

Czech Technical University in Prague

Faculty of Electrical Engineering

Department of Control Engineering



Diesel engine modelling and control

Master's thesis

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ZADÁNÍ DIPLOMOVÉ PRÁCE

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Obor: Technická kybernetika

Název tématu: Modelování a návrh řízení vznětového automobilového motoru

Zásady pro vypracování:

1. Nastudujte a popište problematiku vznětových motorů používaných v automobilech.
2. Vytvořte matematický model vznětového motoru a srovnajte s modely získanými při experimentální identifikaci.
3. Popište úlohy návrhu řízení, které jsou v současné době se vznětovými motory řešeny a uveďte známé technické obtíže.
4. Navrhněte jednoduchý regulátor (například PI, PID) pro úlohu řízení tlaku ve válci.
5. Zvolte vhodnou z pokročilejších metod návrhu robustních regulátorů (Hinf, μ , LPV) jako alternativu k existujícímu návrhu. Srovnajte výsledné chování uzavřené regulační smyčky.

Seznam odborné literatury:

- [1] Heywood, J.B.: *Internal Combustion Engines Fundamentals*. McGraw-Hill, 1988.
[2] Kiencke, U.; Nielsen, L.: *Automotive Control Systems: For Engine, Driveline, and Vehicle*. Springer, 2. vydání, 1998.

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Anotace

Vznětové motory hrají stále důležitější roli v automobilovém průmyslu. Pro dnešní výrobce těchto motorů je optimalizace spotřeby paliva a vypouštěných emisí klíčová, ale bohužel i velice drahá a časově náročná. Počítačová simulace vznětového motoru pomáhá tyto náklady výrazně snižovat. Cílem této diplomové práce je modelování a řízení procesů ve válci vznětového motoru.

První část se zabývá modelem vytvořeným v programu *Matlab-Simulink*. Tato část zahrnuje podrobný popis sacích a výfukových mechanismů, termodynamiky válce, spalovacích procesů i popis parametrů sloužících ke sledování výkonu motoru.

Druhá část se pak zaměřuje na návrh regulátoru pro řízení výstupního kroutícího momentu. Je navrženo několik vhodných regulátorů typu PI a jeden Hin regulátor. Výsledné chování uzavřené regulační smyčky je následně srovnáno.

Annotation

The diesel engines play an important role in today's automotive world. The computer simulation of the processes in these engines is the key for reducing costs in the development of less polluting and more efficient combustion engines. The objective of the thesis is to model and control processes in the cylinder of the diesel engine.

In the first part the detailed description of a *Matlab-Simulink* model of a cylinder is given. This includes the description of cylinders thermodynamics, geometry and mass flows, combustion process and engine performance.

The second part of this thesis is focused on a design of a controller for the control of engines output torque. Several appropriate PI controllers and H controller are suggested. The behavior of the closed-loop is then compared.

Prohlášení

Prohlašuji, že jsem svou diplomovou práci vypracoval samostatně a použil pouze podklady (literaturu, projekty, SW atd.) uvedené v příloženém seznamu.

V Praze dne 15.1.2008



.....
podpis

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I would like to thank Jiří Navrátil and Pavel Deutsch from Ricardo Prague for helping me to understand the combustion engines. My thanks also belongs to my supervisor Zdeněk Hurák for his time and support.

Table of Contents

1	Introduction.....	1
1.1	Motivation.....	1
1.2	Goals.....	1
1.3	History of the project.....	2
2	Diesel engine fundamentals.....	3
2.1	Introduction.....	3
2.2	Four-stroke diesel engine cycle.....	3
2.3	Thermodynamics basics	4
2.3.1	Enthalpy.....	4
3	Experimental data.....	4
3.1	Engine data.....	4
3.2	Chemical data	5
4	Cylinder model.....	6
4.1	Introduction.....	6
4.2	Model schematics.....	7
4.3	Implementation tool	8
4.4	Model description.....	8
4.4.1	Engine geometry.....	10
4.4.1.1	Piston position.....	10
4.4.1.2	Piston speed.....	11
4.4.1.3	Combustion chamber volume	11
4.4.2	Mass flow.....	12
4.4.2.1	Mass flow through valve.....	12
4.4.3	Thermodynamics.....	13
4.4.3.1	Cylinder thermodynamics description.....	13
4.4.3.2	Heat transfer.....	16
4.4.4	Combustion.....	17
4.4.4.1	Stoichiometric combustion.....	17
4.4.4.2	Pollutant formation in diesel engines.....	18
4.4.4.3	Vibe Function.....	19

4.4.5	Engine performance.....	21
4.4.5.1	Mean effective pressure.....	21
4.4.5.2	Torque.....	21
4.4.5.3	Air-fuel ratio.....	22
4.5	Model validation.....	22
4.5.1	Results	23
4.5.2	Comparison with other models.....	25
5	Control	26
5.1	Introduction	26
5.2	Control objectives.....	26
5.3	Mathematical description of cylinder behavior.....	27
5.3.1	BMEP.....	27
5.3.2	λ	29
5.4	Torque demand.....	30
5.4.1	Feedforward controller.....	30
5.4.2	Feedback controller.....	32
5.4.2.1	PI Controller.....	32
5.4.2.2	H_{∞} Controller.....	35
5.4.3	Simulation results.....	37
6	Conclusion.....	38
7	References.....	39
8	Appendix.....	40
8.1	A. Engine parameters.....	40
8.2	B. Cylinder model documentation.....	41
8.2.1	Cylinder model.....	41
8.2.2	Engine Cylinder.....	41
8.2.3	Controller.....	42
8.2.4	Feedforward controller.....	43
8.2.5	Feedback Controller.....	43
8.2.6	Cylinder.....	44
8.2.7	Cylinder geometry.....	45
8.2.8	External inputs.....	46
8.2.9	Intake flow.....	47
8.2.10	Exhaust flow.....	48

8.2.11	Valve	49
8.2.12	Combustion.....	50
8.2.13	Cylinder Mass.....	51
8.2.14	Cylinder Thermodynamics.....	52
8.2.15	Engine performance.....	53
8.2.16	cylinder_model_init.m.....	54
8.2.17	loadmodedata.m.....	56
8.2.18	interpolateData.m.....	57

1 Introduction

1.1 Motivation

There were more than 50 millions cars produced world wide in 2007. The car production increase by 5% every year. This development of automotive market brings many negative aspects that need to be seriously considered by automotive industry. Firstly, the internal combustion engine has become one of the major polluting contributors to our environment. Secondly, the oil prices increase every year as the oil is being depleted at high rates [8]. These circumstances force the engine manufacturers to develop and apply new ideas for combustions engines. New technologies coming from this development allow to produce less polluting and much more efficient combustion engines. These modern engines face several other challenges: to reduce emissions of pollutant species and fuel consumption, to increase power and safety, to improve drivability and passengers comfort [1].

