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A GENeric Internal combustion Engine model

LiU-Genie

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Sammanfattning

Eftersom efterfrågan på reglering i moderna fordon växer behövs modeller för att utveckla bättre förståelse för processer och motorbeteende. Detta exjobb bidrar med en generisk utökad MVEM-motormodell med SI och CI samt VVA-kompatibilitet med modelleringsstruktur i MATLAB / Simulink-miljö.

Modellen innehåller motorns komponenter luftfilter, kompressor med bypassventil, intercooler, trottle, insugsgrenrör, cylinder, avgasgrenrör, EGR, turbin med wastegate och avgassystem. Modellen håller reda på tryck, massflöden, temperaturer och syrekoncentrationer.

I denna rapport presenteras modeller, parametrisering och de signaler som behöver mätas för att göra det möjligt att parametersätta andra motorer med samma metod. För att validera modellen används en 6-cylindrig 12,7 liters Scania-dieselmotor och en 4-cylindrig 2-liters Volvo bensinmotor. Modellen visar överensstämmelse med uppmätta data för tryck, massflöde, temperatur och syrekoncentration på flera driftpunkter. Det visar att modellen kan hantera de flesta motorer med enbart små justeringar. Den komponentbaserade modellstrukturen gör det också möjligt att ändra och justera de modeller som används till exempel med hjälp av den utökade elliptiska modellen för kompressoreffektivitet eller implementera en dubbel Vibe-funktion för förbränningsberäkningar.

Abstract

As the demand for control in modern vehicles grows, models are needed to develop a better understanding of processes and engine behavior. The thesis contributes with a generic extended MVEM with SI and CI as well as VVA compatibility and modeling structure using MATLAB/Simulink environment.

The model contains the engine components air filter, compressor with bypass valve, intercooler, throttle, intake manifold, cylinder, exhaust manifold, EGR, turbine with a wastegate, and exhaust system, keeping track of pressures, mass flows, temperatures, and oxygen concentrations.

This thesis presents models, parametrization, and signals that needed to be measured to make it possible to parametrize other engines with the same method. A 6-cylinder 12.7 liter Scania diesel engine and a 4-cylinder 2 liter Volvo petrol engine are used to validate the model. The model shows agreement with measured data for pressure, mass flow, temperature, and oxygen concentration on several operational points. That shows the model can handle most engines with just small adjustments. The component-based model structure also enables the possibility to change and adjust the models used by, for example using the extended ellipse model for compressor efficiency or implement a double Vibe function for combustion calculations.

Acknowledgments

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Contents

Notation

Constants

Constant	Description
$R = 8.3143$	Gas constant $[J/molK]$
$R_{air} = 287$	Ideal gas constant for air $[J/KgK]$
$R_{exh} = 280$	Ideal gas constant for exhaust gas $[J/KgK]$
$\gamma_{air} = 1.4$	Specific heat constant for air [-]
$\gamma_{exh} = 1.3$	Specific heat constant for exhaust gas [-]
$cp_{air} = 1005$	Specific heat capacity for air [-]
$cp_{cetane} = 1213.3$	Specific heat capacity for exhaust gas [-]
$cp_{exh} = 1213.3$	Specific heat capacity for exhaust gas [-]
$cp_{isooctane} = 1005$	Specific heat capacity for air [-]
$Q_{lhv,CI} = 44.651e+6$	Lower heating value for CI$[J/Kg]$
$Q_{lhv,SI} = 41.300e+6$	Lower heating value for SI$[J/Kg]$
$A/Fs = 14$	Stochiometric air/fuel ratio $[-]$

Abbreviations

Abbreviation	Description
BDC	Bottom Dead Center For Piston Position
CI	Compression Ignition Engine
CRB	Compression Release Brake
DI	Direct Injection
ECU	Electronic Control Unit
EGR	Exhaust Gas Recirculation
EVO	Exhaust Valve Open
HIL	Hardwear In the Loop
ICE	Internal Combustion Engine
IVC	Intake Valve Close
IVO	Intake Valve Open
$MVEM$	Mean Value Engine Model
PID	Proportional–Integral–Derivative controller
SI	Spark Ignition Engine
SOI	Start of Ignition/Injection on SI/CI engine
TDC	Top Dead Center For Piston Position
TFP	Turbine Flow Parameter
TSP	Turbine Speed Parameter
VGT	Variable Geometry Turbo
VVA	Variable Valve Actuation
VVT	Variable Valve Timing
WG	Wastegate

Subindicies

Index	Description
amb	Ambient
af	Air filter
bc	Before compressor
ac	After compressor
bic	Before intercooler
aic	After intercooler
im	Intake manifold
em	Exhaust manifold
bt	Before turbine
at	After turbine

Variables

Variable	Description
$\dot{m}_i$	Mass-flow on subindex i $[kg/s]$
η_i	Efficiency of index i $[-]$
$\gamma = \frac{c_p}{c_v}$	Specific Heat Ratio $[-]$
ω_i	Angular velocity of index i $[rad/s]$
θ	Crank angle $[rad]$
u	Control Signal $[-]$
a	Crank Radius $[m]$
A_E	Effective Area $[m^2]$
B	Cylinder Bore $[m]$
C_D	Discharge Coefficient $[-]$
c_v	Heat capacity, constant volume $[J/K]$
c_p	Heat capacity, constant pressure $[J/K]$
F	Mass fraction burned $[-]$
f_i	Function for index i
H	Enthalpy $[J]$
L	Piston Stroke $[M]$
l	Connection Rod Length
L_v	Valve lift $[m]$
m_i	Mass on index i $[kg]$
n_{cyl}	Number Of Cylinders $[-]$
p_i	Pressure on index i $[Pa]$
Π_i	Pressure ratio on index i $[-]$
Q_i	Heat energy on index i $[J]$
r_c	Compression Ratio $[-]$
T_i	Temperature on index i $[K]$
U	Internal energy $[J]$
V_c	Clearance Volume $[m^3]$
V_D	Total Engine Displacement Volume $[m^3]$
V_d	Displacement Volume $[m^3]$
W	Work $[J]$

1 Introduction

As control has come to play a more important role in the development of modern vehicles, a good model is needed to get a good base for controller testing. In this chapter, the introduction to this thesis is presented.

1.1 Background

Most engine models are developed with a specific purpose in mind. This means a new engine model has to be developed each time a new engine is investigated or a new fuel is used. This engine model has been developed with the possibility to easily change component equations in mind. This means each engine component is presented with its own equation, parameter estimation and validation. Some components share characteristics and the same equations are used for those components.

1.2 Objectives

The objective of this thesis is to create a generic engine model that captures the dynamics in both SI and CI engines with VVA. The goal is to have a model which clearly indicates how to parameterize the data needed to run the engine model. The model will simulate the engine behavior from the air inlet to the exhaust system, including models for the intercooler, turbo, and EGR.

This objective is then devided into 3 goals.

- Implement a model that can handle both SI and CI combustion with VVA.
- Utilize the ability to easily change model equations simulated.
- Structurally present equations, parameters and validation.

1.3 Delimitations

Items that are outside the scope of this thesis are:

- Port fuel injection
- Fuel spray models
- Thermal stress and solid mechanics that affects geometries
- Validation of EGR model
- Warm-up process of the engine
- Formation of emissions
- Implementation of control system
- Variable compression by varying stroke length
- Sensor function

1.4 Related Research

When it comes to the modeling of engines, there are transfer function models, cycle mean value models, zero- or one-dimensional models, and computational fluid dynamic models [26]. The MVEM is a compromise between the simpler transfer function models and the more detailed zero- or one-dimensional models as they are used for the design of the engine control system, where fast simulation times are necessary [26]. This makes MVEM good when it comes to HIL simulations, as the model has to capture engine behavior [20].

When it comes to modeling, the MVEM approach is used because of the low computational power required due to the usage of nonlinear ordinary differential equations [12], meaning faster than real-time simulations are possible [19]. This makes it perfect for controller development and tuning.

A component-based MVEM was developed in Modelica 2001. This includes a thorough explanation of how the models are structured, the Modelica Language as well as the models and validations of them. The methodology is good and understandable and gives a deeper understanding of the possibilities of the component-based modeling approach [4].

Component-based modeling means that you utilize derived models from literature and build your model one component at a time [8]. Components in engine modeling mean for example the air filter, throttle, intake manifold, and cylinder. Then the models also have to be validated to make sure they capture the dynamics of the system accurately.

Different parts of the engine model assume different kinds of flows. The flow is generally compressible or incompressible. Most gas flows in pipes are incompressible and turbulent in engines. However, over most kinds of valves, a compressible flow should be assumed. An example of a component where compressible flow should be considered is the throttle and the opening of an intake/exhaust valve [11].

There are several open-source engine models already available today. One available model is a modified MVEM for SI engines with EGR using the Simulink environment. Modified means that the usual isothermal models for manifolds were replaced with a model for temperature dependency. The results found were that neglecting the instances that change of the input regime in the manifold air temperature is approximately constant [22].

MVEM can also be used to validate models for control purposes. This can for example be for validation of a control model for electronic throttle [5].

LiU-Diesel is a MATLAB/Simulink model simulating a Diesel Engine with EGR and VGT [28]. Further development of the LiU-Diesel called LiU-Diesel2 adds compatibility of Throttle and Turbocharger with Wastegate [7].

1.5 Reason for Research

The reason a more generic engine model is desired is to facilitate the simulation of different kinds of engine configurations and dependencies without the need to develop a new model. If a more generic engine model can be proven to have a good correlation with real engines using a set of different engines with different displacements, the assumption can be made that the model works for a wide variety of engine configurations.

The model can then be used both in engine simulations and controller developed or even be used online to be able to choose control parameters live directly in the ECU.

This model is going to be validated using a 2.0 L turbocharged petrol Volvo engine and a 12.7 L turbocharged Scania diesel engine.

1.6 Report Outline

In chapter 1 the motivation of the thesis is presented and put into relation to previous work done in the field. Chapter 2 describes the fundamental operating principles of engines. In Chapter 3 a description of how the models are implemented and the algorithms used to estimate the parameters needed to simulate the engine is presented. Chapter 4 contains data collection and model parametrization method including data management as well as initial conditions. In Chapter 5 the results are presented. In Chapter 6 the results are discussed and Chapter 7 contains suggested future work.

2

Fundamental Operating Principles of Engines

In this chapter, the relevant theory needed when modeling a four-stroke CI or SI engine will be discussed.

2.1 Description of Modelling concept

The MVEM has appliances for the analysis and design of control and diagnosis systems. This is because MVEM describes variations that are slower than an engine cycle. The MVEM models are sometimes also referred to as *control oriented models* or *filling and emptying models*. Due to the averaging over one or several cycles for the MVEM modeling, physical sensor and actuator dynamics are important to investigate as those components actuate and measure the actual engine [11].

The MVEM will be based on the "filling and emptying" method for both the intake- and exhaust manifold. This method allows for the possibility to divide volumes into sections as the manifolds are represented as finite volumes [15].

2.2 Four-Stroke Cycle

A four-stroke engine has four strokes described below.

Intake During the intake stroke the intake valve opens and the cylinder is filled with fresh air. For SI engines, the fuel is also injected during this stroke. It can be assumed that the cylinder will have the same pressure as the intake at this stroke as the piston reaches BDC.

Compression The intake valve closes as the piston reaches BDC and the air or air/fuel ratio is starting to compress. Just before the piston hits TDC the fuel is injected into the cylinder for CI engines.

Power The spark is ignited on SI engines and the fuel ignites on CI engines when the piston reaches TDC and the work is carried out during the power stroke. The gasses expand as the flame burn which pushes the piston down towards BDC.

Exhaust As the piston reaches BDC all the work has been carried out and the exhaust valve opens as the piston goes up towards TDC to get rid of the exhaust gasses.

2.3 Engine Geometry

The engine geometry can be described with a small number of parameters. A collection of them can be seen in Table 2.1.

Table 2.1: *Engine geometry parameters*

Model parameter	Notation
Cylinder bore	B
Connection rod length	l
Crank radius	a
Piston stroke	$L = 2a$
Crank angle	θ
Minimum clearance volume	V_c
Displaced volume	$V_d = \frac{\pi B^2 L}{4}$

Using the displacement and minimum clearance volume the compression ratio is

$$r_c = \frac{\text{maximum cylinder volume}}{\text{minimum cylinder volume}} = \frac{V_d + V_c}{V_c} \tag{2.1}$$

The total displacement volume of the engine is the sum of the displacement volume of all cylinders

$$V_D = n_{cyl} V_d \tag{2.2}$$

The instantaneous volume with respect to crank angle θ is

$$V(\theta) = V_d \left(\frac{1}{r_c - 1} + \frac{1}{2} \left(\frac{l}{a} + 1 - \cos(\theta) - \sqrt{\left(\frac{l}{a}\right)^2 - \sin^2(\theta)} \right) \right) \tag{2.3}$$

The derivative of equation (2.3) is

$$\frac{dV(\theta)}{d\theta} = \frac{1}{2} V_d \sin(\theta) \left(1 + \frac{\cos(\theta)}{\sqrt{\left(\frac{l}{a}\right)^2 - \sin^2(\theta)}} \right) \tag{2.4}$$

Equation (2.4) can then be expressed in the time domain using the chain rule

$$\frac{dV(\theta)}{dt} = \frac{dV(\theta)}{d\theta}\frac{d\theta}{dt} = \frac{dV(\theta)}{d\theta}\omega_e \tag{2.5}$$

The wall area of the combustion chamber with respect to crank angle is simplified and calculated as the cylinder wall with a flat disc at the top and the bottom

$$A(\theta) = (l + a - s(\theta))\,\pi B + \frac{2\pi B^2}{4} \tag{2.6}$$

$$s(\theta) = a\cos(\theta) + \sqrt{l^2 - a^2\sin^2(\theta)} \tag{2.7}$$

where $s(\theta)$ is the stroke length.

2.4 Engine Components

The engine consists of multiple components. The components can be seen in Figure 2.1 and the flow is positive when flowing to the right.

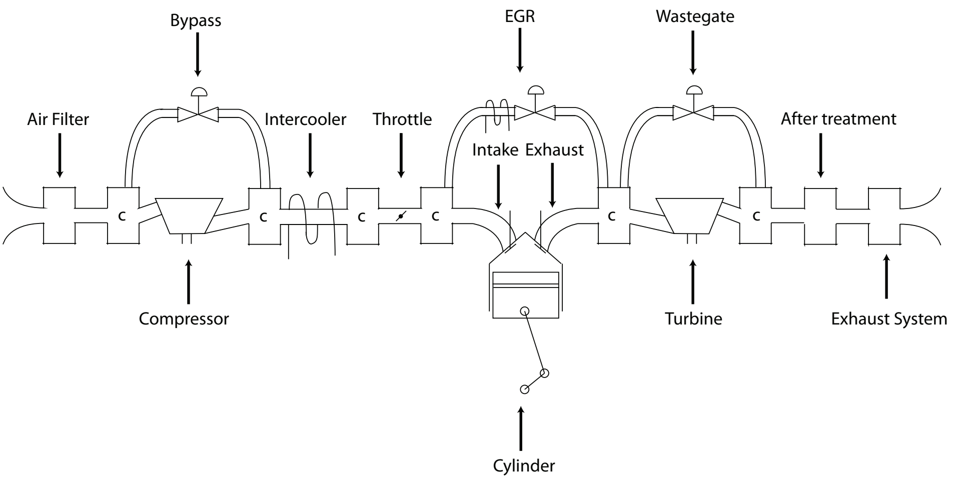

Figure 2.1: *The engine components including air filter, compressor, intercooler, throttle, intake and exhaust, EGR, WG, Turbine, after treatment and the rest of the exhaust system. In this thesis the after treatment and exhaust system is modeled as the "Exhaust system".*

2.4.1 Supercharging

Turbocharging has come to be a bigger part of the control problem for the performance of the supercharged engine. The turbocharger is used to a greater extent for downsizing engines. There are different ways to turbocharge an engine;

mechanical supercharging, turbocharging, two-stage turbocharging - serial, two-stage turbocharging - parallel, engine-driven compressor and turbocharger, and turbocharging with turbo compound [11].

For Turbo control it is important to keep out of maximum speed, surge, choking, and restriction areas. Choke occurs when the pressure ratio is low, on the right line in the compressor map and surge occurs with high-pressure ratios, on the left line in the compressor map while maximum speed is restricted by the top line in the compressor line and restriction of the bottom line. This can be controlled directly using the wastegate for the turbine and indirectly using the blow-off valve for the compressor [11].

2.5 Combustion

The engine combustion is based on the first law of thermodynamics, with combustion being the source of the heat release. For SI engines, the fuel is injected during the intake stroke or early in the compression stroke to create a homogeneous charge before the spark is ignited. For CI engines, the fuel is instead injected just before TDC and there is a delay between when the fuel is injected and when the combustion reaction starts [11]. There are several ways to model combustion. One way is to use the chemistry of the fuel used and an ignition delay associated with the fuel used [24]. Further, it is important to keep track of the available oxygen in the combustion chamber. This depends on the circulated exhaust gasses from the EGR, the gas that is ventilated back from the crankcase as well as how much overlap the intake and exhaust valve has. The mass fraction burned, F can be formulated simply using

$$F = \frac{m_b}{m} \tag{2.8}$$

where m_b and m is the burnt gas mass and total gas mass, respectively, in the combustion chamber [27].

2.5.1 Combustion Modelling

Combustion modeling is often modeled as an s-shaped Vibe function according to

$$x_b(\theta) = \begin{cases} 0, & \theta < \theta_{SOC} \\ 1 - e^{-a\left(\frac{\theta-\theta_{SOC}}{\Delta\theta}\right)^{m+1}}, & \theta \geq \theta_{SOC} \end{cases} \tag{2.9}$$

This function determines the fraction burned, meaning it goes from 0 to 1. $\Delta\theta$ and a are related to combustion duration, m affects the shape and θ_{SOC} denotes the start of combustion.

For heat release calculations, the Vibe function is derived

$$\frac{dx_b\theta}{d\theta} = \frac{a(m+1)}{\Delta\theta}\left(\frac{\theta-\theta_{SOC}}{\Delta\theta}\right)^m e^{-a\left(\frac{\theta-\theta_{SOC}}{\Delta\theta}\right)^{m+1}} \tag{2.10}$$

Combustion duration can then be approximated to $\Delta\theta = \Delta\theta_d + \Delta\theta_b$ [11].

2.5.2 Cylinder Pressure

The cylinder pressure is obtained by using the Matekunas Pressure ratio. The Matekunas pressure ratio is defined as the ratio of cylinder pressure from a fired cycle $p(\theta)$ and the pressure from a motored cycle $p_m(\theta)$

$$PR(\theta) = \frac{p(\theta)}{p_m(\theta)} - 1 \tag{2.11}$$

The pressure ratio is normalized by its maximum value to produce heat release traces as Klein explains the normalized pressure ratio is close to 0.5 when the crank angle suggests 50% burned [18].

$$PR_N(\theta) = \frac{PR(\theta)}{\max\left(PR(\theta)\right)} \tag{2.12}$$

2.6 Thermodynamic States

The essentials when it comes to the thermodynamics of the cylinder are based on the first law of thermodynamics. The equation can be seen in (2.13) , where U is the internal energy, Q is the energy for heat transfer and heat release, W is the work and H is the enthalpy. The cylinders thermodynamic states are illustrated in Figure 2.2 . Further description can be found in [11].

$$\mathrm{d}U = \mathrm{d}Q_{hr} - \mathrm{d}W - \mathrm{d}Q_{ht} + \sum_i \mathrm{d}H_i \tag{2.13}$$

The heat release is calculated according to

$$\frac{dQ_{HR}(\theta)}{d\theta} = m_f Q_{LHV} \eta_{co} \frac{dx_b(\theta)}{d\theta} \tag{2.14}$$

where combustion efficiency is modelled as [11]

$$\eta_{co} = \min\left(1, \lambda\right) \tag{2.15}$$

with the assumption that fuel is burned as long as there is oxygen available.

The heat release over time is then

$$\frac{dQ_{HR}(\theta)}{dt} = \frac{dQ_{HR}(\theta)}{d\theta} \omega_e \tag{2.16}$$

The heat transfer can occur in three ways, convection, conduction, and radiation. In combustion engines, the main heat transfer is from convection, which is the heat transfer between a solid surface and the surrounding gas or liquid [32]. For a combustion engine, convection is the heat from the combustion gas

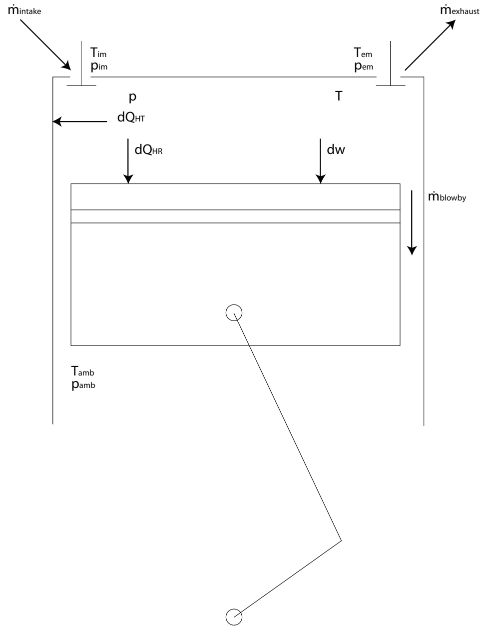

Figure 2.2: *The thermodynamic flows in a cylinder.*

to the cylinder walls. Radiation is other heat transfers happening in a combustion engine [11], but the radiation is assumed small compared to the convection. Therefore only convection is modeled according to Newton's law of cooling

$$\dot{Q}_{HT} = hA\Delta T = hA\left(T - T_w\right) \tag{2.17}$$

where A is the area of the walls in the cylinder, including piston and cylinder head, T is the current temperature of the gas which is assumed homogeneous as well as having the same temperature and T_w is the wall temperature which in this thesis is assumed a constant value of 140 °C for a fully warmed up engine [14].

To find h, the convection heat transfer coefficient of the Woschni method is used [31]. The method is presented in section 3.9.3.

The flows in Figure 2.2vare assumed positive in the arrow directions and negative if the flow goes the other way.

