Modelling of Volumetric Efficiency on a Diesel Engine with Variable Geometry Turbine

Master's thesis
performed in **Vehicular Systems**
Performed for **Scania CV AB**

by

Håkan Bengtsson

Reg nr: LiTH-ISY-EX-3379-2002

16th December 2002

Modelling of Volumetric Efficiency on a Diesel Engine with Variable Geometry Turbine

Master's thesis

performed in **Vehicular Systems**,
Dept. of Electrical Engineering
at **Linköpings universitet**

Performed for **Scania CV AB**
by **Håkan Bengtsson**

Reg nr: LiTH-ISY-EX-3379-2002

Supervisor: **David Elfvik, MSc**
Scania CV AB
Mattias Nyberg, PhD
Scania CV AB
Jonas Biteus, MSc
Linköpings universitet

Examiner: **Professor Lars Nielsen**
Linköpings universitet

Södertälje, 16th December 2002

Avdelning, Institution
Division, Department

Vehicular Systems,
Dept. of Electrical Engineering
581 83 Linköping

Datum
Date

16th December 2002

Språk
Language

☐ Svenska/Swedish
☒ Engelska/English
☐ ______________

Rapporttyp
Report category

☐ Licentiatavhandling
☒ Examensarbete
☐ C-uppsats
☐ D-uppsats
☐ Övrig rapport
☐ ______________

ISBN
—

ISRN
LITH-ISY-EX-3379-2002

Serietitel och serienummer
Title of series, numbering

ISSN
—

URL för elektronisk version
http://www.vehicular.isy.liu.se

Titel Modellering av volymetrisk verkningsgrad på en dieselmotor med variabelgeometriturbin

Title Modelling of Volumetric Efficiency on a Diesel Engine with Variable Geometry Turbine

Författare Håkan Bengtsson
Author

Sammanfattning
Abstract

The objectives for this master's thesis are to develop a model of volumetric efficiency for a diesel engine equipped with a Variable Geometry Turbine, study other existing models and test the models using real engine measurements. In this work measurements from two different Scania diesel engines have been used to identify the parameters in the models and also to validate the models.

Two models have partly been derived in this thesis and two other existing models are presented. The four described models and a fifth model have been implemented in Matlab and tested. Measurements have been used for identifying the parameters in the models and for validation.

Among the five validated models of volumetric efficiency, one model called the Overlap and Residual Gas Model (ORM) is suggested. The model shows very good results in the simulation and the validation. Often the absolute model errors are in the same order as the resolution in the equipment used to measure volumetric efficiency in the engine lab. The suggested model has captured the physical behavior of an engine equipped with Variable Geometry Turbine, which no other model studied have done.

Nyckelord
Keywords

Engine modelling; Compression-Ignition; Heat transfer, Valve overlap, Residual gas.

Abstract

The objectives for this master's thesis are to develop a model of volumetric efficiency for a diesel engine equipped with a Variable Geometry Turbine, study other existing models and test the models using real engine measurements. In this work measurements from two different Scania diesel engines have been used to identify the parameters in the models and also to validate the models.

Two models have partly been derived in this thesis and two other existing models are presented. The four described models and a fifth model have been implemented in Matlab and tested. Measurements have been used for identifying the parameters in the models and for validation.

Among the five validated models of volumetric efficiency, one model called the Overlap and Residual Gas Model (ORM) is suggested. The model shows very good results in the simulation and the validation. Often the absolute model errors are in the same order as the resolution in the equipment used to measure volumetric efficiency in the engine lab. The suggested model has captured the physical behavior of an engine equipped with Variable Geometry Turbine, which no other model studied have done.

Keywords: Engine modelling; Compression-Ignition; Heat transfer, Valve overlap, Residual gas.

Preface

This master's thesis has been performed for Scania CV AB, department of engine development in cooperation with Vehicular Systems at Linköpings universitet.

Thesis outline

This thesis is divided into six chapters.

Chapter 1, Introduction: An introduction to this thesis.

Chapter 2, Theoretical Background: The theoretical background for this thesis.

Chapter 3, Measurements: Describes the measurements, the engine used and the lab environment.

Chapter 4, Modelling: Describes the different models of volumetric efficiency studied in this thesis.

Chapter 5, Validation: Contains a validation for the different models that have been studied in this thesis.

Chapter 6, Conclusions and Future Extensions: The conclusions made for the different models. A discussion about the problems with the models and how to improve they are also covered.

Notation: Contains tables with notations, abbreviations, variables, constants and indices that are used in this thesis.

Acknowledgment

I would like to thank my supervisors at Scania, David Elfvik and Mattias Nyberg, and also Jonas Biteus at Linköpings universitet for all help during this thesis. Other persons that has been helpful are Magnus Petterson, Thomas Åkerblom and Roger Olsson.

Håkan Bengtsson
Södertälje, December 2002

Contents

Chapter 1

Introduction

This master's thesis has been performed for Scania CV AB, department of engine development in Södertälje. Scania was founded in 1891 and is today a leading manufacturer of heavy-duty trucks and busses as well as industrial and marine engines.

Background

The emission requirements of heavy-duty truck diesel engines have been more and more restrictive over the years. In the next few years the admitted emission level will decrease rapidly. There will also be a demand for on-board diagnostics (OBD) system, that must be able to detect any malfunction that increases the emissions. It means that the producer must guarantee that the vehicle not only fulfils the emission requirements when it is manufactured, but also detect faults that increase the emissions over a certain thresholds level [1].

Therefore it becomes more and more important to have a good model describing the engine. One possible way to decrease the emissions is to use more model-based control in the engine control system. Thus new models must be developed and existing models improved.

A good model of the engine can be used for many purposes. It can be used for simulation on a desktop computer and is an useful engineering tool in the development of the engine and especially the engine control system. The engineer can easy see how the emissions will be affected, by simulating different control strategies. This is a complement to the laboratory studies and can reduce the time consuming and costly tests needed in a laboratory.

Another important usage for such a model is in model based control and diagnosis. As the name implies the model is implemented in the engine control system and can be used during operation to achieve better control strategies, that take advantages in the knowledge of how the engine will respond on different control signals. In diagnosis purpose results from the model can be compared to sensor signals. Faults in sensors, actuators or other parts of the engine can be detected and diagnosed in this way. As already mentioned laws will be in force in a few years, that regulates which faults that must be detected during operation.

In previous thesis, Scania has developed a mean value engine model [1, 2, 3]. The engine model is object oriented which makes it easy to replace the sub models to improve the model. The volumetric efficiency model, which describes the breathing efficiency of the engine, is one of the sub models that need to be improved.

Many volumetric efficiency models available today are developed under some assumptions on the pressures in the air path system. New components like the Variable Geometry Turbine[1] (VGT) and waste gate affects pressures in such a way that these assumptions must be reconsidered.

Objective

There are three objectives for this thesis. The objectives are to:

- Evaluate a couple of existing models of volumetric efficiency with the focus on how they work with engines equipped with a VGT.
- Develop a model that works better when pressures can be varied more freely with a VGT.
- Real engine measurements should be used to identify the parameters and for the validation.

The model shall be used in the mean value engine model previously developed at Scania [1, 2, 3].

Method

The model of volumetric efficiency has been developed and implemented in Matlab and Simulink environment. The necessary data for the estimation and the validation have been collected in a test cell in Scania's engine laboratory using a computer measurement system. The engines used for this study are special equipped Scania diesel engines. The engines are used in heavy-duty trucks and buses, has six cylinders and a total engine volume of 12 liters. The main differences from a

[1] In some literature Variable Nozzle Turbine (VNT) is used instead. The main function is the same, but the technical solution and some characteristics may differ.

normal Scania engine are that an Exhaust Gas Recirculation (EGR) system is added. It also has a VGT, with this it is able to change the pressures in exhaust and inlet manifolds more widely as described in Objective. The parameters in the models are found using static measurements from the engines. The models have been validated to see how well they can describe real engines.

Target

This master's thesis is aimed for Scania CV AB and undergraduate or graduate engineers with basic knowledge in vehicular system.

Chapter 2

Theoretical Background

In this chapter the theoretical background which this thesis is based on is presented. Some important conceptions are defined and discussed.

In Section 2.1 mean value engine models are described. Section 2.2 contains a short introduction to the diesel engine. In Section 2.3 the volumetric efficiency is defined. Also the dominating phenomenas that affect the volumetric efficiency are discussed. Section 2.4 describes the air path and divides it into three parts: Inlet; Volumetric; Exhaust. Section 2.5 sums up this chapter.

2.1 Mean Value Models

In mean value engine models things that vary within cycles are not concerned [5, 8]. The signals are often represented as mean values over one or several cycles.

A pure physically based or mixed model of volumetric efficiency often depends on quantities that normally not are available under operation, e.g. the exhaust pressure. Therefore often a pure black-box modelling approach is used to find some dependence of the volumetric efficiency expressed in the main engine variables [4]. An example on such a model is presented in [11].

The objective with this thesis is to develop a MVEM sub model for the volumetric efficiency that is as physically based as possible.

2.2 Diesel Engine

The fuel used in almost all heavy-duty trucks today is diesel. In this thesis only four-stroke diesel engines are studied. The principle after which the diesel engine works is called Compression-Ignition (CI). It means that the diesel is injected into the compressed air and ignites.

There are two important differences between the diesel and the gasoline engines when discussing volumetric efficiency. Diesel engines generally have higher compression ratio than gasoline engines. The other difference is that diesel engines have no throttle in the inlet manifold, which gasoline engines have. Generally diesel engines have higher volumetric efficiency than gasoline engines.

2.3 Volumetric Efficiency

The intake system restricts the amount of air that can be inducted into the cylinder during one cycle. The volumetric efficiency is the parameter describing the effectiveness of the induction process. The induction process is defined as all events taking place between the inlet-valve opening (IVO) and the inlet-valve closing (IVC).

2.3.1 Definitions

There are some different definitions of the volumetric efficiency, η_{vol}, in the literature. One definition often used is described in [5],

$$\eta_{vol} = \frac{120 W_{ic}}{\rho_{im} V_d n_{eng}}, \tag{2.1}$$

where W_{ic} is the mass flow through the intercooler, ρ_{im} is the density in the inlet manifold. The displacement volume for the engine is V_d and the engine speed is n_{eng}. An equivalent definition is

$$\eta_{vol} = \frac{m_{cMix}}{\rho_{im} V_d}. \tag{2.2}$$

The total mass of fresh mixture in the combustion chamber is denoted m_{cMix}. The inlet density, ρ_{im}, is taken in the inlet manifold, which means that only the performance of the inlet valve and the inlet port are considered. Another possibility is to use the atmosphere density [5]. In that case the performance of the total air path would be considered. But this is often not used for a turbo charged engine, where the inlet pressure often is higher than the surrounding atmosphere pressure.

Another definition that slightly differ from the definitions above in characteristics and application is proposed in [10],

$$\eta_{vol} = \frac{m_{cMix} + m_{cRes}}{\rho_{im} V_d}.$$

The mass is divided into two parts, fresh mixture, m_{cMix}, and residual gas, m_{cRes}. In this definition the total mass of residual gas trapped in the cylinder is included in the volumetric efficiency. This definition is practical in some measurement techniques, where the cylinder pressure is used to determine the total mass of gas trapped in the cylinder. In (2.1) and (2.2) only the mass of fresh mixture is considered, not the mass of residual gas.