All the above mentioned issues can be solved only by using advanced electronic control systems. Nowadays, these control systems are mainly based on experimental data. For every type of engine the measurements concerning the fuel economy and pollutants emissions are made. These measurements include a successive changes of each of the many parameters involved [8] in order to cover all the operating conditions of the engine. This is very expensive and time consuming.

There is an obvious alternative to the experimental approach: a computer simulation of the engine processes using a mathematical model. This approach allows to easily try and examine the effects of various designs and setups. The reliable model reduces the need for complex experimental analysis of the engine and consequently reduces the financial costs in the development.

1.2 Goals

There are two primary goals of this thesis. The first aim is to develop and validate a mathematical model. This model should allow to simulate the pressure and temperature in the cylinder during the engine cycle as well as the engine performance parameters. This model is not meant to be used in real-time applications and cannot be directly incorporated into the electronic control unit.

At the same time the model should be easily adjustable to any engine. The second aim is to control the engine performance. Various control strategies will be implemented and compared.

The goals of this thesis can be summed up as it follows:

1. Design and validate a cylinder model for given diesel engine.
2. Design a control algorithm to control engines performance

1.3 History of the project

The topic of this thesis was initially suggested by company Ricardo Prague. Ricardo Group is the leading provider of technology, product innovation, engineering solutions and strategic consulting to world's automotive industries. It develops engines components, electronic hardware and software as well as control systems.

The initial thesis suggestion concerned engines modelling and control as well as real-time estimation of engines parameters. The estimation part was closely connected to another thesis that would have concerned electronic hardware. Due to lack of interest from Ricardo Prague neither the hardware nor the software for parameter estimation were implemented.

Therefore this thesis has moved more towards the non real-time engine modelling and control. Ricardo Prague has provided the essential data and experience. Unfortunately, nobody from Ricardo Prague has directly supervised this thesis and sometimes it was quite difficult to identify particular goals. Also the control problem was not clearly defined from the beginning and this fact posed many problems in the development of this model because it was unclear what is its main purpose.

For the above mentioned reasons the model is as general as possible and thus easily suitable for many control problems. In spite of all the obstacles this thesis brings a significant insight into many processes taking place in diesel engines and also shows the problems connected to diesel engine control. Besides the obvious outcomes of this thesis as is the cylinder model and control algorithms, the thesis also provided me with the unique opportunity to work on many projects in the international company.

2 Diesel engine fundamentals

2.1 Introduction

In this chapter the basics of the four-stroke diesel engines are introduced. The knowledge of these basic concepts is essential in order to fully comprehend the next chapters. Many processes are also explained in the chapter 4.4 that covers the actual model. The source [2] and [4] are recommended for further information on combustion engine fundamentals.

2.2 Four-stroke diesel engine cycle

The four-stroke diesel engine cycle has four important phases. These phases are: *Intake*, *Compression*, *Expansion*, *Exhaust*. The sequence of four-stroke engine phases is indicated by the Figure 2.1.

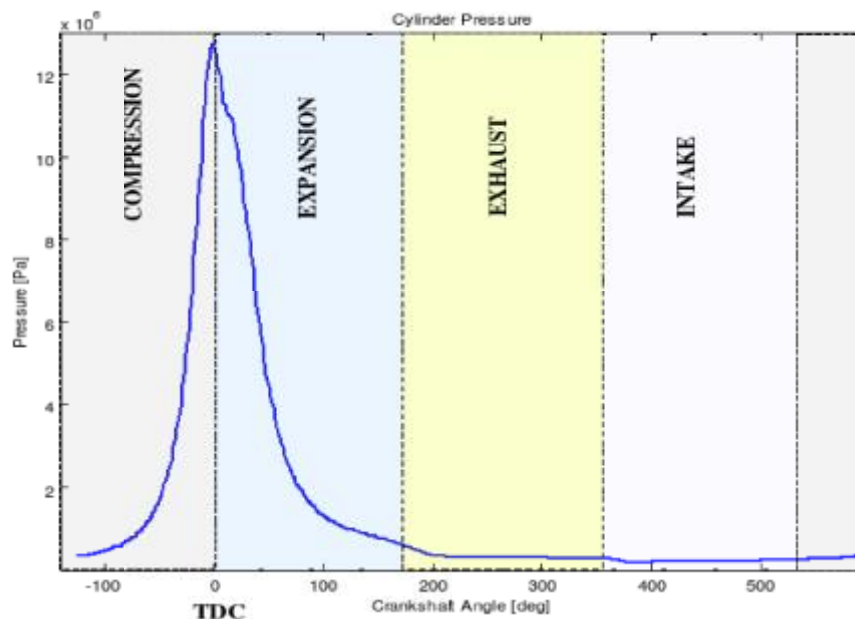


Figure 2.1:

The whole cycle lasts two full revolutions of the crank shaft (720°). The cycle starts with *Intake* phase. During this phase the intake valve opens and the mixture of fresh air and exhaust gases is

sucked into a cylinder. Once the intake valve is closed and the piston moves towards TDC (Figure 2.2), the *Compression* phase takes place. During this phase the gas mixture is compressed by the piston. When the piston approaches the TDC, the pressure rise can be observed as the fuel is injected into the cylinder and starts burning. The combustion occurs with a slight delay after fuel injection. During the *Expansion* phase, the gas expands and does work as the piston is on the way down. In the last *Exhaust* phase the combusted mixture is pressed out from the cylinder and the whole cycle repeats itself.

2.3 Thermodynamics basics

The profound description of the thermodynamics is available in [6]. Its applications and consequences for the engine world are explained in [2] and [4].

2.3.1 Enthalpy

The definition of enthalpy is $H = U + p \cdot V$ where U corresponds to internal energy, p to pressure and V to volume. The term $p \cdot V$ corresponds to the energy associated with work done by the system [16]. For our purposes it is sometimes more useful to define the overall enthalpy by specific enthalpies of present chemical compounds. This brings following formula:

$$H = \sum_i m_i \cdot h_i$$
 where m_i is the mass of chemical i and h_i is the specific enthalpy of this chemical. Note that specific enthalpy is dependent on the temperature.

