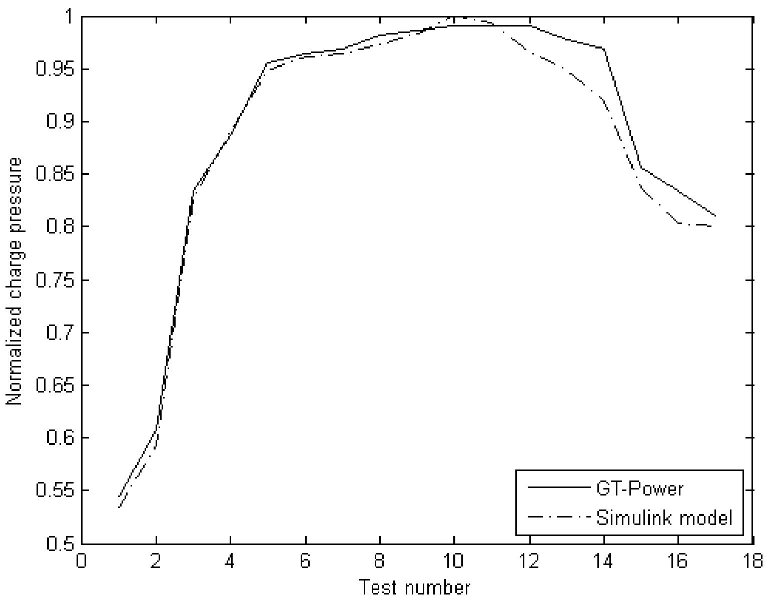

Figure 24 Charge pressure prediction results.

The difference between the results in Figure 25 and Figure 23 depends on the fact that in the complete model test the turbine model input signal is the compressor model output signal which is the turbocharger speed. And since there is a deviation between Simulink model turbocharger speed and the actual speed (data from GT-Power) there will be a slight difference between the two figures.

On the other hand the complete Simulink model results match the one from GT-Power up to a certain test number, which means up to a certain speed at full load. But the shape of the two curves is very similar.

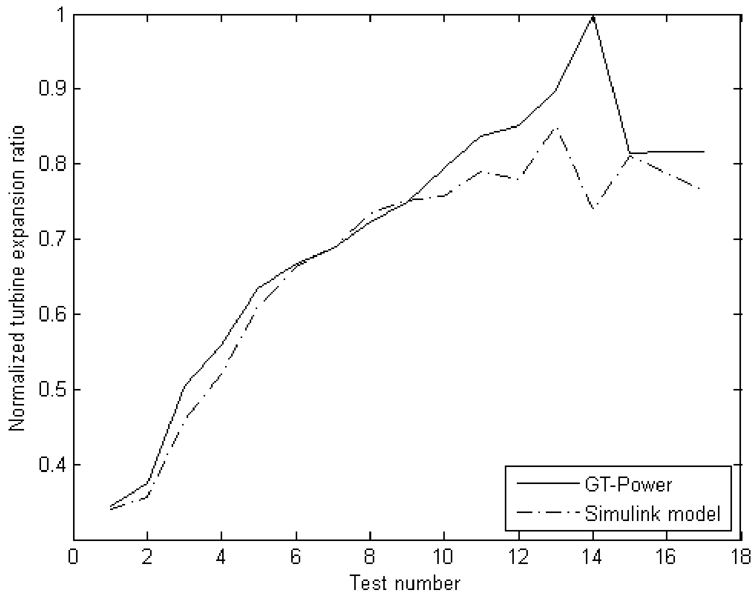

Figure 25 Expansion ratio complete model results.

Figure 25 shows the expansion ratio result from the complete model simulation compared to the reference data and Figure 26 shows the turbo shaft speed from the complete model simulation.