2.7 VVA

Even though the VVA technology adds complexity, it improves things like engine efficiency and emissions. There are four different ways the actuation of VVA can happen: [6]

1. Change the crank angle initiation point, keeping duration and valve lift characteristics constant

2. Change the duration of the valve event while keeping the initiation point the same

3. Modify the characteristics of the lift while keeping the initiation point and duration constant

4. Keeping the valve closed on selected cylinders to change the effective displacement

There are different ways of actuating the VVA system. It can be actuated continuously electro-hydraulically/-magnetically/-mechanically. VVA can also be actuated discretely between two different lift profiles [25].

2.7.1 CRB

Simply explained CRB turns the combustion engine into an air compressor. The air is compressed during the compression stroke then the exhaust valve is hydraulically opened just before TDC meaning there is no energy stored for the downward stroke. As the engine enters the power stroke no pressure remains to act on the piston. This means the expansion is made under low pressure thus carrying out negative work which gives a negative torque. The CRB has greater effects the higher the cylinder pressure is before opening the exhaust valve, but it also puts higher demands on the exhaust valve (since the valve has to withstand the high cylinder pressure and open) and turbocharger (since the air let out will instead be pumped directly into the turbocharger). The CRB can be utilized both every exhaust cycle or even every time the piston is compressing, further increasing the efficiency of the retardation. It is however important that no fuel is injected into the cylinder, as combustion will not occur. CRB is also called *Jake brake* or *Jacobs brake* [13].

2.7.2 Scavenging Effects

Scavenging usually occurs in two-stroke engines when the inlet gasses are pumped through the transfer ports via the crankcase, meaning the fresh inlet gas flows to the exhaust port before the piston seals it up. This effect is negligible in normal four-stroke engines without VVA since the valve overlap is so small. The scavenging effect could happen though if the overlap is high and the valves are open simultaneously over a greater time, as it could be with an engine with VVA and the pressure decreases downstream. This means that if the pressure in the intake is greater than the pressure in the cylinder and in turn is greater than the pressure in the exhaust manifold while both intake and exhaust valves are open, the intake gasses could go directly to the exhaust manifold instead of staying in the cylinder. If instead, the pressure is greater downstream, the opposite could happen and this is called backflow [2].

2.8 Gas Properties

In the modeling, the working fluid in an ICE is assumed to be a pure substance throughout the entire engine. A pure substance means that the gas composition is homogenous. The working gas is assumed to be *air* before combustion and *exhaust* after combustion to reduce the complexity [32].

2.8.1 Working Gases

Air is the occurring gas in the atmosphere, containing several different elements. The most essential element for combustion is oxygen. The other gasses have a negligible impact on combustion effects but take up space. As the most common element in the air is nitrogen, all elements except oxygen are clumped together, creating a common assumption [11]

$$\text{Air} = O_2 + 3.773N_2 \tag{2.18}$$

Exhaust composition is dependent on what fuel is assumed to be burning during combustion. The fuel used is cetane for CI combustion and isooctane for SI combustion. The gas properties used can be seen in Appendix A.1. The burned composition is.

$$\text{Exhaust} = aCO_2 + \frac{b}{2}H_2O + 3.773\left(a + \frac{b}{4}N_2\right) \tag{2.19}$$

2.9 Flow Restriction Modelling

In the engine, each component is associated with some kind of flow restriction. As gas flows through the flow restrictions, pressure drops as a result of for example friction (flow against walls), turning (flow though bends), diffusion (flow area increasing), or acceleration (flow area decreasing) [11].

Flow restrictions are usually divided into categories that represent the physical process associated with the flow restriction. These categories are compressible and incompressible flows. Each flow restriction is then evaluated based on flow velocity, where flow velocities up to Mach 0.2–0.3 (70–100 m/s), for air at ambient conditions, can be considered incompressible. At higher flow velocities, the flow is usually modeled as compressible.

2.9.1 Models for Cylinder Flow

The flow through a poppet valve is usually modeled as compressible flow. This means that the intake and exhaust valve flow is calculated using compressible flow models. The equation for the discharge coefficient (2.20) is then used in equation (2.21). Most often the discharge coefficient C_D is mapped up using a flow rig.

$$C_D = \frac{\text{actual mass flow}}{\text{ideal mass flow}} \tag{2.20}$$

$$A_E = C_D A_R \tag{2.21}$$

The discharge coefficient can however also be mapped using the valve lift ratio [3].

$$L_r = \frac{\text{valve lift}}{\text{maximum valve lift}} = \frac{L}{L_v} \tag{2.22}$$

2.9.2 NASA Polynomials

The specific heat ratio depends on the temperature. NASA polynomial is a database for how the specific heat ratio changes with temperature for different chemical species. The NASA polynomial can be read more in [21].

2.9.3 Blowby

The two main in- and out-flow sources in the cylinder are the intake and exhaust valve. The piston can however not make a perfectly tight seal with the cylinder wall, meaning that some gas will escape through the piston rings down to the crankcase. This gas was previously vented to the atmosphere but due to it being a high source of HC emissions, it is now instead vented back to the intake system [16].

$$\frac{\tilde{c}_p(T)}{\tilde{R}} = a_1 + a_2 T + a_3 T^2 + a_4 T^3 + a_5 T^4 \tag{2.23}$$

Another way of finding varying specific heat ratios is to use the CHEPP package designed for MATLAB [9].

2.10 Parameter Estimation

There are 3 different types of measurement data

- Stationary (also called the engine map)
- Dynamic
- Crankshaft based

From that data, all the parameters can be found mainly using the least-squares method. The least-squares method can be devided into two different categories, linear and non-linear. The method is breifly described in section 4.7. For a more detailed description of the least-squares method, see [1].

3

Mathematical Component Models

In this chapter the models and model parameters used will be presented. Parameter estimation is presented in section 4.7.

3.1 Incompressible Flow model

There are different kinds of flow restrictions among the engine components. The flow can be assumed to be incompressible for the air filter, intercooler, and exhaust system (including the catalyst and muffler). This means that the throttle and cylinder flows will be modeled differently.

The model selected for the incompressible flow restrictions are found in [11]

$$\dot{m}(p_{us}, T_{us}, p_{ds}) = \begin{cases} C_{tu}\sqrt{\frac{p_{us}}{RT_{us}}}\sqrt{p_{us}-p_{ds}}, & \text{if } p_{us}-p_{ds} \geqslant \Delta p_{lin} \\ C_{tu}\sqrt{\frac{p_{us}}{RT_{us}}}\frac{p_{us}-p_{ds}}{\Delta p_{lin}}, & \text{otherwise} \end{cases} \tag{3.1}$$

$$T_{flow} = T_{us} \tag{3.2}$$

The parameters for this model is C_{tu} and Δp_{lin}.

3.2 Control Volumes model

When it comes to control volumes, it can be divided into two different models. One is the isothermal model where it can be assumed that the temperature is constant in the whole control volume. The other one is the adiabatic model which means the heat transfer is often set to zero. All control volumes in the implemented model are adiabatic to also see temperature variations. The model implemented has pressure and temperature as states, p and T [11].

$$\begin{cases} \frac{dT}{dt} = & \frac{RT}{pVc_v}\left[\dot{m}_{in}c_v(T_{in}-T)+R(T_{in}\dot{m}_{in}-T\dot{m}_{out})-\dot{Q}\right] \\ \frac{dp}{dt} = & \frac{RT}{V}(\dot{m}_{in}-\dot{m}_{out})+\frac{p}{T}\frac{dT}{dt} \end{cases} \tag{3.3}$$

The parameter for this model is V, the volume of each control volume. $\dot{m}_{in}$ is the sum of all positive flows into the volume and $\dot{m}_{out}$ is the sum of all negative flows into the volume. T_{in} is the temperature of the gas flowing into the volume and is modeled as a mean value for all the flow flowing into the volume also assuming the same c_v for all flowing fluids in the volume. This means

$$T_{in} = \begin{cases} \frac{\dot{m}_1\cdot T_1}{\dot{m}_{in}} + \frac{\dot{m}_2\cdot T_2}{\dot{m}_{in}} + ... + \frac{\dot{m}_n\cdot T_n}{\dot{m}_{in}} & \dot{m}_{in} > 0 \\ 0 & \text{otherwise} \end{cases} \tag{3.4}$$

where n is the flow with corresponding temperature for inflows into the control volume.

For the isothermal model, the incompressible flow restriction isothermal is

$$\frac{dp}{dt} = \frac{RT}{V}(\dot{m}_{in}-\dot{m}_{out}) \tag{3.5}$$

$$T = T_{in} = T_{out} \tag{3.6}$$

where the only parameter is V the volume of each control volume.

3.3 Compressible Flow model

The flow can be assumed to be compressible over the throttle, bypass, wastegate, and valves. The valve flow function can be seen i section 3.9.1.

Throttle angle has a dynamic behavior, meaning that the angle is modeled as a first-order system.

$$\dot{\alpha}_{th} = \frac{1}{\tau_{th}}\left(\beta_{ped}-\alpha_{th}\right) \tag{3.7}$$

where the time constant τ_{th} is measured doing a step in pedal position β_{ped} measuring the throttle angle α_{th}.

The flow model is based on a compressible flow approach [11]

$$\dot{m}_{at} = \frac{p_{us}}{\sqrt{RT_{us}}}A_e(\alpha_{th})\Psi_{li}(\Pi), \qquad \Pi = \max\left(\frac{p_{ds}}{p_{us}}, \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}}\right) \tag{3.8}$$

$$A_e(\alpha_{th}) = A_0 + A_1\alpha_{th} + A_2\alpha_{th}^2 \tag{3.9}$$

$$\Psi_{li}(\Pi) = \begin{cases} \Psi_0(\Pi) & \text{if } \Pi \le \Pi_{li} \\ \Psi_0(\Pi)\frac{1-\Pi}{1-\Pi_{li}} & \text{otherwise} \end{cases}, \quad \Psi_0(\Pi) = \sqrt{\frac{2\gamma}{\gamma-1}\left(\Pi^{\frac{2}{\gamma}}-\Pi^{\frac{\gamma+1}{\gamma}}\right)} \tag{3.10}$$

where the parameters are Π_{li} that is defining the linear region and A_0, A_1, A_2 are constants for a parameterization method for finding the effective area.

The bypass and wastegate uses the same function, but the function for the effective area is instead a function of the fraction the valves are open, where zero is fully closed and one is fully open.

$$A_e\left([0..1]\right) = C_D A_R f\left([0..1]\right) \qquad A_R = \frac{d_{vavle}\pi}{4} \tag{3.11}$$

3.4 Compressor model

The theory for compressor modelling is presented in section 2.4.1. Compressor flow and efficiency is modeled as [10]

$$\Pi_{c,max} = \left(\frac{u_2^2 \Psi_{max}}{2 c_p T_{af}} + 1\right)^{\frac{\gamma}{\gamma-1}}, \quad u_2 = r_c \omega_{tc} \tag{3.12}$$

$$\dot{m}_{c,corr} = \dot{m}_{c,corr,max}\sqrt{1 - \left(\frac{\Pi_c}{\Pi_{c,max}}\right)^2}, \quad \Pi_c = \frac{p_c}{p_{af}} \tag{3.13}$$

$$\dot{m}_c = \dot{m}_{c,corr}\frac{p_{af}/p_{ref,c}}{\sqrt{T_{af}/T_{ref,c}}} \tag{3.14}$$

$$\eta_c = \eta_{c,max} - \chi^T Q_\eta \chi \tag{3.15}$$

$$Q_{\eta_c} = \begin{bmatrix} Q_{11} & Q_{12} \\ Q_{12} & Q_{22} \end{bmatrix}, \chi = \begin{bmatrix} \dot{m}_{c,corr} - \dot{m}_{c,corr@\eta_{c,max}} \\ \sqrt{\Pi_{c-1} - (\Pi_{c@\eta_{c,max}-1})} \end{bmatrix} \tag{3.16}$$

The parameters for this model is Ψ_{max}, $\dot{m}_{c,corr,max}$, $\eta_{c,max}$, $\dot{m}_{c,corr@\eta_{c,max}}$, $\Pi_{c@\eta_{c,max}}$, Q_{11}, Q_{12}, Q_{22}. Ψ_{max} and $\dot{m}_{c,corr,max}$.

3.5 Turbine model

The theory for turbine modeling is presented in section 2.4.1. Turbine flow and efficiency is modeled as [10]

$$TFP_{model} = k_0\sqrt{1 - \Pi_t^{k_1}}, \quad \Pi_t = \frac{p_{04}}{p_{03}} = \frac{p_{es}}{p_{em}} \tag{3.17}$$

$$\dot{m}_t = \frac{p_{em}}{\sqrt{T_{em}}} TFP_{model} \tag{3.18}$$

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

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

The parameters for this model are k_0, k_1, $\eta_{t,max}$ and BSR_{max}.

3.6 Turbo Dynamics model

The turboshaft friction is modelled according to

$$Tq_{tc,fric} = c_{tc}\omega_{tc} \tag{3.21}$$

where the friction is assumed to be low since the shaft is oil lubricated, meaning $c_{tc} = 1 \cdot 10^{-6}$ Nm/(rad/s). Newton's second law of rotation is then used to model the turbo shaft speed

$$\frac{d\omega_{tc}}{dt} = \frac{1}{J_{tc}}\left(Tq_t - Tq_c - Tq_{tc,fric}\right) \tag{3.22}$$

The parameters for this model are J_{tc}, which is the inertia of the shaft.

3.7 Intercooler model

The Intercooler is modeled as an incompressible flow, see Section 3.1. What differentiates the intercooler from the incompressible flow model is the model for the temperature [11].

$$T_o = T_i - \epsilon\left(\dot{m}_{cool}, \dot{m}_{ic}, ...\right)\left(T_i - T_{cool}\right) \tag{3.23}$$

$$\epsilon = a_0 + a_1\left(\frac{T_i + T_{cool}}{2}\right) + a_2\dot{m}_{air} + a_3\frac{\dot{m}_{air}}{\dot{m}_{cool}} \tag{3.24}$$

where T_o is the temperature of the gas flowing out of the intercooler, T_i is the temperature of the gas flowing into the intercooler, T_{cool} is the temperature of the air surrounding the intercooler, which in this model is set to T_{amb}, $\dot{m}_{air}$ is the mass flow of the gas flowing into the intercooler and $\dot{m}_{cool}$ is the mass flow of the gas cooling the intercooler. It is assumed the intercooler gets enough air to cool the sufficiently, meaning the fraction $\frac{\dot{m}_{air}}{\dot{m}_{cool}}$ is 1. That means

$$\epsilon = a_0 + a_1\left(\frac{T_i + T_{cool}}{2}\right) + a_2\dot{m}_{air} \tag{3.25}$$

The parameters for this model is a_0, a_1 and a_2.

3.8 Intake model

The intake model consists of an adiabatic control volume, see Section 3.2 and a model for the oxygen concentration. [28].

$$X_{Oim} = \frac{m_{Oim}}{m_{totim}} \tag{3.26}$$

where the fraction of X_{Oim} is 1 if there is 1 O_2 for every 3.773 N_2 in the simplifyed air ratio from equation (2.18). The change of oxygen concentration is then

$$\dot{X}_{Oim} = \frac{R_{air} T_{im}}{p_{im} V_{im}} \Big((X_{Oem} - X_{Oim})\, \dot{m}_{egr} + (X_{Oc} - X_{Oim})\, \dot{m}_{th} \Big) \tag{3.27}$$

where X_{Oem} is the oxygen concentration in the exhaust manifold, X_{Oim} is the oxygen concentration in the intake manifold and $X_{Oc} = 1$ and is the constant amount of oxygen in the compressor as there is assumed no residual gas is travelling through the compressor. $\dot{m}_{egr}$ is the mass flow flowing from the EGR and $\dot{m}_{th}$ is the mass flow flowing from the throttle.

3.9 Cylinder model

To fully capture the dynamics of the oxygen concentration, cylinder- temperature, and pressure with the usage of VVA, each cylinder has to be simulated separately with full-cycle cylinder states [17].

3.9.1 Cylinder Flow

The cylinder flow is modelled according to the thermodynamic states displayed in Figure 2.2. This means that there are four different flows: intake, exhaust, CRB, and blowby.

$$\dot{m} = \frac{p_{us}}{\sqrt{R T_{us}}} A_{eff} \Psi(\Pi) \qquad A_{eff} = C_D A_R \tag{3.28}$$

$$h = c_p T_{us} \tag{3.29}$$

where Ψ is modelled according to equation (3.10), A_R is the reference area and h is the specific enthalpy.

The flow is for blowby, the small amount of flow that flows between the cylinder and piston, which is not sealed by the piston rings and assumed an effective area. The effective area for the blowby was therefore used as a tuning parameter to match measured cylinder pressure. For intake and exhaust, the reference area is

$$A_R = \frac{D_v^2 \pi}{4} \tag{3.30}$$

The discharge coefficient is modeled differently for intake and exhaust, as the C_D value is dependent on the lift for the intake and both lift and pressure ratio for the exhaust

$$C_{D,in} = f_{\text{lookup}}\left(L_{in}(\theta)\right) \tag{3.31}$$

$$C_{D,ex} = f_{\text{lookup}}\left(L_{ex}(\theta), Pr\right) \qquad Pr = \frac{p_{cyl}}{p_{em}} \tag{3.32}$$

CRB uses the exhaust valve to brake motion, but only open 1 of the 2 available exhaust valves. This means the effective area for the CRB is the same as the exhaust.

3.9.2 Combustion modelling

Combustion is affected by the amount of available oxygen. Therefore the oxygen concentration $X_{O,cyl}$ is a state. This also gives the $X_{burned} = 1 - X_{O,cyl}$ which is the amount of oxygen that has reacted in the combustion. A perfect combustion is assumed, which means that formation of particles has been neglected in this thesis. Combustion is assumed to be a reaction between air and hydrocarbons. The chemical reaction in the combustion is

$$n_{fuel}C_aH_b + n_{air}\left(O_2 + 3.773N_2\right) + n_{burned}\left(aCO_2 + \frac{b}{2}H_2O + 3.773\left(a + \frac{b}{4}N_2\right)\right) \longrightarrow$$
$$\left(n_{fuel} + n_{burned}\right)\left(aCO_2 + \frac{b}{2}H_2O + 3.773\left(a + \frac{b}{4}N_2\right)\right) +$$
$$+\left(n_{air} - n_{fuel}\left(a + \frac{b}{4}\right)\right)\left(O_2 + 3.773N_2\right) \tag{3.33}$$

where the fuel composition is C_aH_b, the air composition is $O_2 + 3.773N_2$ and the burned composition is $aCO_2 + \frac{b}{2}H_2O + 3.773\left(a + \frac{b}{4}N_2\right)$.

The number of moles in the reaction is calculated at IVC.

$$n_{tot,air} = 1 + 3.773 \qquad n_{tot,burned} = a + \frac{b}{2} + 3.773 \tag{3.34}$$
$$n_{air} = \frac{m_{IVC}X_{O,cyl}}{M_{air}n_{tot,air}} \qquad n_{fuel} = \frac{m_{fuel}}{M_{fuel}} \tag{3.35}$$
$$n_{burned} = \frac{\left(1 - X_{O,cyl}\right)m_{IVC}}{M_{burned}n_{tot,burned}} \tag{3.36}$$

where m_{IVC} is the mass in the cylinder at IVC. The mass in the cylinder is calculated by

$$m_{cyl} = \frac{p_{cyl}V_d}{R_{cyl}T_{cyl}} \tag{3.37}$$

The $X_{O,cyl}$ state is modelled as

$$\frac{dX_{O,cyl}}{dt} = \frac{1}{m_{cyl}}\sum_i\left(X_{O,cyl,i} - X_{O,cyl}\right)\dot{m}_i - C\frac{dx_b}{dt} \tag{3.38}$$

where the Vibe function (2.10) gives $\frac{dx_b}{dt} = \frac{dx_b\theta}{d\theta}\omega_e$ and C is the scaling of the Vibe function to make the Vibe function scale properly to equation (3.33).

The molar fraction x_{air} is the amount of free air that has not or has not yet reacted in the combustion. The amount of free air is directly correlated to the amount of oxygen according to the air assumption (2.18). The scaling factor C is modeled as the amount of air at IVC and the amount of air left after combustion.

$$\tilde{x}_{air,aftrComb} = \frac{\left(n_{air} - n_{fuel}\left(a + \frac{b}{4}\right)\right) n_{tot,air}}{\left(n_{air} - n_{fuel}\left(a + \frac{b}{4}\right)\right) n_{tot,air} + \left(n_{air} - n_{fuel}\left(a + \frac{b}{4}\right)\right) n_{tot,Burned}} \tag{3.39}$$

$$x_{air,aftrComb} = \frac{\tilde{x}_{air,aftrComb} M_{air}}{\left(1 - \tilde{x}_{air,aftrComb}\right) M_{Bunrned} + \tilde{x}_{air,aftrComb} M_{air}} \tag{3.40}$$

This gives the scaling of the Vibe function

$$C = x_{air,IVC} - x_{air,aftrComb} \tag{3.41}$$

Since the gas constant and heat capacity change with oxygen concentration and temperature, NASA polynomials are used to decide the heat capacities $c_{p,Burned}$ and $c_{p,air}$ according to equation (2.23). The gas mass-specific constant and specific heat is calculated as

$$R = (1 - X_{o,cyl}) R_{Burned} + X_{o,cyl} R_{air} \tag{3.42}$$

$$c_p = (1 - X_{o,cyl}) c_{p,Burned} + X_{o,cyl} c_{p,air} \tag{3.43}$$

The ideal gas assumption is used to get the specific heat constant c_v

$$c_v = c_p - R \tag{3.44}$$

The parameters for the IVC event uses intake temperature from the engine map, see Table 4.2 in chapter 4 but the rest of the measurements from the crankshaft based measurements, part of Table 4.5 in chapter 4. The parameters for the Vibe function uses pressures and temperatures from the crankshaft-based measurements, see Table 4.5.

3.9.3 Energy equations

The energy flows in the cylinder are the heat release, heat transfer, the work carried out, and internal energy. The directions of the energies can be seen in Figure 2.2. The heat release is calculated according to equation (2.14).

The heat transfer is calculated using the Woschini method, however the model for the heat transfer coefficient is taken from [11]

$$h = C_0 B^{-0.2} p^{0.8} w^{0.8} T^{-0.53} \tag{3.45}$$

where $C_0 = 1.30 \cdot 10^{-2}$, B is the cylinder bore, p is the pressure in the cylinder, T is the temperature in the cylinder and w is the characteristic velocity equation given by

$$w = C_1 \bar{S}_p + C_2 \frac{V T_{IVC}}{V_{IVC} p_{IVC}} (p - p_m) \tag{3.46}$$

Table 3.1: Constants used in Woschini's model for the heat transfer coefficient.