3 Experimental data

3.1 Engine data

Experimental data were provided by Ricardo Prague. This data come from a turbocharged tractor engine with the displacement volume of 3.3 liters. Exact engine parameters can be found in Appendix A. Following important data were selected as external inputs to the model: *intake manifold pressure and temperature, exhaust manifold pressure and temperature, intake and exhaust valve lift*. The *cylinder pressure and temperature* data are used for the model validation. Another set of data was used to describe the combustion behavior and to find the heat release coefficients for Vibe function.

All these data were available for several working points (different loads) of the engine (except the heat release rate).

3.2 Chemical data

There are several chemical compounds that play a crucial role in the diesel engine cycle. From the internal energy point of view the *Argon* (Ar) , *Oxygen* (O₂), *Nitrogen* (N₂), *Carbon dioxide* (CO₂), *Water vapor* (H₂O) and *Fuel* (C_xH_x) are essential. The specific enthalpies and specific heat values for various temperature conditions need to be known. These data are taken from [2]. The *diesel fuel* (or any other fuel if used with this model) properties need to be also considered. The diesel fuel is dodecane C_{12.5}H_{25.5} and its properties were taken from [5].

4 Cylinder model

4.1 Introduction

The processes in the cylinder of diesel engine are very complex. This is caused by naturally transient and heterogeneous character, controller mainly turbulent mixing of fuel and air [9]. There exist many approaches towards diesel engine modelling. These approaches depends on the use of these models and their accuracy and complexity varies a great extend. For control purposes and for the modelling of complete engine the family of *mean value models* (MVM) is commonly used. The advantages of these models are especially the reasonable precision and low computational complexity. MVM models neglect the discrete cycles of the engine and assume all processes and effects are spread out over the engine cycle [1]. Another family of control oriented models is called *discrete event models* (DEM). The models are useful for air/fuel ratio feedforward control and misfire detection algorithms [1]. The model developed in this thesis is a **discrete event model**.

It is also **one-zone model**. This assumes a homogeneous conditions throughout the cylinder. This is a good approximation for the compression-ignition engines but not for the spark-ignition engines [3]. Another important characteristic is that the air flows outside the cylinder are not modeled. Instead of this the conditions in the intake and exhaust manifold are introduced to the model as the external inputs. This significantly simplifies the structure of the model because the external parts (turbocharger, compressor, manifolds etc.) do not need to be modeled. Such approach is also called **air flow chop** modelling.

The following external data are necessary: pressure and temperature in intake manifold, pressure and temperature in the exhaust manifold and the intake and exhaust valve lift. The model data flow is shown on the Figure 4.1.

4.2 Model schematics

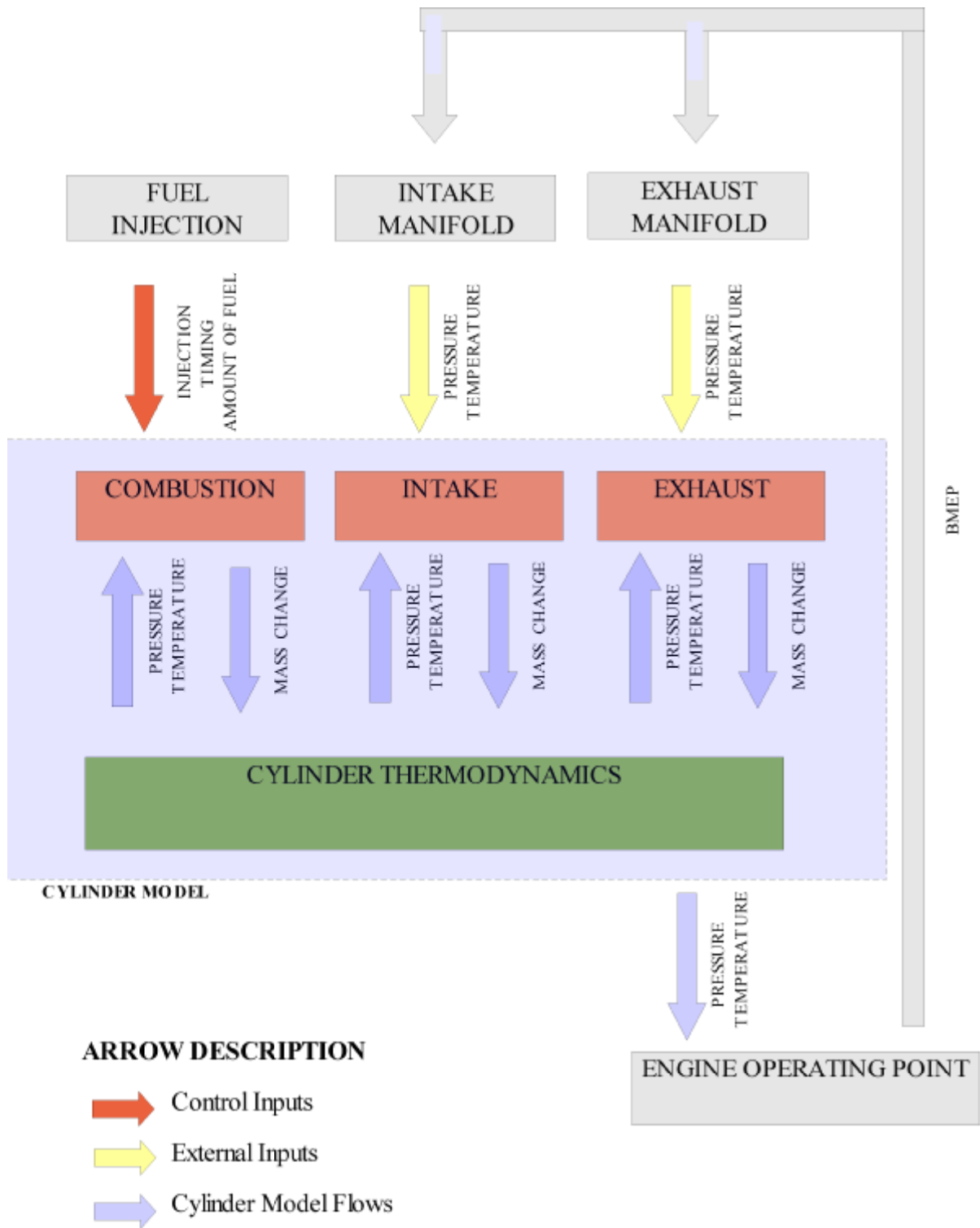


Figure 4.1

4.3 Implementation tool

There are several software tools that allows modelling and graphical representation of a complex physical processes as diesel engine cycle. On the other hand there are just a few like *Matlab* and *Modelica* that are widely used in automotive industry. These two tools differ in a way how the physical processes are represented. *Matlab-Simulink* strictly uses causal description of the process and is more appropriate for real-time and control applications and control . *Modelica* uses the concept of energy transfers between different parts of the physical process without obvious direction and seems to be more suitable tool for modelling of physical processes. However there are two main reasons why is the *Matlab-Simulink* used for the implementation. Firstly this tool is widely spread among academic community (unlike *Modelica*) and secondly it can be easily used for control purposes. In the latest version of *Matlab* the new modelling tool called Simscape is introduced. This tool used *Modelica* concept of modelling and can be also easily linked with other Matlab parts as Simulink and Stateflow.

