Cylinder-by-Cylinder Diesel Engine Modelling - A Torque-based Approach

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

by
Magnus Ramstedt

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Sammanfattning
Abstract

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The model, derived in Matlab/Simulink, is a Cylinder-by-Cylinder Engine Model (CCEM) reconstructing the angle synchronous torque of a diesel engine. To validate the model, it has been parameterised for the DaimlerChrysler engine OM646, a straight turbocharged four cylinder diesel engine, and tested towards measured data from a Mercedes-Benz C220 test vehicle. Due to hardware related problems, validation could only be performed for low engine speeds where the model shows good results. Future work around this theme ought to include further validation of the model as well as implementation on HiL.

Nyckelord
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Esslingen, June 2004
Magnus Ramstedt

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Chapter 1

Introduction

This chapter gives an introduction to the thesis ”Cylinder-by-Cylinder Diesel Engine Modelling - A Torque-based Approach” written at the department of Powertrain Control at DaimlerChrysler Research and Technology in Esslingen, Germany.

1.1 Background

The development of vehicular control systems is a very expensive and resource consuming process. Therefore, researchers are continuously looking to make it more efficient and adaptable to new technologies. Today, Hardware-in-the-Loop (HiL) testing is becoming more popular within the automotive industry since it constitutes both a cheap and flexible alternative. In a HiL system, models of the different vehicle components, such as the engine, are set to run on a software platform. The controller is then physically implemented in an Electronic Control Unit (ECU) and tested together with the models. This brings further demands on the models to be not only accurate but also fast and applicable in real time. Based on software instead of hardware simulations, this method of testing is resource efficient while it at the same time takes physical factors into consideration such as real time effects of the ECU. Another advantage is that it allows the testing of single hardware components before the entire system exists physically.

For most control applications, a Mean Value Engine Model (MVEM) capturing course of events over one or more engine cycles is sufficient. However, when building control systems for certain aspects of the powertrain, for example backlash caused by play between gears, it might be interesting to investigate how individual combustion pulses affect the drive line. Therefore a Cylinder-by-Cylinder Engine Model (CCEM), describing each cylinder individually, could prove a useful alternative to the otherwise more common

MVEMs. A CCEM is also needed to model cylinder individual phenomenon such as misfire when developing systems for diagnostics.

1.2 Objectives

The aim of this thesis is to create a complete (without Exhaust Gas Recirculation), physically based, cylinder-by-cylinder diesel engine model for possible use on a HiL testbed in the future development and testing process of engine and powertrain control systems. The model should focus on the torque generation and, in order to keep the parameterisation complexity down, describe the torque from a crankgear point of view. Also, the model complexity as well as the calculation time should be kept to a minimum in order to render as good conditions as possible for future control.

1.3 Approach

The engine theory is used to implement a model in MatLab/Simulink which describes the main torque working on the crankshaft. It is parameterised for OM646, a straight four cylinder DaimlerChrysler diesel engine. The model is tested and evaluated for different scenarios, and the results are then compared to measurements made with a test vehicle in order to determine the accuracy.

1.4 Outline

The basic outline of the thesis is as follows:

Chapter 2 - Compression Ignited Engine and Modelling
This chapter explains the fundamental compression ignited engine theory. The basic structure of the diesel engine is presented as well as the individual engine parts. The theory of engine modelling is also covered.

Chapter 3 - The Model
Focusing on the torque generation, this chapter describes the model and the theory behind it. It also explains the implementation and parameterisation process.

Chapter 4 - Simulation and Validation
The model is tested for different scenarios to determine its function. A presentation as well as an interpretation of the results is made. The simulated data are then validated towards measured data from a test vehicle.

Chapter 5 - Conclusions

This chapter covers conclusions drawn from the results described in the previous chapter. A discussion follows.

Chapter 6 - Further Work
Future improvements and possible extensions to the model are proposed.

Chapter 2

Compression Ignited Engine and Modelling

This chapter aims to describe the structure and function of the Compression Ignited (CI) engine, normally diesel engine. It also attempts to introduce the reader to the different types of engine models and modelling methods available.

2.1 Compression Ignited Engine

Today CI engines are used in a number of different applications, such as cars, boats and power plants. Depending on the field of application, the CI engine has a certain structure and contains certain components which may differ from field to field. Operating range, torque and emission requirements are examples of factors that decide the constitution of the engine. A basic four-cylinder diesel engine used within the automotive industry, such as the one modelled in this thesis, often has the following structure and components as seen in Figure 2.1. For more information on the theory behind the CI engine, see for example [1] or [2].

Air Filter

This is where the air enters the engine. The main task of the air filter is to separate the incoming air from rough particles. It also has noise, pressure and temperature reducing functions. The temperature change through the air filter is proportional to the air flow $\dot{m}_{0_{in}}$ as follows, where T_0 is the air temperature within the air filter, T_{amb} the air temperature outside the engine and c a constant:

$$T_0 = T_{amb} + c\dot{m}_{0_{in}} \tag{2.1}$$

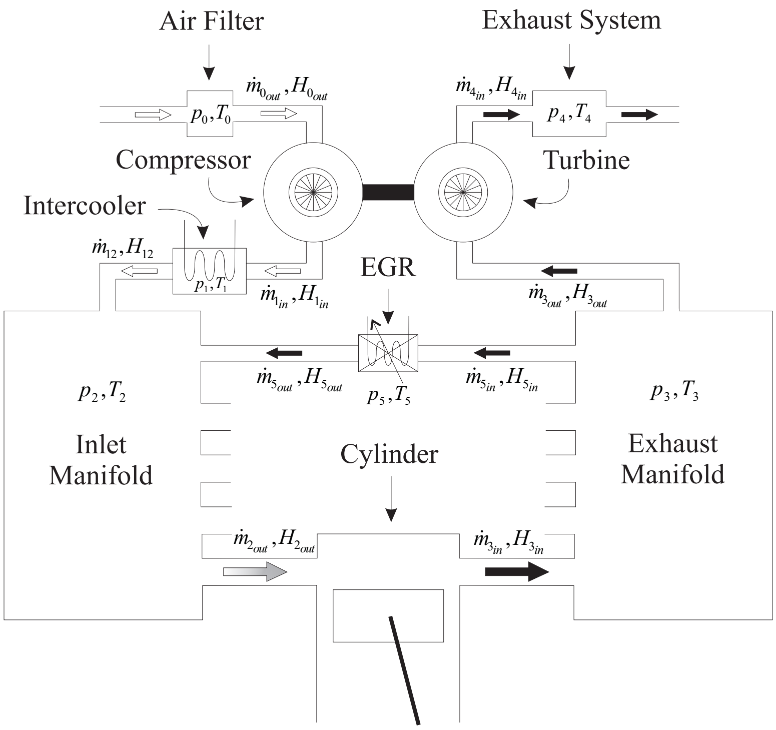

Figure 2.1: The structure of a basic diesel engine

The pressure drop is described through the ideal gas law.

Compressor

The compressor constitutes one of the two main parts of the turbocharger. It is driven by the turbine through the turbo shaft and helps building up pressure in the inlet manifold to enable the charge of more air into the cylinders. The following two equations describe the operation of the compressor.

$$M_{comp}\omega_{tc} = \dot{H}_{1_{in}} - \dot{H}_{0_{out}} \tag{2.2}$$

$$\frac{T_1}{T_0} = 1 + \frac{1}{\eta_{comp}}\left(\left(\frac{p_1}{p_0}\right)^{\frac{\gamma-1}{\gamma}} - 1\right) \tag{2.3}$$

The first equation (2.2), where M_{comp} is the compressor torque and ω_{tc} the angular velocity of the turbocharger, states that the power transfer through the turbo shaft equals the enthalpy increase through the compressor. This describes the main work process of the turbocharger. Due to friction and other side effects associated with the compressor, the air is heated and the compression therefore not ideal. This is considered in equation (2.3) where η_{comp} is the isentropic efficiency and γ the specific heat ratio for air.

Intercooler

When the air passes the intercooler, energy is transferred in order to drop the gas temperature. Since cold air has a greater density than warm air, this enables the charge of more air into the cylinder and further, more energy obtained from the engine per cycle. The intercooler also helps preventing knock during combustion. The cooling process has the following physical equation.

$$T_2 = T_1 - \eta_{cool}\,(T_1 - T_{cool}) \tag{2.4}$$

T_{cool} is the temperature of the cooling medium and η_{cool} the intercooler efficiency.