Fresh mixture is the new gas that is introduced to the cylinder through the inlet valve, i.e. not including EGR. In a diesel engine the fresh mixture contains of air, which e.g. contains of water vapor, oxygen and nitrogen. The gas left in the charge from previous cycle is called residual gas. The charge is the content in the cylinder when all valves are closed.

In this thesis a slightly modified version of (2.2) is used,

$$\eta_{vol} = \frac{m_{cMix} + m_{cEgr}}{\rho_{im} V_d}. \tag{2.3}$$

The difference is a wider interpretation of which mass trapped in the cylinder to be considered. Besides the fresh mixture also the mass from EGR, m_{cEgr}, is counted here. If the engine not is equipped with an EGR system, the two definitions in (2.3) and (2.2) are equal. However the mass of residual gas and the mass of back flow are not counted. Back flow occurs when the intake and the exhaust valves are open at same time and exhaust gas can flow from the exhaust port to the inlet port.

There are two reasons why (2.3) is preferred. The main reason is that it fits better into the MVEM. This is because the mass flow from the fresh mixture and the EGR are merged in the inlet manifold. The second reason is that it gives a simpler model. If the mass of EGR must be separated from the mass of fresh mixture one more variable must be used in the model.

2.3.2 Different Phenomenas

In this section some important phenomenas that affect the volumetric efficiency will be discussed.

Heat Transfer

Heat transfer can be described by the total increase of temperature, ΔT, in the gas [13]. A model describing this is the formula of heat transfer for gas flowing through a tube,

$$\Delta T = \eta \left(T_s - T_g \right), \tag{2.4}$$

where η is a constant, T_s is the surface temperature of the cylinder wall and T_g is the mean gas temperature.

The effect of the heat transfer will decrease the volumetric efficiency. In Section 4.3 a model that uses ΔT to describe the heat transfer is presented.

As (2.4) shows, metal temperature and gas temperature in the inlet manifold will affect ΔT. When the gas temperature in the cylinder is higher than in the inlet manifold, volumetric efficiency will decrease. This is because the density of the gas will decrease when it gets hotter. If the density of the gas in the combustion chamber decreases, the mass of the gas in the combustion chamber will decrease. The metal temperature can vary a lot during engine operation. Thus, T_s is a dynamic effect.

A good dynamic model of T_s is probably the best way to achieve a dynamic volumetric efficiency model that compensates for the temperature effects in transients.

Residual Gas

Residual gas will require a volume in the cylinder, which otherwise could have been filled with fresh mixture. Thus less residual gas will increase the volumetric efficiency.

The volume of the combustion chamber when the exhaust valve closes is constant. Pressure of the gas in the cylinder when the exhaust valve closes strongly depends on the exhaust pressure, p_{em}. Thus the amount of residual gas also will depend on the exhaust pressure just before the exhaust valve closes.

The heat transfer from hot residual gas to cool fresh mixture often is presumed to decrease the volumetric efficiency. This has been found not to be true [13]. The reason is that the compression of cooling residual gas equals the expansion of fresh mixture that is heated in an idealized process. No mass is flowing out from or in to the combustion chamber during this process.

Engine design also affects the amount of residual gas. Higher compression ratio, r_c, will reduce the volume available for the residual gas and therefore increase the volumetric efficiency. Diesel engines generally have much higher compression ratio then gasoline engines. The effect of residual gas will thus be smaller on diesel engines.

Valve Overlap

In Figure 2.1 the valve lift is plotted against the crank angle. As seen the inlet valve starts to open before the exhaust valve is completely closed. Thus flow between the exhaust and the inlet can occur, which is called back flow.

If p_{em} is greater then p_{im} gas will flow from the exhaust manifold back into the cylinder and push out gas to the inlet manifold. During

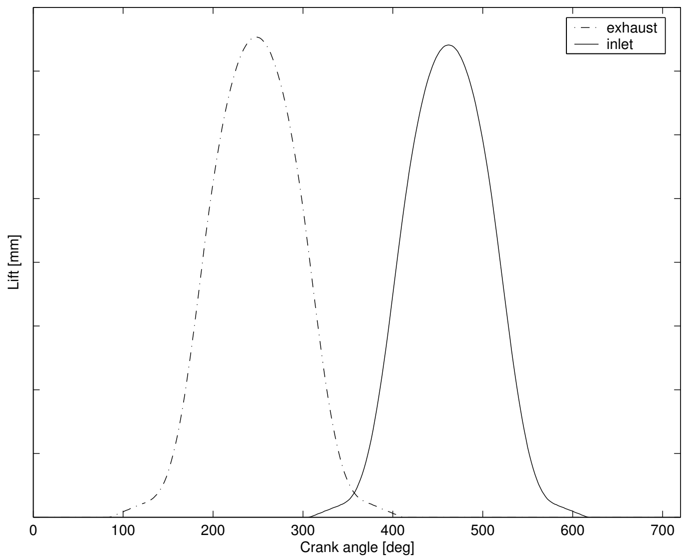

Figure 2.1: *Exhaust and inlet theoretically lift curve. (0 mm cold clearance.)*

intake stroke a combination of fresh mixture and the exhaust gas the were pushed back into inlet manifold by the back flow are inducted into the combustion chamber. Therefore the mass of the exhaust gas will take room from the fresh mixture. Thus back flow will decrease the volumetric efficiency.

If p_{em} is smaller then p_{im} gas will flow from the inlet manifold to the combustion chamber and push out gas through the exhaust port. Assuming that the gas pushed out contains exhaust gas infers that the back flow in this case will increase the volumetric efficiency.

Large overlap is often used in turbo charged diesel engines. Flow from the valve overlap is used to reduce the turbine temperature [13].

Fuel Injection

In diesel engines fuel injection occurs after the inlet valve is closed. Thus there is no direct effect from the fuel injection on the volumetric efficiency.

2.4 Air Path

In the combustion process fuel and oxygen are needed to produce torque. Besides that, heat and exhaust gas are produced. In the laws there are restrictions on how much emissions that are admitted. The emissions are e.g. nitric oxides and smoke. The emissions depend among other on how the air to fuel ratio was in the combustion [8]. Therefore it is important to know how the amount of air that can be inducted depends on different factors. Often the volumetric efficiency is used to calculate how much air that is inducted in the cylinder. The volumetric efficiency is important both in the design of the engine and also in the electronic control system, e.g. to know how much fuel to inject.

In this thesis the objective is to study and build a model of volumetric efficiency. To understand what happens to the air during its path through the engine, this section will describe the different parts of the air path system.

The air path has in this study been divided into three parts: Inlet, Volumetric and Exhaust.

2.4.1 Inlet Part

The inlet part of the air path describes how the air is affected in the path from the atmosphere to the inlet manifold. Effect of the EGR system is included both in the inlet and the exhaust parts. The effects in inlet part will affect the amount of air that can be inducted into the cylinder. But according to the definition of volumetric efficiency in (2.3), only the performance of the inlet port and valve are included. Therefore the inlet part of the air path does not directly affect the volumetric efficiency.

This section of the air path is included in the mean value engine model (MVEM) described in [3], except the EGR system. An almost similar model with EGR system included is presented in [1, 2]. A schematic schedule which shows the inlet part of the air path is presented in Figure 2.2.

Air Filter

The first component in the air path is the air filter, which cleans the air from pollutants. Over the air filter there will be a small pressure drop.

Compressor

A turbo charger has two separate components that affect the mass flow through the engine: the compressor and the turbine. The compressor

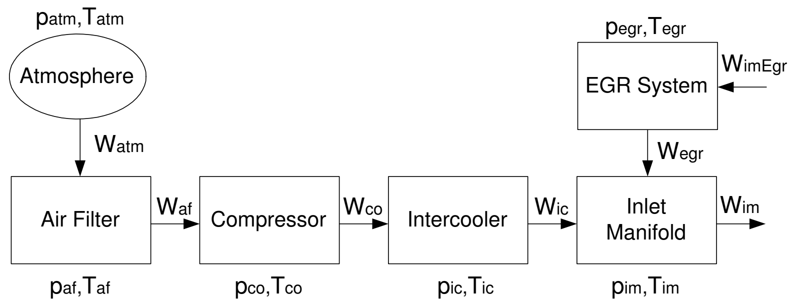

Figure 2.2: *A schedule of the flow through the inlet part. Arrows shows the positive flow direction.*

pumps the fresh mixture into the inlet manifold and thus increases the density of the air. The gas temperature after the compressor will increase according to thermodynamical laws. The energy needed to drive the compressor comes from the turbine via a connecting turbine shaft.

If the engine is equipped with a VGT the geometry of the turbine can be controlled, which gives an extra degree of freedom to the system. In Figure 2.3 the difference between an ordinary turbine and a VGT can be seen. With an ordinary turbine only one position is possible and thus only one line. In The pressure in the inlet manifold and the exhaust manifold are plotted with three different VGT control signals. As the figure shows, the pressure quotient over the engine, $\frac{p_{em}}{p_{im}}$, is approximately constant for a fix VGT position. But the constant differs with the control signal.

The pressure quotient over the engine has been assumed constant in some models of volumetric efficiency. The figure clearly shows that the quotient significantly varies with the VGT control signal,

$$\frac{p_{em}}{p_{im}} = f\left(u_{vgt}\right), \tag{2.5}$$

where u_{vgt} is the control signal to the VGT. Therefore it is important to have a model of volumetric efficiency that works with different pressure quotient.

Intercooler

The intercooler cools the air heated by the compressor and therefore increase the density [13]. There are a lot of thin pipes that the air flows through. Around these pipes, air with lower temperature is streaming

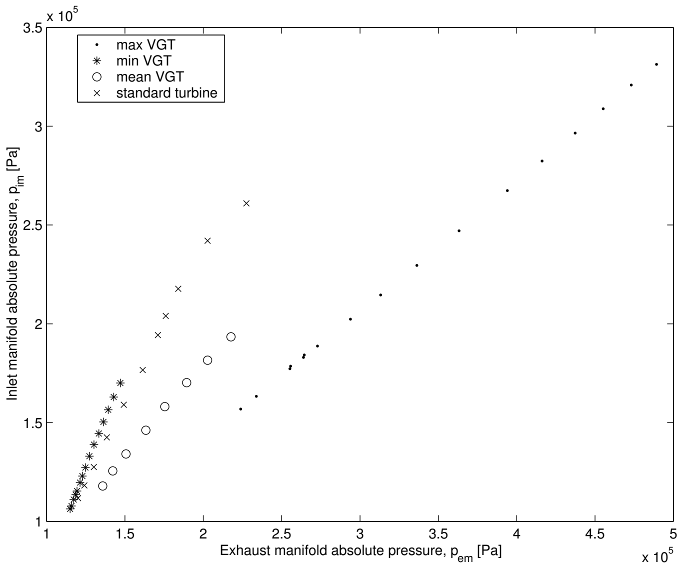

Figure 2.3: *Measurements with three different control signals to the VGT and one with a standard turbine.*

and cooling the air inside the pipes. Thus it also occurs a pressure drop over the intercooler.

Inlet Manifold

The inlet manifold is the volume placed just before the inlet port. It only acts as a volume and no pressure drop occurs over it. The function is to divide the airflow from the intercooler to the different inlet ports leading to each combustion chamber. A principal sketch over the inlet manifold is seen in Figure 2.4. In the definition of volumetric efficiency, (2.3), the pressure and temperature of the air in the inlet manifold are used.

EGR System

If the engine is equipped with an EGR system, exhaust gas is added to the fresh mixture and mixed. The purpose is to affect the combustion process so that the temperature at combustion is lower which will give less nitric oxides in the exhaust gas [8].