The diesel engine model described below is implemented in Simulink and requires Matlab R14 or higher.

4.4 Model description

The model structure can be divided into five important parts that interact with each other. The first part is the *Engine geometry*. In this part all the subsystems implementing the piston movement, cylinder volume and crank angle generation are covered. The second part concerns the *Mass flow* subsystems and includes the description mass flow through the intake and exhaust valve. The third part describes the *Combustion* subsystems and gives brief overview about the combustion processes. The fourth part in a greater detail describes the *Cylinder thermodynamics* including the derivation of cylinder pressure and temperature development. The last part shows the commonly used *Engine performance* parameters.

The Figure 4.2 shows the data flows between these parts:

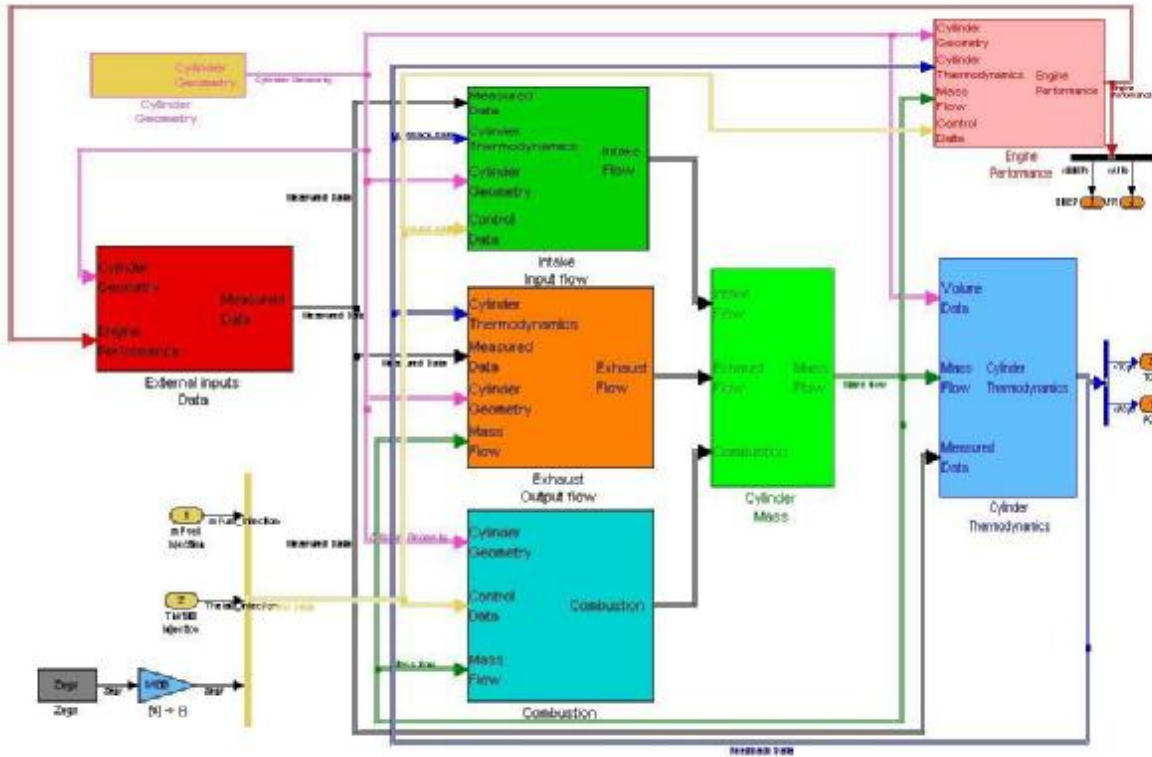


Figure 4.2:

This chapter of this thesis gives the theoretical background for all these blocks. For the complete documentation, please refer to Appendix B.

4.4.1 Engine geometry

4.4.1.1 Piston position

The engine geometry is shown in Figure 1. The vertical position of the piston X is completely determined by the crank angle θ .

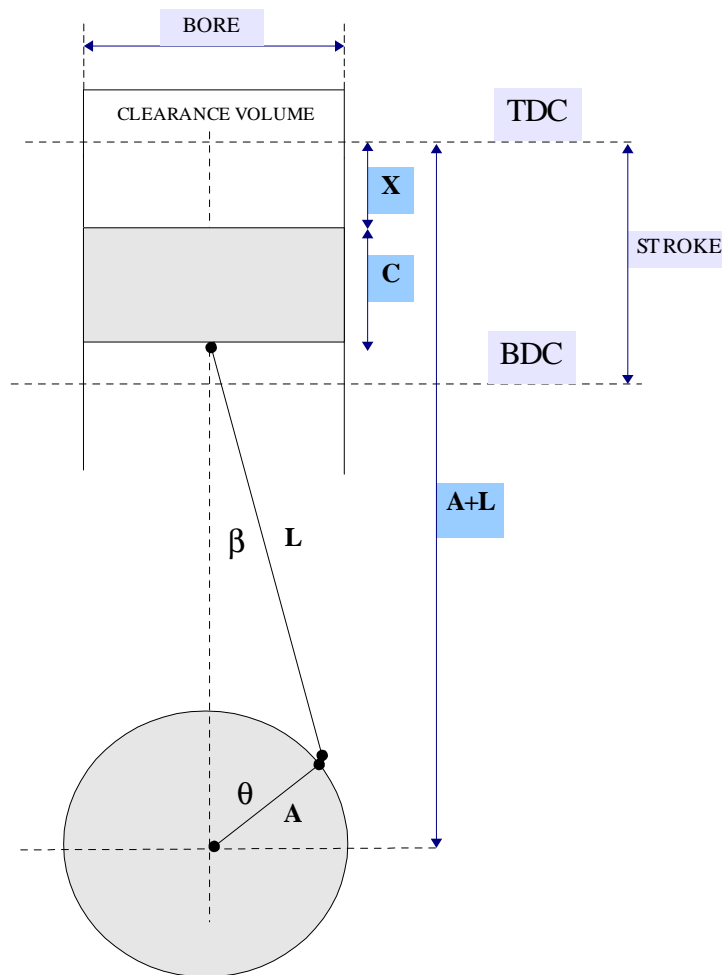


Figure 4.3: Engine Geometry

The piston position X can be derived as function of crank angle θ using cosine theorem. Variables used in the following equations corresponds to the Figure 4.2.