Inlet Manifold

In the inlet manifold, air is mixed with recirculated exhaust gas from the exhaust manifold. The pressure builds up as the gases are gathered before flowing into each cylinder when the respective inlet valve opens. The physical equation for the pressure differentiation within the inlet manifold describes this process,

$$\dot{p}_2 = \frac{RT_2}{V_2}\,(\dot{m}_{12} + \dot{m}_{5_{out}} - \dot{m}_{2_{out}}) \tag{2.5}$$

where the index V_2 is the volume of the inlet manifold and R the gas constant.

Cylinder

The operation of the cylinder is normally described through its cycle, as illustrated in Figure 2.2. For a four stroke engine, two revolutions of the crankshaft are needed to complete one cycle. During the cycle, the piston moves continuously back and forth between its top and bottom positions, Top Dead Center (TDC) and Bottom Dead Center (BDC). In order to gain energy needed during the operating cycle, each cylinder works with a $180°$ phase difference to eachother. The following strokes constitute the four stroke cycle:

Intake $TDC - BDC \Rightarrow \theta : 0° \rightarrow 180°$
The inlet valve opens and the piston moves down to BDC, creating a suction effect of air from the inlet manifold into the cylinder. This effect is caused and maintained by a pressure difference between the two volumes and the increasing cylinder volume as the piston moves downwards.

Compression $BDC - TDC \Rightarrow \theta : 180° \rightarrow 360°$
The inlet valve closes and mechanical work from the crankshaft pushes the

piston towards TDC, compressing the air to create high pressure and temperature within the cylinder. Fuel is injected into the cylinder for about $20 - 40°$, starting approximately $15 - 30°$ before TDC.

Expansion $TDC - BDC \Rightarrow \theta : 360° \rightarrow 540°$
Because of the high temperature and pressure, the air-fuel mixture self-ignites shortly before TDC during the compression stroke. This initiates the combustion process, pushing the piston down to BDC and inducting the main work on the crankshaft.

Exhaust $BDC - TDC \Rightarrow \theta : 540° \rightarrow 720°$
During the expansion stroke, about $50°$ before the piston reaches BDC, the exhaust valve opens. The burnt gases within the cylinder then flow out to the exhaust manifold; first because of pressure difference and later as they are swept out by the piston moving towards TDC. At the end of the exhaust stroke, as the pressure difference between the two volumes is levelled out, the exhaust valve closes and a new cycle begins.

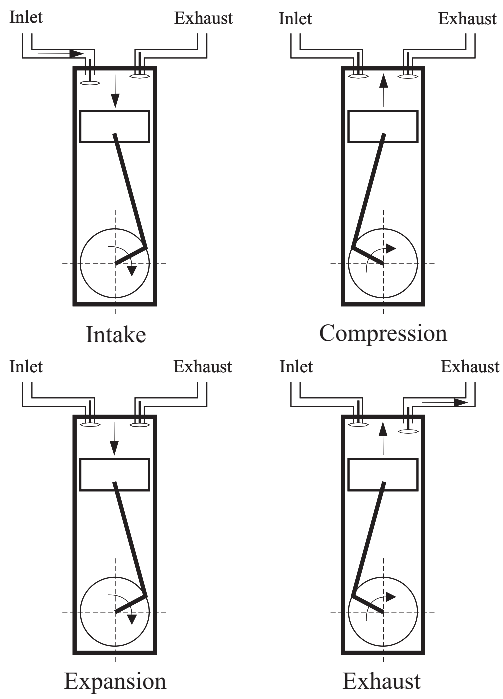

Figure 2.2: The four stroke cycle

Exhaust Manifold

In the exhaust manifold, exhaust gas from the cylinders is gathered after the combustion process. Pressure is built up and the gas is pushed towards the turbine and/or the EGR-system. Like the inlet manifold, the exhaust manifold is defined by the pressure differentiation, where V_3 is the manifold volume.

$$\dot{p}_3 = \frac{RT_3}{V_3}\left(\dot{m}_{3_{in}} - \dot{m}_{3_{out}} - \dot{m}_{5_{in}}\right) \tag{2.6}$$

Turbine

The turbine constitutes one of the two main parts of the turbocharger. It is driven by exhaust gas from the exhaust manifold and is itself the power source of the compressor through the turbo shaft connection. The rotational acceleration of the turbo shaft is derived through Newton's second law describing the power transfer within the turbocharger as follows:

$$\dot{\omega}_{tc} = \frac{M_{turb} - M_{comp}}{J_{tc}} \tag{2.7}$$

M_{turb} and M_{comp} is the torque of the turbine and the compressor respectively and J_{tc} is the inertia of the turbocharger. The main work of the turbine is characterised by equations similar to the ones describing the compressor.

$$M_{turb}\omega_{tc} = \dot{H}_{4_{in}} - \dot{H}_{3_{out}} \tag{2.8}$$

$$\frac{T_4}{T_3} = 1 + \eta_{turb}\left(\left(\frac{p_4}{p_3}\right)^{\frac{\gamma-1}{\gamma}} - 1\right) \tag{2.9}$$

For some turbochargers, the angle of the wings on the turbine wheel is adjustable to better control the power transfer through the turbocharger. This function is called Variable Turbine Geometry (VTG).

Exhaust System

The exhaust system has a number of functions. The engine noise should be reduced and through the catalyst, the temperature of the exhaust gas should be lowered and the gas itself be cleaned enough to meet emission standards.

Exhaust Gas Recirculation (EGR)

As a consequence of emission regulations, the amount of oxides of nitrogen ($\mathrm{NO_x}$) in the exhaust gas must be held down. Since $\mathrm{NO_x}$ is formed at high temperature, one way to fulfil this requirement is to limit the amount of excess air available during combustion. This is done by moving exhaust gas from the exhaust manifold to the inlet manifold with the help of the EGR system. That way a mix of fresh air and exhaust gas will enter the cylinder, keeping down the combustion temperature. The EGR contains a valve and a cooler. The cooling function can be described in the same way as for the intercooler.

2.2 Modelling

There are numerous ways of describing reality through a model. Some are more complex than others and the different approaches may differ in both structure and accuracy. Which one to choose therefore very much depends on the particular situation and especially the field of application. Here follows a summary of the main engine model classifications. More information on engine modelling can be found in [2], [3] and [4].

Equations vs. Black Box

There are a number of approaches available when deciding on the basis of a model. Physical equations theoretically describing the system is the most common method since it creates a general model working for many operating areas. Its drawback is that reality might be difficult to describe correctly in theory. Also, such a model is often resource consuming. Another common approach is to base the model entirely on measurements. The measured data is stored as a table of two or more dimensions in a so called black box and then fetched when needed depending on one or more input signals. This approach often provides an accurate result since it is based directly on empiricism, however it is only defined for a limited region. A combination of both approaches, where the main basis of the model rests on physical equations and black boxes are used to model certain complexities, is also common.

Single-Zone vs. Multi-Zone

The combustion process can be described with varying complexity and accuracy. Normally the degree of complexity is decided by the number of zones in which the cylinder has been divided. An engine model therefore is either single-zone or multi-zone. In a single-zone model the gas mixture within the cylinder is considered to be homogenous for each sample. It is also assumed to be made up strictly of ideal gases. In a multi-zone model, for example the two-zone model, the gases are still considered ideal. However, the homogenous approach has been replaced by a heterogenous one. Here the cylinder is also divided into two zones, one containing injected fuel and the other surrounding air. Each zone itself is homogenous and no heat transfer occurs between the two zones. The simplicity of the single-zone model is its biggest advantage. This makes it fast and therefore applicable in realtime systems. The multi-zone model is more complex and more accurate compared to the single-zone model. A multi-zone model is often needed for combustion chamber design, but for most aspects of control design a single-zone model is good enough.

Mean Value vs. In-Cycle Variation

When it comes to the cylinder, two main approaches can be found. The most common approach is to model all cylinders as one, describing the total engine torque as a mean value over one or more engine cycles. The result of this approach is called Mean Value Engine Model (MVEM). An alternative to the MVEM is a model describing the in-cycle variation torque. One such example is the Cylinder-by-Cylinder Engine Model (CCEM). Unlike the MVEM, it describes each cylinder individually and generates for example a torque signal with each individual combustion pulse present. Normally a MVEM is sufficient enough for use in processes such as control system design. However, for some aspects of powertrain control a model illustrating the in-cycle variations of the torque could prove useful. An example of such an aspect could be backlash caused by play between gears, thus affected by individual combustion pulses, see [3].

