Mean value modelling of a poppet valve EGR-system

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

by
Claes Ericson

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Titel Medelvärdesmodellering av EGR-system med tallriksventil

Title Mean value modelling of a poppet valve EGR-system

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Sammanfattning
Abstract

Because of new emission and on board diagnostics legislations, heavy truck manufacturers are facing new challenges when it comes to improving the engines and the control software. Accurate and real time executable engine models are essential in this work. One successful way of lowering the NOx emissions is to use Exhaust Gas Recirculation (EGR). The objective of this thesis is to create a mean value model for Scania's next generation EGR system consisting of a poppet valve and a two stage cooler. The model will be used to extend an existing mean value engine model. Two models of different complexity for the EGR system have been validated with sufficient accuracy. Validation was performed during static test bed conditions. The resulting flow models have mean relative errors of 5.0% and 9.1% respectively. The temperature model suggested has a mean relative error of 0.77%.

Nyckelord
Keywords

mean value engine modelling, EGR, poppet valve

Abstract

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Outline

Chapter 1 describes the background and the objectives of the thesis.

Chapter 2 gives a description of the measurement setup and related problems and also has a brief section on the working process.

Chapter 3 presents the existing model and modifications. The new EGR models are also introduced.

Chapter 4 describes the parameter tuning process.

Chapter 5 evaluates the EGR models using static measurements.

Chapter 6 discusses the results presented and possible future work.

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Claes Ericson
Södertälje, June 2004

Contents

Chapter 1

Introduction

This master's thesis was performed at Scania CV AB in Södertälje. Scania is a worldwide manufacturer of heavy duty trucks, buses and engines for marine and industrial use. The work was carried out at the engine software development department, which is responsible for the engine control and the on board diagnostics (OBD) software.

1.1 Background

Because of new emissions legislation, both within the European Union and the United States, heavy truck manufacturers are facing new challenges when it comes to improving the engines and the control software. Besides the requirements of substantially lowered emissions, new legislation such as Euro 4 and 5 also requires advanced On Board Diagnostics (OBD) systems. The OBD system has to meet certain demands, for example faults resulting in emission levels higher than the legislative limits must be detected.

In order to meet these goals, it is important to have models of the engine with sufficient accuracy. The models are used for improved model based control and for model based diagnosis.

One successful way of lowering the NOx emissions is to use Exhaust Gas Recirculation (EGR). This means that some of the exhaust gas is circulated back into the intake manifold. The amount of NOx emissions produced during the combustion is closely related to the peak temperature. By reducing the amount of fresh air in the intake manifold and replacing it with exhaust gas, a lower peak temperature during combustion is achieved resulting in decreased NOx. In order to be able to inject the exhaust gas into the intake manifold, it must have a sufficiently high pressure. This can be achieved in several ways, for example by using a venturi, turbo compound or as used in this thesis; a variable geometry turbocharger (VGT). By adjusting the vanes in the VGT a high exhaust pressure is achieved.

1.1.1 Existing Work

There are several types of engine models of different complexity, in this thesis the focus is on mean value engine models (MVEM:s). In such a model, all signals are mean values over one or several cycles. Although some fast dynamics are excluded, the performance is sufficient for the intended purposes. The big advantage with a MVEM is that the computational effort required is small compared to other types of engine models making real-time performance possible if care is taken to ensure low complexity.

At Scania David Elfvik [3] produced the first physical model. Jesper Ritzén [14] simplified the model and improved the real time performance, Manne Gustafson and Oscar Flärdh [5] extended the model with turbo compound.

1.2 Problem Formulation

Previous modelling work at Scania has been successful, providing low mean value errors while maintaining real time executability. The EGR-system has not been properly modelled so far though, earlier MVEM:s [13] show much higher errors when the current EGR submodels are added.

The current EGR-system used in Scania engines uses a pneumatically driven butterfly valve without any position sensor or position feedback. The problem with butterfly valves in general is that it's not possible to close the valve completely, thus 0% EGR is not achievable which is desired in certain situations, for example during transients. Also the lack of position feedback makes precise control of the amount of EGR difficult. One possible solution to both of these problems is to replace the butterfly valve with a poppet valve using an electric actuator. Because of the electric controller, position feedback is possible, and thanks to the mechanical properties of the valve, 0% EGR is achievable. The disadvantage with most poppet valves is that they cause a higher pressure drop than the butterfly valve. The poppet valve has to be modelled and added to the MVEM.

The EGR-cooler is also updated and needs to be added to the MVEM. On the test engine a two-stage cooler is used, first the usual shell and tube heat exchanger using the engine cooling water as a coolant, followed by an air-cooled cross-flow heat exchanger (basically a smaller version of the charge air cooler).

1.3 Objectives

The objectives of this thesis is to create a mean value model for the new EGR system consisting of a poppet valve and a two stage EGR-cooler. The model should be:

- Physical
- Accurate
- Modular

1.4 Delimitations

The exhaust brake that the engine is equipped with is not modeled. During calibration and validation, the exhaust brake has not been active.

1.5 Target Group

The target group of this work is primarily employees at Scania CV and M.Sc./B.Sc. students with basic knowledge in vehicular systems and thermodynamics.

Chapter 2

Method

In this chapter the working process will be described briefly, and the measurements will be described.

2.1 Working Process

Theoretical studies First of all earlier works such as masters theses, articles and books related to engine modelling were studied in addition to some thermodynamics and fluid mechanics.

Modelling Physically and/or empirically based models were developed and implemented in Matlab/Simulink.

Measurement Measurements were planned and performed in an engine test bed.

Tuning The parameters in the earlier developed models were tuned using the optimization software Lsoptim [4].

Validation Using a set of data separate from the one used during parameter setting measured outputs are compared with simulated outputs.

The modelling and parameter setting parts were reworked many times during the course of work after measurements had shown limitations in earlier versions of the models.

2.2 Measurements

All measurements were performed in an engine test bed. In the test bed configuration the engine is equipped with many temperature and pressure sensors in addition to the production sensors.

2.2.1 Measurement setup

Measurements were performed both at steady state and continuous conditions. Some of the sensors have very slow dynamics however, so care had to be taken in order not to misinterpret the resulting data. During steady state measurements there is a stabilization phase of 30 seconds and after that the software calculates a mean value of the variables for another 30 seconds. Continuous measurements were performed at a sampling rate of 10Hz. For data collection the standard measurement system of the test bed was used most of the time. In order to validate some of the measurements, the ATI Vision measurement system was used.

2.2.2 Measured quantities

Temperatures are measured using 5mm pt-100 temperature sensors. The 5mm diameter of the sensors makes them very ”bulletproof” and capable of withstanding high temperatures, but on the other hand makes them quite slow. However, during steady state measurements this is not an issue. When using ATI Vision, 3mm K-elements were used instead (because of the lack of inputs for pt-100 sensors). For pressure measurements, the standard sensors of the test bed were used. All pressure sensors are mounted perpendicular to the flow and consequently it is the static pressure that is measured.

The position of the poppet valve was measured using the built in position sensor of the Eaton valve. The output of this sensor is an analogue signal between 0 and 5 volts, directly proportional to the valve lift.

The position of the vanes in the VGT was measured using the built in position sensor of the Holset turbocharger. The output of this sensor is an analogue signal between 0 and 5 volts.

A flange manufactured by Holset was used to determine the air flow into the engine. The sensor is installed a couple of meters before the intake, which makes it unusable for dynamic measurements but ok for steady state.

The gas flow through the EGR system was determined using an EGR rate variable calculated in the test bed computer. The variable is calculated by the comparison of the CO_2 rate in the intake manifold with the CO_2 rate in the exhaust gas using a HORIBA exhaust gas analyzing system. The fact that the exhaust gases (which are led through pipes to the adjacent room where the analyzing system is located) are used to determine the EGR rate implies a delay of several seconds making it usable only in steady state.

Table 2.1: Measured variables

Variable	Description
p_{amb}	Ambient pressure [bar]
p_{im}	Inlet manifold pressure [bar]
p_{em1}	Exhaust manifold pressure 1 [bar]
p_{em2}	Exhaust manifold pressure 2 [bar]
p_{avalve}	Pressure after EGR valve / before EGR water cooler [bar]
p_{awater}	Pressure after EGR water cooler / before EGR air cooler [bar]
p_{aair}	Pressure after EGR air cooler / intercooler [bar]
T_{amb}	Ambient temperature [K]
T_{im}	Inlet manifold temperature [K]
T_{em1}	Exhaust manifold temperature 1 [K]
T_{em2}	Exhaust manifold temperature 2 [K]
T_{avalve}	Temperature after EGR valve / before EGR water cooler [K]
T_{awater}	Temperature after EGR water cooler / before EGR air cooler [K]
T_{aair}	Temperature after EGR air cooler / intercooler[K]
T_{cool}	Cooling water temperature [K]
n_{trb}	Turbine speed [rpm]
n_{eng}	Engine speed [rpm]
u_{egr}	EGR valve position [V]
u_{vgt}	VGT vane position [V]
M_{eng}	Engine torque [Nm]
δ	Injected fuel [mg/inj]
α	Fuel injection timing [deg]
x_{egr}	EGR rate [%]
W_{air}	Air mass flow [kg/s]

2.2.3 Measurement problems

In general it is difficult to measure temperatures and pressures in an engine due to the unfriendly environment. This section will describe some of the most important phenomena encountered while working on this thesis.