The engine's geometry can be described by following equation

$$L \cdot \cos(\beta) + A \cdot \cos(\theta) + X + C = A + L + C$$

Equation 4.1

to get rid of the term $\cos(\beta)$ the following equality is used

$$L \cdot \sin(\beta) = A \cdot \sin(\theta)$$

Equation 4.2

and by geometric transformation on the Equation 4.2, the $\cos(\beta)$ is expressed as function of crank angle

$$\cos(\beta) = \sqrt{1 - \left(\frac{a}{l}\right)^2 \cdot \sin^2(\theta)}$$

Equation 4.3

and by combination of equation 4.1 and 4.3, the piston position as function of crank angle is finally obtained:

$$X(\theta) = A + L - \sqrt{l^2 - a^2 \cdot \sin^2(\theta)} - A \cdot \cos(\theta)$$

Equation 4.4

4.4.1.2 Piston speed

The instantaneous piston speed $v_P = \frac{dX}{dt} = \frac{dX}{d\theta} \frac{d\theta}{dt} = \frac{dX}{d\theta} \omega_E$ where ω_E is the crank angle

Equation 4.5

angular velocity [3]. Now by using the equation 4.4 and 4.5 we get the instantaneous piston speed.

$$\frac{dX}{d\theta} = \frac{A^2 \cdot \sin(2 \cdot \theta)}{\sqrt{(L^2 - A^2 \cdot \sin^2(\theta))}} + A \cdot \sin(\theta) = A \cdot \sin(\theta) \cdot \left(1 + \frac{2 \cdot A \cdot \cos(\theta)}{\sqrt{(L^2 - A^2 \cdot \sin^2(\theta))}} \right)$$

Equation 4.6

4.4.1.3 Combustion chamber volume

The combustion chamber volume can be at any crank position defined as

$V(\theta) = V_c + \frac{\pi \cdot BORE}{4} \cdot X(\theta)$. The V_c is called clearance volume and corresponds to the cylinder volume at TDC. The $X(\theta)$ is defined by the Equation 4.4. The total displacement volume V_D is expressed as $V(\pi) - V(0)$ and that equals to $\frac{\pi \cdot BORE^2}{4} \cdot L$ [3]. The engines compression ratio r is defined as $r = 1 + \frac{V_D}{V_c}$ [5].

4.4.2 Mass flow

4.4.2.1 Mass flow through valve

The following equation set is taken from [1]. The flow equation through valve for compressible fluids obtained from thermodynamic relationships for isentropic expansion is following:

$$\dot{m}(t) = C_D \cdot A_R(t) \frac{P_{INP}(t)}{\sqrt{R C T_{INP}(t)}} \cdot \Psi \left(\frac{P_{INP}(t)}{P_{OUT}(t)} \right)$$

Equation 4.7

where the function $\Psi \left(\frac{P_{INP}(t)}{P_{OUT}(t)} \right)$ in Equation 5.1 is defined as

$$\Psi \left(\frac{P_{INP}(t)}{P_{OUT}(t)} \right) = \sqrt{\kappa \cdot \left(\frac{2}{\kappa + 1} \right)^{\left(\frac{\kappa + 1}{\kappa - 1} \right)}} \quad \text{for } P_{OUT} < P_{CR}$$

$$\Psi \left(\frac{P_{INP}(t)}{P_{OUT}(t)} \right) = \left(\frac{P_{INP}(t)}{P_{OUT}(t)} \right)^{\frac{1}{\kappa}} \cdot \sqrt{\left(\frac{2 \cdot \kappa}{\kappa - 1} \right) \cdot \left(1 - \frac{P_{INP}(t)}{P_{OUT}(t)} \right)^{\left(\frac{\kappa - 1}{\kappa} \right)}} \quad \text{for } P_{OUT} \geq P_{CR}$$

where $P_{CR} = \left(\frac{2}{\kappa + 1} \right)^{\left(\frac{\kappa}{\kappa - 1} \right)} \cdot P_{INP}$

The term $A_R(t)$ in the Equation 4.7 corresponds to the reference area of the valve. This area can be approximated by $A_R(\theta) = \pi \cdot (ValveDiameter) \cdot (ValveLift(\theta))$ [2]. The valve lift is an external input to the model and has to be measured on the engine. The valve diameter is given by engine's specifications available in Appendix A.

The term C_D expresses the discharge coefficient. This coefficient was set empirically to achieve the best correspondence between the measured and simulated flows.

4.4.3 Thermodynamics

4.4.3.1 Cylinder thermodynamics description

This laws are taken from the following sources [7] and [4] and the more profound explanation can be also found in [2].

The cylinders thermodynamics is based on the first law of thermodynamics that can be written as $\Delta U = \Delta Q - \Delta W$ where ΔQ corresponds to the small amount of heat added and ΔW corresponds to the small amount of work done by the cylinder. The ΔU is often called the change of internal energy. By rearranging the equation and introducing the enthalpy and heat transfer, the following equations can be obtained for each chemical compound present in the cylinder:

$$\frac{dU}{d\theta} = \frac{dH_{INP}}{d\theta} - \frac{dH_{OUT}}{d\theta} + \frac{dQ_{TR}}{d\theta} - \frac{dW}{d\theta}$$

Equation 4.8

$$\frac{dH}{d\theta} = \left(\frac{dm}{d\theta}\right) \cdot h + m \cdot \left(\frac{dh}{d\theta}\right)$$

Equation 4.9

by injecting the Equation 4.9 to the Equation 4.8

$$\frac{dU}{d\theta} = \left(\frac{dm_{INP}}{d\theta}\right) \cdot h_{INP} + m_{INP} \cdot \left(\frac{dh_{INP}}{d\theta}\right) - \left(\frac{dm_{OUT}}{d\theta}\right) \cdot h_{OUT} + m_{OUT} \cdot \left(\frac{dh_{OUT}}{d\theta}\right) + \frac{dQ_{TR}}{d\theta} - p \cdot \left(\frac{dV}{d\theta}\right)$$

Equation 4.10

and the specific enthalpy change can be neglected in the Equation 4.10

$$\frac{dU}{d\theta} = \left(\frac{dm_{INP}}{d\theta}\right) \cdot h_{INP} - \left(\frac{dm_{OUT}}{d\theta}\right) \cdot h_{OUT} + \frac{dQ_{TR}}{d\theta} - p \cdot \left(\frac{dV}{d\theta}\right)$$