	Gas Exchange	Compression	Combustion and Expansion
C_1	6.18	2.28	2.28
C_2	0	0	0.00324

where $\bar{S}_p = \frac{2aN_e}{60}$ is the mean piston speed, p_m is the motored pressure and the constants C_1 and C_2 is dependent on what stroke the engine has and can be seen in Table 3.1.

The motored pressure is modeled with a polytope, where κ is the polytropic exponent.

$$p_m(\theta) = \begin{cases} p(\theta), & \text{if } \theta \le \theta_{SOC} \\ p\left(\theta_{SOC}\right)\left(\frac{V(\theta_{SOC})}{V(\theta)}\right)^{\kappa} & \text{if } \theta > \theta_{SOC} \end{cases} \tag{3.47}$$

where the polytropic exponent κ is set to a constant value, optimized from a motored cycle. Another way to get the motored pressure is to use the pressure from a measurement in an engine test cell.

The power is calculated using Newton's second law of motion.

$$\dot{W} = p_{cyl}\frac{dV}{dt} \tag{3.48}$$

and the internal energy is modelled

$$u = c_v T_{cyl} \tag{3.49}$$

3.9.4 State equations

The temperature and pressure states are T_{cyl} and p_{cyl}.

$$\dot{T}_{cyl} = \frac{R_{cyl}T_{cyl}}{p_{cyl}V c_{v,cyl}}\left(\dot{m}_{intake}\left(h_{intake} - u\right) - \sum_i \dot{m}_i\left(h_i - u\right) + \frac{dQ_{HR}(\theta)}{dt} - \dot{Q}_{HT} - W\right) \tag{3.50}$$

$$\dot{p}_{cyl} = \frac{R_{cyl}T_{cyl}}{V}\left(\dot{m}_{intake} - \sum_i \dot{m}_i\right) + \frac{p_{cyl}}{T_{cyl}}\dot{T}_{cyl} - \frac{W}{V} \tag{3.51}$$

where i is the flows for exhaust, CRB, and blowby.

3.9.5 Generated Engine Torque

A simple instantaneous torque model that neglects friction is used in this thesis [11]

$$M_{e,i}(\theta) = \sum_{j=1}^{n_{cyl}}\left(p_{cyl,j}\left(\theta - \theta_j^{\text{offset}}\right) - p_{amb}\right) A\, L\left(\theta - \theta_j^{\text{offset}}\right) \tag{3.52}$$

where θ_j^{offset} is the crank angle offset for cylinder j, A is the area and $L(\theta)$ is the crank lever. Note that the product of the area and the crank lever is equivalent to the volume derivative with respect to crank angle as can be seen in equation (2.4).

$$A\,L(\theta) = \frac{dV(\theta)}{d\theta} = \frac{dV(\theta)}{dt}\frac{1}{\omega_e} \tag{3.53}$$

The average torque is of interest for example to not exceed any loads for the driveline. The average torque is then calculated over four strokes [11].

$$M_e = \frac{1}{4\pi}\int_0^{4\pi} M_{e,i}(\theta)\,d\theta - M_f \tag{3.54}$$

where M_f denotes the friction, which has been neglected in this thesis.

3.10 Exhaust Manifold model

The exhaust manifold is modelled the same way as the intake manifold, see Section 3.8. The difference is just in the model for the oxygen concentration. For the exhaust manifold, the oxygen concentration is

$$X_{Oem} = \frac{m_{Oem}}{m_{totem}} \tag{3.55}$$

where the fraction of X_{Oem} is 1 if there is 1 O_2 for every 3.773 N_2 in the simplifyed air ratio from equation (2.18).

$$\dot{X}_{Oem} = \frac{R_{exh}T_{em}}{p_{em}V_{em}}\left((X_{Oe} - X_{Oem})\,\dot{m}_{eo}\right) \tag{3.56}$$

where X_{Oe} is the oxygen concentration from the cylinder, X_{Oem} is the oxygen concentration from the exhaust manifold and $\dot{m}_{eo}$ is the mass flow of the gas flowing out of the cylinder.

3.11 Exhaust Gas Recirculation model

The model for the EGR valve consists of multiple sub models that fully capture the behavior. The EGR mass flow is compressible, much like the throttle, see Section 3.3. The equations are [28]

$$\dot{m}_{egr} = \frac{p_{em}}{\sqrt{T_{em}R_{exh}}}A_{egr}\Psi_{egr}, \qquad \Psi_{egr} = 1 - \left(\frac{1-\Pi_{egr}}{1-\Pi_{egropt}} - 1\right)^2 \tag{3.57}$$

$$\Pi_{egr} = \begin{cases} \Pi_{egropt} & \text{if } \frac{p_{im}}{p_{em}} < \Pi_{egropt} \\ \frac{p_{im}}{p_{em}} & \text{if } \Pi_{egropt} \le \frac{p_{im}}{p_{em)}} \le 1 \\ 1 & \text{if } 1 < \frac{p_{im}}{p_{em}} \end{cases}, \qquad x_{egr} = \frac{\dot{m}_{egr}}{\dot{m}_c + \dot{m}_{egr}} \tag{3.58}$$

$$A_{egr} = A_{egrmax} f_{egr}\left(u_{egr}\right) \tag{3.59}$$

$$f_{egr}\left(u_{egr}\right) = \begin{cases} c_{egr1} u_{egr}^2 + c_{egr2} u_{egr} + c_{egr3} & \text{if } u_{egr} \le -\frac{c_{egr2}}{2c_{egr1}} \\ c_{egr3} - \frac{c_{egr2}^2}{4c_{egr1}} & \text{if } u_{egr} > -\frac{c_{egr2}}{2c_{egr1}} \end{cases} \tag{3.60}$$

where the parameters are the optimal value Π_{egropt} for the maximum value of the funtion Ψ_{egr} in equation (3.57) and the coefficients c_{egr1}, c_{egr2} and c_{egr3} in the polynomial for the effective area. $\dot{m}_{egr}$ can not be measured directly because the mass flow is highly dependent on oxygen concentration and temperature. $\dot{m}_{egr,meas}$ is instead assumed by a function of the flow from the throttle and the EGR fraction x_{egr}. This gives $\dot{m}_{egr,meas} = \frac{\dot{m}_{th}}{1-x_{egr}}$.

3.12 Complete model

The complete Simulink model can be seen in Figure 3.1 and is run by first running the init.m file setting up the neccecary model input as well as loading the parameter struct summarized in section 3.12.1.

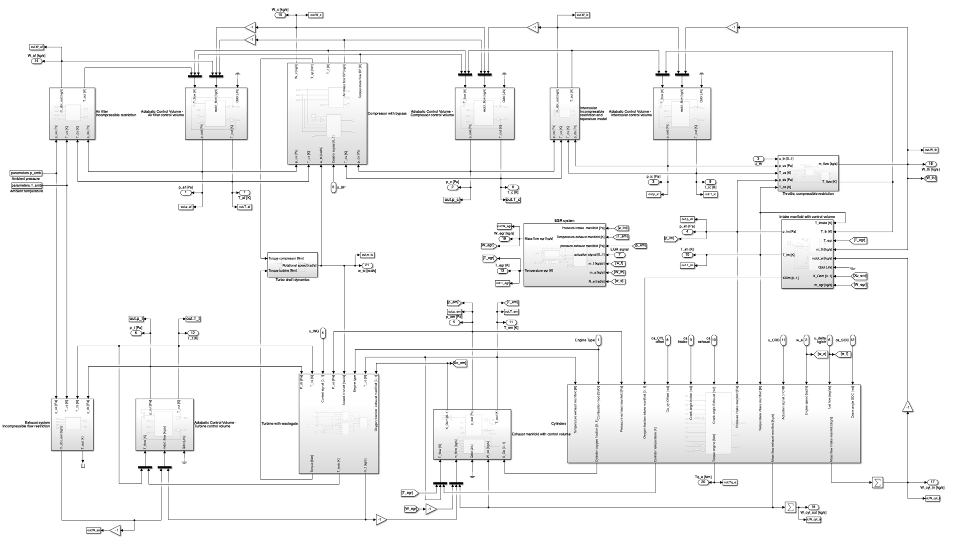

Figure 3.1: *The layout of the Simulink model*

3.12.1 Parameterization

To be able to run this model, a struct containing the parameters are needed using the parameterization discussed in chapter 3. The struct of parameters can be seen in Appendix B. The validation of the models can be seen in Section 4.7.

4

Data Acquisition

In this chapter, the measurements needed to run the model are presented as well as the inputs to the model. Four different kinds of measurements are necessary to make, dynamic measurements, stationary measurements (from the engine map), crankshaft-based measurements, and physical measurements.

4.1 Dynamic Measurements

Dynamic measurements are needed for systems with dynamical behavior. In this thesis, only one (1) measurement is needed for the dynamic systems and can be seen in Table 4.1.

Table 4.1: *List of dynamical measurements.*

Measurement	Description
Step response of throttle angle	To find time constant τ_{th} of the throttle

4.2 Stationary Measurements

Stationary measurements are measurements done where the engine has been given time to stabilize which means that all dynamic behavior has had time to subside. Measurements are usually carried out for a couple of seconds and then return an average value for that operating point. The stationary measurements are usually made in a series, going from one operating point to the next. An example of a complete engine map can be seen in Figure 4.1. The stationary measurements from the engine map needed can be seen in Table 4.2.

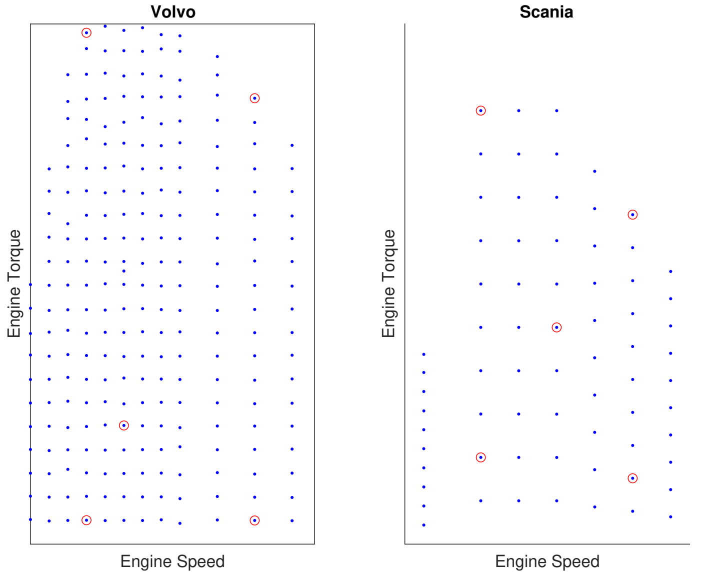

Figure 4.1: *A display of the different operating points of the simulated engines, with the chosen operating points marked with a red ring.*

Table 4.2: *List of stationary measurements from the engine map.*

Measurement	Description
p_{amb}	Ambient pressure
T_{amb}	Ambient temperature
$\dot{m}_f$	Fuel mass flow
$\dot{m}_{air}$	Air mass flow
p_{bc}	Pressure before compressor
p_{ac}	Pressure after compressor
T_{ac}	Temperature after compressor
p_{aic}	Pressure after intercooler
p_{at}	Pressure after turbine
T_{at}	Temperature after turbine
ω_{tc}	Turbo shaft speed
N_e	Engine speed
T_{im}	Temperature intake manifold
T_{bic}	Temperature before intercooler
T_{aic}	Temperature after intercooler
p_{im}	Pressure intake manifold
IVC	Intake valve close crank angle
α_{th}	Throttle angle

If the engine is turbocharged, two additional maps are needed. The maps needed are the compressor- and turbine maps. The measurements needed from the compressor map can be seen in Table 4.3 and the measurements needed from the turbine map can be seen in Table 4.4.

Table 4.3: *List of stationary measurements from the compressor map.*

Measurement	Description
$N_{c,corr}$	Corrected speed
$\dot{m}_{c,corr}$	Corrected mass flow
$p_{c,ref}$	Reference pressure compressor
$T_{c,ref}$	Reference temperature compressor
p_{af}	Pressure air filter
T_{af}	Temperature air filter
Π_c	Pressure ratio compressor
η_c	Compressor efficiency

Table 4.4: *List of stationary measurements from the turbine map.*

Measurement	Description
TSP	Turbine speed parameter
TFP	Turbine flow parameter
T_{em}	Temperature exhaust manifold
Π_t	Pressure ratio turbine
η_t	Turbine efficiency

For the Scania engine, two complete engine maps were available and for the Volvo engine, only one engine map was available.

4.3 Crankshaft-based Measurements

Crankshaft-based measurements are used when cycle variations are of interest. The usual sampling rate is therefore between 0.1 and 1 sample/degree. The crankshaft-based measurements needed can be seen in Table 4.5.

Table 4.5: *List of crankshaft based measurements.*

Measurement	Description
θ	Crank angle
p_{im}	Pressure intake manifold
p_{cyl}	Pressure in cylinder

Additionally to the measurements listed in Table 4.5 , the engine settings seen in Table 4.6. needs to be measured.

Table 4.6: *List of settings used in crankshaft based measurements.*

Measurement	Description
$\theta(IVC)$	Crank angle for IVC event
$\theta(SOI)$	Crank angle for SOI event

Both engines had complete crankshaft-based measurements for all points in the engine map.

4.4 Physical Measurements

There are some parameters and model behavior that depends on geometries and other constants related to the engine. These constants can be seen in Table 4.7.

Table 4.7: *List of engine constants.*

Measurement	Description
B	Cylinder bore
a	Crank radius
s	Piston stroke
l	Connecting rod length
V_d	Displacement volume
r_c	Compression ratio
n_{cyl}	Number of cylinders
L_{Valves}	Lift profiles
$C_{D,\text{Valves}}$	Flow discharge for valves
$C_{D,wastegate}$	Flow discharge for wastegate
$C_{D,bypass}$	Flow discharge for bypass valve
V_{af}	Volume air filter control volume
V_{ac}	Volume compressor control volume
V_{aic}	Volume intercooler control volume
V_{im}	Volume intake manifold control volume
V_{em}	Volume exhaust manifold control volume
V_{at}	Volume turbine control volume
$d_{turbine}$	Diameter turbine
$d_{wastegate}$	Diameter wastegate
$d_{compressor}$	Diameter compressor
d_{bypass}	Diameter bypass valve

4.5 Initial Conditions

Some additional signals are measured along with the stationary measurements in the engine map. These signals are used to achieve a better estimate of the initial condition. The signals and their description can be seen in 4.8. In this thesis, the corresponding available measurements are used as initial conditions to speed up simulation time.

Table 4.8: *List of measurements for more correct initial condition.*

Measurement	Description
p_{af}	Pressure air filter
T_{af}	Temperature air filter
p_{ac}	Pressure after compressor
T_{ac}	Temperature after compressor
p_{aic}	Pressure after intecooler
T_{aic}	Temperature after intercooler
p_{im}	Pressure intake manifold
T_{im}	Temperature intake manifold
p_{em}	Pressure exhaust manifold
T_{em}	Temperature exhaust manifold
p_{at}	Pressure after turbine
T_{at}	Temperature after turbine
ω_{tc}	Speed of turbo shaft

4.6 Model Input Signals

The input signals for the model is

- SI/CI by setting the constant block in Simulink to 1/0
- Engine speed [RPM]
- Desired throttle angle [rad]
- Wastegate actuation [0..1]
- Bypass actuation [0..1]
- Fuel flow [kg/str]
- EGR actuation [0..1]
- Cylinder offset, with cylinder 1=0 [rad]
- Intake cam angle offset [Deg]
- Exhaust cam angle offset [Deg]
- CRB actuation [1/0]
- SOI [rad]

4.7 The Parametrization for the Subsystems

The model consists of several subsystems, as can be seen in Figure 3.1. The description of each subsystem will include **Inputs**, **Outputs**, **Parameters**, **Parameter Estimation**, **Assumptions**, and **Remarks**.

It was decided that both engines would be run without EGR. This is due to there currently being no available data for any of the engines running with the EGR system used. The Volvo engine does not have CRB compatibility so CRB will not be validated or presented on the Volvo engine.

Further, it was decided that the Scania engine would be run without the throttle model and closed wastegate because that is a beneficial way of running the engine. LiU Diesel 2 compressor and turbine model was used because compressor and turbine maps were not available.

4.7.1 Parameter estimation methods

A system of linear equations are formulated as $Ax = B$ solving for x. A can be either a matrix or a vector and B is a vector.

A non-linear least squares problem are formulated as

$$\min \sum_i (\text{xdata}_i - \text{ydata}_i)^2 \tag{4.1}$$

where i is the length of the data set, xdata is modelled value and ydata is measured value. The minimized sum is then the sought parameter or parameters. In this thesis the non-linear least squares problem is solved using the MATLAB function `lsqcurvefit`, a non-linear solver using the least squares method.

4.7.2 Incompressible Flow Restriction

There are multiple flow restrictions in the engine model. The incompressible flow restrictions include models for

- air filter
- intercooler
- exhaust system

Figure 4.2 shows the component for the incompressible flow.

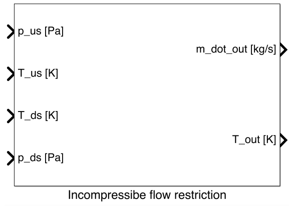

Figure 4.2: *This figure shows the incompressible flow restriction.*

Inputs

p_ds	K	Pressure downstream
p_us	K	Temperature upstream
T_ds	K	Temperature downstream
T_us	K	Temperature upstream

Outputs

m_dot_out	kg/s	Mass flow through the restriction
T_out	K	Temperature through the restriction

Parameters

c_tun	-	Tuning parameter for each component
p_li	Pa	Pressure limit
R	J/kg K	Gas constant

Parameter Estimation

The incompressible flow restrictions for the air filter and intercooler use the gas constant for air and the exhaust system uses the gas constant for exhaust gas.

The tuning parameter c_tun is found by solving a non-linear least squares problem with c_tun and a valdation only parameter for sensor offset called a. The problem is frmulated as

$$\dot{m}^2 R T_{us} = C_{tu}^2 p_{us} \left(p_{us} - (p_{ds} - a)\right)$$

where us is amb and ds is bc and $\dot{m}$ is $\dot{m}_{air}$ for the air filter. For the intercooler, us is ac and ds is aic and $\dot{m}$ is $\dot{m}_{air}$. For the exhaust system, us is at and ds is amb and $\dot{m}$ is $\dot{m}_{air} + \dot{m}_f$.

If p_us = d_ds the Lipschitz condition is not fulfilled as the derivative goes towards infinity, which causes problems for the ODE solver. In this thesis, ODE15s was used. p_li is the linear pressure region and it is assumed p_li ≥ 1000 pa.

The data points used are all available data from the engine map, as seen in table 4.2. The tuning parameters are then validated against the engine map and the results can be seen in Figure 4.3 for the Scania engine and Figure 4.4 for the Volvo engine.

Assumptions

Turbulent flow with no temperature change in the fluid.

Remarks

The equations are valid even for reverse flow, as happens when p_us ≤ p_ds. Upstream and downstream pressure and temperature variables are exchanged. If backflow occurs, p_us ↔ p_ds and T_us ↔ T_ds and the output flow is negative.

Validation Plots

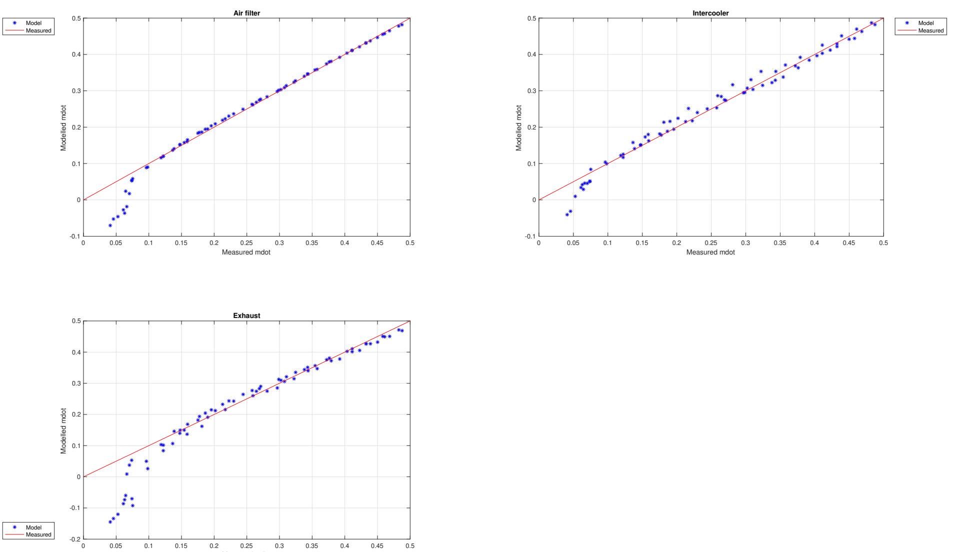

Figure 4.3: *The air filter, intercooler and exhaust system flow validation, validating ctu_{af}, ctu_{in} and ctu_{ex} for the Scania engine.*

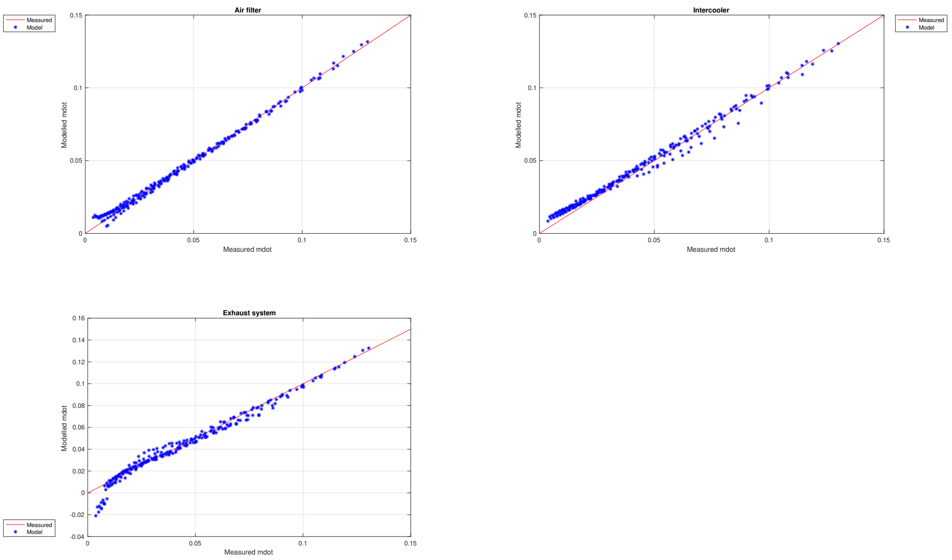

Figure 4.4: *The air filter, intercooler and exhaust system flow validation, validating ctu_{af}, ctu_{in} and ctu_{ex} for the Volvo engine.*

4.7.3 Control Volumes

There are several control volumes in the system and all of them are modeled in the same way. These are

- control volume after the air filter
- control volume after the compressor
- control volume after the intercooler
- intake manifold
- exhaust manifold
- control volume after turbine

There are two available control volumes in the model library, isothermal and adiabatic. Only adiabatic control volumes are used in the presented model. The component for adiabatic control volume is shown in Figure 4.5 and the component for the isothermal control volume is shown in Figure 4.6.