Temperatures

Measuring the temperature of the EGR gas before entering the intake manifold and interpreting the results turned out to be one of the largest challenges of this thesis. Because of the way the combined EGR air cooler / charge air cooler is constructed it is physically difficult to fit separate temperature sensors in order to separate the charge air temperature from the EGR temperature. Because of the high efficiency of the coolers one could assume that the two temperatures are similar though.

The real problem turned out to be the temperatures reported by the T_{aair} temperature sensor. With an ambient temperature of 298K, T_{aair} reported 290K during certain conditions. This is physically impossible since the cooler cannot cool the gases to a lower temperature than ambient. That would indicate an efficiency of more than 100%. The temperature sensors used were calibrated and found to be working properly.

There is a reasonable explanation for these low temperatures however. The exhaust gas contains large amounts of water which could cling onto the temperature sensor in the form of droplets, then evaporate while cooling down the temperature sensor, which in return reports a temperature unrepresentative of the gas temperature. The fact that a wet thermometer reports a lower temperature than a dry is used in for example psychrometers [2]. Comparing T_{aair} with the temperature in the intake manifold T_{im} shows that the temperature of the gas mixture increases by approximately 5 degrees on average, which is what we expect due to heat transfer from the engine block. A five degree increase still means that the temperature in the intake manifold is lower than ambient temperature in some points. The difference is slight during most temperature conditions though and the wet thermometer theory could still hold. One basic problem when measuring such low temperatures is that a difference of a few degrees results in physically impossible results. With the previous EGR systems this was never an issue because the temperatures were much higher, often in the 350-400K region where a two-degree difference is negligible. In order to bring clarity to the situation, another measurement system, ATI vision, with 3mm K-element temperature sensors in place of the pt-100:s was used. The sub-ambient temperatures were not observed with this setup. Possibly the effect of water droplets on the temperature sensor was smaller due to the smaller diameter. These measurements did confirm the first suspicion that the temperature sensor did not measure what was intended, rather than the theory that a new unexplained physical phenomena had been discovered. Using measurement data which could not be fully trusted did impose problems while modelling the EGR air cooler as further discussed in chapter 3.2.2.

Another issue when it came to measuring problems was T_{awater}, the temperature after the EGR water cooler. In several points during low EGR flows, this temperature was substantially lower than T_{cool}, once again indicating an efficiency of over 100%. There are two possible explanations for this phenomena. The temperature sensor could be cooled down because of heat transfer from surrounding pipe walls because of the extremely low flow, resulting in an output more representative of wall temperature than gas temperature. The other theory is that fresh air from the charge air cooler was leaking backwards through the EGR system under these operating conditions, meaning that T_{awater} was actually measuring the temperature of an exhaust gas / charge air mixture.

Pressures

One problem in general with the type of pressure sensors used and mean value type of measuring is that the sensors does not capture the high frequency pressure pulsations originating from the opening and closing of exhaust and inlet valves. Looking at older measurements performed at Scania, this is especially apparent in the exhaust manifold pressure.

The pressure sensor located after the poppet valve, p_{avalve}, is likely to give less than optimal results due to turbulence after the valve and the poor physical location. One indication of this is that the pressure reported from the sensor is lower than the pressure in the intake manifold in many operating conditions, thus indicating a pressure increase over the EGR coolers, which is physically impossible. Looking at p_{awater}, which is better located along a straight pipe, also shows a lower pressure than p_{im} in many points, leading us to believe that there is another explanation than measurement problems. The previously discussed pressure pulsations could be one such explanation. These pulsations will drive a flow through the EGR-system even if the static pressures indicate a flow in the opposite direction.

Chapter 3

Modelling

This chapter describes the modelling of the components in the turbocharged diesel engine. Some of the components in the existing model are modified to work properly with the new components and the model is extended with VGT and EGR.

3.1 Existing Model

In this section, the existing mean value engine model at Scania will be described briefly. The model is for a turbocharged diesel engine without turbo compound and exhaust brake. It has been developed in several steps by combining submodels earlier presented in the master's theses [15] [13], engine modelling literature [8] and others [7]. The final steps in the development of the model was taken by David Elfvik [3] and Jesper Ritzén [14] in their master's theses. The different submodels will be presented following the air/exhaust path through the engine, starting from the intake side. An illustration of the model can be found in figure 3.1 and the inputs and outputs can be seen in figure 3.2.

3.1.1 Compressor

The first component in the air path that is modelled is the compressor, which is stiffly connected to the turbine via the turbine shaft. The modelling of the turbine and the turbine shaft is presented in section 3.1.5 and 3.1.6. Earlier, this model has been presented in [7]. Two output signals are of interest; the torque produced by the compressor and the mass flow through the compressor. The torque is given by:

$$\tau_{cmp} = \frac{W_{cmp} c_{p_{air}} T_{amb}}{\eta_{cmp} \omega_{cmp}} \left(\left(\frac{p_{im}}{p_{amb}} \right)^{\frac{\gamma_{air}-1}{\gamma_{air}}} - 1 \right) \tag{3.1}$$

The flow and the efficiency is modelled by maps provided by the manufacturer. The pressure ratio over and the speed of the compressor are inputs to the maps.

$$W_{cmp} = f_{W_{cmp}}\left(\frac{p_{im}}{p_{amb}}, n_{cmp}\right) \tag{3.2}$$

$$\eta_{cmp} = f_{\eta_{cmp}}\left(\frac{p_{im}}{p_{amb}}, n_{cmp}\right) \tag{3.3}$$

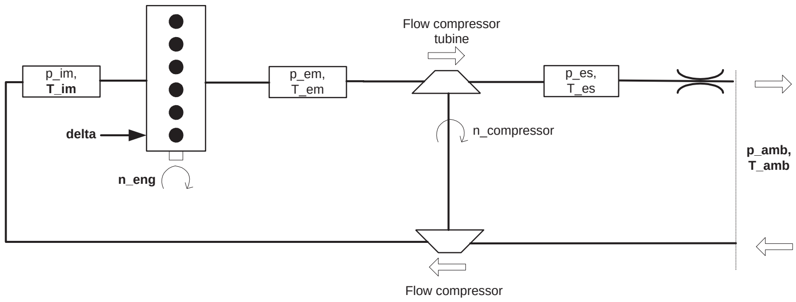

Figure 3.1: Schematic illustration of the existing model, input signals are bold.

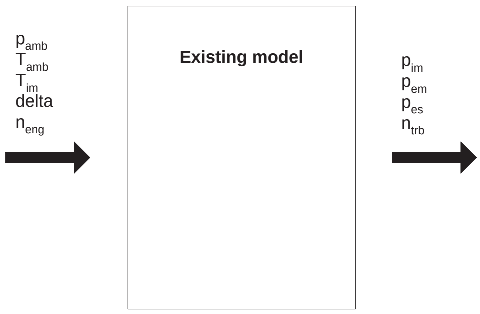

Figure 3.2: Input and output signals of the existing model.

3.1.2 Intake Manifold

The intake manifold has previously been modelled as an isothermal control volume [14]. The isothermal assumption, that the temperature is constant or $\dot{T}$=0 might be reasonable when modelling an engine without EGR. When using EGR however, exhaust gas will be injected into the intake manifold as well as air from the intercooler causing a more fluctuating temperature [6].

Thus $\dot{T}$ must be taken into account. Heat transfer between the intake manifold walls and the air mixture is not taken into account. Because of the air cooler, the temperatures involved are not higher than for a non-EGR engine, thus the addition of the EGR-system does not motivate a change. The volume and temperature in the intake manifold is identical for the exhaust gas and air. Differentiating the ideal gas law:

$$\begin{aligned}
&\dot{p_{im}} = \dot{p_{exh}} + \dot{p_{air}} = \\
&\frac{\dot{m}_{exh} R_{exh} T_{exh}}{V_{im}} + \frac{m_{exh} R_{exh} \dot{T}_{exh}}{V_{im}} + \frac{\dot{m}_{air} R_{air} T_{air}}{V_{im}} + \frac{m_{air} R_{air} \dot{T}_{air}}{V_{im}} = \\
&\qquad \frac{W_{egr} R_{exh} T_{exh}}{V_{im}} + \frac{W_{cac} R_{air} T_{air}}{V_{im}} - \frac{W_{eng,in} R_{im} T_{im}}{V_{im}} + \\
&\qquad\qquad + \frac{m_{exh} R_{exh} \dot{T}_{exh}}{V_{im}} + \frac{m_{air} R_{air} \dot{T}_{air}}{V_{im}}
\end{aligned} \tag{3.4}$$

The internal energy of an ideal gas, assuming that c_v is constant:

$$U_{exh} = m_{exh} u_{exh} = m_{exh} c_{v,exh} T_{exh} \tag{3.5}$$

Differentiate:

$$\begin{aligned}
\frac{dU_{exh}}{dt} &= \dot{m}_{exh} c_{v,exh} T_{exh} + m_{exh} c_{v,exh} \dot{T}_{exh} = \\
&W_{egr} c_{v,exh} T_{exh} - x_{egr} W_{eng,in} c_{v,im} T_{im} + m_{exh} c_{v,exh} \dot{T}_{exh}
\end{aligned} \tag{3.6}$$