Chapter 3

The Model

In this chapter the model is described in detail. The engine to be modelled is a straight four cylinder diesel engine with a VTG turbocharger. In reality, the engine also contains an EGR system. However, since emissions are irrelevant to the desired field of application and since the model should describe the engine at its most effective working point, the EGR is not to be modelled. Also, for the efficiency related reason just mentioned, the VTG function of the turbocharger has been left out.

The chosen model approach is based on a gas path engine model, modelling the path of the gas throughout the engine. Since the gas path engine model is also a Mean Value Engine Model, it has been combined with a cylinder-by-cylinder crank angle based approach of the torque generation to make it cylinder individual. Input signals to the model are the crank angle, engine speed and injected fuel quantity per cylinder.

In the model, each engine block is modelled as either a flow or stagnation component. A flow component describes the flow of gas and enthalpy to and from the block while a stagnation component consider the block to be a volume, calculating the pressure, temperature and fraction of burnt gas within. Each parameter has been given an index number, identifying the volume it is connected to. Further, each flow to and from a volume is identified as either a flow of air or exhaust gas when so is needed. Also, each flow has been given a second index describing the direction of the flow. The model structure can be seen in Figure 3.1. Figure 2.1 in Chapter 2 also helps in the understanding of the model and its parameters. The maps used in the model originates from [7] and [6].

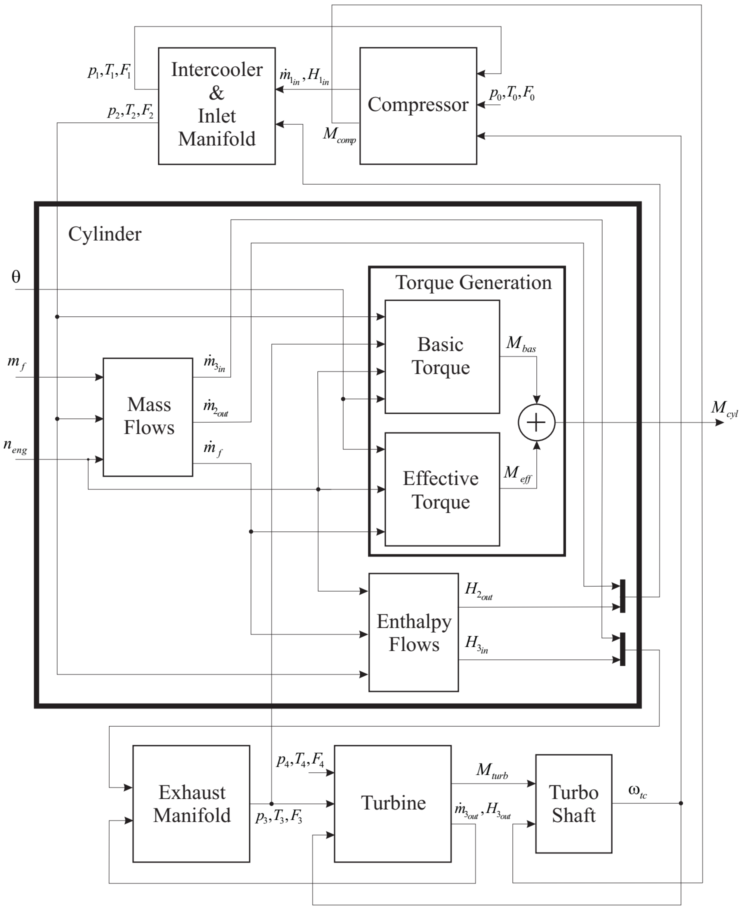

Figure 3.1: The basic structure of the model

3.1 Compressor

The compressor model is a flow component based on equation 2.2 and 2.3. Not all relations within the compressor can be described through physical equations in a satisfying way, hence three maps must be constructed to complete the model. These maps are based on measurements and derive the pressure ratio p_1/p_0, the compressor efficiency η_{comp} as well as the corrected mass flow Φ_{comp} through the compressor. Φ_{comp} is the actual mass flow $\dot{m}_{0_{out}}$ scaled with the pressure p_0 and temperature T_0 of the air filter as follows:

$$\Phi_{comp} = \dot{m}_{0_{out}} \frac{\sqrt{T_0}}{p_0} \tag{3.1}$$

The corrected mass flow is, among other things, needed as input to the efficiency map. Input signals to the compressor block are the pressure and temperature of the intercooler and air filter as well as the angular velocity of the turbo shaft connecting the turbine with the compressor. The flow of air mass $\dot{m}_{1_{in}}$ and enthalpy $\dot{H}_{1_{in}}$ from the compressor together with the compressor torque M_{comp} constitutes the output signals. The mass flow through the air filter and compressor are assumed equal, hence

$$\dot{m}_{0_{out}} = \dot{m}_{1_{in}} \tag{3.2}$$

By combining equation (3.2) and (3.1), and then reversing the latter, $\dot{m}_{1_{in}}$ is formed. The compressor torque M_{comp} can be derived from equation (2.2) where the angular velocity of the turbocharger is the integrated acceleration described in equation (2.7). Since the temperature and enthalpy flow ratio through the compressor have the following relation due to thermodynamical laws,

$$\frac{T_1}{T_0} = \frac{\dot{H}_{1in}}{\dot{H}_{0out}} \tag{3.3}$$

the outgoing enthalpy of the compressor is formed by scaling the incoming enthalpy with the temperature ratio described by equation (2.3). The enthalpy going out of the air filter and into the compressor is derived from the basic formula of calorimetrics,

$$\dot{H}_{0_{out}} = c_{p,a} \dot{m}_{0_{out}} T_0 \tag{3.4}$$

where $c_{p,a}$ is the specific heat capacity for air and T_0 the temperature before the compressor.

3.2 Intercooler and Inlet Manifold

The intercooler and the inlet manifold are considered two stagnation components connected to each other. The connection is such that no pressure difference occurs between the two volumes. Therefore

$$\dot{p}_1 = \dot{p}_2 \tag{3.5}$$

must be fulfilled, where $\dot{p}_1$ and $\dot{p}_2$ is the pressure derivative within the intercooler and the inlet manifold respectively. Input signals to the intercooler/inlet manifold block are the flow of air mass and enthalpy connected to the compressor and the cylinder block. Output signals are the pressure, temperature and fraction of burnt gas within the intercooler and the inlet manifold $p_1, T_1, f_{1,e}$ and $p_2, T_2, f_{2,e}$. Through thermodynamical laws, the pressures and temperatures can be calculated as follows:

$$T_{12} = \frac{\Delta U_{12}}{m_{12,a} c_{p,a}} \tag{3.6}$$

$$p_{12} = \frac{\Delta m_{12,a} R_a T_{12}}{V_{12}} \tag{3.7}$$

ΔU_1 and ΔU_2 are the changes in energy caused by the enthalpy flows respectively within the intercooler and the inlet manifold, while $\Delta m_{1,a}$ and $\Delta m_{2,a}$ represent the change in air mass caused by the mass flows. R_a is the gas constant for air and V_1 and V_2 is the intercooler and the inlet manifold volume respectively. Each volume has the same pressure, temperature and gas mixture throughout its interior. Since no EGR is modelled, the gas consists exclusively of fresh air; hence $f_{1,e}$ and $f_{2,e}$ are null.

3.3 Cylinder

The cylinder block provides a model for each individual cylinder and its torque generation, enthalpy and mass flow. The individual cylinder torque is divided into a basic and an effective part. The basic torque M_{bas} describes mass and gas forces working on the crank shaft outside the combustion process, while the effective torque M_{eff} mainly deals with gas forces during combustion. The two torques are then added to form the total cylinder torque M_{cyl}, see Figure 3.2. In the same way each individual cylinder torque, as well as enthalpy and mass flow, is added to create the total engine torque, enthalpy and mass flow out of the cylinder block. Input signals to the cylinder block are the engine speed, injected fuel quantity per cylinder and crank angle as well as the pressure, temperature and fraction of burnt gas for the inlet and exhaust manifolds. The mass and enthalpy flows of both manifolds constitute the output signals. Each cylinder has been modelled to peak for different crank angles evenly spread over the engine cycle. More information on the approach used for the cylinder model can be found in [5]. Here follows a description of each component comprising the individual cylinder model.