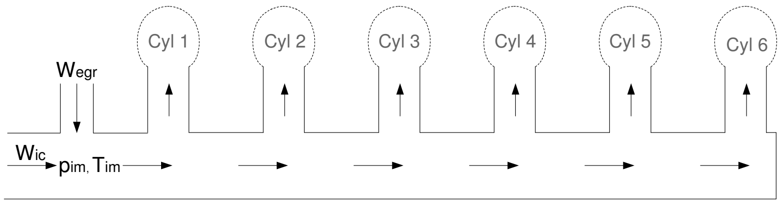

Figure 2.4: *A sketch over the inlet manifold.*

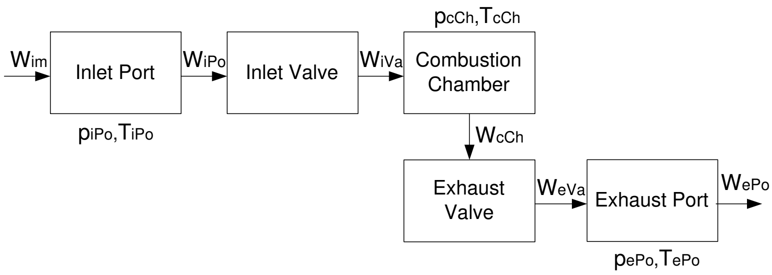

Figure 2.5: *A schedule of the flow through the volumetric part. Arrows shows the positive flow direction.*

2.4.2 Volumetric Part

The effects on the air between the inlet manifold and the exhaust manifold in this thesis has been defined as the volumetric part of the air path. The MVEM in [3] contains no volumetric part in this form. Instead the volumetric efficiency takes care of these effects. This is where the model of volumetric efficiency comes in. A schematic schedule which shows the volumetric part of the air path is presented in Figure 2.5.

Inlet Port

When the air flows through the inlet port, heat is exchanged between the metal and the gas [13]. This will rise the temperature on the gas while passing through the inlet port. This phenomena is called heat transfer and will affect the temperature in the gas when it flows into the combustion chamber.

Inlet Valve

The inlet valve controls the gas flow from the inlet manifold into the combustion chamber. The valve is controlled via the camshaft, which

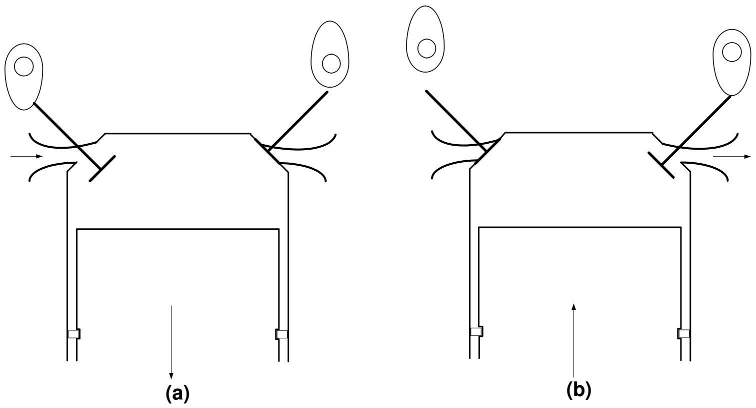

Figure 2.6: *A sketch of the volumetric part of the air path. Arrows shows the positive flow direction and the piston movement.* **(a)** *Intake stroke.* **(b)** *Exhaust stroke.*

lifts the valve from its closed position. The camshaft has a theoretical curve profile, which describes how the valve lift depends on the crank angle. For the engine used in the measurements, the theoretically curve with no cold clearance is shown in Figure 2.1. A simple sketch over the volumetric part is shown in Figure 2.6.

The valve head is placed in the combustion chamber and therefore the valve will be very hot [14]. Hot metal expands, which means that the valve stem will be longer when the temperature of the material rises. Thus there must be a clearance. This means that the actual clearance depending on the temperature must be subtracted from the theoretical curve to get the actual lift.

The flow trough a valve is often expressed by the equation for compressible flow through a flow restriction, which is derived from the one-dimension isentropic flow [5],

$$\dot{m} = \frac{C_D A_R p_0}{\sqrt{RT_0}} \Psi\left(\frac{p_T}{p_0}\right), \tag{2.6}$$

where $\dot{m}$ denotes mass flow through the valve and

$$\Psi\left(\frac{p_T}{p_0}\right) = \begin{cases} \sqrt{\gamma}\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{2(\gamma-1)}} & \text{if } \frac{p_T}{p_0} \le \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}} \\ \left(\frac{p_T}{p_0}\right)^{\frac{1}{\gamma}} \sqrt{\frac{2\gamma}{\gamma-1}\left[1-\left(\frac{p_T}{p_0}\right)^{\frac{\gamma-1}{\gamma}}\right]} & \text{else.} \end{cases}$$

Here C_D is an experimentally determined discharge coefficient, p_0 and T_0

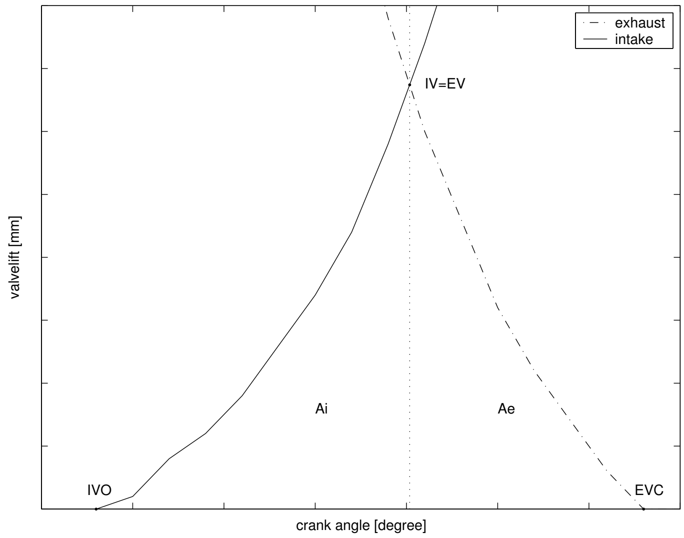

Figure 2.7: *Overlap between the inlet and the exhaust valves curves.*

are the upstream pressure and the temperature. The constant A_R is the reference area, which varies with the actual lift. The variable p_T is the pressure at the throat, i.e. where the flow area is smallest. The constant γ is the specific heat ratio taken for the gas that flows through the valve.

When the condition $\frac{p_T}{p_0} \leq \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}}$ is valid the flow is choked. This means that the maximum flow rate is reached and the flow has become saturated.

Combustion Chamber

In the combustion chamber the temperature will be very high and the cylinder walls will be much hotter than the gas during the intake. Thus if heat transfer from the metal to the gas is considered significant, this will be much larger than the energy that is transferred in the inlet port. The volume of the combustion chamber varies with the crank angle.

Exhaust Valve

The exhaust valve is similar to the inlet valve. One difference is that it is harder to cool the exhaust valve and thus more clearance normally is needed. For the exhaust valve (2.6) is also valid.

The dominating flow direction is from the intake port via the combustion chamber to the exhaust port. But there are two phenomena

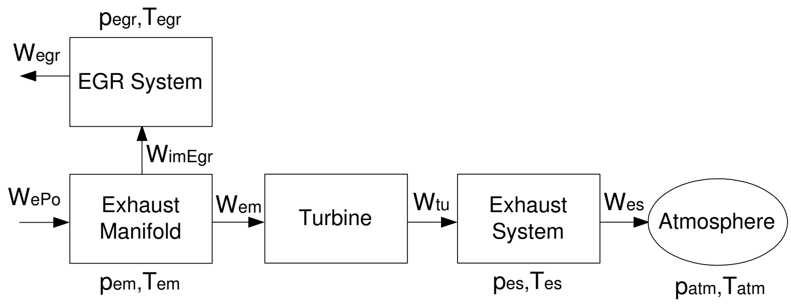

Figure 2.8: *A schedule of the flow through the exhaust part. Arrows shows the positive flow direction.*

that not follow this and are very important in the study of volumetric efficiency: back flow and residual gas.

As can be seen in Figure 2.1 and 2.7 the inlet valve opens before the exhaust valve has closed. If the pressure in the inlet manifold is below the pressure in the exhaust manifold the flow will go in the negative direction.

The residual gas is the exhaust gas that remains in the cylinder after the blow out. If the pressure in the cylinder is higher than the inlet manifold pressure, gas will flow out in the inlet port when the inlet valve opens.

The mass of back flow and residual gas takes room from the fresh mixture and thus they will decrease the volumetric efficiency. The back flow and the residual gas depends on the properties of the gas directly after the exhaust valve.

2.4.3 Exhaust Part

All effects that take places from the exhaust manifold to the atmosphere are in this thesis called the exhaust part of the air path. Also the EGR system is included in the exhaust part. A schematic schedule which shows the exhaust part of the air path is presented in Figure 2.8.

EGR System

The EGR system involves pipes, EGR coolers and a variable valve used to control the amount of EGR flow. When the EGR is added to the air in the inlet manifold it affects the temperature and the pressure there. The definition of volumetric efficiency in (2.3) includes both the mass of fresh mixture and EGR. The physical constants used in (2.6) are approximately equal for exhaust gas and fresh mixture. Thus all the

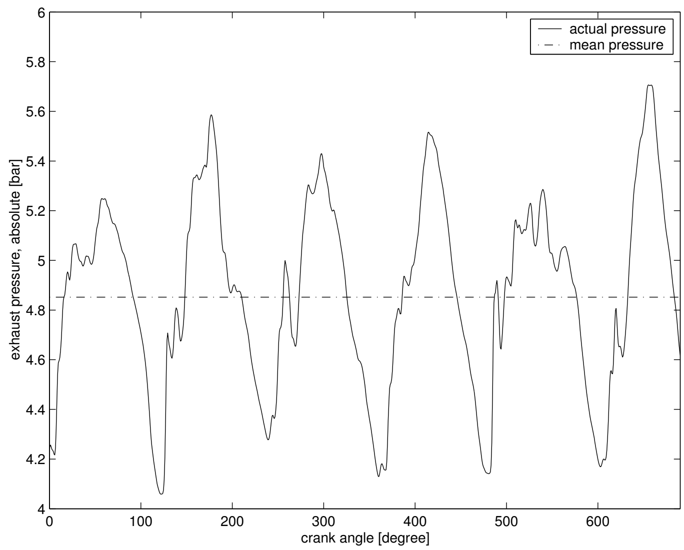

Figure 2.9: *Simulated exhaust manifold pressure over one cycle.*

components in the volumetric part will not behave different with EGR then with fresh mixture.

Exhaust Manifold

After the exhaust port there is an exhaust manifold. The function is opposite to the inlet manifold: To bring the flow from each exhaust port into one pipe. In [3] the exhaust manifold is modelled as a volume with no pressure drop.

The pressure that is important in the study of volumetric efficiency is the one just before the exhaust valve closes. Another measurement related problem occurs when the exhaust valve opens. Inside the combustion chamber gas at very high pressure is confined. When the exhaust valve opens the gas expands quickly out from the combustion chamber and causes a pressure peak in the exhaust manifold.

The same phenomena is present in the inlet manifold also. But the amplitude is larger in the exhaust manifold and thus more important to have in mind. In fact this phenomena is present in all measurements. But it is only the exhaust manifold pressure that has caused problems during this thesis.