Energy conservation for an open system gives:

$$\frac{dU_{exh}}{dt} = \dot{H}_{exh,in} - \dot{H}_{exh,out} - \dot{Q} \tag{3.7}$$

where

$$\dot{H}_{exh,in} = W_{egr} c_{p,exh} T_{exh} \tag{3.8}$$

and

$$\dot{H}_{exh,out} = x_{egr} W_{eng,in} c_{p,exh} T_{exh} \tag{3.9}$$

where

$$x_{egr} = \frac{m_{exh}}{m_{air} + m_{exh}} \tag{3.10}$$

also, if heat transfer is neglected, $\dot{Q} = 0$. Combining 3.6 and 3.7:

$$\dot{T}_{exh} = \frac{1}{m_{exh} c_{v,exh}} (W_{egr} T_{exh} R_{exh} - x_{egr} W_{eng,in} R_{im} T_{im}) \tag{3.11}$$

Equations 3.5 through 3.11 can be derived analogously for the air fraction. Combining 3.4 with 3.11 (and its air counterpart):

$$\dot{p}_{im} = \frac{W_{egr}}{V_{im}}\left(R_{exh}T_{exh} - \frac{R_{exh}^2}{c_{v,exh}}T_{exh}\right) + \frac{W_{cac}}{V_{im}}\left(R_{air}T_{air} - \frac{R_{air}^2}{c_{v,air}}T_{air}\right) - \frac{W_{eng,in}}{V_{im}}\left(R_{im}T_{im} - \frac{R_{exh}}{c_{v,exh}}x_{egr}R_{im}T_{im} - \frac{R_{air}}{c_{v,air}}(1-x_{egr})R_{im}T_{im}\right) = \frac{1}{V_{im}}\left(W_{egr}R_{exh}\gamma_{exh}T_{exh} + W_{cac}R_{air}\gamma_{air}T_{air} - W_{eng,in}R_{im}\gamma_{im}T_{im}\right) \tag{3.12}$$

This relation is often referred to as an adiabatic control volume. The states chosen are pressure and the mass in the intake manifold separated into exhaust gas and air. Another possibility would be to include a temperature state and exclude the mass states. The advantage using mass states is that the EGR-rate can be easily calculated according to 3.10.

The temperature in the intake manifold can be calculated using the ideal gas law:

$$T_{im} = \frac{p_{im}V_{im}}{(m_{exh} + m_{air})R_{im}} \tag{3.13}$$

Summarizing the Intake Manifold model:

$$\dot{p}_{im} = \frac{1}{V_{im}}(R_{air}\gamma_{air}W_{cac}T_{cac} + R_{exh}\gamma_{exh}W_{egr}T_{egr} - R_{im}\gamma_{im}W_{eng,in}T_{im}) \tag{3.14}$$

$$\dot{m}_{exh} = W_{egr} - x_{egr}W_{eng,in} \tag{3.15}$$

$$\dot{m}_{air} = W_{cac} - (1 - x_{egr})W_{eng,in} \tag{3.16}$$

3.1.3 Engine

The engine submodel consists of two submodels, one for the flow through the engine and one for the temperature of the exhaust gases.

Engine Flow Model

During the intake phase of the cylinder cycle, air fills the cylinders. The air mass-flow into the engine depends on many different factors, but the most important are engine speed, intake manifold pressure and temperature. Volumetric efficiency, η_{vol}, is the ratio between the volume inducted into the engine and the volume ideally inducted (the displaced volume every cylinder cycle). The density is assumed to be the same in the intake manifold as in the cylinders during the intake phase, therefore the volumetric efficiency is also the ratio between actual and ideally inducted mass. The air mass flow into the engine is ideally:

$$\dot{m}_{ideal} = \frac{V_d n_{eng} p_{im}}{2RT_{im}} \tag{3.17}$$

Consequently the actual amount inducted into the engine is:

$$\dot{m} = \eta_{vol} \frac{V_d n_{eng} p_{im}}{2RT_{im}} \tag{3.18}$$

Previously a volumetric efficiency model has been used which is based on a look-up table with engine speed and intake manifold temperature as inputs:

$$\eta_{vol} = f_{\eta_{vol}}(n_{eng}, T_{im}) \tag{3.19}$$

In this thesis another model, presented in [16] has been used with good performance:

$$\eta_{vol} = C_{vol} \frac{r_c - \left(\frac{p_{em}}{p_{im}}\right)^{1/\gamma_{air}}}{r_c - 1} \tag{3.20}$$

where r_c is the compression ratio and C_{vol} is a constant.

During the exhaust phase, the exhaust gases are pressed out of the cylinder and into the exhaust manifold. The flow out of the engine equals the sum of the flow into the engine and the amount of fuel injected.

$$W_{eng_{out}} = W_{eng_{in}} + W_{fuel}, \tag{3.21}$$

where

$$W_{fuel} = \frac{\delta n_{eng} N_{cyl}}{120} \tag{3.22}$$

Exhaust Gas Temperature

Two different models for the exhaust gas temperature have previously been used at Scania. In the first one the exhaust gas temperature is modelled as an ideal Otto cycle, earlier presented in [15]. A non-linear equation system has to be solved in every time step. The equations are:

$$T_{em} = T_1 \left(\frac{p_{em}}{p_{im}}\right)^{\frac{\gamma_{exh}-1}{\gamma_{exh}}} \left(1 + \frac{q_{in}}{c_v T_1 r_c{}^{\gamma_{exh}-1}}\right)^{\frac{1}{\gamma_{exh}}} \tag{3.23}$$

The specific energy of the charge per mass is:

$$q_{in} = \frac{W_{fuel} q_{HV}}{W_{eng_{In}} + W_{fuel}} (1 - x_r). \tag{3.24}$$

The residual gas fraction is:

$$x_r = \frac{1}{r_c}\Big(\frac{p_{em}}{p_{im}}\Big)^{\frac{1}{\gamma_{exh}}}\left(1 + \frac{q_{in}}{c_v T_1 {r_c}^{\gamma_{exh}-1}}\right)^{-\frac{1}{\gamma_{exh}}} \tag{3.25}$$

The model is complete with:

$$T_1 = x_r T_{em} + (1 - x_r)T_{im}. \tag{3.26}$$

The problem with this model is the poor performance during low load conditions, that is when W_{fuel} is small.

The second model used is a static model, presented in [12]. The equation is:

$$T_{em} = T_{im} + \frac{Q_{LHV} h(W_{fuel}, N_{Eng})}{c_{p,exh}(W_{eng,in} + W_{fuel})} \tag{3.27}$$

$h(W_{fuel}, N_{Eng})$ is a look-up table. This model will neglect some dynamics, but has proven to be a better overall choice.

An improved model for the exhaust gas temperature would be desirable, but this is beyond the scope of this thesis.

3.1.4 Exhaust Manifold

The exhaust manifold is modelled as an isothermal control volume, that is by differentiating the ideal gas law and neglecting $\dot{T}$. Why is it reasonable to assume that $\dot{T}$ is negligible in this case? Compared to the intake manifold there is only one flow into the exhaust manifold, thus there will be no mixing of gases with different temperatures, flows and thermodynamic properties as in the case with the intake manifold. The fact that a second outward flow for the EGR system is added doesn't change the temperature properties of the exhaust manifold compared to earlier non-EGR MVEM:s [14] presented at Scania. The inward flow is equal to the flow out of the engine and the outward flow is the flow through the turbine plus the flow through the EGR-valve.

$$W_{in} = W_{eng_{out}} \tag{3.28}$$

$$W_{out} = W_{trb} + W_{egr} \tag{3.29}$$

The model for the exhaust manifold will contain one state only with a single parameter V_{em}:

$$\dot{p}_{em} = \frac{R_{exh} T_{em}(W_{eng_{out}} - W_{trb} - W_{egr})}{V_{em}} \tag{3.30}$$

3.1.5 Turbine

The turbine is modelled analogously to the compressor, however since the turbocharger has a variable geometry, the control signal u_{vgt} also comes into play.

The torque equation is essentially the same as for the compressor, but for expanding instead of compressing the gas. The torque is given by:

$$\tau_{trb} = \frac{W_{trb} c_{p_{exh}} T_{em} \eta_{trb}}{\omega_{trb}} \left(1 - \left(\frac{p_{em}}{p_{es}} \right)^{\frac{1-\gamma_{exh}}{\gamma_{exh}}} \right) \tag{3.31}$$

As for the compressor, the mass flow and efficiency are modelled by maps provided by the manufacturer.

$$W_{trb} = f_{W_{trb}} \left(\frac{p_{em}}{p_{es}}, n_{trb}, u_{vgt} \right) \tag{3.32}$$

$$\eta_{trb} = f_{\eta_{trb}} \left(\frac{p_{em}}{p_{es}}, n_{trb}, u_{vgt} \right) \tag{3.33}$$

The temperature after the turbine is modelled as:

$$T_{trb_{out}} = \left(1 + \eta_{trb} \left(\left(\frac{p_{em}}{p_{es}} \right)^{\frac{1-\gamma_{exh}}{\gamma_{exh}}} - 1 \right) \right) T_{em} \tag{3.34}$$

3.1.6 Turbine Shaft

The turbine shaft connects the turbine and the compressor. By use of Newton´s second law the derivative of the turbine shaft speed can be modelled as:

$$\dot{\omega}_{trb} = \frac{1}{J_{trb}} \left(\tau_{trb} - \tau_{cmp} \right) \tag{3.35}$$

The same approach has previously been used in for example [7] and [13].