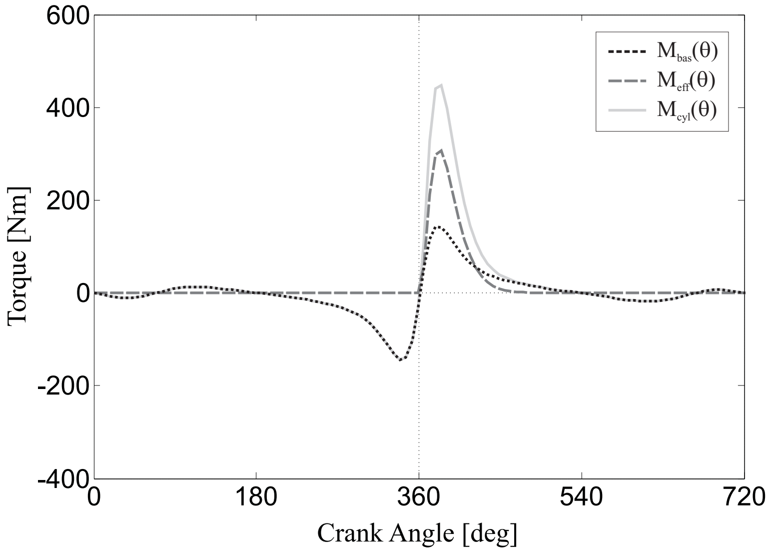

Figure 3.2: The outline of the cylinder torque

3.3.1 Mass Flows

Within the cylinder, four mass flows are calculated; the flow of air $\dot{m}_{2_{out}}$ and fuel $\dot{m}_f$ into the cylinder as well as the flow of air $\dot{m}_{3,a_{in}}$ and exhaust gas $\dot{m}_{3,e_{in}}$ out of the cylinder. Since no EGR is modelled, the flow of exhaust gas into the cylinder is null. Together, the engine speed and the loaded air mass before compression $m_{c,bcp}$ compose $\dot{m}_{2_{out}}$ as a mean value over time.

$$\dot{m}_{2_{out}} = \frac{2 \cdot m_{c,bcp} N_{eng}}{n_r} \tag{3.8}$$

The loaded air mass before compression is described by the theoretically practical intake mass m_{th} and the volumetric efficiency η_{vol}. η_{vol} is mapped while the ideal gas law, using the maximum cylinder volume, forms m_{th}.

$$m_{c,bcp} = \eta_{vol} m_{th} = \eta_{vol} \frac{p_2 V_{disp}}{R_a T_2} \tag{3.9}$$

Both the air and exhaust gas flowing out of the cylinder are based on the total mass flow out of the cylinder $\dot{m}_{3_{in}}$ and the fraction of air within the gas leaving the cylinder $f_{c,a}$. $\dot{m}_3$ is described in the same way as $\dot{m}_{2_{out}}$, beside the effects of the injected fuel quantity m_f.

$$\dot{m}_{3_{in}} = \frac{2\,(m_{c,bcp} + m_f)\,N_{eng}}{n_r} \tag{3.10}$$

To form $f_{c,a}$, the amount of air used in the combustion process is subtracted from the flow of air entering the cylinder. (A/F) is the air/fuel ratio of the flows entering the cylinder. The remaining flow is then divided by the total mass flow into the cylinder.

$$f_{c,a} = \frac{\dot{m}_{2_{out}} - (A/F)\dot{m}_f}{\dot{m}_{2_{out}} + \dot{m}_f} \tag{3.11}$$

The fuel mass flow is derived according to the same principle as the gas mass flows in equation (3.8) and (3.10).

$$\dot{m}_f = \frac{2 \cdot m_f N_{eng}}{n_r} \tag{3.12}$$

This leads to the following intuitive expressions for the flow of air and exhaust gas out of the cylinder.

$$\dot{m}_{3,a_{in}} = f_{c,a}\dot{m}_{3_{in}} \tag{3.13}$$

$$\dot{m}_{3,e_{in}} = (1 - f_{c,a})\,\dot{m}_{3_{in}} \tag{3.14}$$

3.3.2 Enthalpy Flows

Two enthalpy flows are calculated within the cylinder block; the flow going in $\dot{H}_{2_{out}}$, and the flow going out $\dot{H}_{3_{in}}$ of the cylinder. $\dot{H}_{2_{out}}$ is formed in the same way as the equivalent of the compressor in equation (3.4).

$$\dot{H}_{2_{out}} = c_{p,a}\dot{m}_{2_{out}}T_2 \tag{3.15}$$

Out of the cylinder the enthalpy flow is made up of three parts: the ingoing enthalpy, energy released when burning the fuel and heat leaving the cylinder with gases after combustion. The latter is defined by using the gas energy proportion Z_{ge} to scale the total energy released. Z_{ge} is mapped as a function of engine speed and fuel proportion Z_f, where also the fuel ratio itself is based on a map. The enthalpy flow out of the cylinder, where $c_{p,f}$ is the specific heat capacity for fuel, T_f the fuel temperature and Q_{HV} the fuel heating value, can be written as follows:

$$\dot{H}_{3_{in}} = \dot{H}_{2_{out}} + c_{p,f}\dot{m}_f T_f + Z_{ge}Q_{HV}\dot{m}_f \tag{3.16}$$

3.3.3 Basic Torque

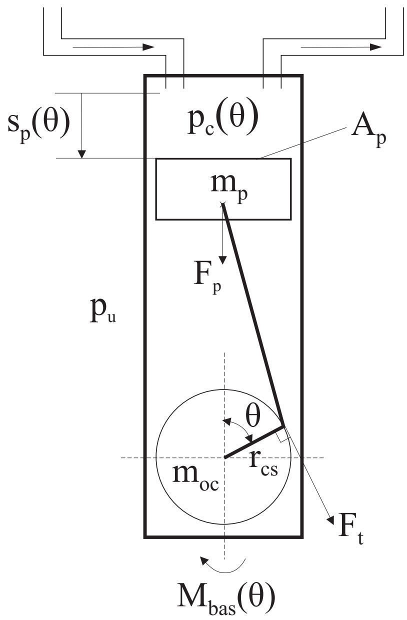

Figure 3.3: Mechanics within the cylinder

Not considering the forces created within the cylinder through combustion, two forces further affect the piston; gas and mass forces. The gas force F_g is created through pressure differences between the gas pressure within the cylinder $p_{bas}(\theta)$ and outside the cylinder p_{amb}. Together, the mass of the piston m_p and the oscillating part of the connecting rod m_{oc} contributes to the mass force F_m. The total force on the piston F_p has the following equation,

$$F_p(\theta) = F_g(\theta) + F_m(\theta) = (p_{bas}(\theta) - p_{amb})\, A_p - (m_p + m_{oc})\, \ddot{s}_p(\theta) \tag{3.17}$$

where $\ddot{s}_p(\theta)$ is the second derivative of the piston stroke $s_p(\theta)$. The contribution of the mass force is negative since it works as a normal force against the gas pressure. The piston stroke can, through geometrical and trigonometrical conditions of the crankgear, be defined as a function dependent on the crank angle. Further simplification through Taylor expansion [5], leads to the following expression for the piston stroke and its second derivative.

$$\begin{aligned} s_p(\theta) &= r_{cs}\left[1 - \cos\theta + \frac{L_{s/cr}}{4}\,(1 - \cos 2\theta)\right] \Rightarrow \\ \ddot{s}_p(\theta) &= r_{cs}\omega_{eng}^2 \left(\cos\theta + L_{s/cr}\cos 2\theta\right) \end{aligned} \tag{3.18}$$

The engine angular velocity ω_{eng} is assumed to be constant within each working cycle. r_{cs} is the radius of the crankshaft and forms, together with the length of the connecting-rod l_{cr}, the stroke-to-connecting-rod ratio.

$$L_{s/cr} = \frac{r_{cs}}{l_{cr}} \tag{3.19}$$

The basic gas pressure $p_{bas}(\theta)$ within the cylinder, is a function defined in three different ways over the engine cycle depending on the crank angle.

$$p_{bas}(\theta) = \begin{cases} p_{c,int} & \forall\, \theta \in [0^\circ, 180^\circ] \\ p_{c,bcp}\left(\frac{V_{c,bcp}}{V_c(\theta)}\right)^\alpha & \forall\, \theta \in [180^\circ, 540^\circ] \\ p_{c,exh} & \forall\, \theta \in [540^\circ, 720^\circ] \end{cases} \tag{3.20}$$

The expression for $\theta = [0^\circ, 180^\circ]$ describes the intake stroke, $\theta = [180^\circ, 540^\circ]$ the high pressure loop (compression and expansion), and $\theta = [540^\circ, 720^\circ]$ the exhaust stroke. The basic gas pressure during the high pressure loop is the same as the pressure for a motored cycle, where $V_c(\theta)$ is the cylinder volume as a function of the crank angle and $V_{c,bcp}$ is the volume at the beginning of compression. Since the volume just before compression is the maximum volume of the cylinder, hence

$$V_{c,bcp} = V_{disp} + V_{clear} \tag{3.21}$$

where V_{disp} is the displaced volume and V_{clear} the clearance volume. The polytropic exponent α, used to describe the combustion process, is individual for each engine and must be decided through testing.