In Figure 2.9 a simulation showing this phenomena is presented. The model used is partly validated against a similar engine as used in this thesis.

Turbine

The turbine is the second part of the turbo. The purpose is to utilize some of the energy wasted with the exhaust gas and to transform it into torque, used to drive the compressor.

In [3] a function describing the mass flow through the turbine is used. The function depends on the quotient $\frac{p_{em}}{p_{es}}$, the turbine speed and the control signal, u_{vgt}, that controls the geometry of the VGT.

Exhaust System

Between the atmosphere and the turbine there is an exhaust pipe leading the exhaust gas and a muffler that should reduce the noise. Over the muffler there is a pressure drop. In future engines there also can be a catalyst used for after treatment of pollutants.

2.5 Summary

In this chapter the theoretical background for this thesis has been presented. The volumetric efficiency has been defined and some other possible definitions are discussed. The definition used in this thesis includes mass of EGR in the same way as mass of fresh mixture.

Volumetric efficiency is affected by a lot of different phenomenas. Some of these have been discussed. A model of how the heat transfer can be described as the total increase of temperature is discussed and how this will affect the volumetric efficiency. The volumetric efficiency is also affected by the amount of residual gas trapped in the combustion chamber. The amount of residual gas has been coupled to the compression ratio, r_c. Another important effect that is discussed is the valve overlap.

To understand volumetric efficiency it is good to have a sketch over the complete air path. This has been described and divided into three different parts: Inlet, Volumetric and Exhaust. The volumetric efficiency shall model all effects taking place in the volumetric part of the air path.

Also, the concept of MVEM is discussed and how the model of volumetric efficiency shall be used in an existing MVEM. Shortly the advantage of a model that is physically based is discussed.

Chapter 3

Measurements

Measurements in Scania's engine lab have been a significant part of the work behind this thesis. In this chapter the measurements performed for this thesis will be described. Also some technical data for the Scania diesel engines used in the measurements are presented.

Section 3.1 contains a description of the engines used. In Section 3.2 measuring of volumetric efficiency is discussed. Section 3.3 cover some problems related to measuring that has been significant. In Section 3.4 the experiments performed are presented. Section 3.5 sums up this chapter.

3.1 The Engine

The data used in this thesis have been collected from two specially equipped Scania diesel engines in Scania's engine lab. The engines are used in heavy-duty trucks and buses, has six cylinders and a total engine volume of 12 liters.

One of the engines was equipped with a VGT. It also had an EGR system, but it was plugged. The other one was equipped with an ordinary turbine and an EGR system.

The most important sensors when studying volumetric efficiency are the mass flow of air, temperatures and pressures in the inlet and the exhaust manifolds, air to fuel ratio, VGT position, EGR, load and engine speed. In Figure 3.1 a principal sketch over the engine is presented. Mass flow of air, W_{im}, is measured a bit away from the engine.

In the engine lab all data are collected with a computer system. It is also possible to see and save all variables available in the engine control system. The engine is controlled through changing variables in the engine control system and the speed of the electric brake connected to the engine.

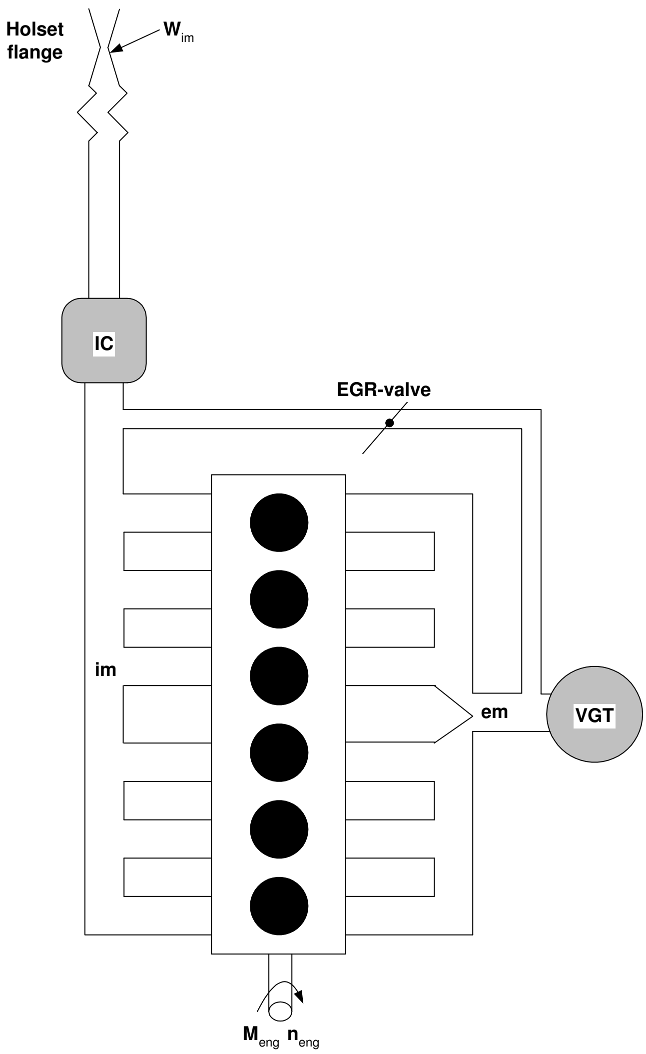

Figure 3.1: *A principal sketch over the engine.*

3.2 Measure of Volumetric Efficiency

The volumetric efficiency is not possible to measure directly. It must be calculated from other measurements using the definition of volumetric efficiency. Most important is the mass flow of air, W_{im}. The airflow is measured in an arrangement that is not placed directly on the engine.

In volumetric efficiency the flow into the combustion chamber is the flow of interest. Therefore a short delay will be seen in transients. Another problem is when the pressure in the inlet manifold is changed. If the pressure rises, more air must be stocked in the inlet manifold. Thus all air measured will not flow into the combustion chamber and the measured volumetric efficiency will be higher than the actual one for a short while. If the pressure in the inlet manifold falls the mass of stocked air in the inlet manifold must decrease. Therefore more air will flow into the combustion chamber than into the inlet manifold for a while. The measured volumetric efficiency will be lower than the actual one.

The air flow is measured with a so called Holset flange [9]. When the air is flowing through the flange a pressure drop occurs. In the flange, the differential pressure is measured which is used to calculate the airflow. The resolution in the pressure measurement is 0.043 mbar. When calculating volumetric efficiency from the air flow, i.e. the differential pressure, this resolution in the pressure cause another resolution in the volumetric efficiency that varies with airflow. For low flows the resolution are poor and for high flows the resolution are better.

When results from the measurements are presented it is important to have the resolution in mind. In some points the resolution is worse than the magnitude of volumetric efficiency, which is shown later in this thesis.

All sensors have a resolution. Thus the resolutions for all sensors used to measure variables in the volumetric efficiency must be considered. But the sensor that dominates the resolution in the measured volumetric efficiency is the airflow sensor. Thus only the effect from the pressure measurement in the Holset flange is plotted in the resolution of the measured volumetric efficiency.

3.3 Other Measurement Problems

The exhaust manifold is a problematic environment to place sensors in. The temperature varies a lot and the exact position of the sensors are important. The exhaust manifold pressure is used when modelling volumetric efficiency. When the exhaust valve opens, a lot of exhaust gas flows into the exhaust manifold quickly. This result in high pressure peaks. The sensor gives the mean value, but of most interest is the pressure before the exhaust valve is closed. Thus this is a potential source of errors.

When EGR is present in the system, the sum of EGR and airflow must be considered. As a matter of principle the EGR is no problem for volumetric efficiency, but when measuring the flow into the cylinder it adds an extra uncertainty. Thus the EGR system was plugged on the engine with VGT.

Another problem concerning the measure of airflow is leakage. There are a lot of places where the air can leak out of the system. To detect if a leakage occurs two different ways of measuring the air to fuel ratio are used, directly measure of air and fuel and another method that uses the concentrations of different molecules in the exhaust gas. If the values from these two methods differ significant it indicates that a leakage may have occurred.

3.4 Experiments

In this section the experiments will be described. Two different engines have been used.

3.4.1 Experiments with VGT

In Figure 2.3 the static points used for identifying parameters and validation are plotted in the $p_{im} - p_{em}$ plane. Only the three lines from different VGT positions that can be seen are used, i.e. not the line from the standard turbine. Along one line the load is changed, by varying the amount of injected diesel. All data are from the engine speed 1167 rpm.

Data from five different transients have been collected, where steps in load, VGT position and both load and VGT position have been tested. In the transient measurements all sensors that are available in the static measurements are not logged, due to technical limitations in the computer measurement system. But all variables that are important for the volumetric efficiency are included.

3.4.2 Experiments with Standard Turbine

The experiment consists of two separate data matrices. There is a larger data matrix, which contains data for all combinations of 21 engine speeds and 11 loads. The smaller data matrix contains all combinations of 17 engine speeds and four loads. No transient measurement has been used from this engine.

3.5 Summary

In this chapter the measurements used in this thesis have been described. Problems related to the measurements are discussed. The main problem is the resolution in the airflow sensor and how it affects the calculated volumetric efficiency. At low airflows the resolution are poor. At higher airflows the problem are smaller.

Another problem of importance when studying volumetric efficiency is the fluctuating exhaust pressure. Only the mean value of the pressure is measured, which not is the pressure of interest. Also, to avoid an extra unnecessary uncertainty the EGR system was plugged.

Finally the data used for identifying the parameters and the validation has been described. Two different engines have been used, one with a VGT and one with an ordinary turbine.

Chapter 4

Modelling

This chapter will go through the modelling part and define the different models that have been derived, studied and compared. Details and derivation of the models are not concerned. The details are presented in Appendix A and B or in the reference where the model is published.

Collecting data is very time consuming. Therefore the approach has been to neglect the engine speed and focus has been to build a model that works good for one engine speed. The amount of data needed is then reduced as well as the model complexity.

By estimating the parameters for the different engine speeds and interpolate between these, a model that works on all speeds is achieved [7].

Section 4.1 presents an existing model based on the ideal cycle. In Section 4.2 another existing regression model is presented. Section 4.3 describes a model derived in this thesis. The model has a term for additional heat transfer. In Section 4.4 another model derived in this thesis is presented. Valve overlap and residual gas are used in the model. Section 4.5 sums up this chapter.

4.1 Ideal Cycle Model

In previous master's thesis performed at Scania, a model of volumetric efficiency based on the ideal engine cycle combined with a lookup table is presented [2, 3]. The model has the quotient $\frac{p_{em}}{p_{im}}$ as variable. The model is

$$\eta_{vol} = \eta_{vol}^{exh} f_{\eta_{vol}^{map}}(n_{eng}, p_{im}) \tag{4.1}$$

where

$$\eta_{vol}^{exh} = \frac{r_c}{r_c - 1} - \frac{1}{\gamma_{air}(r_c - 1)} \left[\left(\frac{p_{em}}{p_{im}} \right) + (\gamma_{air} - 1) \right]. \tag{4.2}$$

In the model, volumetric efficiency is assumed only affected by following:

1. Effects from fuel type, fuel to air ratio, fraction of fuel vaporized in the intake system and fuel heat of vaporization;

2. Mixture temperature as influenced by heat transfer;

3. Compression ratio;

4. Intake and exhaust manifolds design;

5. Intake and exhaust ports design;

6. Intake and exhaust valves geometry, size, lift and timings;

7. Ratio of exhaust to inlet manifold pressure;

8. Engine speed.

The effects from 1 to 6 are seen either as constant or unknown. The function $f_{\eta_{vol}^{map}}(n_{eng}, p_{im})$ shall cover all these factors. The function has the engine speed as a variable, which means that it also involve the effects from 8. Effects from 7 is covered by (4.2), which is based on ideal cycle assumptions [5].