3.1.7 Exhaust System

As above, the pressure is modelled using a standard control volume, assuming the temperature variations are slow. The flow into the volume equals the flow through the turbine and the flow out of the volume equals the flow through the exhaust pipe.

$$\dot{p}_{es} = \frac{R_{exh} T_{es}}{V_{es}} (W_{trb} - W_{es}) \tag{3.36}$$

The flow out of the volume is modelled using a quadratic restriction, with the restriction constant k_{es} [1]:

$$W_{es}^2 = \frac{p_{es}}{k_{es} R_{exh} T_{es}} (p_{es} - p_{amb}) \tag{3.37}$$

The parameters k_{es} ad V_{es} are estimated from measurement data.

3.2 EGR

This section describes the modelling of the EGR-system.

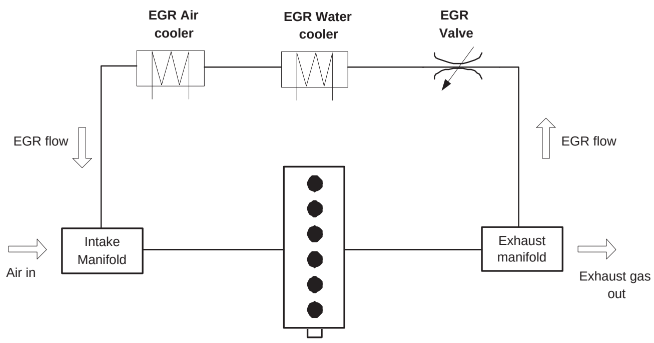

Figure 3.3: The EGR configuration.

3.2.1 Flow Model

The flow through the EGR-valve is modelled using the equation for compressible isentropic flow through a restriction, as described by Heywood [8]:

$$W_{egr} = A_{egr} \frac{p_{em}}{\sqrt{T_{em} R}} \Psi \left(\frac{p_{avalve}}{p_{em}}, \gamma_e \right)$$

$$\Psi \left(\frac{p_{avalve}}{p_{em}}, \gamma_e \right) =$$

$$\begin{cases} \sqrt{\frac{2\gamma_e}{\gamma_e - 1} \left(\left(\frac{p_{avalve}}{p_{em}} \right)^{\frac{2}{\gamma_e}} - \left(\frac{p_{avalve}}{p_{em}} \right)^{\frac{\gamma_e + 1}{\gamma_e}} \right)} & \text{if } \frac{p_{avalve}}{p_{em}} \geq \left(\frac{2}{\gamma_e + 1} \right)^{\frac{\gamma_e}{\gamma_e - 1}} \\ \sqrt{\gamma_e \left(\frac{2}{\gamma_e + 1} \right)^{\frac{\gamma_e + 1}{\gamma_e - 1}}} & \text{else} \end{cases} \tag{3.38}$$

This relation is based on the assumptions that the flow is isentropic and that the gas is ideal. The maximum flow occurs when the velocity at the throat equals the velocity of sound which occurs at the critical pressure ratio

$$\frac{p_{avalve}}{p_{em}} = \left(\frac{2}{\gamma_e + 1}\right)^{\frac{\gamma_e}{\gamma_e - 1}} \tag{3.39}$$

A_{egr} is the effective flow area of the EGR valve. In the ideal case A_{egr} for a poppet valve is a function of valve lift only:

$$A_{egr} = k1 + k2u_{egr} + k3u_{egr}^2 \tag{3.40}$$

For small valve lifts the area will be limited by the flow around the circumference of the valve head (first case, figure 3.4), a fairly linear function of lift. For higher lifts the area will be increasingly limited by the flow area around the rod holding the valve head (second case, figure 3.4), evening out the area, therefore making the second degree polynomial a reasonable description of A_{egr}.

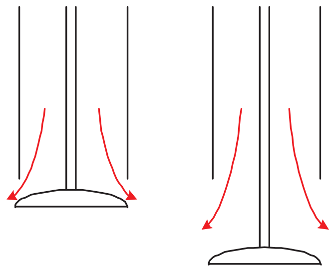

Figure 3.4: Illustrating the linear and evening-out phase of the effective flow area.

There will also be a pressure drop over the water cooler. However, as discussed in chapter 2.2.3 measuring the pressure after the valve / before the water cooler is difficult. Measurements in a blow-rig have shown a pressure drop over the cooler of about 0.08 bar at the highest occurring flows with a typical quadratic relation to the flow. The air cooler imposes a restriction of similar size, but once again this is difficult to verify. Two different model structures are suggested, the single restriction model and the two stage restriction model.

The single restriction model

This model uses equation 3.38 to describe the total pressure drop over the EGR system using p_{em} and p_{im} for inputs. This would however completely neglect the dynamics of the gas volumes present in the EGR system.

Measurement data from the test engine have shown that equations 3.38 and 3.40 does not give an accurate enough description of the EGR flow in the single restriction model. There are two different options which could explain this phenomena. Either the measurements are incorrect or the model structure / choice of inputs to the model is at fault. As discussed in chapter 2.2.3, pressure pulsations in the exhaust manifold could be one explanation. Ricardo consulting [10] suggested that the pulsations could be accounted for by using an additive compensation factor on the exhaust pressure in the form of a two-dimensional look-up table using engine speed and load as inputs:

$$p_{em,comp} = p_{em} + h(n, \delta) \tag{3.41}$$

To test this concept the desired exhaust pressure, that is the pressure which would give a perfect match to measured EGR flow assuming an effective flow area according to equation 3.40, was calculated backwards from equation 3.38 using measurement data. However, no signs were visible that the difference between desired and actual exhaust pressure is correlated to neither load nor engine speed. Another possibility would be to use a black-box type of compensation factor on the effective flow area:

$$A_{egr,comp} = f(p_{em}, p_{im}, U_{egr})A_{egr} \tag{3.42}$$

Looking at measurement data there is a connection between pressure quotient over the valve and diverging calculated effective flow areas. For a given valve position, the lower the pressure quotient, the higher the calculated effective flow area. Taking another look at the measurement data it was noted that for high valve lifts not only the pressure quotient, but also the absolute pressure has an effect on calculated A_{egr}. Increasing exhaust pressure with identical pressure quotient results in a larger calculated effective flow area. Therefore the following relation for the effective flow area is suggested:

$$A_{egr,comp} = (\frac{p_{im}}{p_{em}})^{a1}(a2 + U_{egr}p_{em}^{a3})A_{egr} \tag{3.43}$$

where a1, a2 and a3 are constants.

The polynomial version of the effective flow area as well as the black-box compensated version will be evaluated further in chapter 4.1.

The two stage restriction model

Another option is to ignore the pressure measurements p_{avalve} and p_{awater} and introduce one quadratic restriction for both coolers:

$$W_{egr,out} = \sqrt{\frac{p_{avalve}}{k_1 R_{exh} T_{awater}}(p_{avalve} - p_{im} + k_2)} \tag{3.44}$$

where k_1 is the restriction constant and k_2 is a compensation constant for pressure pulsations. k_1 and k_2 will be calculated to optimize the overall performance of the flow through the EGR system. An isothermal control volume is introduced between the EGR valve restriction and the cooler restriction in order to model the dynamics:

$$\dot{p}_{avalve} = \frac{R_{exh} T_{avalve}}{V_{egr}} (W_{egr} - W_{egr,out}) \tag{3.45}$$

where V_{egr} is a constant. The temperature after the water cooler is used to calculate the density in the quadratic restriction. This is one of several choices that could be made, the reason this choice was made is that this temperature is somewhat an average of the temperature in the EGR system.

The single restriction and two stage restriction models will be compared further in chapter 4.1.

3.2.2 Temperature Model

There will be a slight temperature difference over the valve. One possible model is based on isentropic compression:

$$T_{avalve} = (\frac{p_{avalve}}{p_{em}})^{\frac{\gamma-1}{\gamma}} T_{em} \tag{3.46}$$

In reality however, it was difficult to measure the temperature after the valve, making validation very difficult. Especially during low gas flows in the EGR system the temperature diverge from physically reasonable values, probably due to heat transfer from the surrounding walls. Either way the temperature drop will be slight, less than 20 degrees in most cases which will not affect the temperature after the EGR-coolers significantly. Therefore no temperature model for the valve is included, making the temperature after the valve identical with the exhaust manifold temperature in the model.