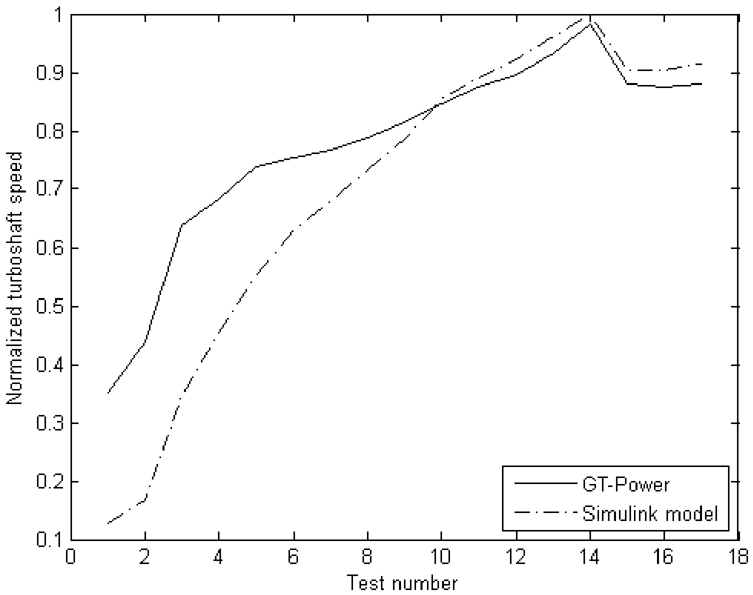

Figure 26 Turbo shaft speed complete model results.

4.3 Online Model Verification

The resulting Simulink model presented in Figure 19 was also tested online in the engine test cell. The variables that in the offline verification were data from the GT-Power simulation are in the online verification switched to the physical engine measurements.

By varying the engine speed and engine load the Simulink model was verified in a large area of engine operation points. A sweep of engine speed where made both with constant load and with varying load.

To verify the model at as many possible engine operations points as possible test runs with EGR was executed.

All tests show that the charge pressure prediction works properly with only small deviations from the actual charge pressure. The complete Simulink model also works in the online tests but it is hard to conclude if the results are correct or not, since the physical engine was not equipped with a turbocharger and this was, as said before, the reason to the previously done offline tests.

5 Discussion

This discussion will be built up in the same order as the simulations are conducted in the model. First each part will be discussed individually and at the end how they function together will be discussed.

The required charge pressure model presented in Section 3.2.2 and Section 3.3.2 shows good results in both offline and online testing, with and without EGR, although some deviations are seen at higher engine speeds. A possible reason for this could be the assumption made in Section 3.2.2, that 85 per cent of the volume in the cylinder is exchanged each cycle. In reality this figure is not constant but affected by a number of parameters such as engine speed, ratio of exhaust to inlet manifold pressures, port design etc. (Heywood, 1988). To improve the charge pressure model this figure could have been exchanged to some form of look-up table to better fit different driving conditions. Also the temperature in the inlet is assumed to be constant, which may affect the results. A similar solution with a look-up table could be used in this case. To validate how much these two parameters affect the model output an analysis could be made where the values are varied and the output recorded.

In the compressor model the inlet pressure is assumed to be the same as atmospheric pressure, which means that all pressure losses over the intake system, such as air filter, piping, etc are neglected since they are assumed to be relatively small and due to that the intake system of the one-cylinder engine is not comparable to the intake system of a multi cylinder engine. Also the interpolation script may have some improvement potential to get a more accurate result. One way to improve could be by analyzing the data handling and setting up more conditions for how to interpolate. Although the results from both online and offline simulations shows good potential and predicts the characteristics of the compressor well.

The turbine model predicts the expansion ratio over the turbine with results that has the same behaviour as the reference data. As in the case with the compressor speed the expansion ratio results captures the characteristics of the reference data results. Although there is some deviations and the reason for this could again be the data handling. This could be investigated more by, for example, letting the input compressor speed be the reference compressor speed and then tune the exhaust mass flow to get the correct expansion ratio and vice versa. This could give more information of which of the parameters that affects the results the most.