Adiabatic Control Volume

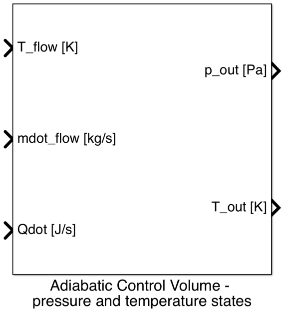

Figure 4.5: *This figure shows the adiabatic control volume.*

Inputs

mdot_flow	kg/s	Mass flow
Qdot	J/s	Heat flow into the system
T_flow	K	Temperature

Outputs

p_out	Pa	Pressure in the control volume
T_out	K	Temperature in the control volume

Parameters

c_v	J/K	Specific heat coefficient
p_init	K	Initial pressure
R	J/kg K	Gas constant
T_init	K	Initial temperature
V	m^3	Volume

Parameter Estimation

The volume V corresponds to the volume of hoses and manifolds between each component.

T_init and p_init are the initial temperature and pressure, which in this thesis is set to the measured value for each operating point.

R and c_v are the gas constant and the specific heat coefficient. The control volumes for the air filter, compressor, intercooler, and intake manifold use the gas constant and specific heat coefficient for air and the exhaust manifold, and the turbine uses the gas constant and specific heat coefficient for exhaust gas.

Assumptions

- The fluid is a prefect gas
- No gas dynamics

Remarks

The equations are valid even for reverse flow. A negative value of the input flow is considered as an output flow for the control volume.

Isothermal Control Volume

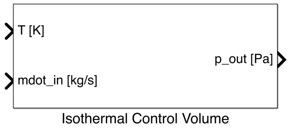

Figure 4.6: This figure shows the isothermal control volume.

Inputs

mdot	kg/s	Mass flow
T	K	Temperature

Outputs

p_out	Pa	Pressure in the control volume

Parameters

c_v	J/K	Specific heat coefficient
p_init	K	Initial pressure
R	J/kg K	Gas constant
V	m^3	Volume

Parameter Estimation

The same parameter estimation applies for the isothermal control volume as for the adiabatic for the parameters V, p_init, R, and c_v.

Assumptions

The same assumptions are made for the isothermal control volume as for the adiabatic control volume.

Remarks

The same remarks are made for the isothermal control volume as for the adiabatic control volume.

4.7.4 Compressible Flow Restriction

The flow restrictions that are compressible are

- bypass
- throttle
- valves
- wastegate

The effective area is for bypass and wastegate modeled as a fraction of how open the valve is. For the cylinder valves, the effective area is determined by the lift profiles and the tabulated values of the discharge coefficient and lastly, the effective area for the throttle is modeled using a polynomial with throttle plate angle. Figure 4.7 shows the component for the incompressible flow.

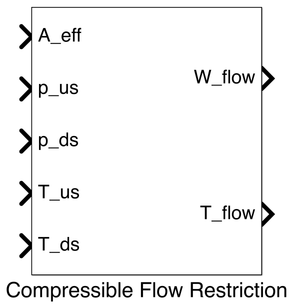

Figure 4.7: *This figure shows the compressible flow restriction.*

Inputs

A_eff	m^2	Effective area
p_us	Pa	Pressure upstream
p_ds	Pa	Pressure downstream
T_us	K	Temperature upstream
T_ds	K	Temperature downstream

Outputs

mdot_flow	kg/s	Mass flow
T_flow	K	Temperature

Parameters

gamma	-	Specific heat constant
R	J/kg K	Gas constant

Parameter Estimation

The compressible flow restrictions for bypass and throttle use the gas constant for air, the cylinder valves use the NASA polynomial to find the gas constant using the oxygen concentration, temperature, and various molecular masses to determine the gas composition. Further calculations of the NASA polynomial can be seen in Section 3.9.2. The wastegate uses the gas constant for exhaust gas.

γ is the specific heat ratio, $\frac{c_p}{c_v}$, $\left(\frac{\text{Heat capacity, constant volume}}{\text{Heat capacity, constant pressure}}\right)$. Bypass and throttle use the specific heat ratio for air, the cylinder valves use the NASA polynomials to determine the specific heat ratio and the wastegate uses the specific heat ratio for exhaust gas.

Assumptions

Isenthalpic restriction with no temperature change over the restriction.

Remarks

The equations are valid even for reverse flow, see assumptions for the incompressible flow restriction. This model does not fulfill the Lipschitz condition when p_us = p_ds, which can create problems in some cases.

4.7.5 Throttle plate area

Throttle flow is modeled as compressible flow, as seen in the section compressible flow restriction. This model describes the throttle angle dynamics and the effective area for the throttle. Figure 4.8 shows the component for the incompressible flow.

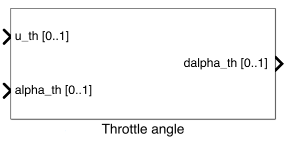

Figure 4.8: *This figure shows the throttle angle flow.*

Inputs

alpha_th	-	Current throttle angle
u_th	-	Desired throttle angle

Outputs

dalpha_th	rad/s	Angle change

Parameters

A0, A1, A2	-	Coefficients in the effective area for the throttle
tau_th	s	Rise time of throttle
delay	s	Throttle angle delay

Parameter Estimation

tau_th is found doing a dynamic step response in throttle angle. The rise time is then the time it takes for the throttle amplitude going from 10% to 90% of the desired output amplitude.

The delay is associated with the throttle dynamics. The throttle has both delay- and rise-time dynamics to reach the desired throttle angle. In this thesis, the delay for the throttle is 0.001 seconds.

The coefficients A0, A1, and A2 for the throttle effective area are found by solving a system of linear equations. This model gives good estimations of the flow rate for pressure ratios not so close to 1. For parameter estimations only Π_{th} < 0.8 are used.

The tuning parameters are then validated against the measurements in the engine map and the results can be seen in Figure 4.9 for the Volvo engine. The Scania engine is not validated since it was decided the Scania model should not be using the throttle.

Assumptions

Same as the compressible restriction.

Validation Plots

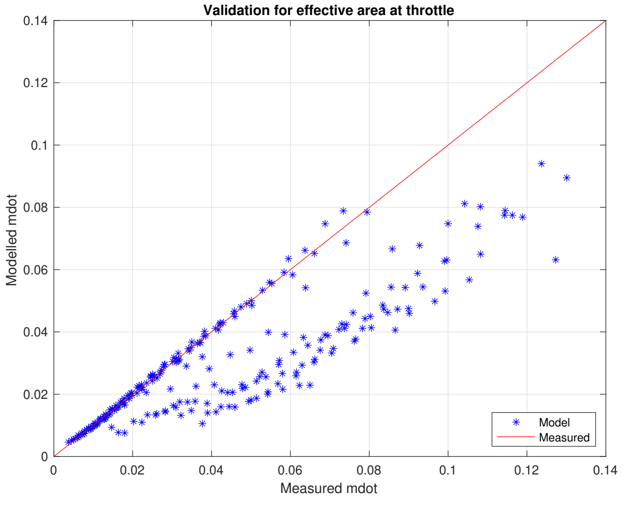

Figure 4.9: *Throttle flow validation, validating A_0, A_1 and A_2 for the Volvo engine.*

4.7.6 Compressor

The parameter estimation for the compressor function presented in section 3.4 used for the Volvo engine is presented below. The parameters for the Scania engine which used the LiU-Diesel compressor model was already fitted and given in measurement data. Figure 4.10 shows the component for the compressor block.

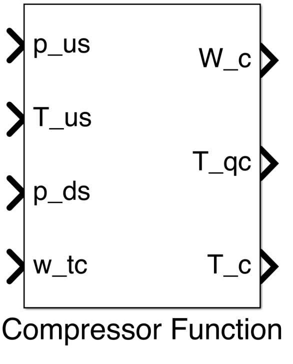

Figure 4.10: *This figure shows the compressor.*

Inputs

p_us	Pa	Pressure upstream
p_ds	Pa	Pressure downstream
T_us	K	Temperature upstream
w_tc	rad/s	Rotational speed of turbo shaft

Outputs

T_qc	Nm	Torque from compressor
T_c	K	Temperature in compressor
W_c	kg/s	Mass flow

Parameters

cp_air	-	Heat capacity
etac_max	-	Maximum compressor efficiency
etac_min	-	Minimum compressor efficiency
gamma_air	-	Specific heat constant
mccorr_etamax	kg/s	Parameter for compressor efficiency place in map
mccorr_max	kg/s	Parameter for corrected mass flow
Pc_ref	Pa	Reference pressure
PI_c_etamax	-	Parameter for compressor efficiency place in map
Psi_max	-	Parameter for maximum compressor pressure ratio
q11, q12, q21, q22	-	Coefficients for compressor efficiency matrix
r_c	m	Compressor wheel radii
Tc_ref	K	Reference temperature

Parameter Estimation

To get the parameters for the compressor model, the compressor map is needed with the measurements listed in Table 4.3. p_{af} and T_{af} are optimal and the mean value from the engine map can also be used.

For heat capacity and specific heat constant, the values for air are used since the gas flowing through the compressor is assumed to be 100% air. Pc_ref and Tc_ref are taken directly from the measurement in the map and are used to calculate the corrected quantities. The compressor radii r_c is either measured physically or given in the map or on a drawing.

Psi_max and mccorr_max which are found by plugging (3.12) into (3.13) using $\omega_{tc} = N_{c,corr}\sqrt{T_{af}/T_{c,ref}} \cdot \frac{2\pi}{60}$ and solving a non-linear least squares problem with Psi_max and mccorr_max as optimization variables.

etac_max, mccorr_etamax, PI_c_etamax, q11, q12, q21 and q22 which are found by putting equation (3.16) into (3.15) by solving a non-linear least squares problem with etac_max, mccorr_etamax, PI_c_etamax, q11, q12, q21 and q22 as optimization variables. Since the Q-matrix is symmetrical, q12 = q21.

The compressor efficiency is calculated as an ellipse. For modeling reasons etac_min = etac_max/2 to avoid negative efficiency.

The tuning parameters are then validated against the compressor map. The results can be seen in Figure 4.11. Since the parameters for the Scania engine were already fitted, the validation was not performed on the Scania engine.

Assumptions

Angular velocity w_tc and mass flow are assumed to always be greater than 0.

Validation Plots

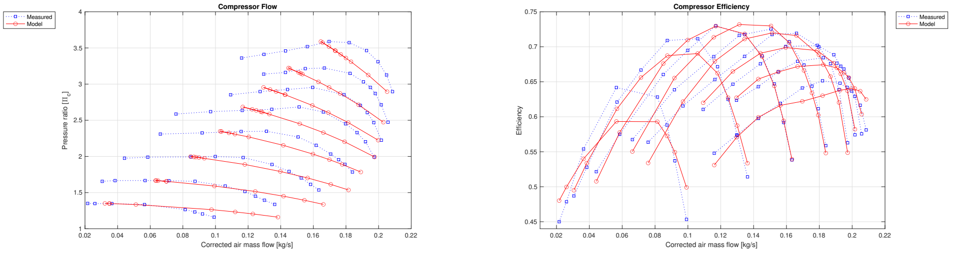

Figure 4.11: *The compressor flow validation, validating Ψ_{max} and $\dot{m}_{c,corr,max}$ for compressor flow and $\eta_{c,max}$, $\dot{m}_{c,corr@\eta_{c,max}}$, $\Pi_{c@\eta_{c,max}}$, Q_{11}, Q_{12} and Q_{22} for compressor efficiency for the Volvo engine.*

4.7.7 Bypass and Wastegate model

The flow is based on the compressible flow restriction model. Figure 4.12 shows the component for bypass and wastegate.

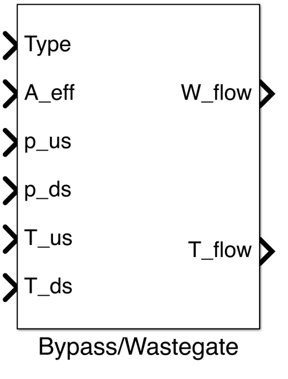

Figure 4.12: *This figure shows the bypass and wastegate.*

Inputs

A_eff	m^2	Effective area
p_us	Pa	Pressure upstream
p_ds	Pa	Pressure downstream
T_us	K	Temperature upstream
T_ds	K	Temperature downstream
Type	-	Parameter deciding bypass or wastegate

Outputs

W_flow	kg/s	Mass flow
T_flow	K	Temperature

Parameters

gamma	-	Specific heat constant
R	J/kg K	Gas constant
delay	s	Wastegate position delay

Parameter Estimation

The input parameter Type is [1/0] deciding if the component is bypass or wastegate. This affects the specific heat and gas constant. Specific heat and gas constant for air is used for bypass and specific heat and gas constant for exhaust gas is used for the wastegate.

The delay is associated with the wastegate dynamics. The wastegate has delay dynamics to reach the desired position. In this thesis, the delay for the wastegate is 0.001 seconds.

Assumptions

Same as compressible flow restriction.

4.7.8 Turboshaft Dynamics model

The turboshaft dynamics are modeled according to Newton's second law for rotating systems.

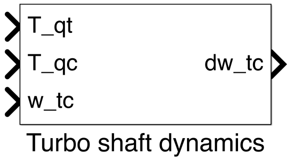

Figure 4.13: *This figure shows the turbo shaft dynamics.*

Inputs

T_qc	Nm	Torque from compressor
T_qt	Nm	Torque from turbine
w_tc	rad/s	Rotational speed of turbo shaft

Outputs

dw_tc	rad/s^2	Change of rotational speed of turbo shaft

Parameters

c_tc	Nm/(rad/s)	Friction coefficient of turbo shaft
J_tc	kgm^2	Inertia of turbo shaft
w_init	rad/s	Initial rotational speed

Parameter Estimation

The friction coefficient c_tc is set to $1 \cdot 10^{-6}$ Nm/(rad/s) for an oil lubricated shaft.

The inertia J_tc can either be measured using a test rig [29] or used as a tuning parameter while validating the complete model. In this thesis, the inertia of the turboshaft was previously measured using the test rig.

w_init is the initial rotational speed of the turboshaft. In this thesis, the initial rotational speed is the measured value for each operating point.

4.7.9 Intercooler

The intercooler uses the incompressible flow restriction previously described earlier. The difference is the model for the temperature as the intercooler is cooling the charged air. The input and output of the intercooler are the same as for the incompressible flow restriction. The only difference is the parameters for the temperature model.

Figure 4.2 shows the component for the incompressible flow.

Inputs

p_ds	K	Pressure downstream
p_us	K	Temperature upstream
T_ds	K	Temperature downstream
T_us	K	Temperature upstream

Outputs

m_dot_out	kg/s	Mass flow through the restriction
T_out	K	Temperature through the restriction

Parameters

a0, a1, a2	-	Coefficients in the polynomial for the intercooler effectiveness
c_tun	-	Tuning parameter for each component
p_li	Pa	Pressure limit
R	J/kg K	Gas constant
T_cool	K	Temperature of the air surrounding the intercooler

Parameter Estimation

c_tun, p_li, and R are described in the incompressible flow restriction.

T_cool is the temperature of the air flowing past the intercooler. In this thesis, T_cool is set to the minimum temperature available in the map after the intercooler and therefore assumed to be the temperature surrounding the intercooler.

a0, a1 and a2 and by plugging equation (3.25) into (3.23) and solving a non-linear least squares problem with a0, a1 and a2 as optimization variables. The tuning parameters are compared to the engine map and results can be seen in Figure 4.14 for the Scania engine and Figure 4.15 for the Volvo engine.

Validation Plots

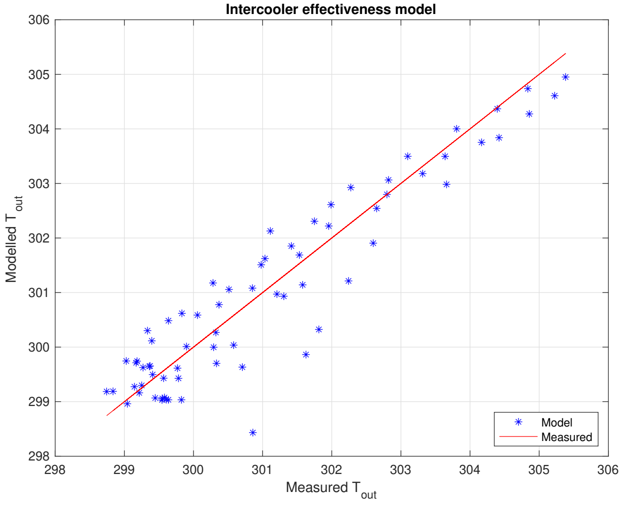

Figure 4.14: *The intercooler effectiveness validation, validating a_0, a_1 and $_2$ for the Scania engine.*

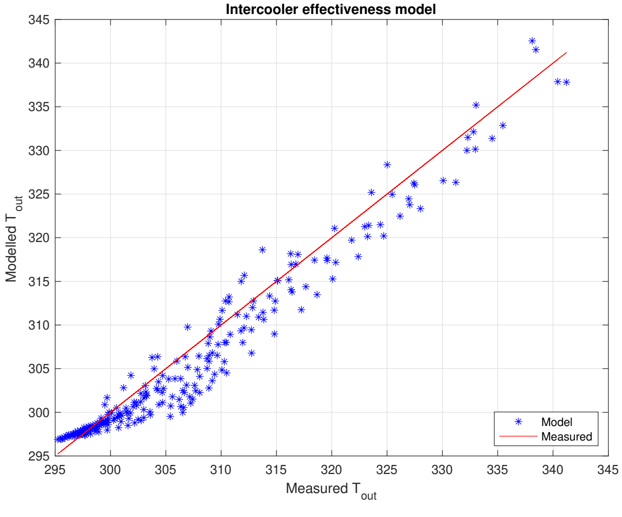

Figure 4.15: *The intercooler effectiveness validation, validating a_0, a_1 and $_2$ for the Volvo engine.*

4.7.10 Intake and Exhaust manifold

The intake and exhaust manifold consists of an adiabatic control volume and a function for oxygen concentration. Figure 4.16 shows the component for intake manifold and FIgure 4.17 shows the component for exhaust manifold.

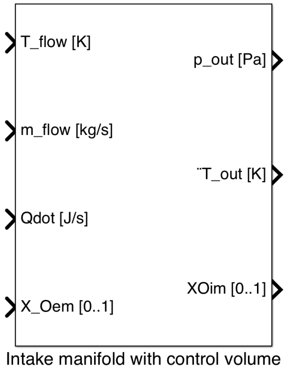

Figure 4.16: *This figure shows the intake manifold.*

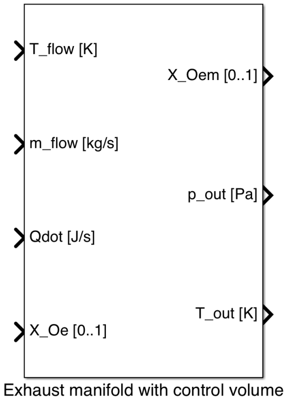

Figure 4.17: *This figure shows the exhaust manifold.*

Inputs

mdot_flow	kg/s	Mass flow
Qdot	J/s	Heat flow into the system
T_flow	K	Temperature
X_Oem / X_oe	-	Oxygen fraction from exhaust manifold/engine

Outputs

p_out	Pa	Pressure in the control volume
T_out	K	Temperature in the control volume
X_Oim / X_oem	-	Oxygen fraction from intake manifold/exhaust manifold

Parameters

c_v	J/K	Specific heat coefficient
p_init	K	Initial pressure
R	J/kg K	Gas constant
T_init	K	Initial temperature
V	m^3	Volume
X_Oc	-	Oxygen fraction from compressor

Parameter Estimation

Parameter estimation for c_v, p_init, R, T_init, and V is described in the adiabatic control volume previously described above.

X_Oc is the constant fraction of air at the compressor and is assumed to be 100% in this thesis.

4.7.11 Cylinder

The cylinder contains models for cylinder flows, the vibe function and oxygen concentration, energy equations, the cylinder states, and the generated engine torque. All parameters for the cylinder model are presented. The cylinder component can be seen in Figure 4.18.