The cooling of the EGR gas is made by a two-stage cooler, first the standard heat exchanger using the engine cooling water as coolant and secondly an air cooler. The heat exchanger effectiveness is defined as:

$$\epsilon = \frac{\text{actual heat transfer}}{\text{maximum possible heat transfer}} \tag{3.47}$$

or equivalently

$$\epsilon = \frac{\Delta\text{T(minimum fluid)}}{\text{Maximum temperature difference in heat exchanger}} \tag{3.48}$$

The minimum fluid is the one experiencing the largest temperature difference over the heat exchanger. Assuming that the coolant has the lower temperature drop (this is normally the case) 3.48 can be rewritten as:

$$T_{out} = T_{in} + \epsilon(T_{cool} - T_{in}) \tag{3.49}$$

One way of describing the efficiency is the effectiveness-NTU method. NTU is short for number of transfer units:

$$N = \frac{UA}{C_{min}} \tag{3.50}$$

where U is the overall heat transfer coefficient, A is the effective area and $C_{min} = \dot{m}c_p$ of the minimum fluid. Thus, NTU is indicative of the size of the heat exchanger. Holman [9] has listed the efficiencies as a function of NTU for various types of heat exchangers.

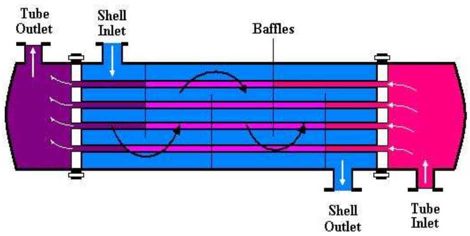

Figure 3.5: Shell and tube heat exchanger.

The type of water cooler used is a shell and tube heat exchanger with one shell pass and one tube pass, figure 3.5. The flow inside the heat exchanger is because of the internal structure a combination of counterflow and cross-flow. This is somewhat difficult to model. One assumption could be that the flow is mainly counterflow. The following effectiveness relation is valid for the counterflow case:

$$\epsilon_{water} = \frac{1 - e^{-N(1-C)}}{1 - Ce^{-N(1-C)}} \tag{3.51}$$

where $C = \frac{C_{min}}{C_{max}}$

Alternative, more simple ways of modelling the efficiency were also considered. Plotting the measured effectiveness against EGR-flow showed a fairly constant efficiency (close to 1) for low flows and a linear relation for higher flows:

$$\epsilon_{water} = \begin{cases} k1 & \text{if } W_{egr} \leq k2 \\ k3 + k4W_{egr} & W_{egr} > k2 \end{cases} \tag{3.52}$$

A third way of modelling the water cooler was suggested by Ricardo Consulting [10]:

$$\epsilon_{water} = e^{k1W_{egr} + k2W_{egr}^2} \tag{3.53}$$

This relation was also derived on a purely empirical basis. The linear model proved to offer the best performance out of the three models suggested and was therefore chosen. The water cooler will suffer from fouling [11] which will lower the efficiency over time by up to 20%. This behavior has not been modelled.

The prototype air cooler on the test engine is a cross flow cooler, identical to the charge air cooler, only smaller. The NTU-efficiency is given by:

$$\epsilon_{air} = 1 - e^{\frac{e^{-CN^{0.78}} - 1}{CN^{-0.22}}} \tag{3.54}$$

As discussed in chapter 2.2.3, the temperature after the air cooler is difficult to measure properly. Possibly because of this the NTU model was a poor fit to measurement data and therefore rejected. Superior results were achieved with a constant efficiency:

$$\epsilon_{air} = k1 \tag{3.55}$$

3.3 Extended Models

In this section the new EGR models will be integrated with the modified version of the existing model.

The models are schematically depicted in figure 3.6 and 3.7.

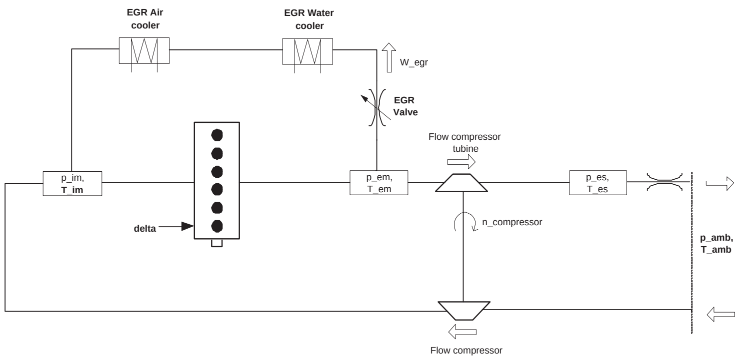

Figure 3.6: Extended model with single restriction EGR model.

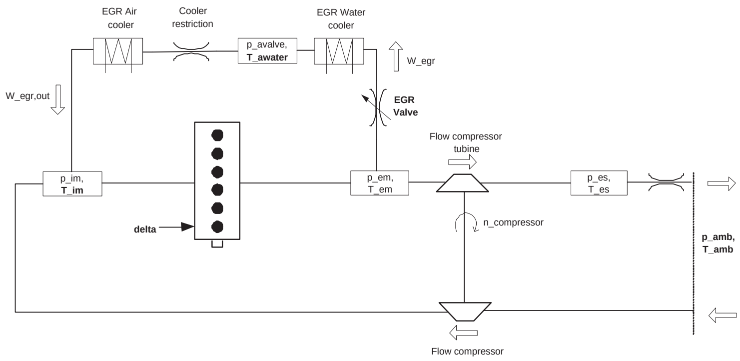

Figure 3.7: Extended model with two stage restriction EGR model.

Table 3.1: Input signals

Signal	Description
p_{amb}	Ambient pressure
T_{im}	Inlet manifold temperature
T_{cac}	Charge air temperature
T_{amb}	Ambient temperature
n_{eng}	Engine revolution speed
δ	Injected fuel
u_{egr}	EGR valve position
T_{cool}	EGR cooling water temperature
u_{vgt}	VGT vane position

Table 3.2: Output signals

Signal	Description
p_{im}	Intake manifold pressure
p_{em}	Exhaust manifold pressure
p_{es}	Exhaust system pressure
n_{trb}	Turbine speed
x_{egr}	EGR rate

Chapter 4

Tuning

All parameters were tuned using a set of 51 static points (separate from the data set used in chapter 5 for validation). Different operating conditions were used, 1200, 1500 and 1900rpm at 25, 50, 75 and 100% load using manual adjustment of the VGT vane position and EGR valve position in order to cover the whole range of EGR flows and valve positions. The entire engine model has not been tuned, only the new EGR model. For optimization, Lsoptim [4] was used.

4.1 Flow Model

The ”measured” effective flow area was calculated backwards from equation 3.38 using the measured value of W_{egr}.

4.1.1 Single restriction model

First, the polynomial model for effective flow area (equation 3.40) is tuned. The results are shown in figure 4.1. The mean absolute relative error to parameter setting data is quite high, which is obvious from the figure. The mean relative error is 57.5%.

Table 4.1: Tuned parameters for the polynomial model

Variable name	Value
k1	-12.4064
k2	34.1357
k3	-6.4102

The results of the tuning of the black-box effective area function (equation 3.43) are shown in figure 4.2.

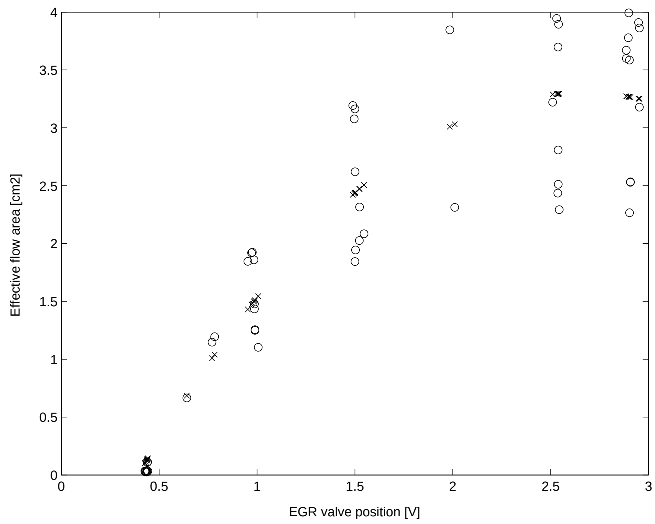

Figure 4.1: Measured effective area (o) and fitted model data (x), polynomial model.

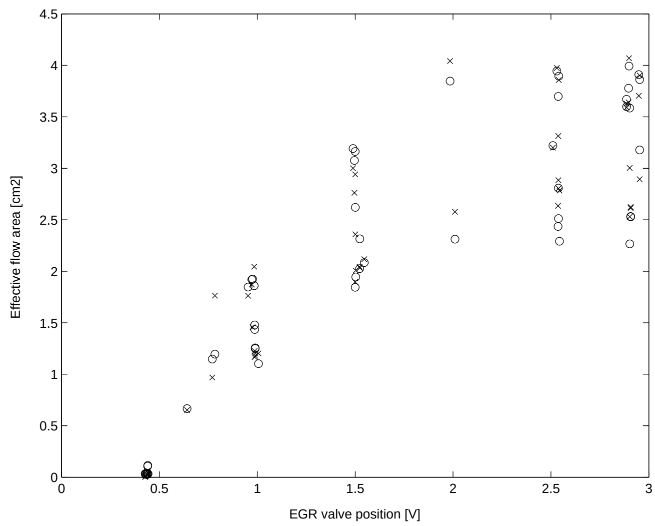

Figure 4.2: Measured effective area (o) and fitted model data (x), black-box model.

The fit to measured data is clearly improved compared to the polynomial model. The mean relative error is 18.9%, and for $u_{egr} > 0.5V$, it's 8.2%.