Also the complete Simulink model test confirms the sub model testing. Since the required charge pressure test procedure is the same for the sub model as for the complete model the results are the same because the input signals are the same and hence the input data for the compressor model will have a small counted error. Although since these errors are that small, the result from the individual simulation (Figure 22) of the compressor and the compressor result in the complete simulation (Figure 26) looks almost the same. But still there is a difference between the compressor speed results from the Simulink model compared to the compressor speed results from GT-Power. Although this difference in compressor speed does not reflect the difference in the resulting expansion ratios from the turbine model (Figure 23 shows the results with GT-Power compressor speed and Figure 25 shows the results with the Simulink model compressor speed). This could mean that the data used is not that sensitive to the compressor speed and some more investigation should be done to determine if this is true.

A problem during the verification of the models has been that the data available from real engine tests and GT-Power simulations are as previously mentioned limited. The available data has been full load sweeps over the engine speed span, which means that the models only have been verified at full load and therefore needs verification in different driving conditions.

6 Conclusion

One part of the purpose of this master thesis was to design tailored models using Simulink in order to determine the boundary conditions for the turbo system of a single cylinder engine and to feed set point values to the engine control. This has been achieved with satisfying results with model for required charge pressure which has proven to work in offline as well as online simulations.

The model should also be able to evaluate whether or not a specific turbo system was suitable for the engine at different driving conditions. This has been done by modelling the compressor and turbine of a turbocharger from performance data measured on the turbocharger by the manufacturer. It is possible for the user to change which turbocharger that should be examined by changing the input data to the model and hence testing different systems. This model feature has given positive results by capturing the characteristics of the turbocharger in simulations, although improvements are needed to increase the reliability of the results.

The model has been implemented in VCC's single cylinder automation system and is verified to function together with the rest of the automation system.

7 Future work

In this chapter some suggestions for future work in the area will be presented.

- As mentioned in discussion a further improvement would be to validate the model in other conditions than full load, which would require a real engine installed in a dyno or further simulations in GT-Power to produce data to compare against.
- The interpolation strategy for the compressor and turbine datasets could be evaluated further and developed to give a higher accuracy.
- To make the model more useful for future engine development, the scope could be widened to include also multi-stage turbo systems, variable geometry turbo systems, mechanically driven compressors and different combinations of these.
- Additional sub models can be added to include parameters that has been neglected or assumed to be constant, such as pressure drop in the intake system and temperatures in the intake system.
- When considering improvements to the model it could also evaluated what improvements and drawbacks there could be with an integration of the engine simulation software GT-Power in the PUMA system.

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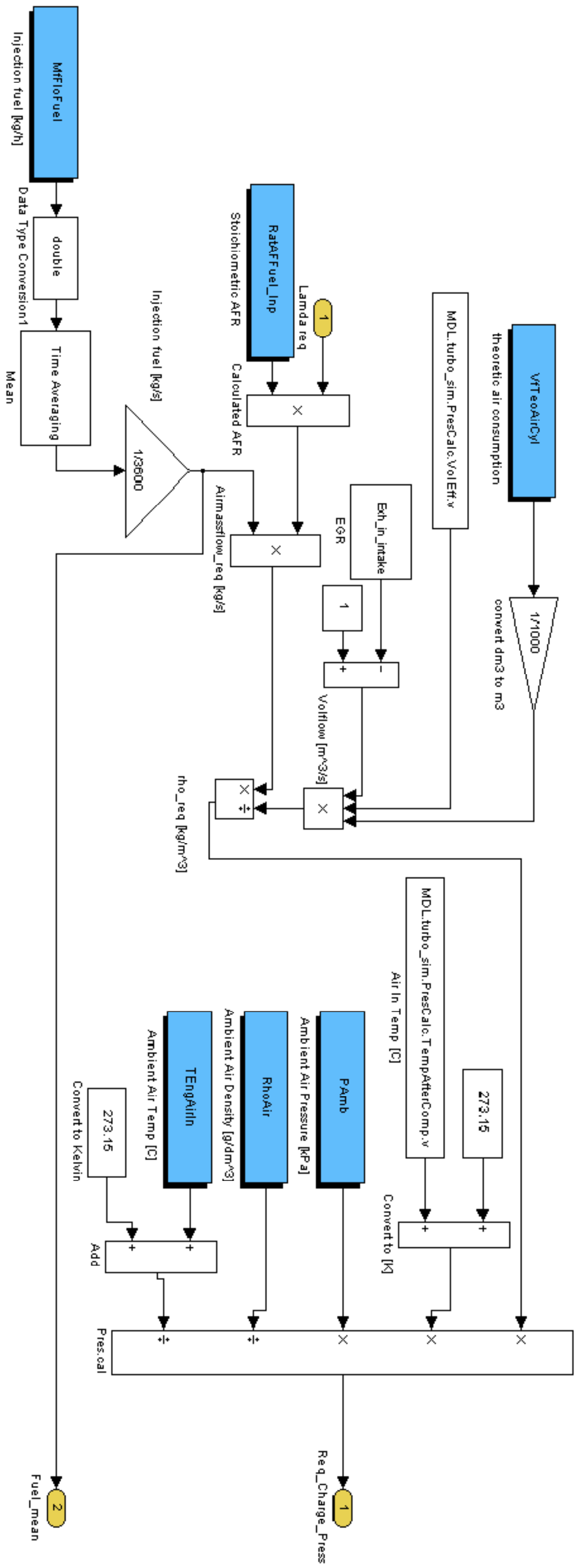
Appendices