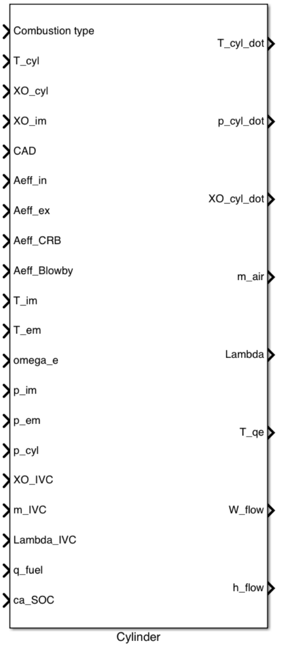

Figure 4.18: *This figure shows the cylinder.*

Inputs

Aeff_Blowby	m^2	Effective area blowby
Aeff_CRB	m^2	Effective area CRB
Aeff_ex	m^2	Effective area exhaust valves
Aeff_in	m^2	Effective area intake valves
CAD	rad	Crank angle
ca_SOC	rad	Crank angle for SOI event
Combustion type	-	Parameters deciding SI or CI
Lambda_IVC	-	Lambda in cylinder at IVC
m_IVC	kg	Mass air in cylinder at IVC
omega_e	rad/s	Engine speed
p_cyl	Pa	Pressure in cylinder
p_em	Pa	Pressure in exhaust manifold
p_im	Pa	Pressure in intake manifold
q_fuel	kg/stroke	Fuel flow
T_cyl	K	Temperature in cylinder
T_em	K	Temperature in exhaust manifold
T_im	K	Temperature in intake manifold
XO_cyl	-	Oxygen concentration in cylinder
XO_im	-	Oxygen concentration in intake manifold
XO_IVC	-	Oxygen concentration at IVC

Outputs

h_flow	kJ/kg	Specific enthalpy
Lambda	-	Lambda in cylinder
m_air	kg	Mass air in cylinder
p_cyl_dot	Pa/s	Change in pressure in cylinder
T_cyl_dot	K/s	Change in temperature in cylinder
T_qe	Nm	Generated engine torque
W_flow	kg/s	Cylinder flows intake, exhaust, blowby and CRB
XO_cyl_dot	-	Change in oxygen concentration in cylinder

Parameters

a	m	Crank radius
a_coal	-	Number of coal atoms in fuel
a_vibe	-	Parameter for combustion duration
AFs	-	Stoichiometric air/fuel ratio
B	m	Engine bore
b_hydrogen	-	Number of hydrogen atoms in fuel
C_0, C_1, C_2	-	Coefficients in the heat transfer
Deltatheta	rad	Parameter for combustion duration
l	m	Connecting rod length
m_vibe	-	Shape of burning profile
MCO2	kg/mol	Molar mass CO2
Mfueltype	kg/mol	Molar mass for the fuel used
MH2O	kg/mol	Molar mass H2O
MN2	kg/mol	Molar mass N2
MO2	kg/mol	Molar mass O2
n	-	Exponent in heat transfer
p_amb	Pa	Ambient pressure
p_IVC	Pa	Pressure in cylinder at IVC
QLHV	J/kg	Lower heating value of fuel
r_c	-	Compression ratio
R_tilde	J/mol K	Gas constant
S	m	Engine stroke
T_amb	K	Ambient temperature
T_IVC	K	Temperature in cylinder at IVC
Tw	K	Cylinder wall temperature
V_d	m^3	Cylinder Volume
V_IVC	m^3	Volume in cylinder at IVC

Parameter Estimation

AFs is the stoichiometric air/fuel ratio and QLHV is the lower heating value of the fuel used. p_amb and T_amb is the ambient pressure and temperature. R_tilde is the gas constant.

Tw is the cylinder wall temp, assumed to be constantly 140 °C for a warm engine [14]. n is the exponent for the heat transfer, which is the specific heat constant gamma_air. C_0, C_1, and C_2 are the coefficients to decide the Woschini [11].

Further, for the heat transfer, parameters from the IVC are needed. Those are V_IVC, T_IVC, and p_IVC which is the volume, temperature, and pressure in the cylinder at IVC. The temperature is assumed to be the temperature in the intake since temperature measurements from the cylinder are not available.

To parameterize the vibe function and IVC event, measurement data needs to be crankshaft-based. A list of the signals can be seen in Table 4.5. a_vibe, m_vibe, and Deltatheta are the vibe parameters. The vibe function is over parameterized which means all vibe parameters can not be uniquely determined. To get the burned fraction, Matekunas pressure ratio is used to get the cylinder pressure. The fraction burned is then calculated as a fraction between the pressure in the cylinder and the maximum pressure in the cylinder according to

$$mfb = \frac{p_{cyl}}{max\left(p_{cyl}\right)}$$

where mfb is the fraction mass fuel burned. It is assumed that all fuel is burned when the cylinder reaches maximum pressure. In this thesis, m_vibe was set to 1 for both engines. Deltatheta is then the crank angle difference to go from 10 % fuel burned, mfb=0.1, to 90 % fuel burned, mfb=0.9. a_vibe is then found by solving a non-linear least-squares problem from SOC to mfb=1.

In this thesis, all simulations were run with the same parameterization. This means that IVC and Vibe parameters only were taken for one operating point for each engine. The parametrized operating point for the Scania engine is 2500 Nm and 1100 RPM. The parametrized operating point for the Volvo engine is 350 Nm and 2000 RPM. Both operating points are considered high load for each engine.

MCO2, Mfueltype, MH2O, MN2, and MO2 are the molar masses for the substances in the engine. All molar masses can be seen in Table A.1. a_coal is the number of coal atoms in the fuel and b_hydrogen is the number of hydrogen atoms in the fuel. For Cetane, a_coal is 16 and b_hydrogen is 34 and for Isooctane a_coal is 8 and b_hydrogen is 18.

4.7.12 Turbine

The parameter estimation for the turbine function presented in section 3.5 used for the Volvo engine is presented below. The parameters for the Scania engine which used the LiU-Diesel turbine model was already fitted and given in measurement data. Figure 4.19 shows the component for the turbine block.

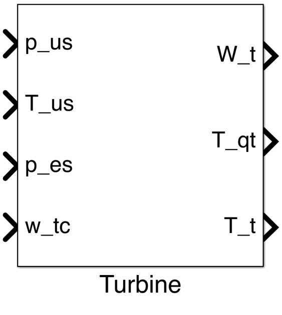

Figure 4.19: *This figure shows the turbine*

Inputs

p_us	Pa	Pressure upstream
p_ds	Pa	Pressure downstream
T_us	K	Temperature upstream
w_tc	rad/s	Rotational speed of turbo shaft

Outputs

T_qc	Nm	Torque from compressor
T_c	K	Temperature in compressor
W_c	kg/s	Mass flow

Parameters

BSR_max	-	Turbine efficiency parameter
cp_gas	-	Heat capacity
etat_max	-	Maximum turbine efficiency
etat_min	-	Minimum turbine efficiency
gamma_gas	-	Specific heat constant
k0, k1	-	Turbine flow parameter
r_t	m	Turbine wheel radii

Parameter Estimation

To get the parameters for the turbine model, the compressor map is needed with the measurements listed in Table 4.4.

For heat capacity and specific heat constant, the values for exhaust are used since the gas flowing through the turbine is assumed fully combusted. The turbine radii r_t is either measured physically or given in the map or on a drawing.

k0 and k1 are found by using equation (3.17) and solving a least-squares problem with k0 and k1 as optimization variables.

etat_max and BSR_max are solved by using equation (3.19) and plugging that into equation (3.20) using $\omega_{tc} = TSP\sqrt{T_{em}} \cdot \frac{2\pi}{60}$ and solving a non-linear least squares problem with etat_max and BSR_max as optimization variables.

The turbine efficiency is calculated as a parabolic function. For modeling reasons, etat_min is the minimum turbine efficiency from the turbine map divided by 2 to avoid negative efficiency.

The tuning parameters are then validated against the turbine map. The results can be seen in Figure 4.20. Since the parameters for the Scania engine were already fitted, the validation was not performed on the Scania engine.

Validation Plots

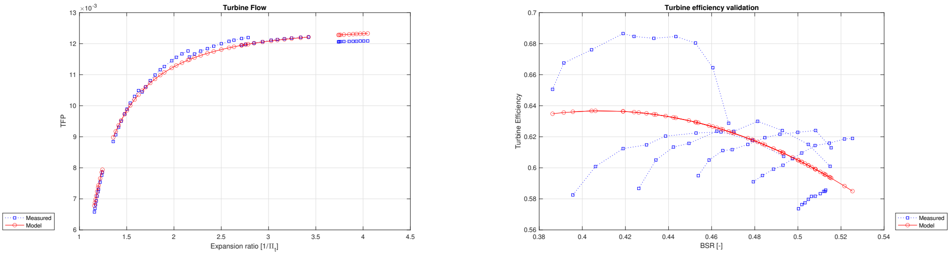

Figure 4.20: *The turbine flow validation, validating k_0 and k_1 for turbine flow and $\eta_{t,max}$ and BSR_{max} for turbine efficiency for the Volvo engine.*

4.7.13 Adiabatic mixer

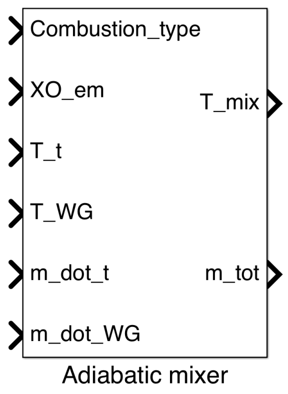

Figure 4.21: *This figure shows the adiabatic mixer.*

Inputs

Combustion_type	-	Parameters deciding SI or CI
m_dot_t	kg/s	Mass flow in turbine
m_dot_WG	kg/s	Mass flow in wastegate
T_t	K	Temperature from turbine
T_WG	K	Temperature from wastegate
XO_em	-	Oxygen concentration in exhaust manifold

Outputs

m_tot	kg/s	Total mass flow in turbine and wastegate
T_mix	K	Temperature of the flow from turbine and wastegate

Parameters

R_tilde	J/mol K	Gas constant

Assumptions

- NASA polynomials are used to determine ideal gas constant and specific heat constant for the mixed flow
- The flow is only in one direction.
- Adiabatic mixing, no heat transfer.
- The mixing is done at constant pressure.

4.8 Regulators

To validate the Volvo model, two regulators were implemented to get the right pressures. The two regulators were the throttle regulator, regulating the intake pressure and the second regulator was the wastegate regulator, regulating the compressor pressure. The regulators use a PID controller. The parameters for the PID controller was tuned manually for each operating point as the desired behaviour was just to reduce oscillations in angle or position change.

The throttle regulator uses the desired intake pressure as a reference and creates an error between reference and simulated intake pressure to increase or decrease the throttle angle to reduce the error. The modeled value for intake pressure is going through a low pass filter to reduce oscillations. The gain is used to reduce the amplitude of the error going into the PID controller. The regulator can be seen in Figure 4.22.

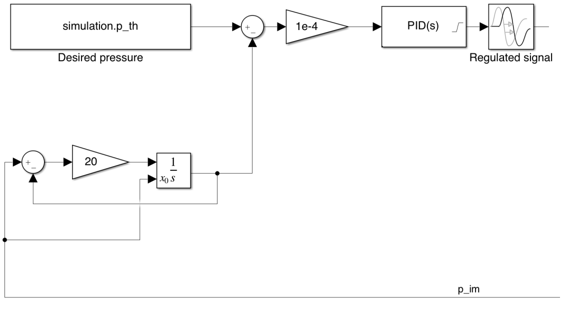

Figure 4.22: *The implemented throttle regulator with low pass filter for measured signal*

The same approach was used for the wastegate regulator. Measured compressor pressure was used as a reference and the regulator regulated the error between modeled and reference compressor pressure. A low pass filter was also used in that controller to reduce oscillations and a gain to reduce amplitude. The regulator can be seen in Figure 4.23.

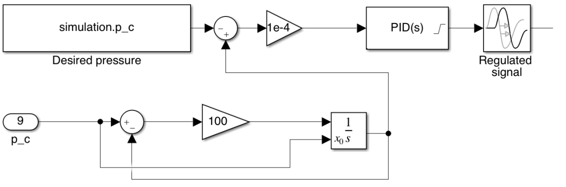

Figure 4.23: *The implemented wastegate regulator with low pass filter for measured signal*

5

Results

In this chapter, the model performance is presented through model agreement simulations of the model.

Some components were not simulated for the Volvo and Scania engine, as described in section 4.7. This meant some inputs were kept the same for each iteration. The inputs that were kept the same for the Scania engine were throttle angle, wastegate, bypass, and EGR actuation and were set to zero for the validation. The inputs that were kept the same for the Volvo engine were bypass, EGR, and CRB actuation and were set to zero for the validation. By setting inputs to zero means has the same effect as if they were non-existent.

To validate the model, five different operating points are presented in this chapter. These operating points are high and low load with high and low engine speeds, as well as a point with medium load and medium engine speed. Symmetrical drag and CRB operating points are presented in the Appendix C.

In Tables 5.2 to 5.15 below as well as Tables C.2 to C.5 in Appendix C the parameters are pressure, temperature, turbo shaft speed, engine torque, mass flows and oxygen concentration. The parameters for pressure are after the compressor, p_c, in the intake manifold, p_im, and after the exhaust manifold, p_em. The parameters for the temperature are after the compressor, T_c, in the intake manifold, T_im, after the exhaust manifold, T_em, and after the turbine, T_t. The parameter for the turbo shaft speed is w_tc, the parameter for generated engine torque is Tq_e. The parameters for mass flow are all mass flows in the model. The parameter for oxygen concentration is after the exhaust manifold, Xo_em, but a reference for this measurement is only available for the Scania engine. The units used to validate the model is *bar* for pressure, °*C* for temperature, *kRPM* for turbo speed, *Nm* for generated torque, *g/s* for mass flows and % for oxygen concentration. Each operating point is further investigated under its respective section.

Results are presented when the model has reached a steady value. The simulation time was set to 10 seconds which sufficiently made sure all dynamics were gone at the end of the simulation. Measured data was investigated on a cycle to cycle basis. The simulation reached steady state after about 50 cycles. The results presented below are taken at the 9.5-second mark for the cycle data and a mean of the last 50 cycles for the Volvo engine and the last 25 cycles for the Scania engine, for mean and max error. This is due to difference in engine speed between the engines. The cycle data is over 720 degrees, which is a complete cycle for the four-stroke engine.

In Figure 5.1 the measurement for T_c is presented in the region data is analyzed for the mean operating point for both the Scania and Volvo engine to illustrate the meaning of modeled, measured, mean error. Mean error is the percentage error between the measured signal and the mean of the measured signal according to

$$E = \frac{\text{mean(modelled value)} - \text{measured value}}{\text{measured value}} \tag{5.1}$$

where E is the error. A positive error means modeled data is larger than measured and modeled data is less than measured for negative error.

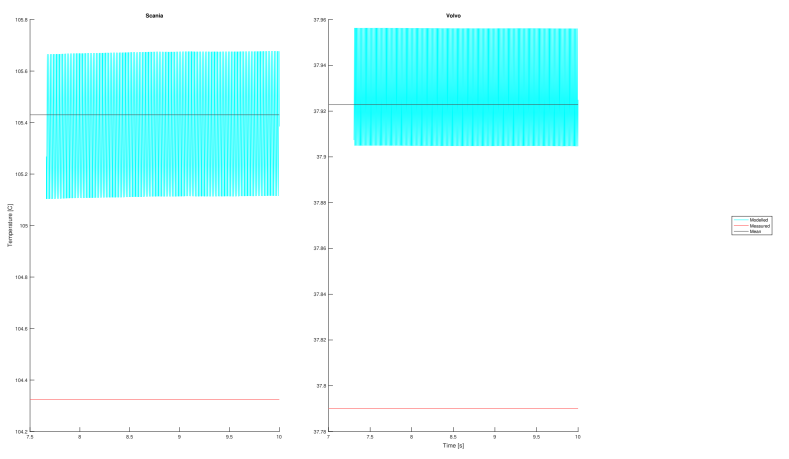

Figure 5.1: *Simulated and measured T_c with mean errors demonstrated for the mean operating point for both the Scania and Volvo engine.*

5.1 High Load and High Engine Speed

Here the results for different parameters are presented for the high load and high engine speed operational point. The non-constant parameters needed to run this operational point for the different engines are presented in Table 5.1. In Table 5.2 and 5.3 mean value data is presented for different parameters and in Figure 5.2 and 5.3 cycle data is presented for the Scania and Volvo engine respectively.

Table 5.1: *Operating points run for Scania and Volvo engine of the high load and high engine speed operating point.*

Parameter	Scania	Volvo
Engine Speed [RPM]	1700	4000
Throttle Angle [Deg]	-	38.24
Fuel Flow [mg/stroke]	203.58	94.46
SOI [Deg]	-5.22	10.00
Crank Angle Intake Offset [Deg]	30.12	-32.66
Crank Angle Exhaust Offset [Deg]	-30.11	0.00

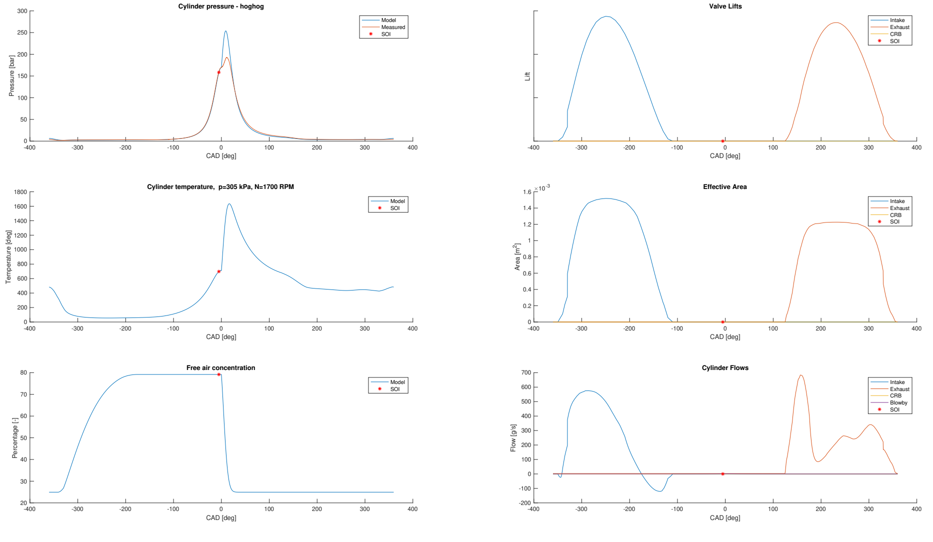

Figure 5.2: *Cylinder pressure, temperature, oxygen concentration, valve lift, effective area for the valves and the cylinder flows over one complete cycle on the high load and high engine speed operating point for the Scania engine.*

Table 5.2: *Mean error and max error for different parameters for the Scania engine on the high load and high engine speed operating point.*

Parameter	Modelled value	Measured value	Units	Mean error [%]
p_c	2.96	3.13	bar	-5.63
p_im	2.86	3.05	bar	-6.21
p_em	3.21	3.21	bar	-0.09
T_c	185.05	178.18	C	3.86
T_im	32.04	30.93	C	3.60
T_em	485.23	494.63	C	-1.90
T_t	386.51	408.55	C	-5.39
w_tc	102.16	99.71	kRPM	2.46
Tq_e	1627.15	1900.04	Nm	-14.36
Lambda	1.71	1.89	-	-9.66
W_af	490.00	474.82	g/s	3.20
W_c	490.00	474.82	g/s	3.20
W_ic	489.92	474.82	g/s	3.18
W_cyl_in	489.86	474.82	g/s	3.17
W_cyl_out	523.12	647.82	g/s	-19.25
W_t	523.13	647.82	g/s	-19.25
W_es	523.18	647.82	g/s	-19.24
Xo_em	25.67	10.36	%	15.31

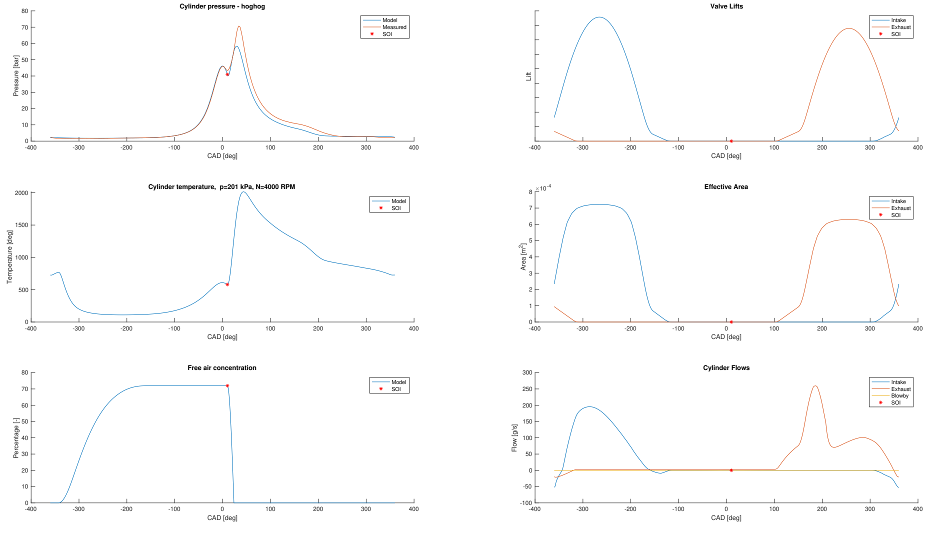

Figure 5.3: *Cylinder pressure, temperature, oxygen concentration, valve lift, effective area for the valves and the cylinder flows over one complete cycle on the high load and high engine speed operating point for the Volvo engine.*

Table 5.3: *Mean error and max error for different parameters for the Volvo engine on the high load and high engine speed operating point.*

Parameter	Modelled value	Measured value	Units	Mean error [%]
p_c	2.15	2.15	bar	0.00
p_im	2.03	2.01	bar	0.69
p_em	2.87	2.69	bar	6.62
T_c	145.58	153.63	C	-5.24
T_im	63.72	72.11	C	-11.63
T_em	1011.99	956.62	C	5.79
T_t	921.46	815.31	C	13.02
w_tc	156.71	187.00	kRPM	-16.20
Tq_e	245.20	300.02	Nm	-18.27
Lambda	0.54	0.74	-	-26.43
W_af	112.80	130.09	g/s	-13.29
W_c	112.80	130.09	g/s	-13.29
W_ic	112.80	130.09	g/s	-13.29
W_th	112.79	130.09	g/s	-13.30
W_cyl_in	112.79	130.09	g/s	-13.30
W_cyl_out	137.40	133.24	g/s	3.12
W_t	137.40	133.24	g/s	3.12
W_es	137.40	133.24	g/s	3.12

5.2 High Load and Low Engine Speed

Here the results for different parameters are presented for the high load and low engine speed operational point. The non-constant parameters needed to run this operational point for the different engines are presented in Table 5.4. In Table 5.5 and 5.6 mean value data is presented for different parameters and in Figure 5.4 and 5.5 cycle data is presented for the Scania and Volvo engine respectively.