Table 4.2: Tuned parameters for the black-box model

Variable name	Value
a1	-3.0038
a2	3.5947
a3	0.3082
a4	-1.9045
a5	4.9567
a6	-1.1311

4.1.2 Two stage restriction model

In this section a combination of manual tuning and Lsoptim has been used for practical reasons. The basis for this optimization work has been to optimize the EGR flow to measured data for the complete EGR system, using measured p_{em} and p_{im} (static pressures) for inputs and ignoring the unreliable p_{avalve} and p_{awater}. During the optimization process, only static data was used, and therefore V_{egr} cannot be estimated and also $W_{egr} = W_{egr,out}$.

The constants k_1 and k_2 in the cooler restriction were optimized to give a p_{avalve} which will suit equation 3.38 optimally.

Table 4.3: Tuned parameters for the cooler restriction

Variable name	Value
k1	-0.0002
k2	0.07

Note that k_1 has a negative value. In combination with the value given for k_2 this means that the cooler restriction will give a pressure drop for low flows and a pressure increase for high flows as shown in figure 4.3.

The negative value of k_1 results in an inverse quadratic restriction which is obvious in figure 4.3. At first glance this might seem like a non-physical behavior, but we should remember that the pressures modelled are static pressures and not total pressures. A simple example illustrates this:

$$p_{avalve} + \frac{1}{2}\rho_{exh} v_{egr}^2 = p_{im} + \frac{1}{2}\rho_{im} v_{im}^2 \tag{4.1}$$

or equivalently

$$p_{avalve} + \frac{W_{egr}^2 R_{exh} T_{em}}{2A_{egr}^2 p_{avalve}} = p_{im} + \frac{W_{im}^2 R_{im} T_{im}}{2A_{im}^2 p_{im}} \tag{4.2}$$

The pressure sensor p_{im} is located in the middle of the intake manifold, therefore it's reasonable to assume that the air/exhaust gas flow passing the sensor (resulting in the corresponding dynamic pressure) is approximately half of the

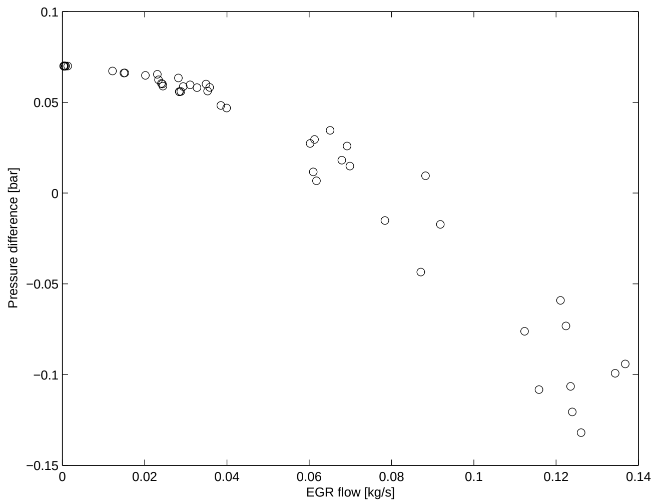

Figure 4.3: Pressure difference vs. EGR flow.

total intake manifold flow. In order to get a qualitative feeling for the impact of the dynamic pressures, p_{im} and p_{avalve} are both assumed to be in the 2 bar range, an exhaust temperature of 700K, an intake temperature of 300K, an EGR flow of 0.14kg/s (the highest practically occurring flow), a 28% EGR rate resulting in an air flow of 0.50kg/s and finally the areas were assumed to be $7cm^2$ and $24cm^2$ respectively. This results in:

$$p_{avalve} - p_{im} = -0.16bar \tag{4.3}$$

which confirms that an increase in the static pressure over the EGR coolers is possible because of the decrease in dynamic pressure.

Using the calculated pressure p_{avalve} from the cooler restriction model as an input to equation 3.38 the dispersion in effective flow area is reduced compared to the single restriction model. The simple non-compensated polynomial function for effective flow area is now a much better fit, see figure 4.4.

The mean absolute relative error while fitting the model to parameter setting data is 22.1%, and for $u_{egr} > 0.5V$, it's only 5.1%.

4.2 Temperature model

4.2.1 Water cooler

The linear model suggested in chapter 3.2.2 was tuned using Lsoptim. The results are shown in table 4.5 and the fitting of the model to parameter setting data in figure 4.5.

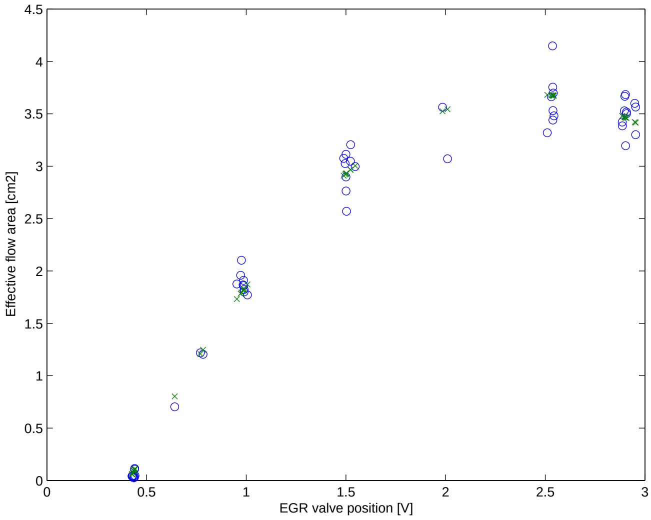

Figure 4.4: Measured effective area (o) and fitted model data (x), two stage restriction model.

Table 4.4: Tuned parameters for the two stage restriction model

Variable name	Value
k1	-0.0002
k2	0.07
a1	-16.7628
a2	44.5691
a3	-9.2529

Table 4.5: Tuned parameters for the linear water cooler model

Variable name	Value
k1	0.99
k2	0.025
k3	1.0488
k4	-2.2834

The model is a reasonably good fit to measurement data. One factor which could affect the result is fouling which will degrade the efficiency of the cooler [11]. The 51 static points used for parameter setting was measured on two separate occasions which is obvious in figure 4.5.

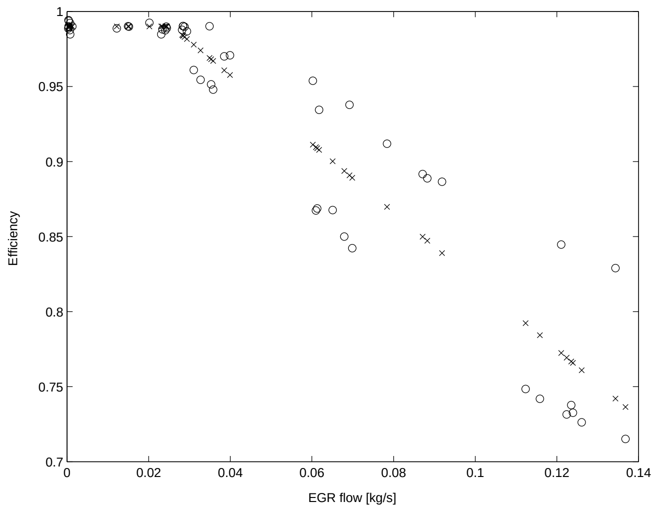

Figure 4.5: Measured cooler efficiency (o) and fitted model data (x).

4.2.2 Air cooler

The constant efficiency of the air cooler was tuned to 0.97.

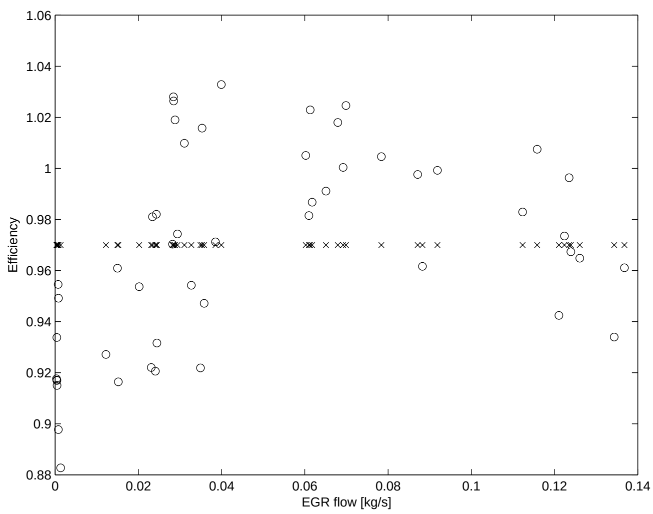

Figure 4.6: Measured cooler efficiency (o) and fitted model data (x).

Although the model fit to parameter setting data doesn't look very good, the relative error is not great.

Chapter 5

Validation

In this chapter, the models are validated using a set of 28 static points collected in the engine test bed. Error is measured by the measures mean error and maximal error. Also, the error distribution is analyzed using histogram plots.

$$\text{mean relative error} = \frac{1}{n}\sum_{i=1}^{n}\frac{|\hat{x}(t_i) - x(t_i)|}{|x(t_i)|} \tag{5.1}$$

$$\text{maximum relative error} = \max_{1\leq i\leq n}\frac{|\hat{x}(t_i) - x(t_i)|}{|x(t_i)|}, \tag{5.2}$$

where $x(t_i)$ is the measured quantity, $\hat{x}(t_i)$ is the simulated quantity and n is the number of samples.