Since the pressures during intake and exhaust, $p_{c,int/exh}$, are of less importance to the torque generation, they are considered constant in order to simplify the model. Both pressures are therefore static relations between, respectively, charge-air pressure p_2 and exhaust pressure p_3 as well as valve flow losses Δp_{int} and Δp_{exh}.

$$p_{c,int} = p_2 - \Delta p_{int} \tag{3.22}$$
$$p_{c,exh} = p_3 + \Delta p_{exh} \tag{3.23}$$

The valve flow losses are, through the following relations, dependent on the gas velocities w_{int} and w_{exh} through each valve cross-section.

$$\Delta p_{int} = 0.08\left(\frac{w_{int}}{100}\right)^2 \tag{3.24}$$

$$\Delta p_{exh} = 0.12 \left(\frac{w_{exh}}{100}\right)^2 \tag{3.25}$$

By using the piston speed

$$v_p = 2 \cdot s_{p,\mathit{eff}} N_{eng} \tag{3.26}$$

dependent on the engine speed and the effective piston stroke

$$s_{p,\mathit{eff}} = 2 \cdot r_{cs} \tag{3.27}$$

as well as the piston surface A_p and the maximum valve cross-sections $A_{int/exh}$, the gas velocities are determined.

$$w_{int/exh} = \left(\frac{A_p}{A_{int/exh}}\right) v_p \tag{3.28}$$

The calculation of the pressure at the beginning of compression $p_{c,bcp}$ is based on the first law of thermodynamics. First the fresh charge-air mass $m_{c,bcp}$ is calculated according to equation (3.9). To consider the effects of heat transfer in and around the valve, the gas temperature $T_{c,bcp}$ is then approximated by the following formula according to [8].

$$T_{c,bcp} = 85K + \frac{5}{6} T_2 \tag{3.29}$$

The cylinder pressure can then be written as follows:

$$p_{c,bcp} = \frac{m_{c,bcp} R_a T_{c,bcp}}{V_{disp} + V_{clear}} = \frac{r_{cp} - 1}{r_{cp}} p_2 \eta_{vol} \left(\frac{85^\circ K}{T_2} + \frac{5}{6}\right) \tag{3.30}$$

where the compression ratio

$$r_{cp} = \frac{V_{disp} + V_{clear}}{V_{clear}} \tag{3.31}$$

In order to calculate the correct torque, the tangential force F_t working perpendicular on the crankshaft is formed with F_p through the help of trigonometry. F_t is then multiplied with the crankshaft radius r_{cs} to form the basic torque.

$$M_{bas}(\theta) = F_t(\theta) r_{cs} = F_p(\theta) \sin\theta \left(1 + \frac{L_{s/cr} \cos\theta}{\sqrt{1 - L_{s/cr}^2 \sin^2\theta}}\right) r_{cs} \tag{3.32}$$

By combining all of the equations above, equation (3.32) can be rewritten in the following way to better suit our crank angle based approach.

$$M_{bas}(\theta) = \begin{cases} \left(p_{c,int}c_1(\theta) + \omega_{eng}^2 c_2(\theta) + c_3\right) c_4(\theta) & \forall\, \theta \in [0°, 180°] \\ \left(p_{c,bcp}c_1(\theta) + \omega_{eng}^2 c_2(\theta) + c_3\right) c_4(\theta) & \forall\, \theta \in [180°, 540°] \\ \left(p_{c,exh}c_1(\theta) + \omega_{eng}^2 c_2(\theta) + c_3\right) c_4(\theta) & \forall\, \theta \in [540°, 720°] \end{cases} \tag{3.33}$$

where

$$c_1(\theta) = \begin{cases} A_p \left(\frac{V_{c,bcp}}{V_{clear}+A_p r_{cs}\left[1-\cos\theta+\frac{L_{s/cr}}{4}(1-\cos 2\theta)\right]} \right)^{\alpha} & \forall\, \theta \in [180°, 540°] \\ A_p & else \end{cases} \tag{3.34}$$

$$c_2(\theta) = -\left(m_{oc} + m_p\right) r_{cs} \left(\cos\theta + L_{s/cr}\cos 2\theta\right) \tag{3.35}$$

$$c_3(\theta) = -p_{amb}A_p \tag{3.36}$$

$$c_4(\theta) = r_{cs}\sin\theta \left(1 + \frac{L_{s/cr}\cos\theta}{\sqrt{1 - L_{s/cr}^2 \sin^2\theta}}\right) \tag{3.37}$$

3.3.4 Effective Torque

The contribution to the total cylinder torque given by forces created during combustion is described by the effective torque M_{eff}. It also takes into consideration the wall-heat losses during the high pressure loop left out in the construction of the basic torque. Since the combustion process is very complex, it is also very difficult to describe in theory. Therefore, the effective torque is approximated through a replacement function.

A replacement function is a function which describes a measured curve in a mathematical way. The following replacement function describing the effective torque as seen in Figure 3.4 has been chosen.

$$M_{eff}(\theta) = \begin{cases} a\left(\theta - 360\right)^2 e^{(-b(\theta-360))} & \forall\, \theta \in [360°, 540°] \\ 0 & else \end{cases} \tag{3.38}$$

where

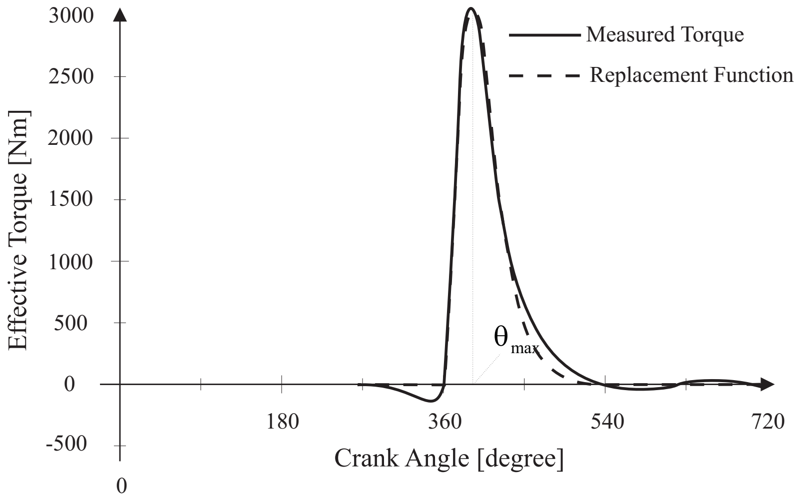

Figure 3.4: Replacement function describing the effective torque

$$a = \frac{4 \cdot 360\overline{M_{c,hp}}}{(\theta_{max} - 360)^3} \tag{3.39}$$

$$b = \frac{2}{\theta_{max} - 360} \tag{3.40}$$

θ_{max} is the angle for which the replacement function has its maximum, in other words where the effective torque is peaking. Initially θ_{max} is set to $395°$ [5]. The parameters a and b are dependent on the working point and determined through the help of two boundary conditions. For more information on this as well as the replacement function, see [5].