Table 5.4: *Operating points run for Scania and Volvo engine of the high load and low engine speed operating point.*

Parameter	Scania	Volvo
Engine Speed [RPM]	900	1750
Throttle Angle [Deg]	-	40.72
Fuel Flow [mg/stroke]	252.47	17.97
SOI [Deg]	-4.93	13.00
Crank Angle Intake Offset [Deg]	-0.01	-48.00
Crank Angle Exhaust Offset [Deg]	-0.24	30.00

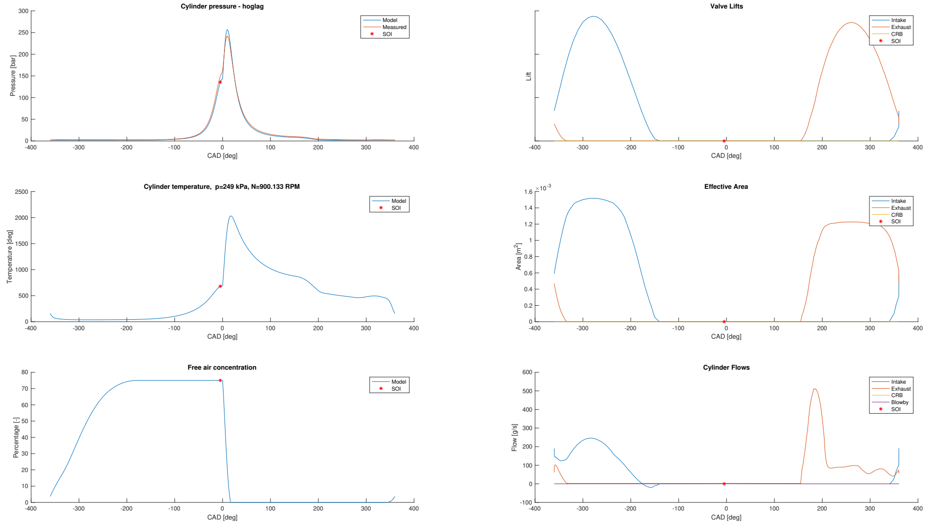

Figure 5.4: *Cylinder pressure, temperature, oxygen concentration, valve lift, effective area for the valves and the cylinder flows over one complete cycle on the high load and low engine speed operating point for the Scania engine.*

Table 5.5: *Mean error and max error for different parameters for the Scania and engine on the high load and low engine speed operating point.*

Parameter	Modelled value	Measured value	Units	Mean error [%]
p_c	2.23	2.52	bar	-11.54
p_im	2.20	2.49	bar	-11.60
p_em	1.88	2.04	bar	-7.70
T_c	143.22	127.10	C	12.68
T_im	26.91	27.07	C	-0.59
T_em	585.80	412.67	C	41.95
T_t	523.26	390.98	C	33.83
w_tc	77.40	80.41	kRPM	-3.75
Tq_e	2044.44	2500.07	Nm	-18.22
Lambda	1.12	1.66	-	-32.19
W_af	244.17	272.99	g/s	-10.56
W_c	244.17	272.99	g/s	-10.56
W_ic	244.16	272.99	g/s	-10.56
W_cyl_in	244.16	272.99	g/s	-10.56
W_cyl_out	265.10	386.63	g/s	-31.43
W_t	265.63	386.63	g/s	-31.29
W_es	265.71	386.63	g/s	-31.28
Xo_em	1.79	8.69	%	-6.90

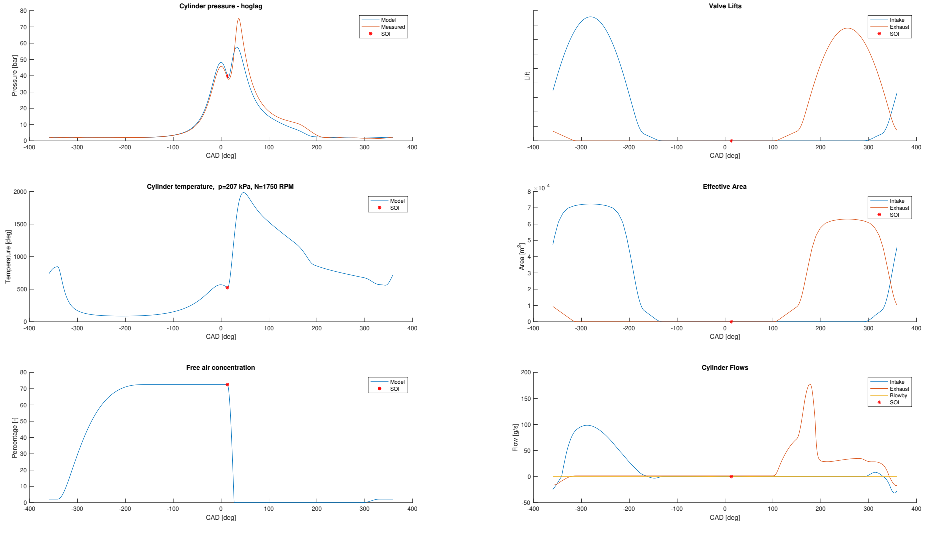

Figure 5.5: *Cylinder pressure, temperature, oxygen concentration, valve lift, effective area for the valves and the cylinder flows over one complete cycle on the high load and low engine speed operating point for the Volvo engine.*

Table 5.6: *Mean error and max error for different parameters for the Volvo engine on the high load and low engine speed operating point.*

Parameter	Modelled value	Measured value	Units	Mean error [%]
p_c	2.10	2.09	bar	0.46
p_im	2.07	2.07	bar	0.23
p_em	2.13	1.87	bar	13.86
T_c	166.02	137.25	C	20.96
T_im	43.18	48.78	C	-11.48
T_em	968.09	902.01	C	7.33
T_t	866.13	762.83	C	13.54
w_tc	138.58	160.21	kRPM	-13.51
Tq_e	283.27	344.07	Nm	-17.67
Lambda	0.56	0.79	-	-28.98
W_af	55.02	64.96	g/s	-15.31
W_c	55.02	64.96	g/s	-15.31
W_ic	55.03	64.96	g/s	-15.29
W_th	55.05	64.96	g/s	-15.27
W_cyl_in	55.08	64.96	g/s	-15.22
W_cyl_out	66.20	66.43	g/s	-0.34
W_t	66.29	66.43	g/s	-0.22
W_es	66.29	66.43	g/s	-0.21

5.3 Low Load and High Engine Speed

Here the results for different parameters are presented for the low load and high engine speed operational point. The non-constant parameters needed to run this operational point for the different engines are presented in Table 5.7. In Table 5.8 and 5.9 mean value data is presented for different parameters and in Figure 5.6 and 5.7 cycle data is presented for the Scania and Volvo engine respectively.

Table 5.7: *Operating points run for Scania and Volvo engine of the low load and high engine speed operating point.*

Parameter	**Scania**	**Volvo**
Engine Speed [RPM]	1700	4000
Throttle Angle [Deg]	-	6.66
Fuel Flow [mg/stroke]	48.23	8.58
SOI [Deg]	-6.04	-7.50
Crank Angle Intake Offset [Deg]	30.03	-15.38
Crank Angle Exhaust Offset [Deg]	-30.00	0.00

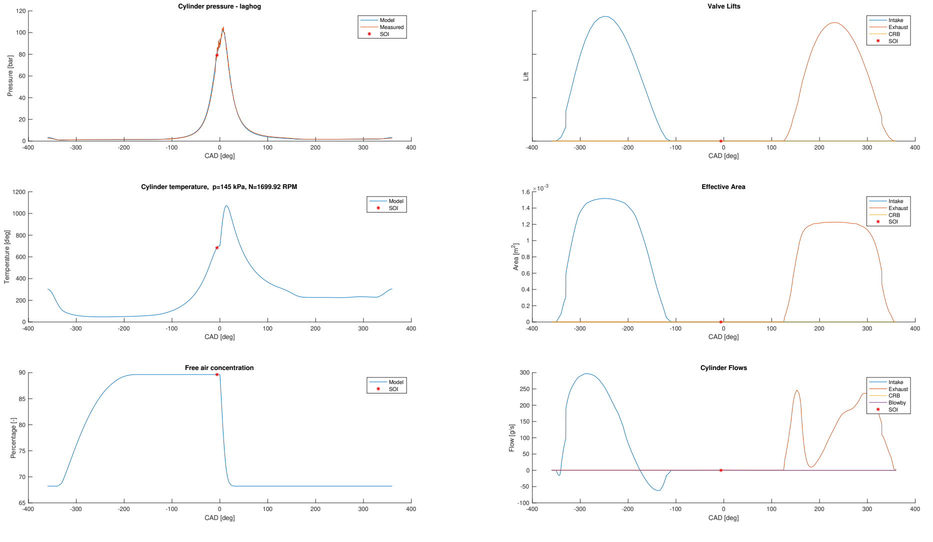

Figure 5.6: *Cylinder pressure, temperature, oxygen concentration, valve lift, effective area for the valves and the cylinder flows over one complete cycle on the low load and high engine speed operating point for the Scania engine.*

Table 5.8: Mean error and max error for different parameters for the Scania engine on the low load and high engine speed operating point.

Parameter	Modelled value	Measured value	Units	Mean error [%]
p_c	1.49	1.49	bar	-0.06
p_im	1.45	1.45	bar	0.01
p_em	1.57	1.46	bar	7.61
T_c	85.51	69.44	C	23.13
T_im	27.93	27.49	C	1.58
T_em	236.51	266.57	C	-11.28
T_t	203.21	228.31	C	-10.99
w_tc	57.51	54.82	kRPM	4.90
Tq_e	362.47	379.93	Nm	-4.60
Lambda	4.24	3.80	-	11.63
W_af	250.97	225.70	g/s	11.20
W_c	250.97	225.70	g/s	11.20
W_ic	250.87	225.70	g/s	11.15
W_cyl_in	250.82	225.70	g/s	11.13
W_cyl_out	258.87	266.69	g/s	-2.93
W_t	258.80	266.69	g/s	-2.96
W_es	258.83	266.69	g/s	-2.95
Xo_em	68.35	15.82	%	52.53

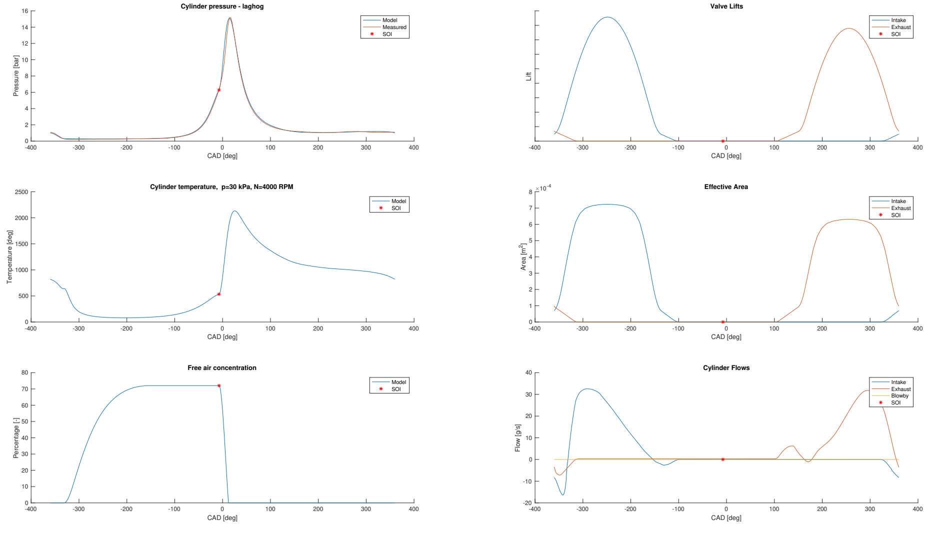

Figure 5.7: Cylinder pressure, temperature, oxygen concentration, valve lift, effective area for the valves and the cylinder flows over one complete cycle on the low load and high engine speed operating point for the Volvo engine.

Table 5.9: *Mean error and max error for different parameters for the Volvo engine on the low load and high engine speed operating point.*

Parameter	Modelled value	Measured value	Units	Mean error [%]
p_c	1.04	1.04	bar	0.00
p_im	0.30	0.30	bar	0.46
p_em	1.06	1.05	bar	0.71
T_c	38.96	38.81	C	0.40
T_im	26.20	30.93	C	-15.29
T_em	975.68	775.31	C	25.84
T_t	976.45	727.07	C	34.40
w_tc	26.24	22.22	kRPM	18.08
Tq_e	34.05	15.95	Nm	113.56
Lambda	0.94	0.99	-	-5.15
W_af	16.07	15.88	g/s	1.17
W_c	16.07	15.88	g/s	1.17
W_ic	16.07	15.88	g/s	1.17
W_th	16.07	15.88	g/s	1.17
W_cyl_in	16.07	15.88	g/s	1.17
W_cyl_out	18.31	16.17	g/s	13.22
W_t	18.31	16.17	g/s	13.24
W_es	18.31	16.17	g/s	13.25

5.4 Low Load and Low Engine Speed

Here the results for different parameters are presented for the low load and low engine speed operational point. The non-constant parameters needed to run this operational point for the different engines are presented in Table 5.10. In Table 5.11 and 5.12 mean value data is presented for different parameters and in Figure 5.8 and 5.9 cycle data is presented for the Scania and Volvo engine respectively.

Table 5.10: *Operating points run for Scania and Volvo engine of the low load and low engine speed operating point.*

Parameter	Scania	Volvo
Engine Speed [RPM]	900	1750
Throttle Angle [Deg]	-	3.31
Fuel Flow [mg/stroke]	54.81	7.65
SOI [Deg]	-4.30	-11.5
Crank Angle Intake Offset [Deg]	0.16	-13.56
Crank Angle Exhaust Offset [Deg]	-0.47	10.27

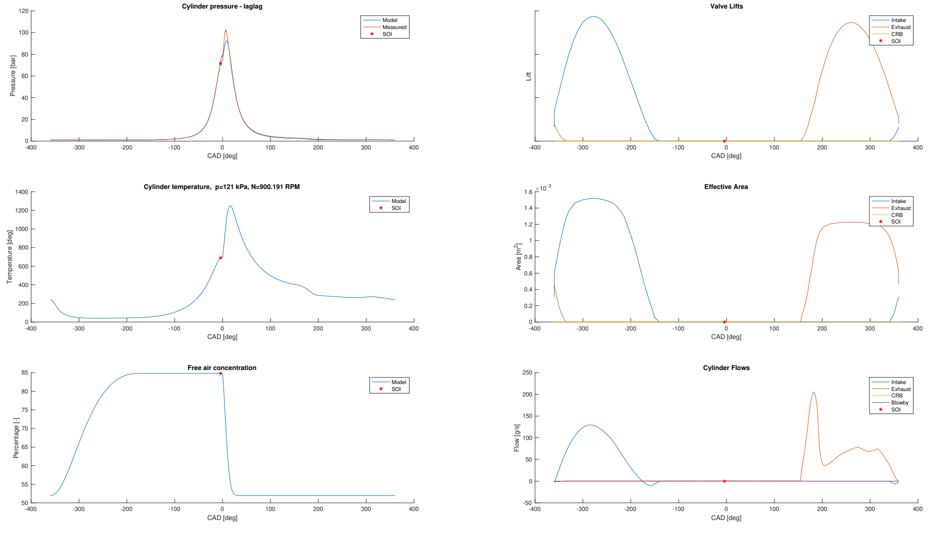

Figure 5.8: *Cylinder pressure, temperature, oxygen concentration, valve lift, effective area for the valves and the cylinder flows over one complete cycle on the low load and low engine speed operating point for the Scania engine.*

Table 5.11: *Mean error and max error for different parameters for the Scania engine on the low load and low engine speed operating point.*

Parameter	Modelled value	Measured value	Units	Mean error [%]
p_c	1.16	1.23	bar	-5.28
p_im	1.15	1.21	bar	-4.85
p_em	1.18	1.13	bar	3.87
T_c	55.78	49.30	C	13.14
T_im	26.33	26.31	C	0.06
T_em	292.46	237.23	C	23.28
T_t	281.95	221.51	C	27.29
w_tc	33.43	37.20	kRPM	-10.13
Tq_e	473.21	500.22	Nm	-5.40
Lambda	3.03	3.59	-	-15.41
W_af	117.76	128.32	g/s	-8.23
W_c	117.76	128.32	g/s	-8.23
W_ic	117.75	128.32	g/s	-8.24
W_cyl_in	117.75	128.32	g/s	-8.24
W_cyl_out	121.98	153.00	g/s	-20.28
W_t	121.29	153.00	g/s	-20.07
W_es	121.35	153.00	g/s	-20.03
Xo_em	55.31	15.41	%	36.90

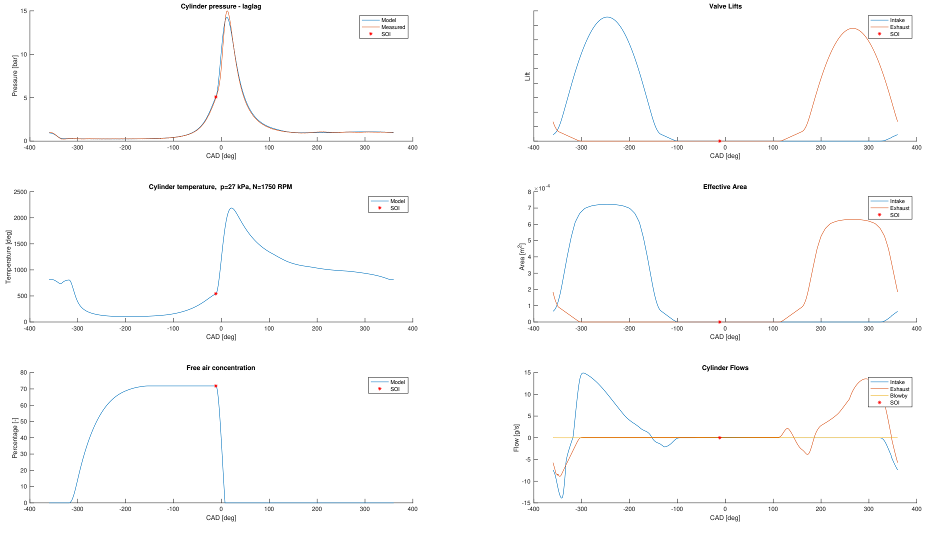

Figure 5.9: *Cylinder pressure, temperature, oxygen concentration, valve lift, effective area for the valves and the cylinder flows over one complete cycle on the low load and low engine speed operating point for the Volvo engine.*

Table 5.12: *Mean error and max error for different parameters for the Volvo engine on the low load and low engine speed operating point.*

Parameter	Modelled value	Measured value	Units	Mean error [%]
p_c	1.02	1.02	bar	0.72
p_im	0.27	0.27	bar	0.48
p_em	1.00	1.02	bar	-1.99
T_c	34.34	39.70	C	-13.51
T_im	24.95	34.36	C	-27.38
T_em	938.20	574.48	C	63.31
T_t	949.49	497.55	C	90.84
w_tc	14.62	7.55	kRPM	93.67
Tq_e	28.31	16.10	Nm	75.82
Lambda	0.90	0.99	-	-9.48
W_af	4.28	6.21	g/s	-31.13
W_c	4.28	6.21	g/s	-31.13
W_ic	4.28	6.21	g/s	-31.13
W_th	4.28	6.21	g/s	-31.13
W_cyl_in	4.28	6.21	g/s	-31.12
W_cyl_out	5.15	6.32	g/s	-18.55
W_t	5.15	6.32	g/s	-18.57
W_es	5.15	6.32	g/s	-18.56

5.5 Mean

Here the results for different parameters are presented for the mean operational point. The non-constant parameters needed to run this operational point for the different engines are presented in Table 5.13. In Table 5.14 and 5.15 mean value data is presented for different parameters and in Figure 5.10 and 5.11 cycle data is presented for the Scania and Volvo engine respectively.