Equation 4.11

From the enthalpy definition $H = U + p \cdot V$ the following equation can be deduced by

rearranging the terms and derivation with respect to crank angle

$$\frac{dU}{d\theta} = \frac{dH}{d\theta} - V \cdot \frac{dp}{d\theta} - p \cdot \frac{dV}{d\theta}$$

Equation 4.12

by using another enthalpy definition that uses specific enthalpies of chemicals $H = m \cdot h$

$$\frac{dU}{d\theta} = m \cdot \frac{dh}{d\theta} + h \cdot \frac{dm}{d\theta} - V \cdot \frac{dp}{d\theta} - p \cdot \frac{dV}{d\theta}$$

Equation 4.13

and by introducing the temperature derivate instead of specific enthalpy derivative to Equation 4.13

$$\frac{dU}{d\theta} = m \cdot \frac{dh}{dT} \frac{dT}{d\theta} + h \cdot \frac{dm}{d\theta} - V \cdot \frac{dp}{d\theta} - p \cdot \frac{dV}{d\theta}$$

Equation 4.14

where $\frac{dh}{dT} = c_p$ is known as specific heat constant and finally the following equation is obtained

$$\frac{dU}{d\theta} = m \cdot c_p(T) \cdot \frac{dT}{d\theta} + h(T) \cdot \frac{dm}{d\theta} - V \cdot \frac{dp}{d\theta} - p \cdot \frac{dV}{d\theta}$$

Equation 4.15

To eliminate the term $V \cdot \frac{dp}{d\theta}$ from Equation 4.15 we can use the ideal gas law

$$p \cdot V = \frac{m}{M} \cdot r \cdot T$$

and by substitution $(k = \frac{m}{M})$ and derivation with respect to θ , the following equation is obtained

$$p \cdot \frac{dV}{d\theta} + V \cdot \frac{dp}{d\theta} = k \cdot \left(m \cdot \frac{dT}{d\theta} + T \cdot \frac{dm}{d\theta} \right)$$

Equation 4.16

by rearranging Equation 4.16, the following expression is obtained:

$$V \cdot \frac{dp}{d\theta} = k \cdot \left(m \cdot \frac{dT}{d\theta} + T \cdot \frac{dm}{d\theta} \right) - p \cdot \frac{dV}{d\theta}$$

Equation 4.17

by introducing Equation 4.17 to the Equation 4.15 the following equation is obtained:

$$\frac{dU}{d\theta} = m \cdot c_p(T) \cdot \frac{dT}{d\theta} + h(T) \cdot \frac{dm}{d\theta} - k \cdot \left(m \cdot \frac{dT}{d\theta} + T \cdot \frac{dm}{d\theta} \right) - 2 \cdot p \cdot \frac{dV}{d\theta}$$

Equation 4.18

by comparing the Equations 4.11 and 4.18 the cylinder temperature differential can be deduced:

$$\frac{dT}{d\theta} = \frac{1}{c_p(T) \cdot m - k \cdot m} \left(h_{INP} \cdot \left(\frac{dm_{INP}}{d\theta} \right) - h_{OUT} \cdot \left(\frac{dm_{OUT}}{d\theta} \right) + \frac{dQ_{TR}}{d\theta} - h(T) \cdot \frac{dm}{d\theta} + k \cdot T \cdot \frac{dm}{d\theta} - p \cdot \frac{dV}{d\theta} \right)$$

Equation 4.19

This temperature change has to be consider for all the chemical compounds present in the

cylinder. The total number of moles in the cylinder can be described by $k = \sum_i k_i$ where

$i \in (O_2, N_2, Ar, CO_2, H_2O)$. The internal energies of the other chemicals as CO, NO, NO₂, H₂ etc. are negligible and they are not taken into account. Therefore the overall cylinder temperature derivative is derived:

$$\frac{dT_{CYL}}{d\theta} = \frac{\sum_i h_{INPi} \cdot \left(\frac{dm_{INPi}}{d\theta} \right) - \sum_i h_{OUTi} \cdot \left(\frac{dm_{OUTi}}{d\theta} \right) - \sum_i h_i(T) \cdot \frac{dm_i}{d\theta} + k_i \cdot T_{CYL} \cdot \frac{dm_i}{d\theta} - p_{CYL} \cdot \frac{dV_{CYL}}{d\theta} + \frac{dQ_{TR}}{d\theta}}{\sum_i c_{Pi}(T) \cdot m_i - k_i \cdot m_i}$$

where $k_i = \frac{m_i}{M_i}$ and $i \in (O_2, N_2, Ar, CO_2, H_2O)$.

Then the cylinder pressure is obtained using ideal gas law.

$$p_{CYL} = \frac{T_{CYL}}{V_{CYL}} \sum_i r_i \cdot m_i$$

4.4.3.2 Heat transfer

Typically the 20-35% of the fuel energy is passed on by the heat transfer to the engine coolant [11]. The primary heat transfer mechanism from the cylinder gases to the surrounding areas is the convection through cylinder wall. The heat loss to the wall is generally given by:

$$\frac{dQ_{TR}}{dt} = h_C \cdot A \cdot (T_{CYL} - T_{WALL})$$

where A corresponds to the convection area derived from the cylinder geometry and T_{WALL} is the average temperature of surrounding area. The convection heat transfer coefficient h_C can be estimated by several empirical approaches as Woschni, Annand, Hohenberg, Eichelberg [4] heat transfer correlations. The brief overview of these some of them is given below. Further informations can be also found in [11], [10] and [4]. The best results shall be obtained with Woschni coefficient.

When simulating heat transfer in crank angle domain the energy transfer becomes

$$\frac{dQ_{TR}}{d\theta} = \frac{dQ_{TR}}{dt} \cdot \frac{dt}{d\theta} = \frac{dQ_{TR}}{dt} \cdot \frac{60}{2 \cdot \pi \cdot N} \quad \text{where } N \text{ [rpm] is the engine speed.}$$

Eichelberg heat transfer coefficient

The Eichelberg heat transfer coefficient is given by [4]: $h_C = 7.8 \cdot 10^{-11} \cdot u_P^{0.3333} \cdot p^{0.5} \cdot T^{0.5}$

Hohenberg heat transfer coefficient

The Hohenberg heat transfer coefficient is given by [2]:

$$h_C = C_1 \cdot V^{-0.06} \cdot p^{0.8} \cdot T^{-0.4} \cdot (u_P^{0.3333} + C_2)^{0.8}$$

The coefficients C1 and C2 for diesel engines are equal to $C_1 = 13 \cdot 10^{-3}$, $C_2 = 1.4$

Woschni heat transfer coefficient

The Woschni heat transfer coefficient is given by [3]: $h_C = 3.26 \cdot 10^{-3} \cdot B^{-0.2} \cdot p^{0.8} \cdot T^{-0.55} \cdot u_P^{0.8}$

There also exist more complicated formula for this coefficient given by [2].