The mean cylinder torque during the high pressure loop $\overline{M_{c,hp}}$, used to form a, is based on the combustion efficiency η_{vol} mapped as a function of engine speed and fuel ratio. Together with the fuel heating value, the fuel mass flow and the engine speed, $\overline{M_{c,hp}}$ can be calculated in the following way according to [**?**].

$$\overline{M_{c,hp}} = \frac{30\eta_{vol}Q_{HV}\dot{m}_f}{N_{eng}\pi} \tag{3.41}$$

3.4 Exhaust Manifold

The exhaust manifold is, like the intercooler and the inlet manifold, modelled as a stagnation component with the flow of gas and enthalpy connected to the cylinder and turbine as input signals. The pressure p_3, temperature T_3 and fraction of burnt gas $f_{3,e}$ within the manifold constitute the output signal. Since the exhaust manifold contains both air and exhaust gas, the following equations describing temperature and pressure have been expanded compared to the ones formed for the inlet manifold.

$$T_3 = \frac{\Delta U_3}{(m_{3,a_{in}} c_{p,a} + m_{3,e_{in}} c_{p,e})} \quad (3.42)$$

$$p_3 = \frac{(\Delta m_{3,a} R_a + \Delta m_{3,e} R_e) T_3}{V_3} \quad (3.43)$$

$c_{p,e}$ and R_e is the specific heat capacity and gas constant for exhaust gas respectively. The expression for the fraction of burnt gas is formed as a ratio between the different gas masses as follows:

$$f_{3,e} = \frac{m_{3,e_{in}}}{(m_{3,a_{in}} + m_{3,e_{in}})} \quad (3.44)$$

3.5 Turbine

Like the compressor, the turbine is modelled as a flow component. Input signals to the turbine block are the pressure, temperature and fraction of burnt gas within the exhaust manifold and the exhaust system, as well as the angular velocity of the turbo shaft. With the help of physical equations and two maps, the flow of air mass $\dot{m}_{3,a_{out}}$, exhaust gas mass $\dot{m}_{3,e_{out}}$ and enthalpy $\dot{H}_{3_{out}}$ through the turbine, as well as the turbine torque M_{turb} are derived as output signals. The two maps used in the turbine model describe the turbine mass flow parameter Φ_{turb} and the turbine efficiency η_{turb}. Like Φ_{comp} in the compressor model, Φ_{turb} describes the total mass flow through the turbine scaled with temperature and pressure,

$$\Phi_{turb} = \dot{m}_{3_{out}} \frac{\sqrt{T_3}}{p_3} \quad (3.45)$$

from which $\dot{m}_{3_{out}}$ can easily be derived. The specific mass flows of air and exhaust gas through the turbine are then formed with the help of the fraction of burnt gas $f_{3,e}$ within the exhaust manifold.

$$\dot{m}_{3,a_{out}} = (1 - f_{3,e})\,\dot{m}_{3_{out}} \quad (3.46)$$
$$\dot{m}_{3,e_{out}} = f_{3,\mathrm{e}}\dot{m}_{3_{out}} \quad (3.47)$$

Like for the compressor, the enthalpy flow from the exhaust manifold $\dot{H}_{3_{out}}$ going into the turbine is calculated through the basic formula of calorimetrics.

$$\dot{H}_{3_{out}} = c_{p,a}\dot{m}_{3_{out}}T_3 \quad (3.48)$$

To form the turbine torque M_{turb}, the enthalpy flow $\dot{H}_{4_{in}}$ leaving the turbine and going into the exhaust system has to be calculated. This is done through equation (2.9) according to the same principle as for the compressor, described earlier in this chapter. The turbine torque can then be formed according to equation (2.8).

Chapter 4

Simulation and Validation

This chapter presents and describes results acquired through simulations made with the model. For the following scenarios the model has been parameterised for the DaimlerChrysler engine OM646, a straight four cylinder diesel engine. Most simulations covered in this chapter have been run for the engine speed $N_{eng} = 1078rpm$ and the injected fuel quantity $m_f = 27.01mg$ to fit measured data.

4.1 Simulated Individual Cylinder Torque

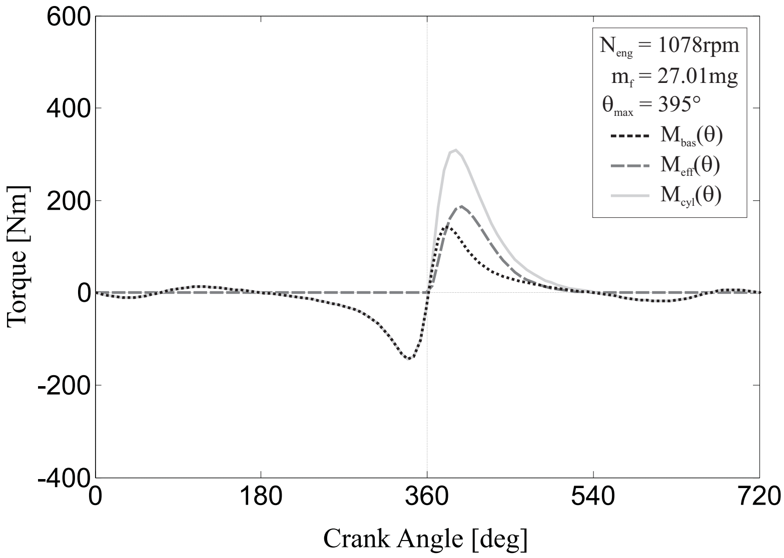

Figure 4.1: The simulated torque for one cylinder

By looking at the simulated torque for one cylinder displayed in Figure 4.1, the basic and effective torques forming the total cylinder torque as well as the basic structure of the operating cycle can be identified. The basic torque describing mass and gas forces outside combustion contributes over the entire cycle while the effective torque is active only during the expansion stroke. Just before the end of the cycle, it can be seen that the torque increases a bit. This is when the inlet valve opens and the flow of air between the two manifolds, through the cylinder, creates a short but positive work on the piston. Then as the piston reaches TDC, it turns and produces a negative torque as it is pulled down by the rotation of the crankshaft. The pressure then rises as more air enters the cylinder, and the torque turns positive. At $180°$ the compression stroke begins and the torque reaches the lowest point of the cycle as plenty of work is needed to compress the air within the cylinder. Shortly before $360°$, fuel is injected and ignited producing positive work characterised by the peaking torque curve. Then at $540°$ the exhaust stroke begins and the torque becomes negative since work is needed to push the exhaust gas out of the cylinder. For a theoretical description of the four stroke cycle, see Chapter 2.

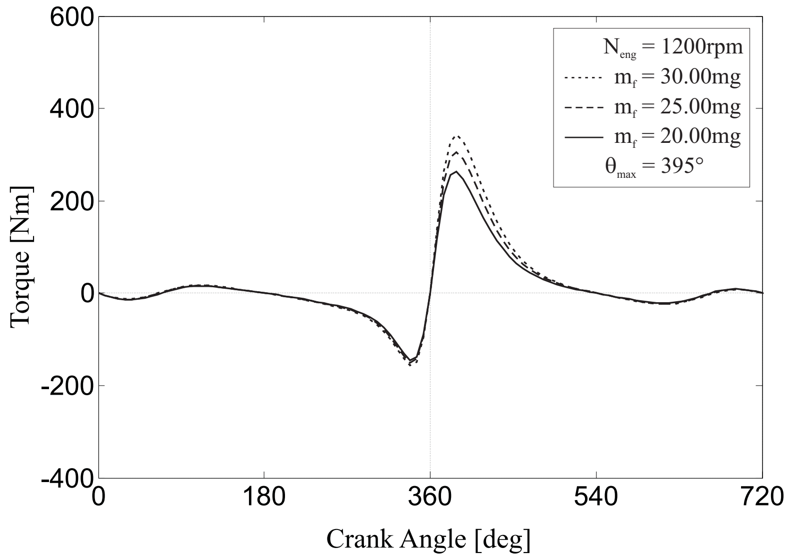

Figure 4.2: The simulated individual cylinder torque for $N_{eng} = 1200rpm$ and different injected fuel quantities

In Figure 4.2, 4.3 and 4.4 the simulated individual cylinder torque over one cycle for different engine speeds and injected fuel quantities is displayed.

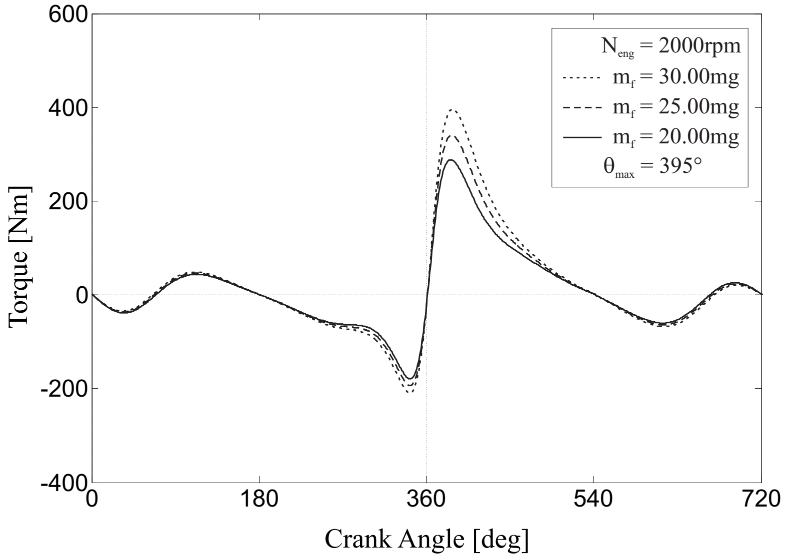

Figure 4.3: The simulated individual cylinder torque for $N_{eng} = 2000rpm$ and different injected fuel quantities