4.2 Regression Model

A pure black box model describing a relationship between volumetric efficiency and engine operating conditions is presented in [11]. The model is a regression model with six parameters. The parameters have been chosen to reflect the physically phenomena that affect the volumetric efficiency. The complete model is

$$\begin{aligned}\eta_{vol} = A + B\frac{1}{\sqrt{T_{im}}} + C\frac{p_{em}}{p_{im}} + D\frac{T_{im}}{T_{em}} + E\left(\frac{n_{eng}}{T_{im}}\right)^{0.8} + \\ + F\frac{n_{eng}^2}{T_{im}}\left(\frac{T_{im}}{T_{em}} + r_c - 1\right),\end{aligned} \tag{4.3}$$

where A,...,F are the parameters that should be fitted to the data.

This model is based on the observation that physically related functional form of the regression model gives better results. The variables used in the model covers the different physically phenomena that affect the volumetric efficiency. In Table 4.1 the variables used in the model are specified.

Physically Phenomena	**Function**
Intake charge temperature	$f_B\left(\frac{1}{\sqrt{T_{im}}}\right)$
Exhaust and intake pressures	$f_C\left(\frac{p_{em}}{p_{im}}\right)$
Residual gas temperature	$f_D\left(\frac{T_{im}}{T_{em}}\right)$
Heat transfer	$f_E\left(\left(\frac{n_{eng}}{T_{im}}\right)^{0.8}\right)$
Engine speed	$f_F\left(\frac{n_{eng}^2}{T_{im}}\left(\frac{T_{im}}{T_{em}}+r_c-1\right)\right)$

Table 4.1: *Variables in the Regression Model.*

4.3 Heat Transfer Model

The Heat Transfer Model is partly derived in this thesis. Details are presented in Appendix A. The model is based on the indicator diagram and involves the effect of heat transfer. The model is

$$\eta_{vol} = \frac{1}{1+\left(\frac{\Delta T}{T_{im}}\right)}\left(\eta_0 - \eta_1\left(\frac{p_{em}}{p_{im}}\right)\right), \tag{4.4}$$

which has two parameters, η_0 and η_1. Variable ΔT is the total temperature rise in the gas from heat transfer.

This is a simple model and the form is quite similar to the ideal cycle model described in Section 4.1. In this model a term compensating for the heat transfer is added. The problem with this model is to model the heat transfer.

The model is derived from the ideal Otto cycle with two assumptions:

- Fresh mixture and residual gas are ideal gases with the same specific heat and molecular weight.
- There is no appreciable flow through the exhaust valve after the inlet valve starts to open.

If ideal gas is assumed and the specific heat ratio, γ, and the compression ratio, r_c, are assumed constant, the model in the form presented above is received.

The simple model of ΔT used in the simulations for this thesis is

$$\Delta T = a_0 + a_1\delta, \tag{4.5}$$

where δ is the mass of injected diesel in the cylinder. The model has two parameters, a_0 and a_1.

4.4 Overlap and Residual Gas Model

The Overlap and Residual Gas Model, which takes the valve overlap in consideration, is partly derived in this thesis. In Appendix B the derivation of the model is presented. The model (4.6) has three parameters, k_1, k_2 and k_3,

$$\eta_{vol} = k_1 - k_2 \left(\frac{p_{em}}{p_{im}}\right)^{\frac{\gamma+1}{2\gamma}} \frac{\sqrt{\frac{|p_{em}-p_{im}|}{\rho_a}}}{\operatorname{sign}\left(p_{em}-p_{im}\right)} - k_3\lambda\left(\frac{p_{em}}{p_{im}}\right)^{\frac{1}{\gamma}}, \quad (4.6)$$

where λ is the fuel equivalent ratio and ρ_a is the ambient gas density defined at exhaust pressure and inlet temperature.

The idea is to model volumetric efficiency using an existing model of residual gas and back flow. These two phenomena depend on the pressure quotient over the engine and is therefore interesting on a VGT equipped engine. The model is based on the physical flow process during the overlap and a simple ideal cycle model of the thermodynamic states involved.

In the form presented above, engine speed n_{eng}, overlap factor (OF) and compression ratio r_c has been nailed up with the model parameters. It may be possible to use these constants in the future to improve the model, use on engines with different concept or reduce the need for estimating new parameters when changing something in the engine hardware.

If more data were available, it would be interesting to investigate the model where the overlap factor not has been neglected. If the model is good, it could be used on the same engine with different cam profiles without changing the parameters. Another idea is to calculate the actual OF during operation, using a model over the actual valve lift which compensates with the actual clearance. It could perhaps improve the model. These ideas have not been tested due to lack of data to use for the identification.

4.5 Summary

In this chapter four volumetric efficiency models that has been studied are presented. The two first models, Ideal Cycle Model and Regression Model, are existing ones and the other two, Heat Transfer Model and Overlap and Residual Gas Model, have partly been derived in this thesis.

Chapter 5

Validation

In this chapter the models described in Chapter 4 will be validated to see how well they describe the real engine.

In Section 5.1 a description on how data has been used in the validation process is presented. The results in form of tables and figures are also presented. Section 5.2 contains comments that are general for all the models. In Sections 5.3 to 5.6 results for the different models are discussed. In Section 5.7 the validation result for a model not described in this thesis is presented. Section 5.8 sums up this chapter.

5.1 Validation Cases

From the engine with a VGT, the data have been collected at three different VGT positions as seen in Figure 2.3. The EGR system was plugged to get more reliable data for the mass flow. All data used was collected only at one engine speed. At every VGT position the load was changed. The lines contain 17, 15 and 8 different loads.

Collecting data in the engine lab is very time consuming. Thus it is necessary to reduce the experiments. To get more data with different engine speeds, available data for another Scania diesel engine with a standard turbine and an EGR system were used. Two data matrices were available with different load and engine speeds. The larger data matrix contains 21 engine speeds with 11 different loads at each speed. The smaller data matrix contains 17 engine speeds with 4 different loads at each engine speed.

Available data have been used to set up four different cases to see how the models behave in different conditions. Data for identification of the parameters have not been used in the validation to get more reliable results.

Case A. Data from the engine with a VGT has been used. One VGT position was used for the identification and the other two was used for the validation. This case can be used to see if the model can extrapolate when the VGT position is changed. A good result would show that the model has a physically correct behavior for the pressure quotient $\frac{p_{em}}{p_{im}}$.

Case B. Data from the engine with a VGT has been used. From all three VGT positions, every second points along the lines was used for the identification and the other ones for the validation. Thus data from the whole range of available data are used both for identifying the parameters and the validation, but no point is used booth for the identification and the validation. This case shows how good the result can be in the whole range of VGT positions.

Case C. Data from the engine with an EGR system and no VGT has been used. The larger data matrix has been used to identify the parameters for each engine speed. Data from the smaller data matrix was used for the validation. This case shows how good the result can be on an engine without a VGT.

Case D. Data from the engine with an EGR system and no VGT has been used. The larger data matrix has been used to identify the parameters for four engine speeds. The data for all other speeds have been used for the validation. This case shows how good the linear interpolation between different engine speeds are.

In Table 5.1 results for each model and case are presented.

5.2 General Comments

When studying the results it is important to have in mind that the resolution in the measurement of volumetric efficiency often is in the same order of size as the model error. The resolution is clearly seen in the transients as quanted distinct lines. Figure 5.1 shows the phenomena. In the plots with the model errors the resolution also is plotted.

Another observation made is that the model error often is worse in the region where the pressure quotient $\frac{p_{em}}{p_{im}}$ is around 1 or below. In Section 2.4.3 the problem with the pulsating exhaust pressure is discussed. In the region where the quotient is near one, the error in p_{em} can change the effect of back flow and residual gas from negative to positive and vice versa. The observed higher errors in this region may come from this measurement problem. Figure 5.2 shows the typical appearance of the model error.

Case	ICM Mean abs. err.[%]	ICM Max abs. err.[%]	RM Mean abs. err.[%]	RM Max abs. err.[%]	HTM Mean abs. err.[%]	HTM Max abs. err.[%]
A	1.06	2.33	0.81	1.92	0.68	2.08
B	0.69	3.12	0.18	0.65	0.31	0.99
C	0.58	1.53	1.86	11.11	0.33	0.98
D	0.51	1.79	–	–	0.67	1.60
Case	**ORM Mean abs. err.[%]**	**ORM Max abs. err.[%]**	**AM Mean abs. err.[%]**	**AM Max abs. err.[%]**	**RESOLUTION Mean res.[%]**	**RESOLUTION Max res.[%]**
A	0.50	0.94	0.59	1.96	0.14	0.32
B	0.20	0.51	0.33	1.58	0.19	0.39
C	0.44	1.05	0.66	2.57	0.13	0.80
D	0.74	2.39	0.53	2.18	0.20	1.15

Table 5.1: *Static validation data for all models. Resolution shows the resolution in the measurements of volumetric efficiency for the validation data. The unit of volumetric efficiency is percentage. All numbers are absolute errors. The models are: Ideal Cycle Model (ICM), Regression Model (RM), Heat Transfer Model (HTM), Overlap and Residual Gas Model (ORM) and Another Model (AM).*

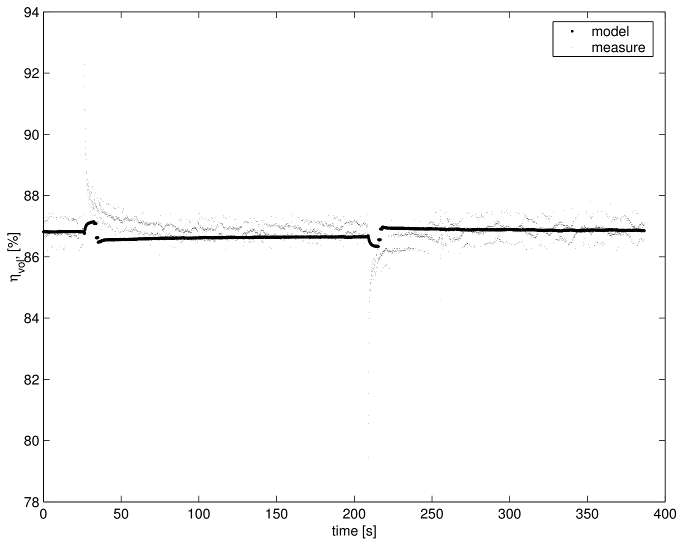

Figure 5.1: *Shows how the resolution can be seen in the transient measurements.*

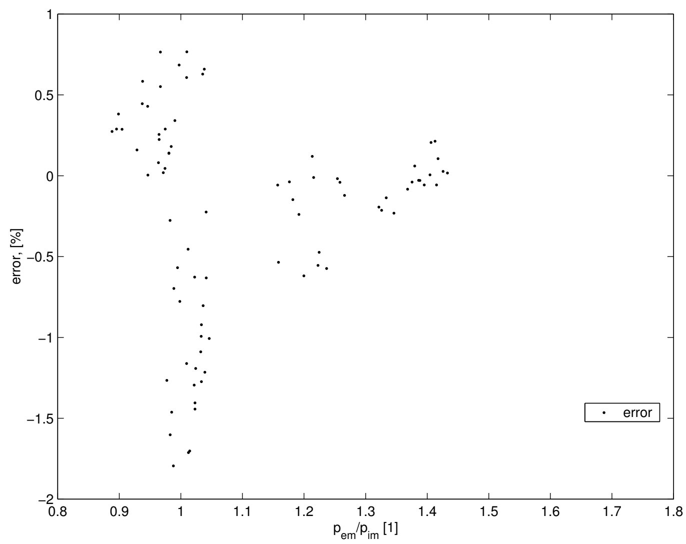

Figure 5.2: *Shows a typical distribution of the errors.*

The mean absolute error used in this thesis is the 1-norm of the error η_{err} defined by

$$\eta_{err} = \eta_{volMe} - \eta_{volSi}. \tag{5.1}$$

Here η_{volMe} is the measured volumetric efficiency and η_{volSi} is the simulated volumetric efficiency.