Table 5.13: *Operating points run for Scania and Volvo engine of the middle operating point.*

Parameter	Scania	Volvo
Engine Speed [RPM]	1300	2250
Throttle Angle [Deg]	-	8.42
Fuel Flow [mg/stroke]	126.49	17.97
SOI [Deg]	-8.04	-8.00
Crank Angle Intake Offset [Deg]	15.06	-48.00
Crank Angle Exhaust Offset [Deg]	-15.00	30.00

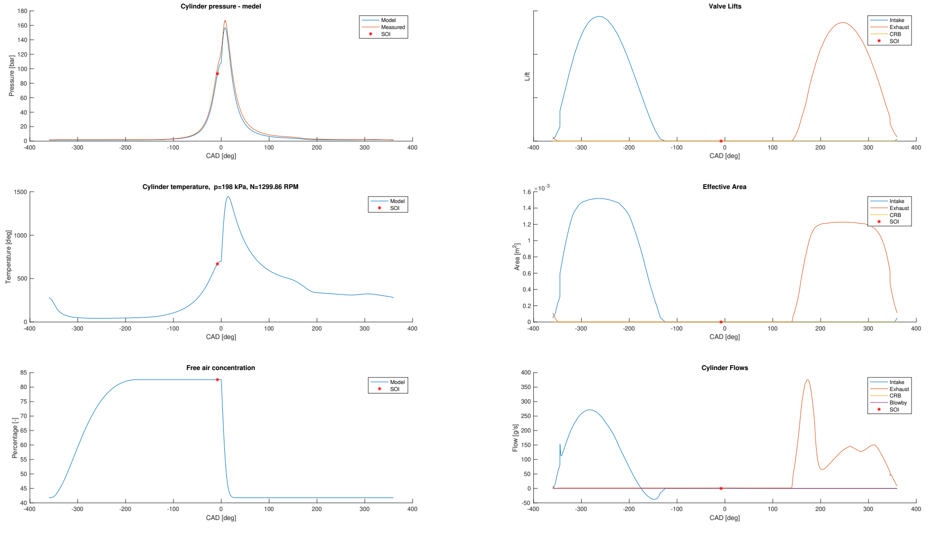

Figure 5.10: *Cylinder pressure, temperature, oxygen concentration, valve lift, effective area for the valves and the cylinder flows over one complete cycle on the mean operating point for the Scania engine.*

Table 5.14: *Mean error and max error for different parameters for the Scania engine on the mean operating point.*

Parameter	Modelled value	Measured value	Units	Mean error [%]
p_c	1.76	2.02	bar	-13.00
p_im	1.72	1.98	bar	-13.02
p_em	1.68	1.80	bar	-7.14
T_c	105.43	104.32	C	1.06
T_im	27.83	26.56	C	4.77
T_em	353.55	371.50	C	-4.83
T_t	310.33	330.88	C	-6.21
w_tc	65.28	70.50	kRPM	-7.40
Tq_e	857.17	1249.92	Nm	-31.42
Lambda	1.85	2.24	-	-17.20
W_af	247.16	266.72	g/s	-7.33
W_c	247.16	266.72	g/s	-7.33
W_ic	247.13	266.72	g/s	-7.34
W_cyl_in	247.12	266.72	g/s	-7.35
W_cyl_out	262.29	348.93	g/s	-24.83
W_t	262.76	348.93	g/s	-24.69
W_es	262.84	348.93	g/s	-24.67
Xo_em	42.29	12.00	%	30.29

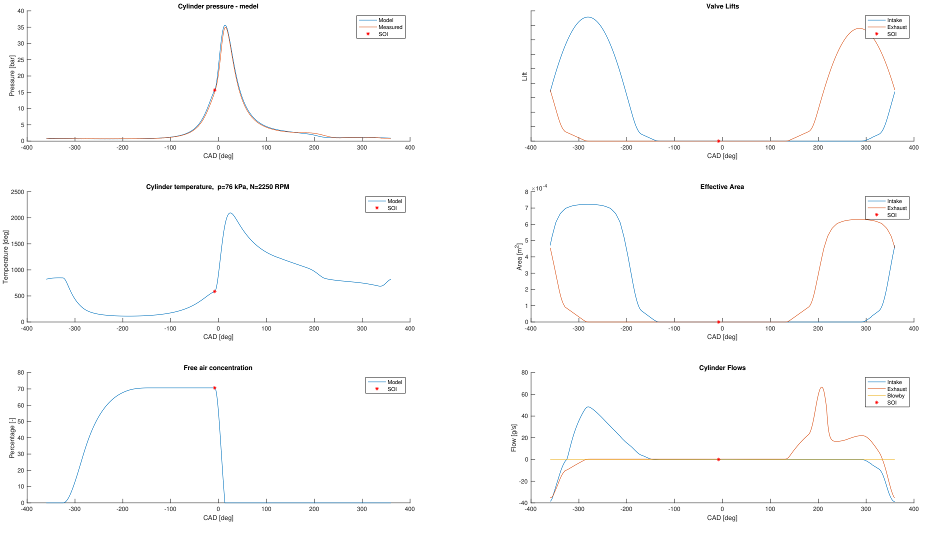

Figure 5.11: *Cylinder pressure, temperature, oxygen concentration, valve lift, effective area for the valves and the cylinder flows over one complete cycle on the mean operating point for the Volvo engine.*

Table 5.15: *Mean error and max error for different parameters for the Volvo engine on the mean operating point.*

Parameter	Modelled value	Measured value	Units	Mean error [%]
p_c	1.03	1.03	bar	0.00
p_im	0.77	0.76	bar	0.95
p_em	1.06	1.03	bar	2.52
T_c	37.92	37.79	C	0.35
T_im	26.55	32.45	C	-18.20
T_em	883.79	665.21	C	32.86
T_t	881.55	637.69	C	38.24
w_tc	24.19	32.84	kRPM	-26.34
Tq_e	99.78	79.86	Nm	24.95
Lambda	1.04	0.99	-	4.60
W_af	15.23	18.75	g/s	-18.77
W_c	15.23	18.75	g/s	-18.77
W_ic	15.23	18.75	g/s	-18.77
W_th	15.23	18.75	g/s	-18.77
W_cyl_in	15.23	18.75	g/s	-18.79
W_cyl_out	17.80	19.09	g/s	-6.78
W_t	17.80	19.09	g/s	-6.77
W_es	17.70	19.09	g/s	-6.77

6 Discussion

During validation of the Volvo engine, it was found that the turbine efficiency minimum was set too low which led to the turbocharger not charging enough. This leads to compressor pressure being too low. This was solved by increasing the minimum turbine efficiency etat_min to the minimum efficiency in the map (instead of the minimum efficiency divided by 2).

Another thing found during validation was that the compression ratio for the engine was decreased from 10.8 to 10.0 for the Volvo engine. This was due to the cylinder pressure for compression and expansion stroke not following the pressure trace from the measured data as it was increasing too much in compression and decreasing too much during expansion.

From measured data, if the values for the ignition angle for the Volvo engine were used, the simulated cylinder pressure indicated a completely incorrect burn profile. The values for SOC for the Volvo engine were instead changed to values that seemed more plausible and the results got much better.

One can see the difference in combustion between SI and CI combustion by looking at the oxygen concentration in the cylinder in Figures 5.2 to 5.11. Combustion in a CI is run lean to decrease chances of creating particles, which also is a reason to use the EGR to reduce the available oxygen [11]. In these simulations, the available oxygen is not combusted fully leading to free air levels not going down to zero after combustion in all operating points. For SI, all oxygen is combusted which means the available oxygen is 0% at the end of combustion. The fill-up of available oxygen also follows the effective area. This also proves that the model can handle both fresh air flow as well as residual gasses.

As seen in the Tables 5.2 to 5.15, the simulated pressures are closer for the Volvo model than the Scania model, but temperatures are closer for the Scania engine than the Volvo engine. This is due to the throttle and wastegate regulators are regulating the pressures to the measured and desired value. The reason for the temperature deviation is that there is no direct correlation between pressure and temperature so even if a regulator is regulating the pressure does not mean the temperature deviation also will improve.

Cylinder pressures are the only reference measurement available with cycle-to-cycle variations. In Figures 5.2 to 5.11 they are used to validate combustion parameters. From Figures 5.2 and 5.3 for the Scania and Volvo engine in the high load and high engine speed operating point the simulated pressure is lower than the measured for both engines. The same applies for the Figures 5.4 and 5.5 for the Scania and Volvo engine in the high load and low engine speed operating point. In Figures 5.6 and 5.7 for the Scania and Volvo engine in the low load and high engine speed operating point, 5.8 and 5.9 for the Scania and Volvo engine in the low load and low engine speed operating point the simulated cylinder pressure is greater than the measured pressures for both engines. This is because the vibe parameters are set for one operating point and not changed when the operating point is changed. As combustion varies depending on load, the vibe parameters should also change. This was however a simplification made in this thesis.

In Figures 5.10 and 5.11 for the Scania and Volvo engine in the mean operating point, the modelled values are close to the measured value. This proves the value of hte vibe parameters are of less importance in low load operational points.

Another aspect differing from measured data is the model for the generated engine torque. This is due to the model not simulating engine friction. The reason most engine torques are lower than measured is because the way the model has been simulated. As the model uses vaiable step lengths, the cycle length in the measured data is often less than 720 degrees. If however the cycle length would be increased by one step, the cycle length would be longer than 720 degrees.

The mass balance in the cylinder was found to not follow m_air + m_fuel - m_blowby = m_exhaust. This is due to the changing values of gas constant and specific heat constant with the use of NASA polynomial. The difference is 0.005 kg per cycle for the Scania engine and 0.0025 kg per cycle for the Volvo engine. The reason that the mass balance was unfufilled is that there was not enough time during the thesis to correctly implement the correct mass balance equations. While doing the parameter estiamtion, it can be of importance to investigate sensor offset. Big engine maps usually has a sensor offset in one or more sensors. The results of not estimating the sensor offset can be seen in Appendix D in Figures D.1 and D.2 for the Scania and Volvo engine respectively.

However, the model agreement is not the most important objective with this thesis, it was to make a generic engine model with VVA compatibility. With generic, the meaning was to have the possibility to easily change the equations used for each component with the addition to remove some components altogether as well as the possibility to interchange between CI and SI combustion. This was completed by making a separate MATLAB equation for each component as each component is represented by one Simulink block. The generality is proven by the fact that the Volvo engine is SI and the Scania engine is CI as well as components being removed for the Scania engine and a different compressor and turbine function was used. That was fulfilled by just changing what equation was run in MATLAB and reroute some signals in Simulink. As an added benefit with the usage of this generic model, equations can easily be changed to improve the model performance of the models used in this thesis.

7

Future work

All models are made to be simple. The objective was to develop a framework that could then be used to further develop more advanced models for each component. Therefore, most of the models will need to be changed in some way.

The most important component to develop is the turbocharger. The compressor functions are presented in the simplest form. The common way to develop this model is to use the extended elliptical model for the compressor efficiency [7], as is done in LiU Diesel 2. That will capture more of the compressor dynamics. The turbine equations can also be altered by adding more terms to the TFP polynomial. More terms in the TFP polynomial will make the flow modeling more accurate.

Another component that can be improved is the throttle. The throttle function can be extended in its function for the effective area by adding more terms to the polynomial. This would help to add a more complex flow pattern, as the polynomial gets bigger by adding more terms.

As for this thesis, the valve flow discharge coefficient is measured when the flow reaches a steady state. The valve lifts are measured on a cold engine. As the metal heats up, the lift profile might change because of the coefficient of thermal expansion. One way to compensate is to use hydraulic lash adjusters, HLA [23]. With the use of HLA, IVO, and EVO, the valve lifts could be more accurately determined.

The valve flow discharge coefficient can be measured in a different way to better represent the actual usage. As for this thesis, the valve flow discharge coefficient is measured in a steady state. It has been concluded that transient experiments of the discharge coefficient would create much more representative data of the discharge coefficient. The transients would be measured with similar valve movements as it is made in the engine [30].

By adding states for piston and wall temperature to ensure better estimates of the heat transfer. It could be possible to obtain T_w estimates by using the engine coolant, measuring the temperature, and knowing the flow rate [14].

To more accurately describe combustion, a double vibe function is often used for CI engines [11]. To parameterize the double vibe function accurately, the heat release needs to be more accurately studied by, for example, looking at the gross heat release and comparing it to the net heat release, and also considering heat transfer. This also means that the SOI needs to be more accurately determined, as ignition delay plays a large role in heat release.

Another thing to look into is the selection of non-linear solver to see if the parameters found are different and if so investigate model performance with different non-linear parameter solvers.

One other important thing is to change the mass balance calculation in the cylinder to make sure all inflows equal all outflows. This can be done by changing the pressure state to a mass state and calculate the cylinder pressure from the ideal gas law.

However, to achieve better correlations with the two engines presented, the turbocharger models should be developed, as they did not match the dynamics of the actual system.

Appendix

A

Model Parameters

This appendix presents some parameter data from the cylinder model.

A.1 Gas properties

In Table A.1, the molecular mass of different substances are shown.

Table A.1: *Molecular masses of different substances from [16]*

Substance	Molecular mass [g/mol]
Carbon, C	12.011
Hydrogen, H_2	2.0160
Cetane, $C_{16}H_{34}$	226.44
Isooctane, C_8H_{18}	114.20
Oxygen, O_2	32.000
Atmospheric Nitrogen, N_2	28.160
Carbon dioxide, CO_2	44.010
Water, H_2O	18.020

.

By using the values in table A.1, the molecular mass is calculated for air and burned fuel.

$$M_{air} = \frac{1}{1+3.773}\left(1 \cdot M_{O_2} + 3.773 \cdot M_{N_2}\right) = 28.96[\text{g/mol}] \tag{A.1}$$

$$\begin{aligned} M_{burned} &= \frac{1}{a + \frac{b}{2} + 3.773\left(a + \frac{b}{4}\right)}\left(aM_{CO_2} + \frac{b}{2}M_{H_2O} + 3.773\left(a + \frac{b}{5}M_{N_2}\right)\right) = \\ &= \begin{cases} 28.81\ [\text{g/mol}] & \text{if fuel is cetane} \\ 13.56\ [\text{g/mol}] & \text{if fuel is isooctane} \end{cases} \end{aligned} \tag{A.2}$$

A.2 Valve Properties

The different engines have different valve properties. Valve lifts for the Scania engine can be seen in Figure A.1 and for the Volvo engine in Figure A.2.

The valve lifts are used to calculate the C_D values for intake, exhaust, and CRB. C_D for the intake is a function of the normalized lift L_v/D_v and can be seen in Figure A.3 for the Scania engine and Figure A.4 for the Volvo engine. C_D for the exhaust is a function of the normalized lift L_v/D_v and pressure ratio P_r and can be seen in Figure A.5 for the Scania engine and Figure A.6 for the Volvo engine. The C_D values have been measured in steady flow experiments.

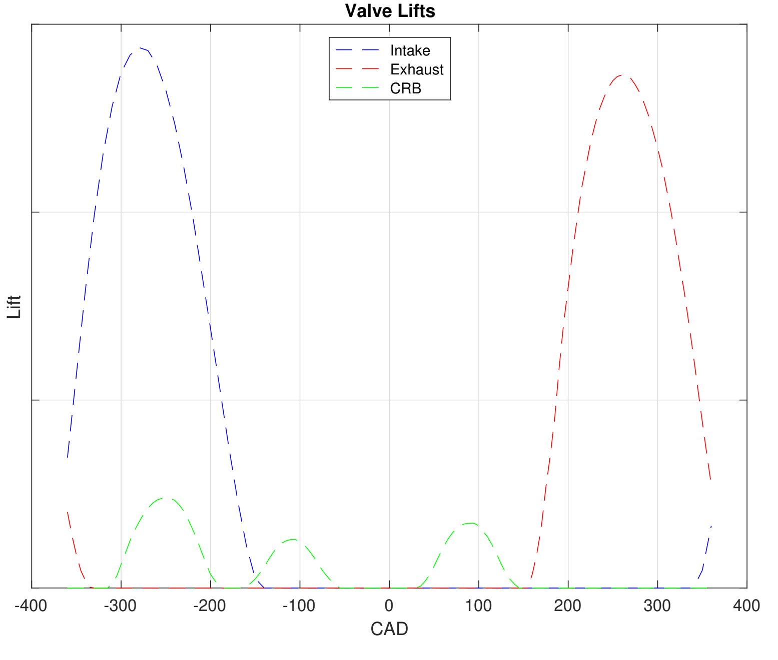

Figure A.1: *Valve lift for intake, exhaust and CRB for the Scania engine. Combustion occurs approximately at 0°.*

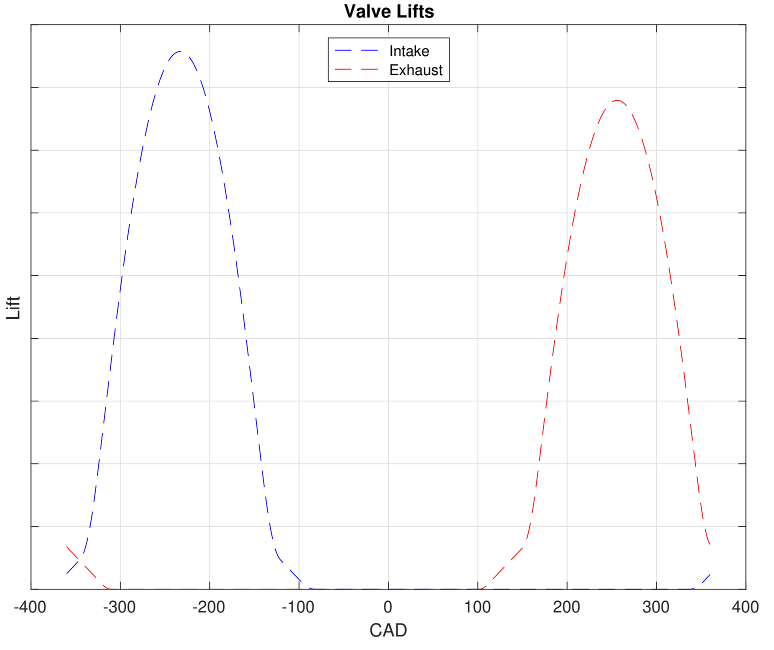

Figure A.2: *Valve lift for intake, exhaust and CRB for the Volvo engine. Combustion occurs approximately at 0 °.*

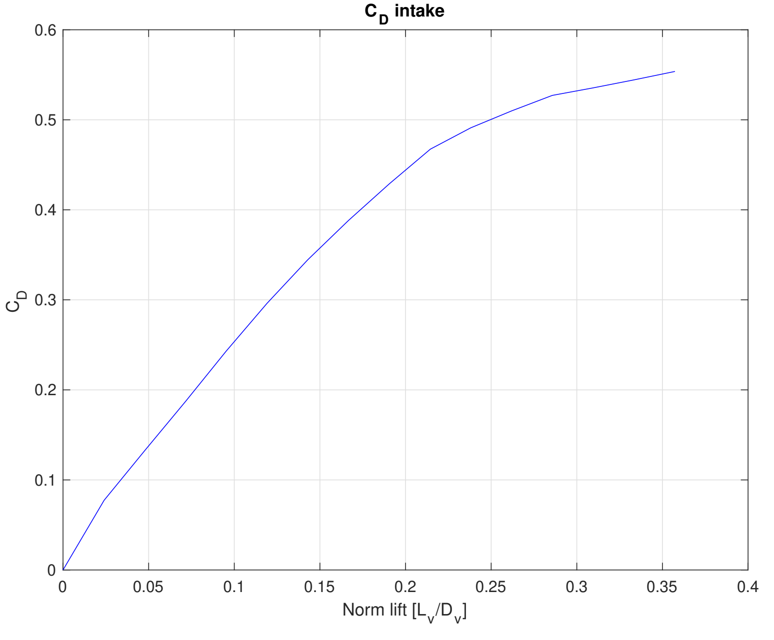

Figure A.3: *C_D value for the intake valve as a function of normalized lift L_v/D_v for the Scania engine.*

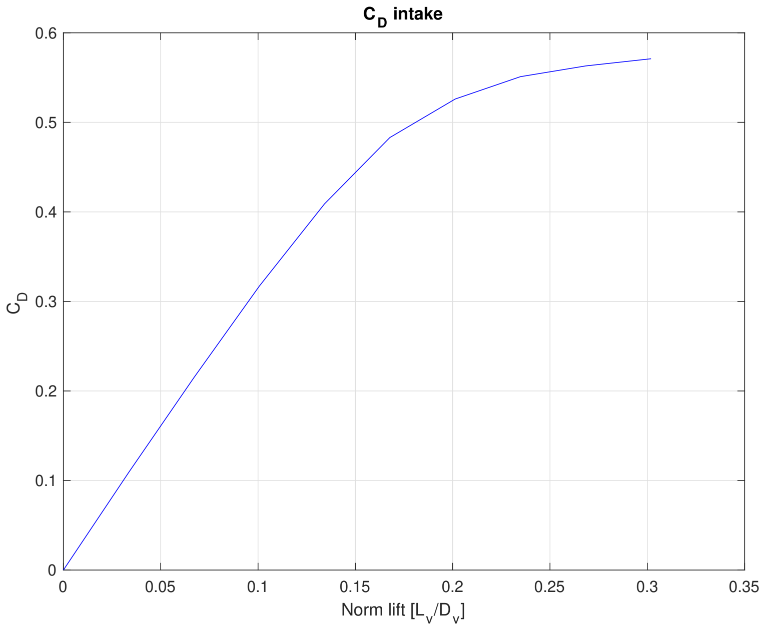

Figure A.4: *C_D value for the intake valve as a function of normalized lift L_v/D_v for the Volvo engine.*

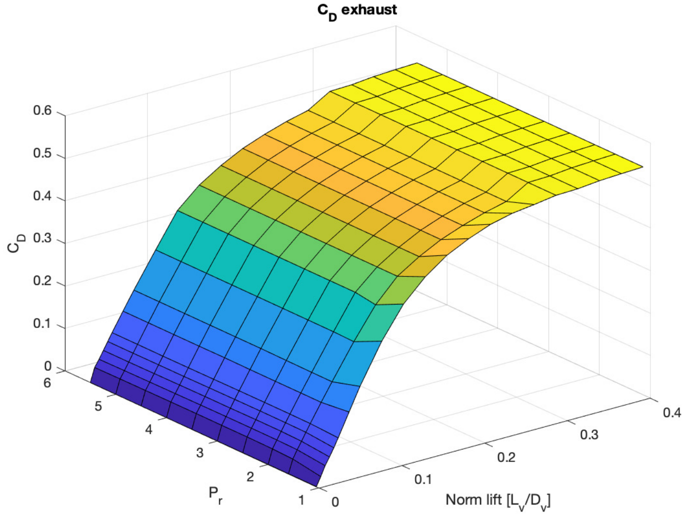

Figure A.5: *C_D value for the exhaust valve as a function of normalized lift L_v/D_v and pressure ratio P_r for the Scania engine.*

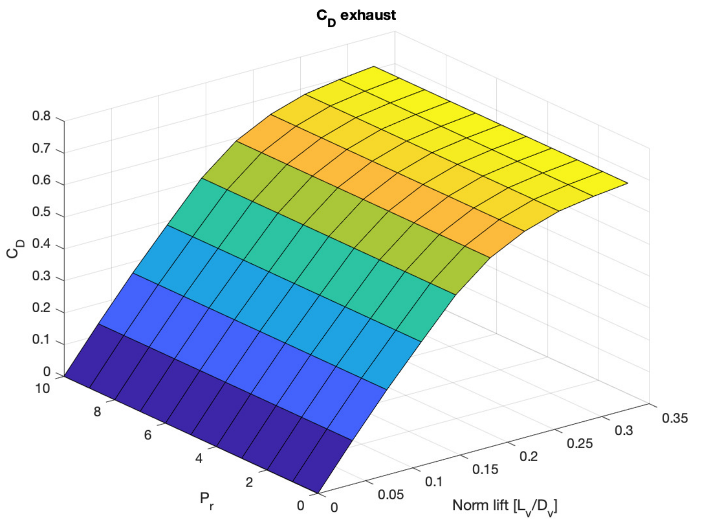

Figure A.6: *C_D value for the exhaust valve as a function of normalized lift L_v/D_v and pressure ratio P_r for the Volvo engine.*

B

Parameters Struct

Below follows a list of the parameters needed to run the model.