5.1 Flow Model

The three different EGR valve model configurations presented are validated in this section.

5.1.1 Single restriction model

The measured and modelled effective areas for the simple polynomial model are shown in figure 5.1 . The fit is not very good, for corresponding mean and max errors, see table 5.1. Note that the relative errors are much lower if the data for a almost fully closed valve ($u_{egr} \leq 0.5$) is excluded. The relative error for these almost closed valve positions is not of great significance because of the extremely low flows. A relative error of 100% in these cases would imply an EGR rate of for example 0.2% instead of the actual 0.1%. This will have a very small impact on the model as a whole. Therefore, the relevant mean relative figure is the one given for $u_{egr} > 0.5$. One sign that the model is flawed can be seen in the histogram of the relative errors (for $u_{egr} > 0.5$), figure 5.2. The error distribution is rather even in contrast to the desired gaussian distribution.

Table 5.1: Single restriction polynomial model validation

Data filter	Relative error (%)	
	mean	max
All data	68.82	452.47
$u_{egr} > 0.5$	18.57	25.94

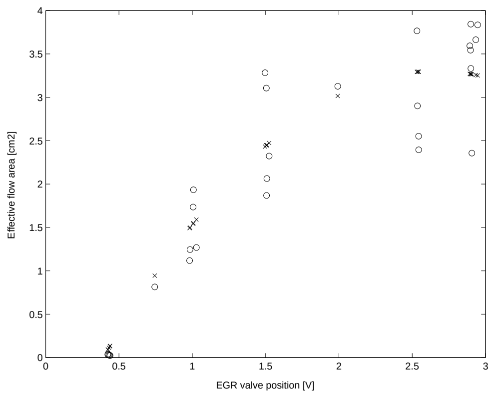

Figure 5.1: Measured effective area (o) and modelled effective area (x) for the single restriction polynomial model

The results for the black-box model are shown in figure 5.3. The correspondence between measured and simulated data is now much better, for relative errors see table 5.2. Also, the error distribution is more gaussian (figure 5.4), although the distribution has a clear shift to the right indicating that the model has some type of systematic error.

Table 5.2: Single restriction blackbox model validation

Data filter	Relative error (%)	
	mean	max
All data	27.58	208.43
$u_{egr} > 0.5$	9.08	23.56

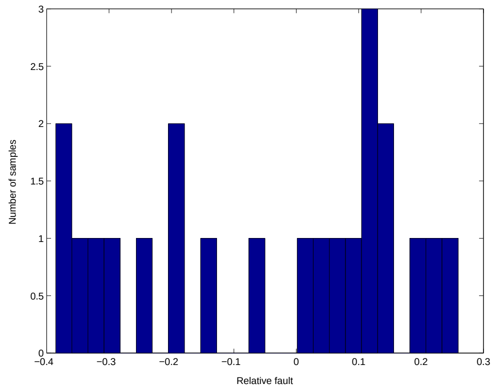

Figure 5.2: Histogram of the relative errors for the polynomial model.

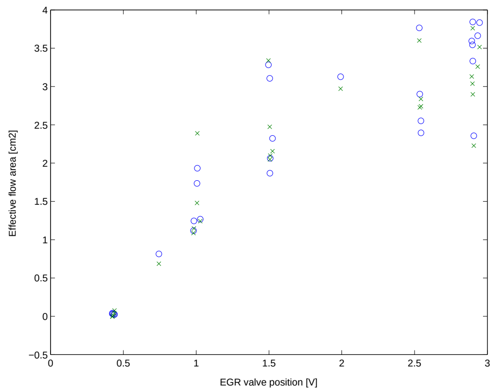

Figure 5.3: Measured effective area (o) and modelled effective area (x) for the single restriction black-box model

5.1.2 Two stage restriction model

The results for the polynomial model in the two stage restriction model are shown in figure 5.5. The fit is good, superior to both single restriction models. For corresponding mean and max errors, see table 5.3. Note that the mean relative error for $u_{egr} > 0.5V$ is only 4.97%. The error distribution for $u_{egr} > 0.5V$ is reasonably gaussian with a few exceptions, see figure 5.6.

5.2 Temperature Model

The water and air cooler temperature models are validated in this section.

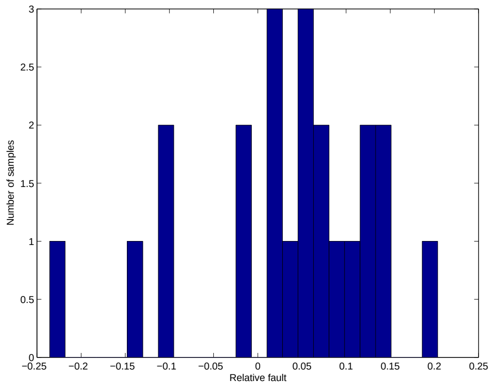

Figure 5.4: Histogram of the relative errors for the black-box model.

Table 5.3: Two stage restriction model validation

Data filter	Relative error (%)	
	mean	max
All data	29.73	313.36
$u_{egr} > 0.5$	4.97	19.60

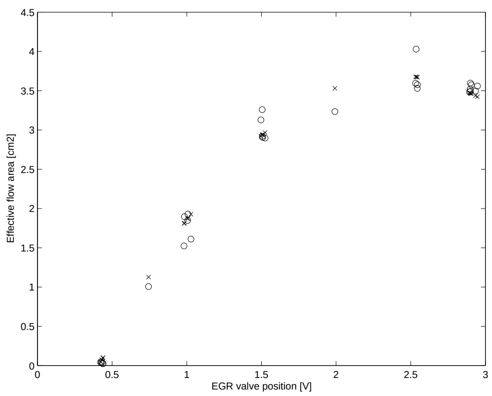

Figure 5.5: Measured effective area (o) and modelled effective area (x) for the two stage model

5.2.1 Water cooler

The water cooler model was validated against measurement data with fair results, see figure 5.7. Figure 5.8 shows the measured and modelled temperature after the water cooler. Relative errors are small in both cases, see table 5.4

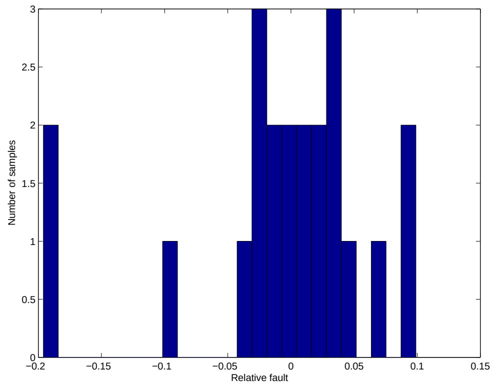

Figure 5.6: Histogram of the relative errors for the two stage model.

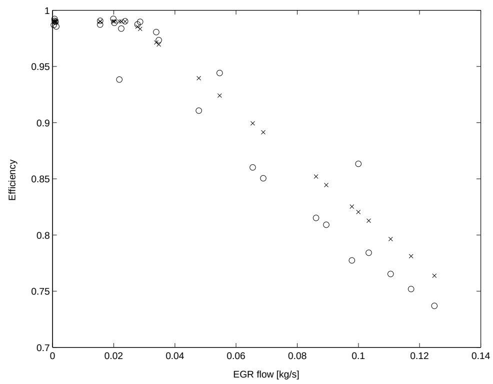

Figure 5.7: Measured efficiency (o) and modelled efficiency (x) for the water cooler.

Table 5.4: Water cooler model validation

Data	Relative error (%)		Absolute error	
	mean	max	mean	max
Efficiency	2.14	6.17	0.0180	0.0516
Temperature	1.32	3.46	5.3731	14.6555

5.2.2 Air cooler

The simple air cooler model gives reasonable accuracy during test bed conditions, although a maximum temperature error of 4.9K is quite high considering the low temperatures involved.

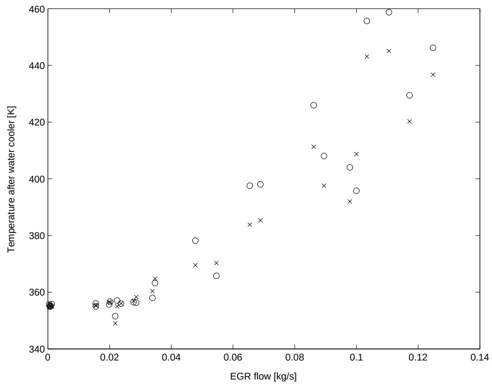

Figure 5.8: Measured temperature after the water cooler (o) and modelled temperature (x).

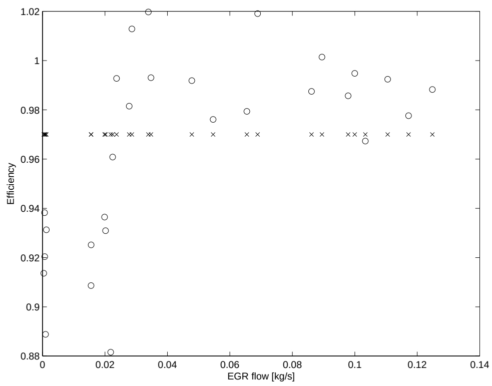

Figure 5.9: Measured efficiency (o) and modelled efficiency (x) for the air cooler.