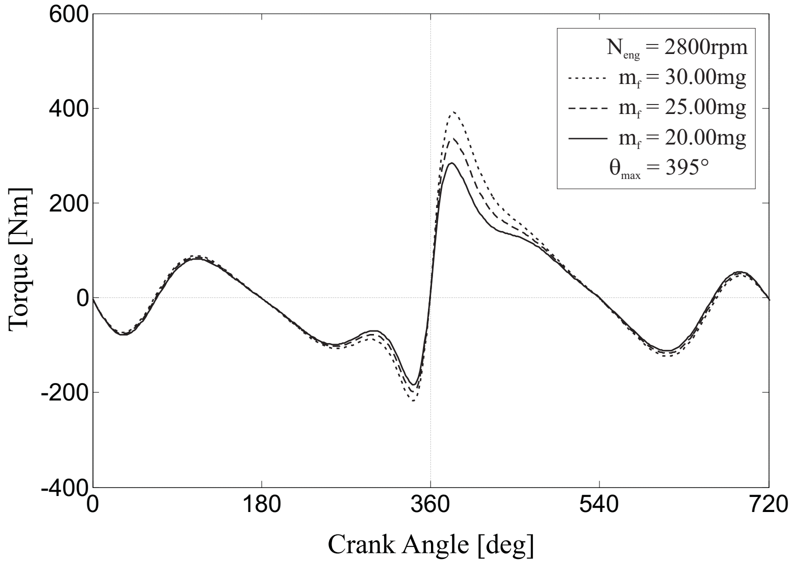

Figure 4.4: The simulated individual cylinder torque for $N_{eng} = 2800rpm$ and different injected fuel quantities

If we compare Figure 4.2, 4.3 and 4.4 we see how the torque increases for increasing engine speed as well as injected fuel quantity. The torque increase for increasing engine speed at constant fuel injection might seem strange since the torque would rather decrease in order to maintain the power of the engine. One possible explanation can be found in the wall-heat losses during the high pressure loop covered by the effective torque. These losses decrease for increasing engine speed and thereby alters the power output of the engine. Whether this change is big enough to explain the result still remains to be said.

4.2 Simulated Total Cylinder Torque

As mentioned in section 3.3, the torque from all four cylinders are added to form the total cylinder torque. In Figure 4.5 the simulated individual torque for all four cylinders is displayed with different lines.

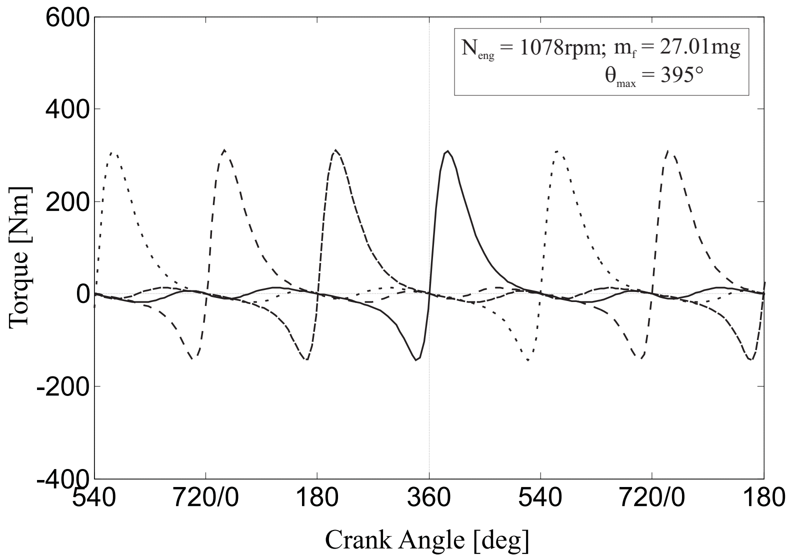

Figure 4.5: The simulated individual cylinder torque for all four cylinders

It can be seen that the individual torque peaks are shifted 180° for each added cylinder to spread out the work over the operating cycle. The result when adding all cylinder torques, the engine torque, can be seen in Figure 4.6.

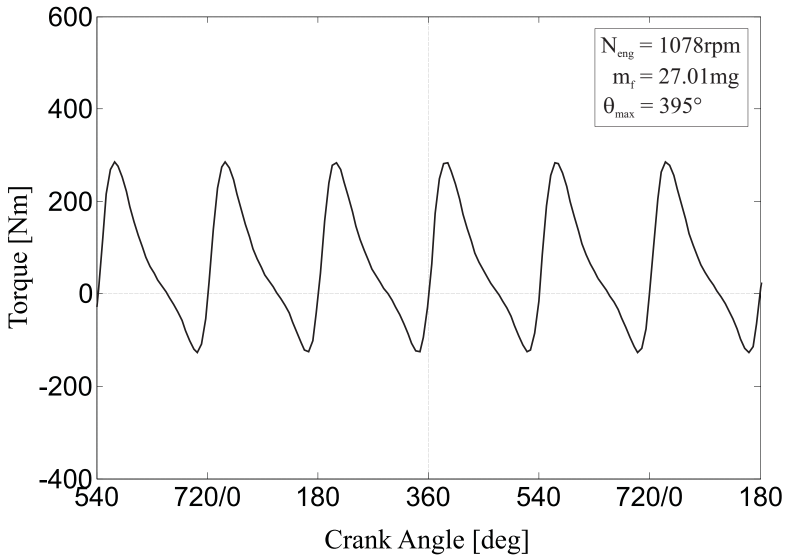

Figure 4.6: The simulated engine torque

4.3 Validation

To validate the simulated data, measured data for the engine torque has been produced with the help of a test car. The test car used is a Mercedes-Benz C220 with the engine OM646. The torque signal has been acquired from a wire strain gauge sensor fitted on the double mass flywheel after the auxiliary devices. To minimise the effects of the auxiliary devices, the air conditioning system has been switched off during the test runs.

The results acquired from these measurements are limited due to hardware related problems concerning the sampling frequency, and therefore only describe the torque for low engine speeds. In Figure 4.7 the measured engine torque for $N_{eng} = 1078rpm$ and $m_f = 27.01mg$ is displayed.

In comparison, the measured torque in Figure 4.7 differs from the simulated torque in Figure 4.6 when it comes to amplitude. The notch seen at the bottom of the first four torque peaks in Figure 4.7 is thought to be a signal processing error. By adjusting the tuning parameter θ_{max} described in Chapter 3, the difference in angle $\Delta\theta$ between the peaks of the effective torque and the basic torque can be minimised. Minimising $\Delta\theta$ leads to increasing amplitude of the cylinder torque. θ_{max} also affects the amplitude of the effective torque directly. In this case, θ_{max} is lowered from $395°$ to $381°$. The effect on the cylinder torque can be seen in Figure 4.8.

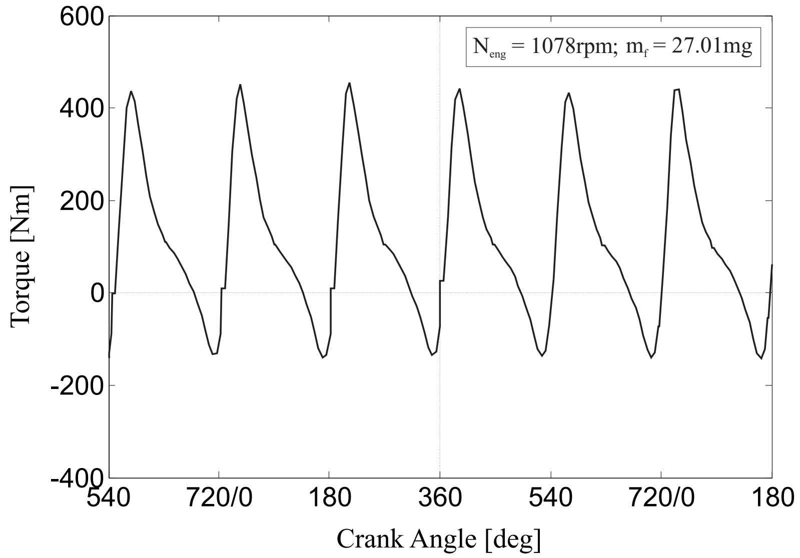

Figure 4.7: The measured engine torque.