The maximum absolute error used in this thesis is the ∞-norm of η_{err}.

As seen in Table 5.1 Case D, linear interpolation between model parameters for different engine speeds works for all models but the Regression Model. The errors for the other models increases moderately. In Case D four engine speeds has been used for the model. How many that are preferred and which engine speeds to use are not concerned in this thesis.

Besides the four models presented in the previous chapter data for a fifth model, not presented in this thesis, are included. The model is called Another Model (AM).

5.3 Ideal Cycle Model

Static validation results for the Ideal Cycle Model (ICM) are presented in Table 5.1. The model gives better results for the engine with a standard turbine than the engine equipped with a VGT.

In Figure 5.3 (a) and (b) the result from Case A is plotted. As seen the simulated points behave in a different way then the measured one. This shows that the model has not captured the correct behavior of the engine with a VGT.

Figure 5.3 (c) and (d) shows the result from Case B. In this case the result is acceptable. But some simulated points differ significantly from the measured ones. A conceivable reason for that and a problem with this model are that with a VGT, a low inlet manifold pressure, p_{im}, can belong to any VGT position. The implementation of the model uses a look-up table, which needs a monotone function to work. In order to get a monotone function in p_{im}, the function sorts the result in a monotone order. Thus the function will mix constants from different VGT positions. Interpolating between two p_{im} values in the table may be valid for a completely other VGT position than the actual one.

The conclusion is that this model not is suitable for engines with a VGT. It may be possible to solve the problem by changing the variable used in the look-up table, but this has not been tested yet.

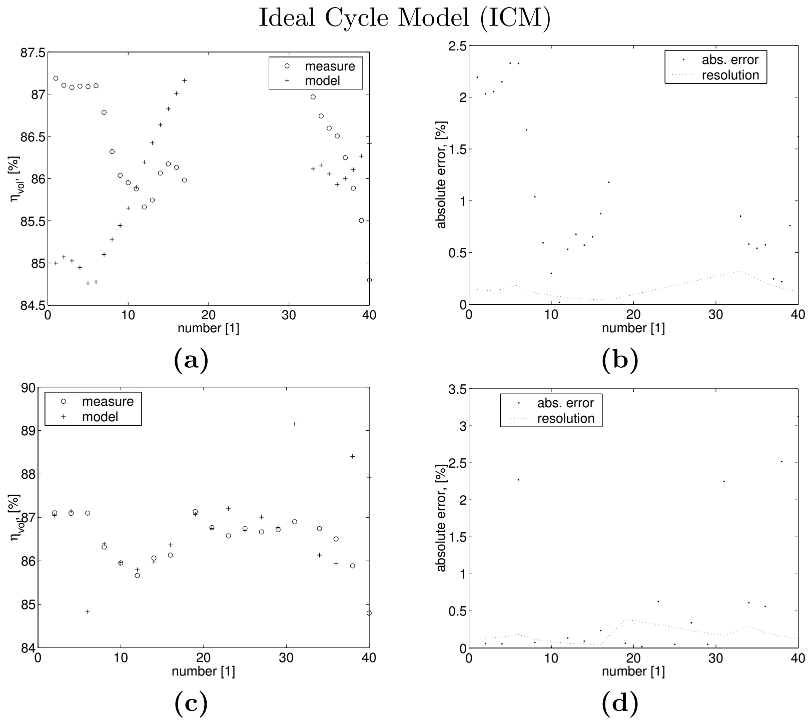

Figure 5.3: *Validation plots for the Ideal Cycle Model. (a) and (b) are from Case A. (c) and (d) are from Case B.*

5.4 Regression Model

Static validation results for the Regression Model (RM) are presented in Table 5.1. The model has shown an instable tendency during test. It can describe volumetric efficiency very good in small regions and give absurd results if the conditions are changed a little. In case D no results are presented. The reason is that the model failed to interpolate between the different engine speeds and gave absurd results. The mean absolute error was 18 000 % and the maximum absolute error was 93 000 %. The model has six parameters, which seems to have no physically coupling, i.e. the model is over parameterized.

In Figure 5.4 (a) and (b) the result from Case A is plotted. In the left half of plot (a) the behavior from the simulation and the measured volumetric efficiency do not match. In the right half the result is good. This agrees to the reasoning above. The points in the right half lies near the ones used for identifying the parameters. In Figure 2.3 the line in the middle is used for identifying the parameters. The points in the right half of plot (a) corresponds to the line to the left in Figure 2.3.

Figure 5.4 (c) and (d) shows the result from Case B. In this case points used to identify the parameters and the validation are not the same, but both are from the whole range of measurements. The result in this case is good.

Within small regions the model can describe the volumetric efficiency well, but if some condition is changed the model likely gives a poor result. This is not a desirable quality.

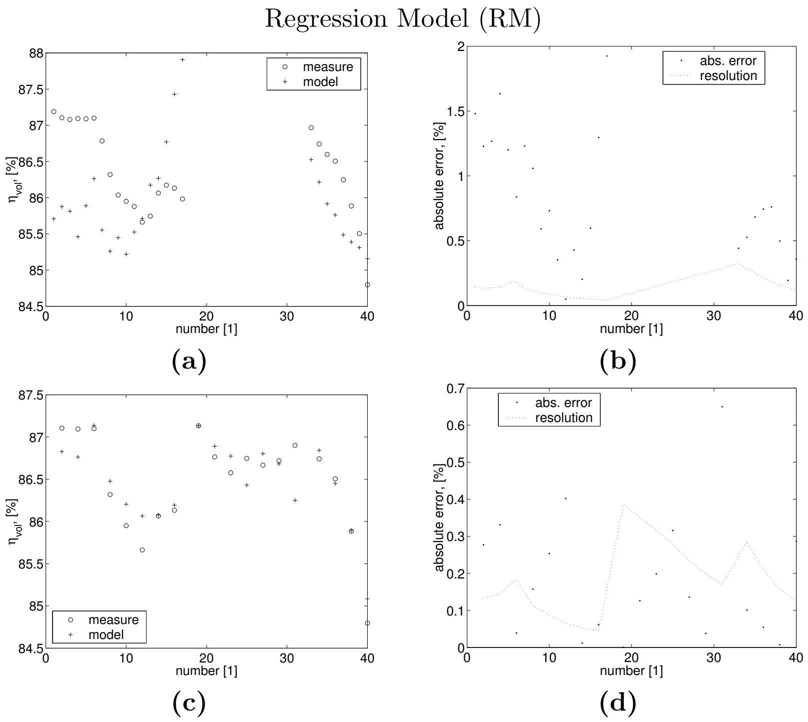

Figure 5.4: *Validation plots for the Regression Model. (a) and (b) are from Case A. (c) and (d) are from Case B.*

5.5 Heat Transfer Model

Static validation results for the Heat Transfer Model (HTM) are presented in Table 5.1. The model shows good results for the engine with a standard turbine and fair results for the engine equipped with a VGT. But in Figure 5.5 (a) and (b) the result from Case A is plotted. The simulated points do not follow the measured. Thus the model has not captured the physical behavior of the volumetric efficiency for the engine equipped with a VGT. Figure 5.5 (c) and (d) shows the result from Case B.

The model used for predicting the temperature rise is very simple. A dynamic heat transfer model is not yet tested, but could perhaps improve the results. Also the dynamic effects that has been seen in the volumetric efficiency can be described with such a model.

Heat Transfer Model (HTM)

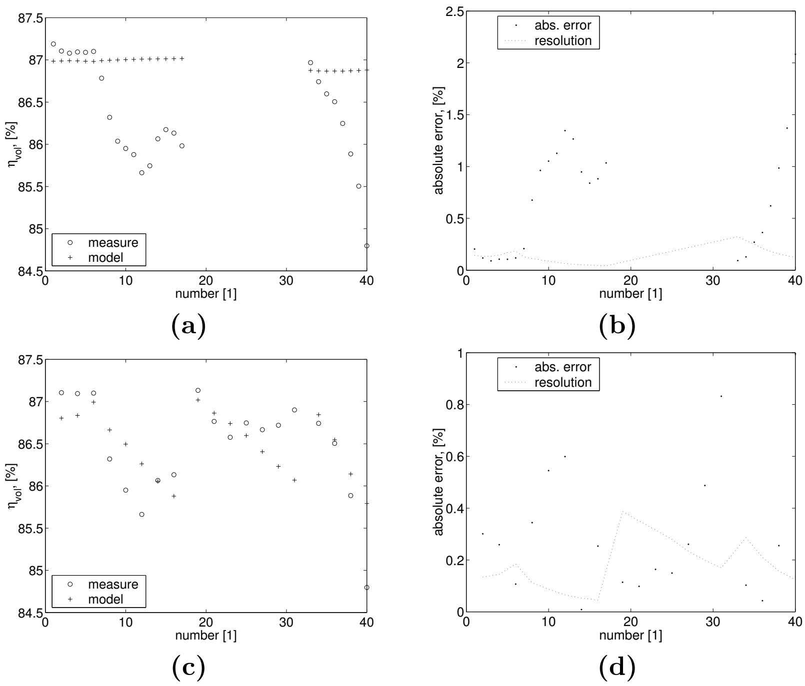

Figure 5.5: *Validation plots for the Heat Transfer Model. (a) and (b) are from Case A. (c) and (d) are from Case B.*

5.6 Overlap and Residual Gas Model

Static validation results for the Overlap and Residual Gas Model (ORM) are presented in Table 5.1. The model shows an over all good behavior. The mean absolute errors in all cases are low. This model has shown the best results for the VGT equipped engine.

In Figure 5.6 (a) and (b) the result from Case A is plotted. The result from the simulation follows the measured fairly well. Note that case A imply extrapolating to VGT positions way out of the range where the parameters were identified. Figure 5.6 (c) and (d) shows the result from Case B. This gives confidence to the models ability to captured the physical behavior of a VGT equipped engine.

In the test where the model extrapolates volumetric efficiency in other VGT positions this model is the only model that shows fair result.

The model has shown to be stable in the sense that the results not change much if the data used for identifying the parameters or the validation data are changed. The three parameters used in the model vary in reasonable ranges.

If the model is parameterized using data from the whole engine operating range, the absolute model errors are not much higher then the resolution of the measured volumetric efficiency. Thus much better results can not be expected without improving the resolution of the measurements.