Table 5.5: Air cooler model validation

Data	Relative error (%)		Absolute error	
	mean	max	mean	max
Efficiency	3.44	10.0	0.0325	0.0884
Temperature	0.77	1.66	2.3179	4.9201

5.3 Sensitivity analysis

One important question is how much the errors in EGR flow and temperature affect the model as a whole, and how much errors in the rest of the model affect the EGR model. These aspects will be discussed in this section.

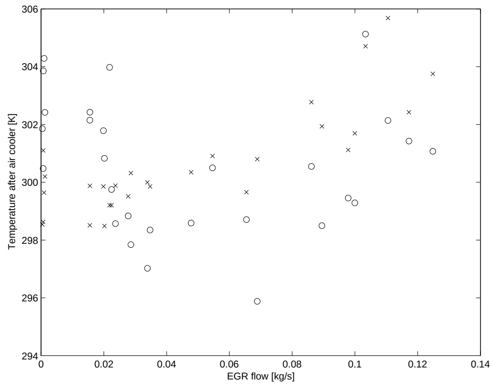

Figure 5.10: Measured temperature after the air cooler (o) and modelled temperature (x).

A complete engine model was implemented in Simulink for this purpose. The constants and look-up tables used in the model were based on previous experience, and not tuned properly for this specific occasion. A perfectly tuned model is not important since only relative errors are of interest. The 13 static operating points of the ESC cycle were chosen to cover a wide range of engine speeds and loads during the sensitivity analysis.

Table 5.6: The ESC cycle

Engine speed [Nm]	Load [%]
500	0
1250	25
1250	50
1250	75
1250	100
1600	25
1600	50
1600	75
1600	100
1950	25
1950	50
1950	75
1950	100

The most important output from the model is intake manifold pressure, p_{im}. Therefore it is interesting to know how much the error in W_{egr} and T_{egr} influences this pressure. Table 5.7 shows the impact of a 5 or 10% error in W_{egr} and a 1% error in T_{egr} on p_{im}. The errors are fairly even in the different operating points, therefore the mean value is given. If the two stage restriction EGR model is used, a typical relative mean error of 5% can be expected and the sensitivity on intake manifold pressure for an error of this size is 0.379%. This can be compared with the intake manifold pressure mean error of 3.4% of a non-EGR model [14], thus 0.379% is not negligible but reasonably low. The sensitivity for errors in T_{aair} is low (table 5.7), considering that the mean error of the simple temperature model is well below 1%. If more work were to be done modelling the EGR system, the focus should be on improving the flow model.

Table 5.7: Sensitivity analysis, p_{im}

Forced error	Resulting mean relative error in p_{im} [%]
5% in W_{egr}	0.379
10% in W_{egr}	0.759
1% in T_{aair}	0.131

As discussed in [5], the exhaust temperature model is seriously flawed. How much will an error in T_{em} influence the EGR temperature and flow? Also, what is the sensitivity for errors in exhaust manifold pressure, p_{em}? Table 5.8 shows the results. The EGR flow is sensitive to errors in exhaust temperature and particularly to errors in p_{em}. Although the exhaust manifold pressure can be tuned to better performance than 5% error, this shows that special care must be taken when tuning a complete engine model with EGR.

Table 5.8: Sensitivity analysis, T_{aair} and W_{egr}

Forced error	Resulting error, T_{aair} [%]	Resulting error, W_{egr} [%]
10% in T_{em}	0.04	2.79
20% in T_{em}	0.08	5.68
5% in p_{em}	0.07	17.49
10% in p_{em}	0.13	26.47

5.3.1 Water cooler fouling

As mentioned in chapter 3.2.2, the EGR water cooler will suffer from fouling, resulting in a decrease in the efficiency of up to 20%. How much will this affect the EGR temperature after the air cooler, and in return what impact will this have on the intake manifold pressure? In order to investigate this the earlier developed cooler models will be used, and T_{aair} will be calculated using the standard model and also using a 10% and 20% decreased water cooler efficiency. The results are shown in table 5.9. With a 20% decrease in efficiency, the resulting mean relative error is 0.56% in EGR temperature. Looking at the sensitivity analysis for errors in T_{aair} (table 5.7), it's clear that a 0.56% error will have a negligible impact on the simulated intake manifold pressure. It is possible that fouling will change the flow properties of the cooler though, this has not been investigated.

Table 5.9: Sensitivity analysis, water cooling fouling

Fouling	Resulting mean relative error, T_{aair} [%]
10%	0.28
20%	0.56

5.4 Summary

Two models for the EGR flow, the single restriction black-box model and the two stage restriction model have been validated with good performance. The mean relative errors are 9.08% and 4.97% respectively (excluding the fully closed position of the valve). This translates into a reasonably low performance degradation of the modelled intake manifold pressure. The mean relative error caused by these flow errors are approximately 0.76% and 0.38% respectively.

The simple temperature models for the water and air EGR coolers were found to be working satisfactorily, at least during test bed conditions. The modelled temperature of the exhaust gas injected into the intake manifold shows a mean relative error of 0.77% to measured data. This low error will have a negligible impact on the mean relative error of p_{im}. The sensitivity analysis also shows that the EGR flow models are sensitive to errors in the exhaust manifold temperature and pressure. Therefore it is important to tune these properly in the full engine model.

Chapter 6

Conclusions and Future Work

6.1 Conclusions

Two models for the EGR flow have been developed and successfully tested. The single restriction black-box model features a low complexity and a mean relative error of 9.08% during static conditions. The two stage restriction model is of higher complexity and has a mean relative error of 4.97%, which is low enough considering that the sensitivity analysis showed that a 5% EGR flow error gives an error in the modelled intake manifold pressure of only 0.38%. Both models give big improvements over the previously used EGR models. The intake manifold model has been modified to take temperature fluctuations into account. The temperature model suggested has also been tested with good performance. The 0.77% mean relative error in EGR temperature has a negligible impact on the model as a whole. Fouling of the water EGR cooler of up to 20% reduced efficiency will also have a very small impact on the complete model.

6.2 Future Work

During the work with this thesis a couple of interesting areas for further investigations have come up. In this section some of them are presented.

The most important future work is to validate the model using dynamic measurement data. In order to do this the entire model must be tuned, there is currently no way to measure the EGR flow dynamically making a validation of the EGR system separate from the rest of the model difficult. A data collecting system with a higher sampling rate than the test bed computer along with faster temperature sensors are also essential.

It would be interesting to validate the EGR cooler temperature models in a truck or in a test bed with climate control to investigate their behavior at ambient temperatures different from 298K.

Investigating pressure pulsations and their influence on exhaust and intake manifold pressure is of great interest if further improvements are to be made on the EGR flow model. Is it possible to compensate the mean value model in order to model these pulsations? A measurement system with a high sampling rate along with high speed pressure sensors would be useful. Measuring dynamic pressure would be useful in order to reinforce the motivation for the inverse quadratic cooler restriction.

In order to make the model more physical and also simplifying the tuning, the turbine and compressor look-up tables could be replaced with physically based equations. Some work has been done in this field [16], although integration with the new EGR models and further validation under dynamic conditions is needed. One possible disadvantage with physically based equations is that limitations could be revealed in the other models.

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Notation

Table 6.1: Symbols used in the report.

Symbol	Value	Description	Unit
c_p	Con	Specific heat capacity at constant pressure	$J/(kg \cdot K)$
c_v	Con	Specific heat capacity at constant volume	$J/(kg \cdot K)$
γ	Con	Ratio of heat capacities, c_p/c_v	$-$
δ	Var	Amount of injected fuel	$kg/stroke$
η	Var	Efficiency	$-$
η_{vol}	Var	Volumetric efficiency	$-$
R	Con	Gas constant, $c_p - c_v$	$J/(kg \cdot K)$
τ	Var	Torque	Nm
n	Var	Rotational speed	rpm
ω	Var	Rotational speed	$\frac{1}{s}$
N_{cyl}	Con	Number of cylinders	$-$
p	Var	Pressure	Pa
$\dot{p}$	Var	Derivative of pressure	Pa/s
q_{HV}	Con	Heating value	J/kg
T	Var	Temperature	K
V	Con	Volume	m^3
W	Var	Mass-flow	kg/s
$\dot{m}$	Var	Mass-flow	kg/s
x_r	Var	Residual gas fraction	$-$
x_{egr}	Var	EGR rate / exhaust gas fraction	$-$
r_c	Con	Compression ratio	$-$
J	Con	Moment of inertia	Nms
V_d	Con	Displacement volume (all cylinders)	m^3

Table 6.2: Abbreviations used in this report.

Abbreviation	Explanation
Con	Constant
Var	Variable
rpm	Revolutions per minute
OBD	On Board Diagnostics
EGR	Exhaust Gas Recirculation

Table 6.3: Indices used in this report.

Index	Explanation
im	Inlet manifold
em	Exhaust manifold
es	Exhaust system
trb	Compressor turbine
cmp	Compressor
eng	Engine
amb	Ambient
exh	Exhaust
cac	Charge air cooler
in	Into the component
out	Out of the component