Comparison of heat transfer approaches

The Figure 4.3 shows the results of these three heat transfer approaches.

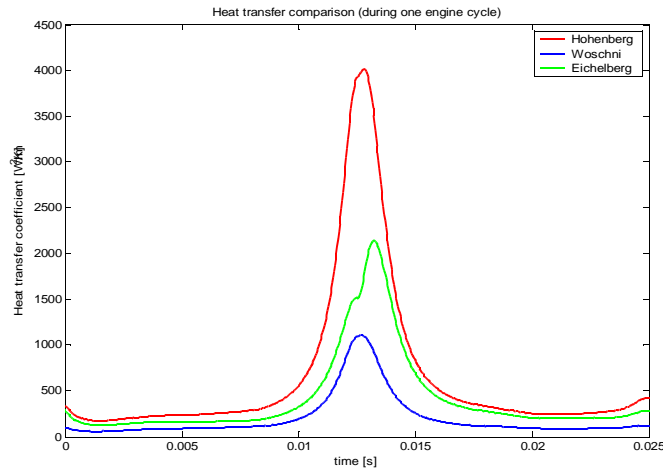


Figure 4.4

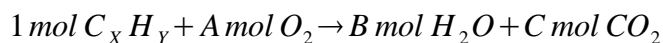
The fact that the radiation heat transfer component is often neglected makes the selection of an appropriate model for the heat losses problematic [10]. However the Woschni coefficient gives the best results for this model.

4.4.4 Combustion

4.4.4.1 Stoichiometric combustion

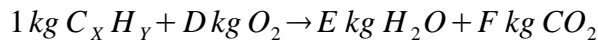
The combustion reactions are very complicated and involve a large number of chemical compounds resulting in many reaction products. The complete list of reactions including their kinetics behavior can be found in [11]. The relevant description is also given by [8] and [9].

The complete combustion of hydrocarbon C_xH_y follows this chemical formula:



The following coefficient can be therefore derived: $A = X + \frac{Y}{4}$, $B = \frac{Y}{2}$, $C = X$.

This can be of course rewritten as a mass balance equation:



The coefficients of this equation are:

$$D = \left(X + \frac{Y}{4} \right) \cdot \frac{M_{O_2}}{(X \cdot M_C + Y \cdot M_H)}; E = \left(\frac{Y}{2} \right) \cdot \frac{M_{H_2O}}{(X \cdot M_C + Y \cdot M_H)}; F = X \cdot \frac{M_{CO_2}}{(X \cdot M_C + Y \cdot M_H)}$$

The standard formula of diesel fuel is $C_{12.5}H_{22.5}$ and the molar mass for various molecules are available in Appendix C. Therefore the exact value of coefficients is:

$$A = 18.125, B = 11.25, C = 12.5, D = 3.359, E = 1.173, F = 3.185$$

The air contains 79% of N₂ and 21% of O₂. Therefore the stoichiometric mass of air needed to burn 1 kg of diesel fuel is proportional to the mass of air that contains 3.243kg of O₂. The total

mass of this air is equal to $m_{AIR} = \frac{3.359}{0.21 \cdot M_{O_2}} \cdot (0.21 \cdot M_{O_2} + 0.79 \cdot M_{N_2}) = 14.43 \text{ kg}$. This corresponds to the values given in [1].

4.4.4.2 Pollutant formation in diesel engines

The complete combustion assumed above is not attained even in perfect thermodynamic equilibrium conditions. The pollutant species are always emitted. The main pollutant species in exhaust gases are nitrogen oxides NO_x, particulate matter PM, carbon monoxide CO and hydrocarbons HC. The primary factor that affects the formation of these pollutants is the AFR. The other important factors are injection timing, cylinder pressure and the EGR rate [1].

The modelling of these pollutant species requires profound knowledge of chemical reaction kinetics and goes beyond the purpose of this text. The model thus considers only complete combustion reactions. Nevertheless further information can be found in [1],[11] and the reactions kinetics data can be found in NIST tables available at [12].

4.4.4.3 Vibe Function

The fuel injected into cylinder does not burn all at once, but the heat released at given time depends on the combustion delays and cylinder temperature. The Vibe function is an empirical function that shows the amount of fuel combusted at given crank angle. This function is given by:

$$W(\theta) = 1 - \exp\left(-a \cdot \left(\frac{\theta - \theta_{IGN}}{\Delta\theta}\right)^n\right)$$
 where a, n are function parameters, $\Delta\theta$ corresponds to

combustion duration in degrees of crank angle and θ_{IGN} to crank angle when the combustion occurred.

These parameters are different for every engine and they have to be set up from the measured data.

Similarly the Vibe function derivative can be derived:

$$\frac{dW(\theta)}{d\theta} = \frac{a \cdot n}{\Delta\theta} \cdot \left(\frac{\theta - \theta_{IGN}}{\Delta\theta}\right)^{n-1} \cdot \exp\left(-a \cdot \left(\frac{\theta - \theta_{IGN}}{\Delta\theta}\right)^n\right)$$

The Vibe function coefficients a, n can be fitted to the measured heat release rate using linear regression. There are four important steps:

1. $W(\theta) = 1 - \exp\left(-a \cdot \left(\frac{\theta - \theta_{IGN}}{\Delta\theta}\right)^n\right)$
2. $\ln(1 - W(\theta)) = -a \cdot \left(\frac{\theta - \theta_{IGN}}{\Delta\theta}\right)^n$
3. $\ln(-\ln(1 - W(\theta))) = \ln\left(a \cdot \left(\frac{\theta - \theta_{IGN}}{\Delta\theta}\right)^n\right)$
4. $\ln(-\ln(1 - W(\theta))) = \ln(a) + n \cdot \ln\left(\frac{\theta - \theta_{IGN}}{\Delta\theta}\right)$

The $W(\theta), \Delta\theta, \theta_{IGN}$ are directly taken from measured data and consequently the coefficients a, n can be determined. On the contrary this method greatly depends on the numerical accuracy as measured heat release rate (or Vibe function if you insist) reaches one. Therefore the matlab function *lsqcurvefit* is used for obtaining the results. The results are shown in the Figure 4.5.

```
Vibe parameters
F(X) = (1 - exp(-aWiebe*((X-Theta0)./ThetaL).^nWiebe))
          Theta0 = 3.322800 ; ThetaL = 58.234200
lsqcurvefit:      aWiebe = 6.082167 ; nWiebe = 1.929291
linear regression: aWiebe = 5.294775 ; nWiebe = 1.735528
```