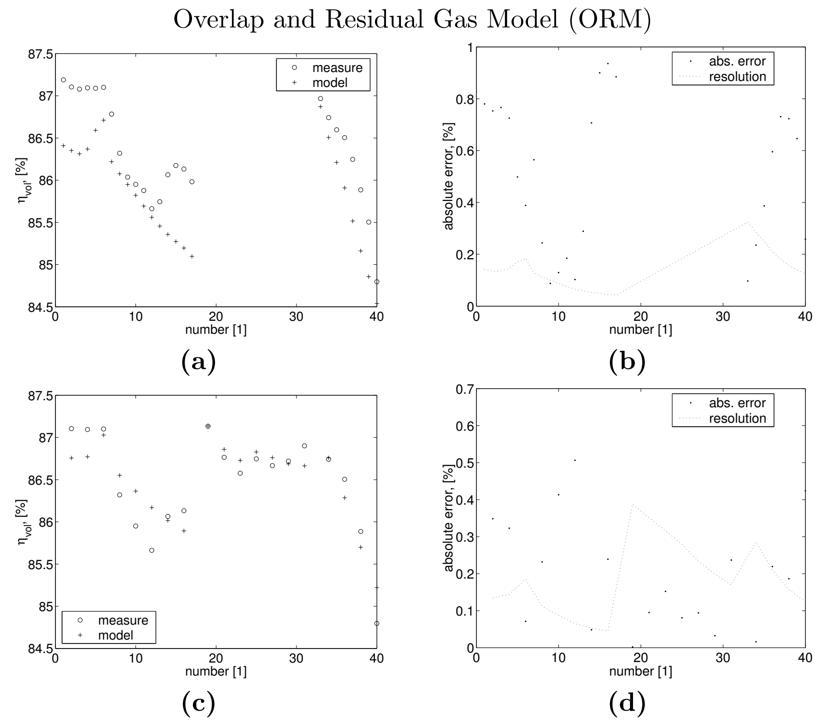

Figure 5.6: *Validation plots for the Overlap and Residual Gas Model. (a) and (b) are from Case A. (c) and (d) are from Case B.*

5.7 Another Model

Besides the four models presented in Chapter 4 results from a fifth model developed at Scania, here called Another Model (AM), are presented in Table 5.1.

Static validation results for Another Model (AM) are presented in Table 5.1. The model shows high errors for all cases.

In Figure 5.7 (a) and (b) the result from Case A is plotted. As seen the simulated points behave in a different way then the measured ones. This indicates that the model has not captured the correct behavior of the engine equipped with a VGT. Figure 5.7 (c) and (d) shows the result from Case B.

The structure of Another Model is not presented in this thesis, but the results are calculated in the same way as the other models. Thus the only conclusion drawn is that a model that shows better results then Another Model is presented in this thesis.

Another Model (AM)

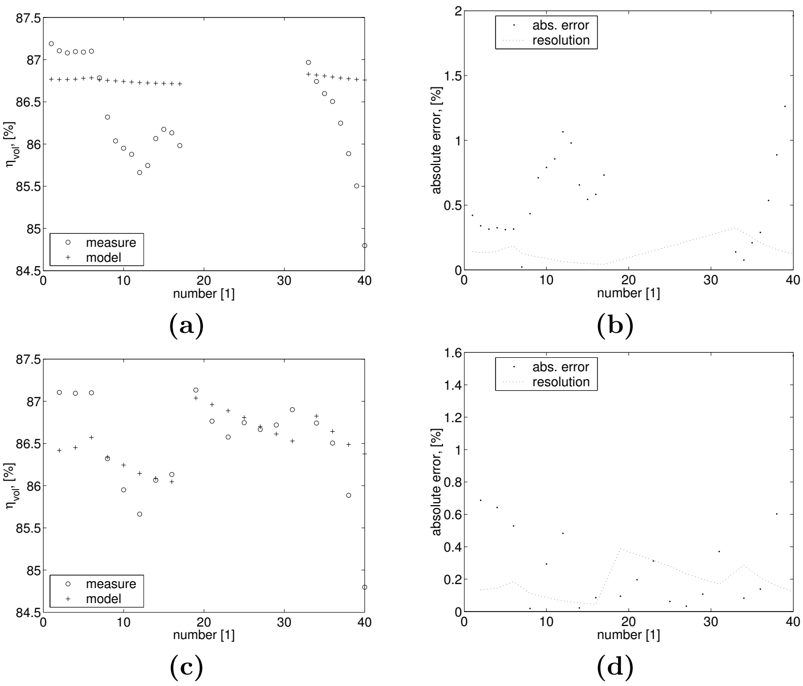

Figure 5.7: *Validation plots for Another Model. (a) and (b) are from Case A. (c) and (d) are from Case B.*

5.8 Summary

In this chapter the models presented in the previous chapter has been validated to see how well they describe the real engine. Also data for a fifth model are included.

The Ideal Cycle Model (ICM) shows good results in the validation. It works best if the engine does not has a VGT. The reason for that is that a part of the model mixes the behavior from different VGT positions.

The Regression Model (RM) has shown an instable tendency. The parameters vary in ranges that not are possible to explain physically. Probably this model is over parameterized. In small regions the model can describe the behavior of volumetric efficiency for the engine equipped with a VGT well, but it fails to extrapolate.

The Heat Transfer Model (HTM) has shown fair results on the VGT engine. But it was not able to extrapolate from a VGT position to another. On the engine with a standard turbine the result is very good.

The Overlap and Residual Gas Model (ORM) has shown an over all good behavior. The results are good for all cases. It has shown to be stable and is the only model that has been able to extrapolate to other VGT positions fairly well.

Another Model (AM) shows high errors for all cases. The behavior of the engine equipped with a VGT is not captured with this model.

The best model among the five studied is the Overlap and Residual Gas Model. If the model is parameterized using data from the whole engine operating range, the absolute model errors are not much higher then the resolution of the measured volumetric efficiency. Thus much better results can not be expected without improving the resolution of the measurements.

Chapter 6

Conclusions and Future Extensions

In this chapter the results obtained in this thesis will be discussed. In Section 6.1 conclusions from the results are drawn. In Section 6.2 suggestions for future extensions are presented. Finally Section 6.3 discusses the fulfilments of the objectives for this thesis.

6.1 Conclusions

The model partly derived in this thesis, called the Overlap and Residual Gas Model (ORM), is suggested as the model of volumetric efficiency. Results and observations for the other four models are presented in Chapter 5.

In Chapter 5 the validation shows that ORM is the only model among the five studied that is able to extrapolate the volumetric efficiency for other VGT control signals. This is one of the most important qualities to look for, since the purpose with this work is to study volumetric efficiency models for engines equipped with a VGT. The model has shown to be stable in the sense that the results not change much if the data used for identifying the parameters or the validation data are changed. The three parameters used in the model vary in reasonable ranges.

In all validation cases the model gives small absolute errors. The results are the best among the five studied models. If the model is parameterized using data from the whole engine operating range, the absolute model errors are not much higher then the resolution of the measured volumetric efficiency. Thus much better results can not be expected without improving the resolution of the measurements.

The model describes the volumetric efficiency for one engine speed. By estimating parameters for a set of engine speeds and interpolate between these a complete model that works on all engine speeds is archived. The validation shows that this approach works well.

6.2 Future Extensions

The time available for this thesis is limited. Thus all ideas and problem have not been finished. In this section two proposals for future extensions of the Overlap and Residual Gas Model are presented.

6.2.1 Exhaust Pressure

In the models studied in this thesis the exhaust pressure, p_{em}, in the exhaust manifold is used as variable. Even in the engine lab only the mean value is measured. In the trucks today there is no sensor for the exhaust pressure. The solution must be either to use a model of the exhaust manifold pressure or adding a sensor.

The mean pressure is probably not the best variable to use in the models. A possible improvement would be to add a model of the exhaust manifold pressure just before the exhaust valve closes. The result plotted in Figure 5.2 shows that such a model could decrease the error around the point where the inlet and the exhaust manifold pressures are equal.

6.2.2 Heat Transfer

In the engine lab the environment is relative constant. Experiments with the purpose to affect temperature in the inlet manifold were not successive. It was not possible to change the inlet manifold temperature more than some degree. Thus effects from changes in the inlet manifold temperature have not been possible to examine. To see if the result is good at other temperature conditions more tests are needed. In engine lab there is a special equipped cell where the temperature can be controlled, which may be used for this purpose. Also data from tests in a real truck at different weather conditions would be a good material to use.

An idea during this thesis has been to implement a dynamic model of the cylinder wall temperature. Such a model could be used to model the heat transfer from the metal to the cylinder charge. The data needed to identify the parameters was not available and thus the idea has not been tested.

This is probably the best way to include the dynamic effects from the heat transfer that can be seen when changing the load in large steps. The inlet manifold temperature could then be compensated dynamically using such a model.

Such a model could be used together with the Overlap and Residual Gas Model. The last variable in the formula could be modified in a similar way that is used in the Heat Transfer Model, by adding a factor $\frac{1}{1+\left(\frac{\Delta T}{T_{im}}\right)}$.

6.3 Objective Fulfilments

The objectives for this thesis are to:

- Evaluate a couple of existing models of volumetric efficiency with the focus on how they work with engines equipped with a VGT.
- Develop a model that works better when pressures can be varied more freely with a VGT.
- Real engine measurements should be used to identify the parameters and for the validation.

Three existing models have been evaluated, with the focus on how they work with an engine equipped with a VGT. First point in the objective thus is fulfilled.

Two alternative models have been developed and evaluated together with the other models. Both the models show good results. The Overlap and Residual Gas Model seems to have captured most of the physical behavior for the engine equipped with a VGT. Thus the second point in the objective is fulfilled.

Data from two real engines have been used for identifying the parameters and the validation. Thus all objectives are fulfilled for this thesis work.

References

[1] M. Adolfsson. Simulation of emisson related faults. Master's thesis LiTH-ISY-EX-3266, Department of Electrical Engineering, Linköpings Universitet, Linköping, Sweden, 2002.

[2] J. Biteus. MVEM of DC12 Scania engine. Internal Scania report, May 2002.

[3] D. Elfvik. Modeling of a diesel engine with vgt for control design simulations. Master's thesis, KTH, Stockholm, Sweden, july 2002.

[4] R. Scattolini G. De Nicolao and C. Siviero. Modelling the volumetric efficiency of ic engines: Parametric, non-parametric and neural techniques. *Control Eng. Practice*, 4(10):1405–1415, 1996.

[5] J.B. Heywood. *Internal Combustion Engine Fundamentals*. McGraw-Hill, Singapore, 1988.

[6] W.K. Cheng J.W. Fox and J.B. Heywood. A model for predicting residual gas fraction in spark-ignition engines. *SAE paper No. 931025*, 1993.

[7] T. Fickenscher L.A. Smith and R.P. Osborne. Engine breathing - steady speed volumetric efficiency and its validity under transient engine operation. *SAE paper No. 1999-01-0212*, 1999.

[8] L. Nielsen and L. Eriksson. *Course Material Vehicular Systems*. Linköping, Sweden, 2000.

[9] M. Nilsson. Mätning av specifika emissioner i en transient körcykel utan cvs. Master's thesis MMK 2000:83 MFM57, Department of Machine Design, KTH, Stockholm, Sweden, 2000. In Swedish.

[10] K. Roussopoulos. A convenient technique for determining comparative volumetric efficiency. *SAE paper No. 900352*, 1990.

[11] H.B. Servati and R.G. DeLosh. A regression model for volumetric efficiency. *SAE paper No. 860328*, 1986.

[12] P. Skogtjärn. Modelling of the exhaust gas temperature for diesel engines. Master's thesis LiTH-ISY-EX-3378, Department of Electrical Engineering, Linköpings Universitet, Linköping, Sweden, 2002.

[13] C.F. Taylor. *The Internal Combustion Engine in Theory and Practice*, volume Vol. 1. The M.I.T. Press, USA, 2nd ed. edition, 1985.