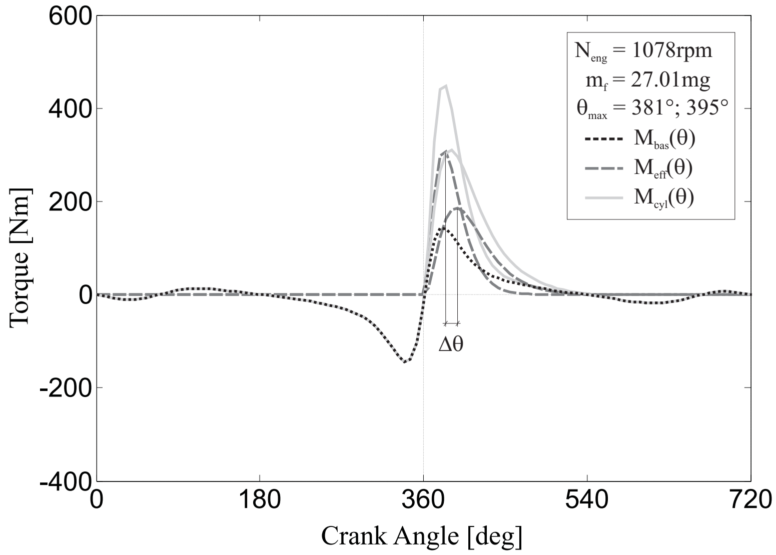

Figure 4.8: The simulated individual cylinder torque for $\theta_{max} = 381°$ as well as $\theta_{max} = 395°$.

By looking at the simulated engine torque for $\theta_{max} = 381°$ displayed together with the measured engine torque in Figure 4.9, one can see that the adjustment of θ_{max} described above brings the amplitude of the simulated engine torque closer to the measured value. However, at the same time it also changes the shape of the simulated torque making it less similar in that aspect. The difference between measured and simulated engine torque, the validation error, is also displayed in Figure 4.9. When examining the results, one should take into consideration the limited amount of measured data available during the validation process as well as the fact that the sensor itself is a source of measuring error to some degree.

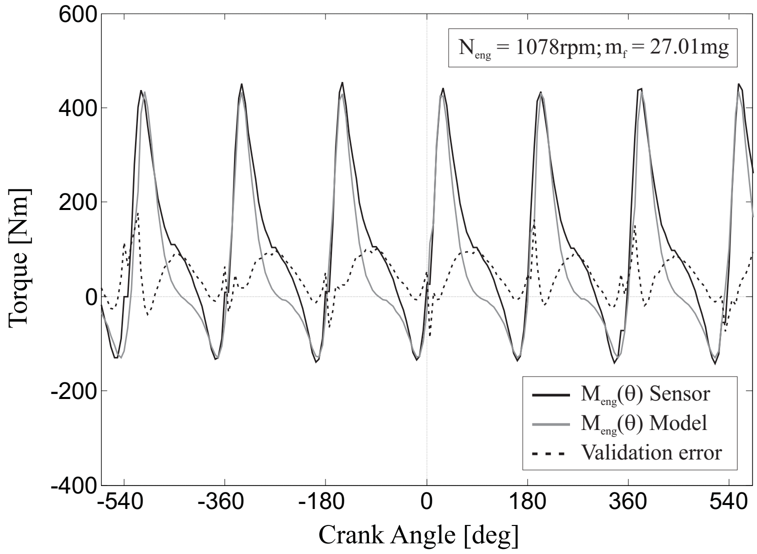

Figure 4.9: The measured and simulated engine torque for $\theta_{max} = 381°$ as well as the validation error.

Chapter 5

Conclusions

A cylinder-by-cylinder diesel engine model describing the angle synchronous engine torque has been constructed. The engine torque produced is the summation of all individual cylinder torques. Each cylinder torque is made up of one basic torque describing gas and mass forces working on the crankshaft outside combustion and one effective torque describing the remaining forces created during combustion through a replacement function. By studying the engine torque, the torque contribution for each individual cylinder can be distinguished and used to trigger functions of control or diagnostics. Through the gas path concept used and the crank angle based approach of the torque generation, the complexity of the model has been held down.

As described in Chapter 4 the simulated torque has been tuned with the help of the parameter θ_{max} and satisfies measured data from the test vehicle well for low engine speeds. For higher engine speeds, more measured data is needed before further conclusions concerning the accuracy of the model can be drawn. For increasing engine speed at constant fuel injection, the modelled torque increases when theory point towards a decrease. An explaination linked to the wall-heat losses during the high pressure loop has been given in Chapter 4.

The model constructed is neither computationally complex nor does it require any special efforts in order to be parameterised. That makes it suitable for Hardware-in-the-Loop implementation.

Chapter 6

Further Work

Even though the task has been undertaken in accordance with the predefined aims, work still exist in order to further improve the model. Three improvements, two towards improved accuracy and one towards Hardware-in-the-Loop implementation, have been recognised and are described below.

The first step towards improved accuracy is to validate the model more. That requires more and better measured data from the test vehicle. Once this has been acquired, other approaches for model improvement may be investigated. One such approach would be to look further into the replacement function describing the effective torque. As described in Chapter 4, the effective torque has been optimised to fit measured data by adapting the parameter θ_{max}. However, it is still unclear which and how different parameters affect the effective torque. The replacement function itself therefore needs to be further evaluated in order to determine whether a more accurate function, dependent on present or new parameters, can be found or not. It would also be of interest giving more thought to the possibility of modelling the effective torque physically.

As it is now, the model describes a constant injection over time. In reality however, the course of injection differs over the injection period. It would be interesting to improve the possibility of controlling the start of injection, maybe through a new parameter describing the time or crank angle at injection begin. In the present model the time or crank angle at which the injection begins is embodied in the parameter θ_{max} and the description of the effective torque. A correct modelling of the different aspects of the course of injection could provide a more accurate model.

One of the aims of the thesis was to create a model suitable for Hardware-in-the-Loop testing. This has been achieved through the low complexity of the model and its low calculation time. However, before implementing the model on a HiL testbed, it must be discretised. The sample rate might also need to be adjusted to fit the specific HiL standards.

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Notation

Variables and parameters

A	Area $[m^2]$
c_p	Specific heat capacity at constant pressure $[J/kgK]$
f	Fraction [-]
F	Force $[N]$
$\dot{H}$	Enthalpy flow $[J/s]$
J	Inertia $[kgm^2]$
l	Length $[m]$
L	Length ratio [-]
m	Mass $[kg]$
$\dot{m}$	Mass flow rate $[kg/s]$
M	Torque $[Nm]$
n_r	Number of cylinders [-]
N	Rotational speed $[rps]$
p	Pressure $[Pa]$
$\dot{p}$	Pressure differentiation rate $[Pa/s]$
Q_{HV}	Fuel heating value $[J/kg]$
r	Radius $[m]$
R	Gas constant $[J/kgK]$
s	Stroke $[m]$
$\ddot{s}$	Acceleration $[m/s^2]$
T	Temperature $[K]$
U	Energy $[J]$
v	Speed $[m/s]$
V	Volume $[m^3]$
w	Mean gas velocity $[m/s]$
Z	Proportion [-]
α	Polytropic exponent [-]
γ	Specific heat ratio for air [-]
η	Efficiency [-]
θ	Crank angle $[deg]$
(A/F)	Air/Fuel ratio [-]
Φ	Corrected mass flow $[kg/s]$
ω	Angular velocity $[rad/s]$
$\dot{\omega}$	Angular acceleration $[rad/s^2]$

Subscript identifiers

0	Air filter
1	Intercooler
2	Inlet manifold
3	Exhaust manifold
4	Exhaust system
5	EGR
a	Air
amb	Ambient
bas	Basic
bcp	Beginning of compression
c	Cylinder
$clear$	Clearance
cs	Crankshaft
cr	Connecting rod
$comp$	Compressor
$cool$	Cooling medium
cp	Compression
$disp$	Displacement
e	Exhaust gas
eff	Effective
eng	Engine
exh	Exhaust
f	Fuel
g	Gas
ge	Gas energy
hp	High pressure loop
int	Intake
m	Mass
oc	Oscillating part of the connecting rod
p	Piston
s/cr	Stroke to connecting rod
t	Tangential
tc	Turbocharger
th	Theoretically practical
$turb$	Turbine
vol	Volumetric

Subscript directions

in	Incoming
out	Outgoing

Abbreviations

BDC	Bottom Dead Centre
$CCEM$	Cylinder-by-Cylinder Engine Model
CI	Compression Ignited
ECU	Electronic Control Unit
EGR	Exhaust Gas Recirculation
HiL	Hardware-in-the-Loop
$MVEM$	Mean Value Engine Model
TDC	Top Dead Centre
VTG	Variable Turbine Geometry