Table B.1: *The parameters struct.*

Parameters			
AFs	**.Compressor**	**.Cylinder**	**.Turbine**
cp_air	A_bypass	Aeff_Blowby	A_wastegate
cp_cetane	Cd_bp	CAD_crb	BSR_max
cp_exh	etac_max	CAD_exh	Cd_wg
cp_isooctane	etaCmin	CAD_in	delay
cv_air	mccorr_etamax	CD_exhaust	eta_t_max
cv_exh	mccorr_max	CD_intake	etaTmin
gamma_air	pc_ref	exhaust_pr	k0
gamma_exh	PI_c_etamax	Lift_crb	k1
p_amb	Psi_max	Lift_exh	rad_turb
Q_lhv	q11	Lift_in	
R_air	q12	MCO2	**.Turboshaft**
R_exh	q21	MH2O	c_tc
R_tilde	q22	MN2	J_tc
SOC	rad_comp	MO2	w_init
T_amb	Tc_ref	n	
Xo_c		Norm_exhaust_lift	**.Throttle**
	.CV	Norm_intake_lift	A0
.Engine	p_init_af	pi_li	A1
a	p_init_c	p_IVC	A2
B	p_init_ex	ref_diam_exh	delay
l	p_init_ic	ref_diam_in	tau_th
n_cyl	p_init_im	T_IVC	
r_c	p_init_t	Tw	
S	T_init_af	V_IVC	
V_d	T_init_c		
	T_init_ex	**.Vibeparameters**	
.Incompressible_flow	T_init_ic	a_vibe	
ctu_af	T_init_im	Deltatheta	
ctu_in	T_init_t	m_vibe	
ctu_ex	V_airfilter		
p_li	V_compressor	**.Intercooler**	
	V_exhaust	a0	
	V_intercooler	a1	
	V_intake	a2	
	V_turbine	T_cool	

C

Model Validation for Symmetrical Drag and CRB

In this Appendix, the model validation is presented. The validation is divided into the two different engines validated. The operating points used to present the results are presented. Two additional operating points are also presented as Symmetrical drag and CRB.

C.1 Symmetrical drag

Symmetrical drag is running the engine without fuel. This is useful to see the engine behavior when combustion is not occurring. However, symmetrical drag was not measured on the Volvo engine so the operating point with the lowest load was chosen as a reference. The non-constant parameters needed to run this operational point for the different engines are presented in Table C.1. The results can be seen in Table C.2 and C.3 for the Scania and Volvo engine. Cylinder pressure, temperature, oxygen concentration, valve lift, effective area for the valves and the cylinder flows over one complete cycle can be seen in Figure C.1 for the Scania engine and Figure C.2 for the Volvo Engine.

Table C.1: *Operating points run for Scania and Volvo engine of the symmetrical drag operating point.*

Parameter	**Scania**	**Volvo**
Engine Speed [RPM]	1800	3000
Throttle Angle [Deg]	-	4.84
Fuel Flow [mg/stroke]	0	7.53
SOI [Deg]	0	-10.00
Crank Angle Intake Offset [Deg]	70.00	-15.62
Crank Angle Exhaust Offset [Deg]	-70.00	9.35

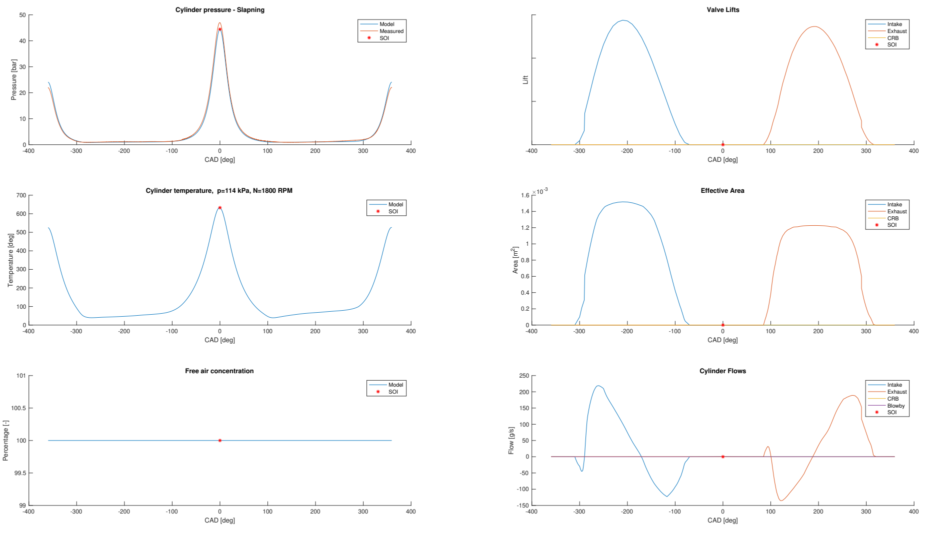

Figure C.1: *Cylinder pressure, temperature, oxygen concentration, valve lift, effective area for the valves and the cylinder flows over one complete cycle on the symmetrical drag operating point for the Scania engine.*

Table C.2: *Mean error and max error for different parameters for the Scania engine on symmetrical drag operating point.*

Parameter	Modelled value	Measured value	Units	Mean error [%]
p_c	1.02	1.14	bar	-10.81
p_im	1.02	1.14	bar	-10.52
p_em	1.04	1.00	bar	3.78
T_c	40.04	56.26	C	-28.84
T_im	25.98	28.81	C	-9.81
T_em	84.27	90.69	C	-7.08
T_t	82.90	178.27	C	-53.50
w_tc	12.67	28.81	kRPM	-56.01
Tq_e	-18.72	-148.19	Nm	87.37
Lambda	65535.00	-111.95	-	65535.00
W_af	61.11	70.97	g/s	-13.89
W_c	61.11	70.97	g/s	-13.89
W_ic	61.17	70.97	g/s	-13.81
W_cyl_in	61.27	70.97	g/s	-13.67
W_cyl_out	61.08	70.97	g/s	-13.93
W_t	61.06	70.97	g/s	-13.97
W_es	61.07	70.97	g/s	-13.95
Xo_em	100.00	100.00	%	0.00

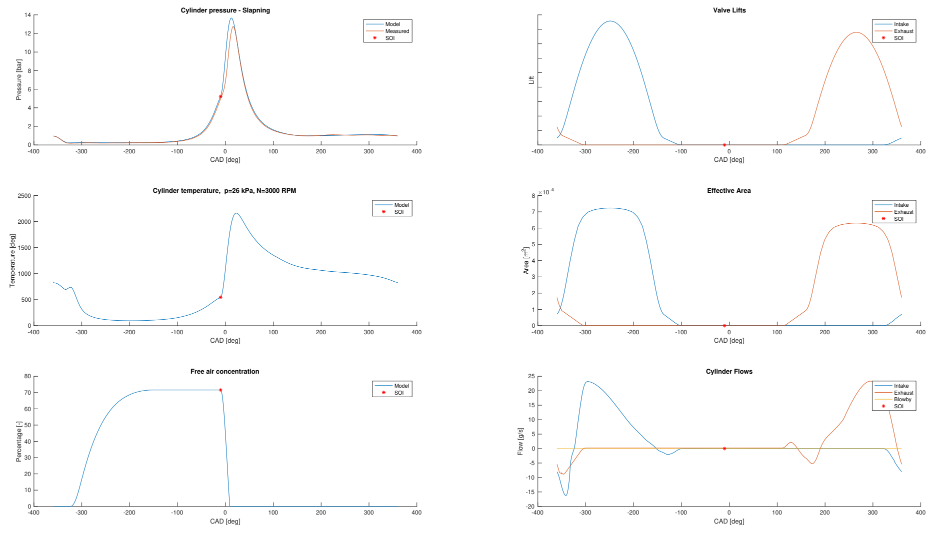

Figure C.2: *Cylinder pressure, temperature, oxygen concentration, valve lift, effective area for the valves and the cylinder flows over one complete cycle on the symmetrical drag operating point for the Volvo engine.*

Table C.3: *Mean error and max error for different parameters for the Volvo engine on symmetrical drag operating point.*

Parameter	Modelled value	Measured value	Units	Mean error [%]
p_c	1.03	1.02	bar	0.95
p_im	0.27	0.26	bar	0.59
p_em	1.02	1.03	bar	-0.27
T_c	36.05	38.79	C	-7.06
T_im	25.50	31.84	C	-19.93
T_em	967.24	706.16	C	36.97
T_t	974.92	631.97	C	54.27
w_tc	19.62	15.96	kRPM	22.87
Tq_e	27.22	13.98	Nm	94.68
Lambda	0.91	0.99	-	-8.63
W_af	8.86	10.44	g/s	-15.15
W_c	8.86	10.44	g/s	-15.15
W_ic	8.86	10.44	g/s	-15.15
W_th	8.86	10.44	g/s	-15.15
W_cyl_in	8.86	10.44	g/s	-15.15
W_cyl_out	10.32	10.63	g/s	-2.84
W_t	10.32	10.63	g/s	-2.84
W_es	10.32	10.63	g/s	-2.84

C.2 CRB

CRB is used as an engine braking method to further improve engine braking efficiency and reduce wear on the brake pads. The non-constant parameters needed to run this operational point for the different engines are presented in Table C.4. The results can be seen in Table C.5. Cylinder pressure, temperature, oxygen concentration, valve lift, effective area for the valves and the cylinder flows over one complete cycle can be seen in Figure C.3.

Table C.4: *Operating points run for Scania engine of the CRB operating point.*

Parameter	**Scania**
Engine Speed [RPM]	1700
Fuel Flow [mg/stroke]	0
SOI [Deg]	0
Crank Angle Intake Offset [Deg]	30.00
Crank Angle Exhaust Offset [Deg]	-76.00

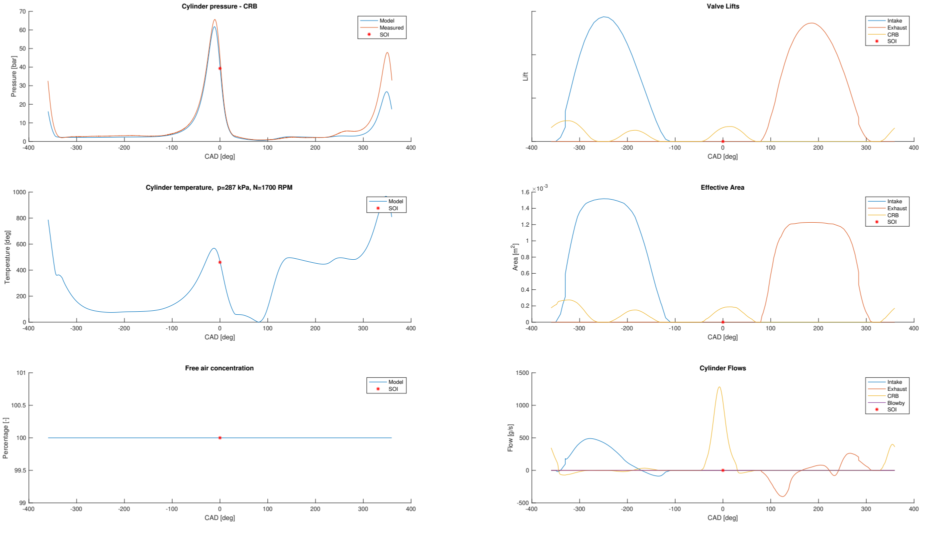

Figure C.3: *Cylinder pressure, temperature, oxygen concentration, valve lift, effective area for the valves and the cylinder flows over one complete cycle on the CRB operating point for the Scania engine.*

Table C.5: *Mean error and max error for different parameters for the Scania engine on CRB operating point.*

Parameter	**Modelled value**	**Measured value**	**Units**	**Mean error [%]**
p_c	2.46	2.93	bar	-15.75
p_im	2.40	2.87	bar	-16.27
p_em	2.46	2.71	bar	-9.25
T_c	152.42	163.50	C	-6.78
T_im	29.60	28.89	C	2.49
T_em	515.49	447.51	C	15.19
T_t	428.16	336.09	C	27.39
w_tc	87.37	93.67	kRPM	-6.72
Tq_e	-1117.16	-1705.44	Nm	34.49
Lambda	65535.00	-2076.46	-	65535.00
W_af	375.17	414.24	g/s	-9.43
W_c	375.17	414.24	g/s	-9.43
W_ic	375.07	414.24	g/s	-9.46
W_cyl_in	374.84	414.24	g/s	-9.51
W_cyl_out	375.44	414.24	g/s	-9.37
W_t	374.86	414.24	g/s	-9.51
W_es	374.77	414.24	g/s	-9.53
Xo_em	100.00	100.00	%	0.00

D
Sensor offset demonstration

There is often a sensor offset in big engine maps. When estiamting parameters it can be useful to assume one of the pressure sensors being accurate and the other being inaccurate. This will need one extra paramter to estimate for validation purposes only. The results of not estimating the pressure sensor offset can be seen in Figures D.1 and D.2 for the Scania and Volvo engine respectively.

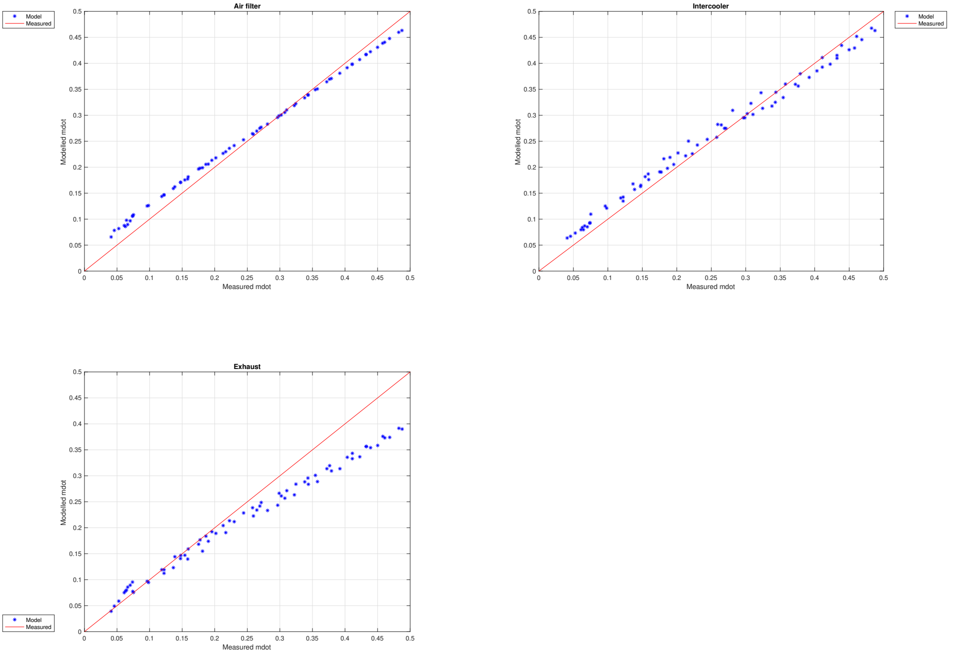

Figure D.1: *The air filter, intercooler and exhaust system flow validation, validating ctu_{af}, ctu_{in} and ctu_{ex} for the Scania engine.*

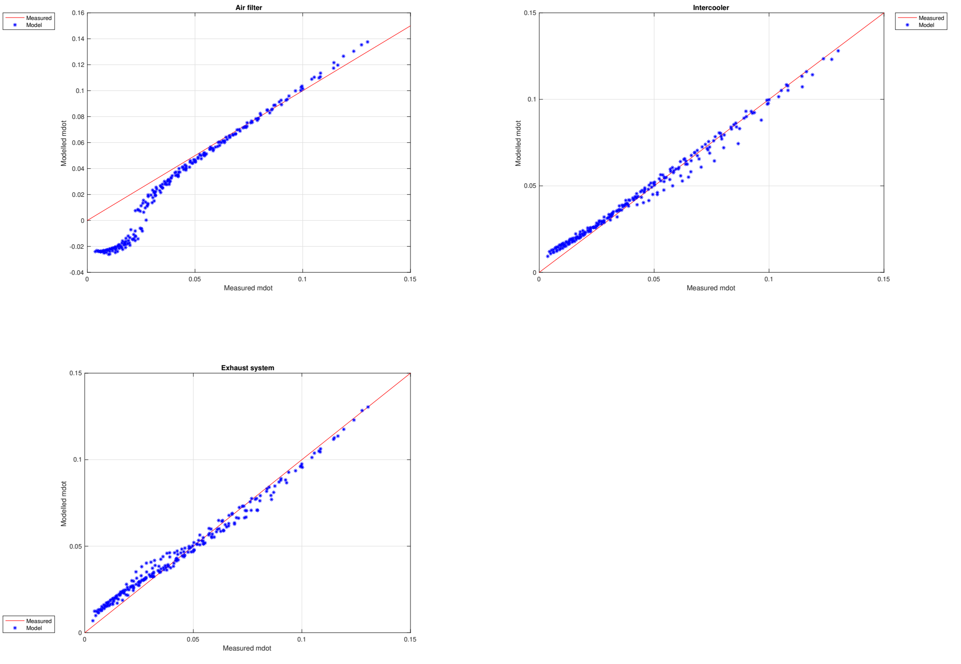

Figure D.2: *The air filter, intercooler and exhaust system flow validation, validating* ctu_{af}*,* ctu_{in} *and* ctu_{ex} *for the Volvo engine.*

Bibliography

[1] Åke Björck. *Numerical methods for least squares problems*. SIAM, 1996.

[2] Gordon P. Blair. *Design and simulation of two-stroke engines*. Society of Automotive Engineers, 1996.

[3] Gordon P. Blair. *Design and simulation of four-stroke engines*. Society of Automotive Engineers, 1999.

[4] Jan Brugård, Lars Eriksson, and Lars Nielsen. Mean value engine modeling of a turbo charged spark ignited engine - A principle study. Technical Report LiTH-R-2370, Department of Electrical Engineering, Linköpings Universitet, SE-581 83 Linköping, Sweden, 2001.

[5] Wen-Po Chaing, Liangtao Zhu, and Ravindra Patankar. Mean value engine modeling and validation for a 4-stroke, single cylinder gasoline engine. *Trends in Applied Sciences Research*, 2(2):124–131, 2007.

[6] Thomas Dresner and Philip Barkan. A review of variable valve timing benefits and modes of operation. Technical report, SAE Technical Paper 891676, 1989.

[7] Kristoffer Ekberg, Viktor Leek, and Lars Eriksson. Modeling and Validation of an Open-Source Mean Value Heavy-Duty Diesel Engine Model. *Simulation Notes Europe (SNE)*, 28(4):198–203, 2018. ISSN 2306-0271. doi: 10.11128/sne.28.4.1044.

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

[9] Lars Eriksson. Chepp – a chemical equilibrium program package for matlab. *SAE Transactions, Journal of Fuels and Lubricants, 2004-01-1460*, 4(113): 730–741, June 2005.

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

[11] Lars Eriksson and Lars Nielsen. *Modeling and Control of Engines and Drivelines*. John Wiley Sons Inc, 2014.

[12] Lars Eriksson, Lars Nielsen, Jan Brugård, Johan Bergström, Fredrik Pettersson, and Per Andersson. Modeling of a turbocharged SI engine. *Annual Reviews in Control*, 26(1):129–137, October 2002.

[13] Neil Fuchs, Zdenek Meistrick, Steven Ernest, Gabe Roberts, and Justin Baltrucki. Development of a high performance two-cycle engine brake for medium and heavy duty diesel engines. *SAE International Journal of Commercial Vehicles*, 6(1), January 2013.

[14] Lino Guzzella and Christopher Onder. *Introduction to modeling and control of internal combustion engine systems*. Springer Science & Business Media, 2009.

[15] John Heywood. *Internal combustion engine fundamentals*. McGraw-Hill, 1988.

[16] John B. Heywood. *Internal Combustion Engine Fundamentals, Second Edition*. McGraw-Hill Education, 2018.

[17] Linus Johansson. Full cycle cylinder state estimation in di engines with vva. Master's thesis, Linköping University, Vehicular Systems, 2019.

[18] Markus Klein. *Single-zone cylinder pressure modeling and estimation for heat release analysis of SI engines*. PhD thesis, Institutionen för systemteknik, 2007.

[19] Xavier Llamas and Lars Eriksson. Control-oriented modeling of two-stroke diesel engines with exhaust gas recirculation for marine applications. *Journal of Engineering for the Maritime Environment*, 223(2):551–574, 2019. ISSN 1475-0902.

[20] Fadila Maroteaux and Charbel Saad. Combined mean value engine model and crank angle resolved in-cylinder modeling with nox emissions model for real-time diesel engine simulations at high engine speed. *Energy*, 88: 515–527, 2015.

[21] Bonnie J McBride. *NASA Glenn coefficients for calculating thermodynamic properties of individual species*. National Aeronautics and Space Administration, John H. Glenn Research Center ..., 2002.

[22] Mehdi Mostofi, A.H. Shamekhi, and M. Gorji-Bandpy. Modified mean value si engine modeling (egr included). *Fuel*, 10:5, 2006.

[23] Patrick Phlips and Andreas Schamel. The dynamics of valvetrains with hydraulic lash adjusters and the interaction with the gas exchange process. Technical report, SAE Technical Paper, 1991.

[24] Youngchul Ra and Rolf D. Reitz. A combustion model for ic engine combustion simulations with multi-component fuels. *Combustion and Flame*, 158 (1), January 2011.

[25] Mark Sellnau and Eric Rask. Two-step variable valve actuation for fuel economy, emissions, and performance. *SAE Transactions*, 112(3), 2003.

[26] Gerasimos Theotokatos, Cong Guan, Hui Chen, and Iraklis Lazakis. Development of an extended mean value engine model for predicting the marine two-stroke engine operation at varying settings. *Energy*, 143(15), January 2018.

[27] Christian Winge Vigild. *The Internal Combustion Engine Modelling: Modelling, Estimation and Control Issues*. PhD thesis, Technical University of Denmark, 8 2002. Project title: Robust control of non-linear systems.

[28] Johan Wahlström and Lars Eriksson. Modeling diesel engines with a variable-geometry turbocharger and exhaust gas recirculation by optimization of model parameters for capturing non-linear system dynamics. *Journal of Automobile Engineering*, 225(7), 2011.

[29] Fredrik Westin. *Accuracy of turbocharged SI-engine simulations*. PhD thesis, Maskinkonstruktion, 2002.

[30] PM Winroth, Christopher L Ford, and Per-Henrik Alfredsson. On discharge from poppet valves: effects of pressure and system dynamics. *Experiments in Fluids*, 59(2):24, 2018.

[31] Gerhard Woschni. A universally applicable equation for the instantaneous heat transfer coefficient in the internal combustion engine. Technical report, SAE Technical paper, 1967.

[32] Yunus A. Çengel, John M. Cimbala, and Robert H. Turner. *Fundamentals of Thermal-Fluid Sciences*. McGraw-Hill Education, forth edition, 2012.