Appendix A Example of cmp-file

```
FORMAT> v7.2
TYPE> compressor
NRPM> 6
NPR> 12
MXRSP> 5908.701
MXRVOL> 0.3220356E-01
DIAW> 0.9032080E-01
DROOP> no
PRMIN> yes
SUPER> no
SURGE> no
PRLT1> no
DAENV> yes
SH-RATIO> 1.4
GAS-CONSTANT> 287.00
PWRMECH> ign
PREF> 100000.0
TREF> 298.0000
COMPTYP> radial
DATA>
ARPM>
.00000 .52531 .69314 .81519 .91461 1.00000
<
APPR>
.00000 .38322 .50566 .59469 .66722 .72951
.78470 .83461 .88040 .92287 .96259 1.00000
<
PMAX>
1.0200 1.7236 2.4169 3.1347 3.8733 4.14140
<
PMIN> {optional}
1.0000 1.0000 .99985 .99753 .96243 .934875
<
MASS>
.00210 .68699 .87547 .95187 .97609 1.0000
.00186 .63206 .84225 .94195 .97481 .99886
.00179 .61450 .83163 .93878 .97440 .99849
.00173 .59165 .82018 .93586 .97403 .99823
.00169 .57086 .80761 .93211 .97354 .99793
.00165 .55300 .79681 .92793 .97275 .99752
.00162 .52671 .77607 .91764 .97056 .99664
.00159 .49935 .75098 .90767 .96605 .99556
.00156 .47135 .72406 .88628 .95595 .99348
.00153 .42926 .68232 .86081 .94065 .99156
.00151 .36600 .62472 .81849 .91663 .98322
.00149 .24085 .45141 .68427 .83809 .97505
<
EFFI>
.20000 .20000 .20000 .20000 .20000 .20000
.20000 .41476 .41726 .41193 .40750 .37978
.20000 .48338 .48668 .47965 .47379 .43722
.20000 .54475 .54113 .53122 .52398 .47899
.20000 .59717 .58933 .57876 .56994 .51265
.20000 .64218 .63074 .61947 .60649 .54114
.20000 .67521 .66550 .65475 .63784 .56747
.20000 .70320 .69460 .68657 .66310 .59086
.20000 .72617 .71932 .70998 .68382 .61072
.20000 .73496 .73527 .72894 .70084 .62914
.20000 .73099 .74107 .73827 .71318 .64005
.20000 .65508 .69492 .72437 .71458 .65000
<
```

Appendix B Required Charge Pressure Model



Appendix C Complete Simulink Model