[14] C.F. Taylor. *The Internal Combustion Engine in Theory and Practice*, volume Vol. 2. The M.I.T. Press, USA, 2nd ed. edition, 1985.

Notation

Notations consist of three tables. Table 6.1 defines the variables and constants used in this report. In Table 6.2 all indices are presented. In Table 6.3 abbreviations and notions used in this report are explained.

Table 6.1: *Variables and constants used in this report.*

Variable /constant	Explanation	Unit
T_x	Temperature in component x	K
p_x	Absolute pressure in component x	Pa
W_x	Mass-flow through component x	kg/s
m_x	Mass of gas in the component	kg
ρ_x	Density in component x	kg/m^3
R	Gas constant	$J/(kg \cdot K)$
γ	Specific heat ratio	$-$
n_{eng}	Engine speed	rpm
M_{eng}	Engine torque	Nm
δ	Mass of injected diesel per cycle	$kg/stroke$
u_{vgt}	VGT control signal	$[0,1]$
χ_{res}	Residual gas fraction	$[0,1]$
r_c	Compression ratio	$-$
V_d	Displacement volume (1 cylinder)	m^3
V_c	Clearance volume (1 cylinder)	m^3
η_{vol}	Volumetric efficiency	%
λ	Air-to-fuel equivalence ratio	$-$
T_s	Cylinder wall surface temperature	K
T_g	Mean gas temperature in the cylinder	K
ρ_a	Ambient gas density, defined at exhaust pressure and inlet temperature.	kg/m^3
η_{err}	Difference between measured and simulated volumetric efficiency	%
η_{volMe}	Measured volumetric efficiency	%
η_{volSi}	Simulated volumetric efficiency	%

Index x is a general index chosen from Table 6.2.

Table 6.2: *Indices used in this report.*

Index	Explanation
af	Air Filter
atm	Atmosphere
cCh	Combustion Chamber
cEgr	EGR in Combustion Chamber
cIvc	In Combustion Chamber at IVC
cIvo	In Combustion Chamber at IVO
cMix	Fresh Mixture in Combustion Chamber
co	Compressor
cRes	Residual Gas in Combustion Chamber
egr	EGR system
em	Exhaust Manifold
ePo	Exhaust Port
es	Exhaust System
eVa	Exhaust Valve
ic	Intercooler
im	Inlet Manifold
imEgr	EGR in Inlet Manifold
iPo	Inlet Port
iVa	Inlet Valve
tu	Turbine

Table 6.3: *Abbreviations and notions used in this report.*

Abbreviation /notion	Explanation
VGT	Variable Geometry Turbine
VNT	Variable Nozzle Turbine
rpm	Revolutions Per Minute
EGR	Exhaust Gas Recirculation
MVEM	Mean Value Engine Model
OBD	On-board diagnostics
CI	Compression-Ignition
SI	Spark-Ignition
TDC	Top Dead Center
BDC	Bottom Dead Center
IVO	Inlet Valve Opening
IVC	Inlet Valve Closing
EVO	Exhaust Valve Opening
EVC	Exhaust Valve Closing
ORM	Overlap and Residual Gas Model
HTM	Heat Transfer Model
RM	Regression Model
ICM	Ideal Cycle Model
AM	Another Model
OF	Overlap Factor
Residual gas	Exhaust gas trapped in the cylinder after the exhaust valves are closed
Charge	Contents in the cylinder when all valves are closed

Appendix A

Heat Transfer Model

In this appendix the model presented in Section 4.3 is derived. The model is based on the indicator diagram and involves heat transfer.

In Section A.1 an existing model is presented. Section A.2 specifies the assumptions and the simplifications made. In Section A.3 the final Heat Transfer Model is presented.

A.1 Volumetric Efficiency from Spring Diagram

In [13] a model based on the spring diagram is derived

$$\eta_{vol} = \frac{1}{1 + \left(\frac{\Delta T}{T_{im}}\right)} \left(\frac{\alpha\,(\gamma - 1)}{\gamma} + \frac{\frac{p_{cIvc} y}{p_{im}} r_c - \frac{p_{cIvo}}{p_{im}}}{\gamma\,(r_c - 1)} \right), \tag{A.1}$$

where α is the ratio of actual work on the piston, defined by

$$\alpha = \frac{1}{p_{im} V_d} \int_x^y p_{cCh}\, dV. \tag{A.2}$$

Also x and y are defined by $x = \frac{V_{cIvo}}{V_d + V_c}$ and $y = \frac{V_{cIvc}}{V_d + V_c}$, where V_{cIvo} and p_{cIvo} are the volume and the pressure in the cylinder at IVO. The pressure in the combustion chamber is denoted p_{cCh}. In the same way V_{cIvc} and p_{cIvc} are the volume and the pressure in the cylinder at IVC.

In this model all effects from the heat transfer are merged into ΔT, which is defined as

$$\Delta T = T_{cCh} - T_{im}. \tag{A.3}$$

Here T_{cCh} is the gas temperature in the combustion chamber when the inlet valve closes.

A.2 Assumptions and Simplifications

The model in A.1 can be simplified if ideal gas and ideal process with heat transfer are assumed. In the ideal process [13]:

- The process follows the Otto cycle.
- Both fresh mixture and residual gas are ideal gases with the same specific heat and molecular weight.
- Inlet pressure, p_{im}, is constant.
- Inlet temperature, T_{im}, is constant.
- Exhaust pressure, p_{em}, is constant.

In the ideal process IVC occurs at BDC, which imply that $y = 1$. Also the ratio of actual work on the piston is constant. In the ideal process also $\frac{p_{cIvcy}}{p_{im}} = 1$ and $\frac{p_{cIvo}}{p_{im}} = \frac{p_{em}}{p_{im}}$ are valid [13].

The parameters are identified for one specific engine in this thesis. Therefore the specific heat ratio, γ, and the compression ratio, r_c, can be assumed constant.

A.3 The Model

By using the assumptions and the simplifications specified in the previous section, the model becomes

$$\eta_{vol} = \frac{1}{1 + \left(\frac{\Delta T}{T_{im}}\right)} \left(\eta_0 - \eta_1 \left(\frac{p_{em}}{p_{im}}\right)\right). \tag{A.4}$$

Here η_0 and η_1 are constants.

Appendix B

Overlap and Residual Gas Model

In this appendix the model presented in Section 4.4 is derived. The effect of residual gas and valve overlap are included in this model.

In Section B.1 an expression relating the volumetric efficiency and the residual gas fraction is derived. Section B.2 presents an existing model of the residual gas fraction. In Section B.3 the Overlap and Residual Gas Model is derived from the result in previous sections.

B.1 Volumetric Efficiency and Residual Gas Fraction

In (2.3) the volumetric efficiency has been defined. The mass trapped in the cylinder, m_{cCh}, is the sum of fresh mixture, EGR and residual gas which gives

$$m_{cCh} = m_{cMix} + m_{cEgr} + m_{cRes}. \tag{B.1}$$

Here m_{cRes} is the mass of residual gas trapped in the cylinder, m_{cMix} is the mass of fresh mixture trapped and m_{cEgr} is the mass of EGR in the cylinder.

Residual gas fraction is the amount of residual gas that is trapped in the cylinder, which is

$$\chi_{res} = \frac{m_{cRes}}{m_{cCh}}. \tag{B.2}$$

Combining (B.1) with (B.2) gives

$$m_{cMix} + m_{cEgr} = (1 - \chi_{res})\, m_{cCh}, \tag{B.3}$$

which expresses how much of the total mass trapped in the cylinder that not is residual gas.

Now by combining (B.3) with the definition of volumetric efficiency in (2.3), the expression becomes

$$\eta_{vol} = (1 - \chi_{res}) \frac{m_{cCh}}{\rho_{im} V_d}. \tag{B.4}$$

This is an expression for volumetric efficiency depending on the residual gas fraction.

In (B.4) the quotient $\frac{m_{cCh}}{\rho_{im} V_d}$ can be simplified as a constant. This means that if the combustion chamber was empty, i.e. at vacuum, the mass inducted into the cylinder is linear to the density of the air in the inlet manifold. Formula (2.6) is valid for flow through the inlet valve. If the flow not has reached critical flow the linear approximation is acceptable. Expression (B.4) then becomes

$$\eta_{vol} = k' (1 - \chi_{res}), \tag{B.5}$$

which relates the volumetric efficiency and the residual gas fraction. But now an expression of χ_{res} is needed.

B.2 A Model of Residual Gas Fraction

In [6] a model of the residual gas fraction is described, which consider valve overlap and the residual gas trapped in the cylinder. The model is tested on a SI engine. But there is no reason why it would not work on a CI engine as well.

B.2.1 Overlap Factor

The back flow during the valve overlap is difficult to model. In the model of residual gas fraction a factor is used to measure how much overlap there is. This is called the overlap factor (OF) and is defined by

$$OF = \frac{D_i A_i + D_e A_e}{V_d}, \tag{B.6}$$

where D_i and D_e are the inner seat diameters of the intake and the exhaust valves. Also A_i and A_e are defined by

$$A_i = \int_{IVO}^{IV=EV} L_i d\theta \tag{B.7}$$

and

$$A_e = \int_{IV=EV}^{EVC} L_e d\theta. \tag{B.8}$$

Here L_i and L_e are the actual valve lift. The curves showing these are shown in Figure 2.1. Figure 2.7 shows the integral areas.

B.2.2 Complete Residual Gas Fraction Model

The complete model of the residual gas fraction is

$$\chi_r = C_1 \frac{\left(\frac{OF}{n_{eng}}\right)\left(\frac{p_{em}}{p_{im}}\right)^{\frac{\gamma+1}{2\gamma}}\sqrt{\frac{|p_{em}-p_{im}|}{\rho_a}}}{\operatorname{sign}\left(p_{em}-p_{im}\right)} + C_2 \frac{\lambda}{r_c}\left(\frac{p_{em}}{p_{im}}\right)^{\frac{1}{\gamma}}. \tag{B.9}$$

Here λ is the air-to-fuel equivalence ratio and ρ_a is the ambient gas density defined at exhaust pressure and inlet temperature.

B.3 Complete Volumetric Efficiency Model

Combining (B.5) and (B.9) gives

$$\eta_{vol} = k' \left(1 - C_1 \frac{\left(\frac{OF}{n_{eng}}\right)\left(\frac{p_{em}}{p_{im}}\right)^{\frac{\gamma+1}{2\gamma}}\sqrt{\frac{|p_{em}-p_{im}|}{\rho_a}}}{\operatorname{sign}\left(p_{em}-p_{im}\right)} - C_2 \frac{\lambda}{r_c}\left(\frac{p_{em}}{p_{im}}\right)^{\frac{1}{\gamma}}\right). \tag{B.10}$$

The approach in this thesis is to build a model that is valid for one engine speed only. Thus n_{eng} is constant. The overlap factor (OF) and the compression ratio r_c also are seen as constants in this thesis. Applying this on (B.10) gives

$$\eta_{vol} = k_1 - k_2 \left(\frac{p_{em}}{p_{im}}\right)^{\frac{\gamma+1}{2\gamma}} \frac{\sqrt{\frac{|p_{em}-p_{im}|}{\rho_a}}}{\operatorname{sign}\left(p_{em}-p_{im}\right)} - k_3 \lambda \left(\frac{p_{em}}{p_{im}}\right)^{\frac{1}{\gamma}}, \tag{B.11}$$

which is the final Overlap and Residual Gas Model.