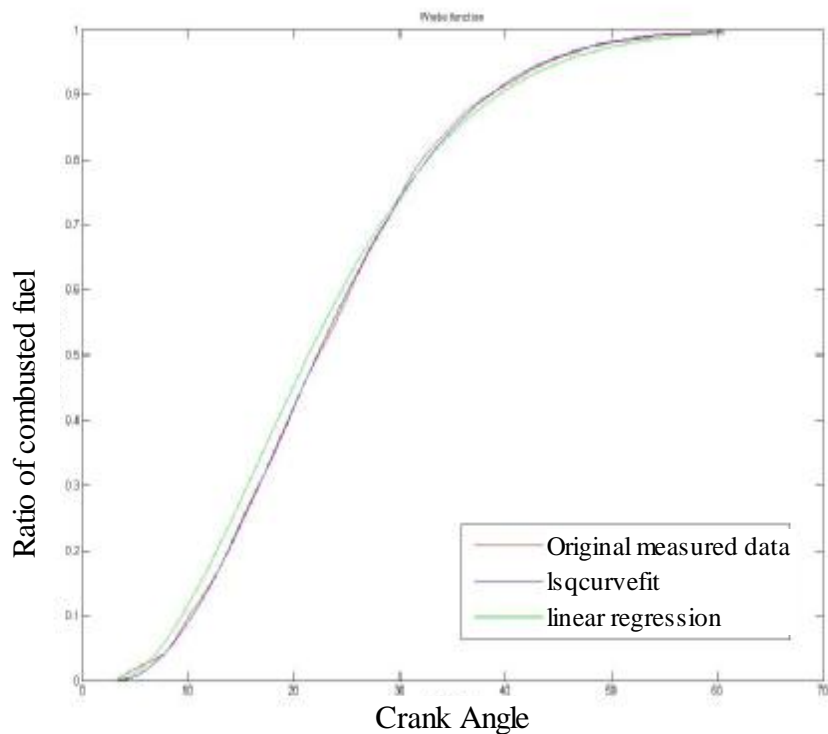


Figure 4.5

4.4.5 Engine performance

4.4.5.1 Mean effective pressure

The useful measure of the engine performance is work per engine cycle divided by the displacement volume. This measure is independent on the engine size and is often called mean

effective pressure (mep) [3]. The mean effective pressure is defined as $\frac{1}{V_D} \int p dV$. There are several types of meps summarized in Table 4.1.

MEP	Description
<i>IMEP</i> Indicated mean effective pressure	Mean effective pressure during expansion and compression phases.
<i>PMEP</i> Pumping mean effective pressure	Mean effective pressure during intake and exhaust phases.
<i>FMEP</i> Friction mean effective pressure	Work used by piston to overcome friction.
<i>BMEP</i> Brake mean effective pressure	The maximum BMEP of the most engines is well known and this parameter is used to compare various engines. The BMEP value is proportional to engines torque. The BMEP value is defined as $BMEP = IMEP - PMEP - FMEP$.

Table 4.1

4.4.5.2 Torque

As was said above, the engine's torque is proportional to the brake mean effective pressure. The

torque can be expressed as $M_M = \left(\frac{i_V \cdot i_P \cdot V_D}{\pi \cdot \tau} \right) \cdot BMEP$ where i_V corresponds to the number of cylinder, i_P to the number of pistons in one cylinder, V_D to displacement volume, τ to

the number of strokes per engine cycle [4].

4.4.5.3 Air-fuel ratio

The air/fuel ratio shows the chemical composition of the mixture in the cylinder. This the key parameter for the combustion behavior and consequently for the pollutants emission.

This parameter is given by $\lambda = \frac{m_{AIR}}{L_T \cdot m_{FUEL}}$ where m_{AIR} is the mass of air, m_{FUEL} is the mass of injected fuel and L_T is the stoichiometric mixing ratio. This ratio is different depends on the type of the fuel used. For common diesel fuel is equal to about 14.8 [4]. Sometimes only the

AFR ratio is used $AFR = \frac{m_{AIR}}{m_{FUEL}}$.

4.5 Model validation

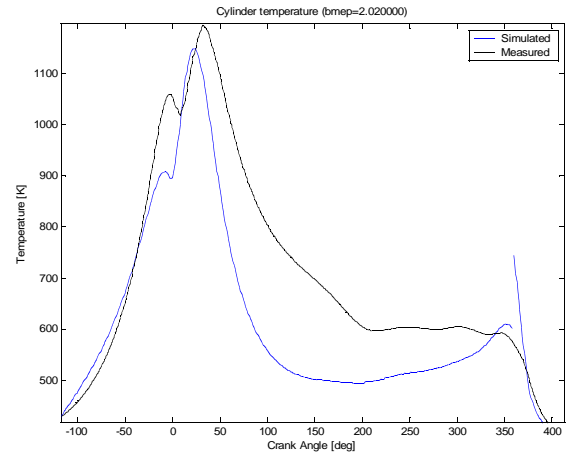
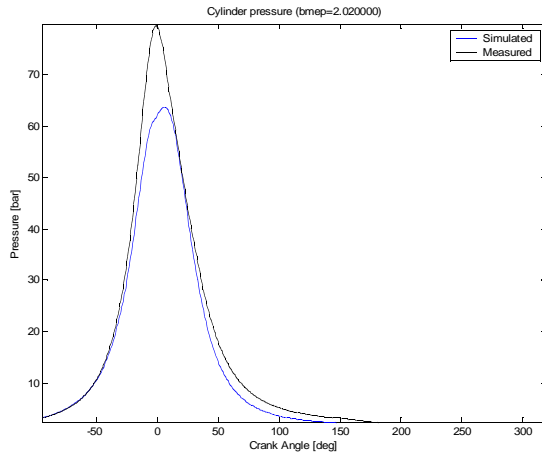
The model validation was done for four operating points. These points cover the range of loads from minimum up to maximum for the engine speed equal to 2400rpm. The different load conditions are represented by different BMEP values. The best results should be achieved for the maximum load.

The validation procedure followed following steps:

1. The injection timing and amount of fuel injected were set to the values corresponding to a given load conditions. These values were available in measured data.
2. The simulation was let to run until all the values were stabilized at some point.
3. The readings of AFR and output BMEP were taken.

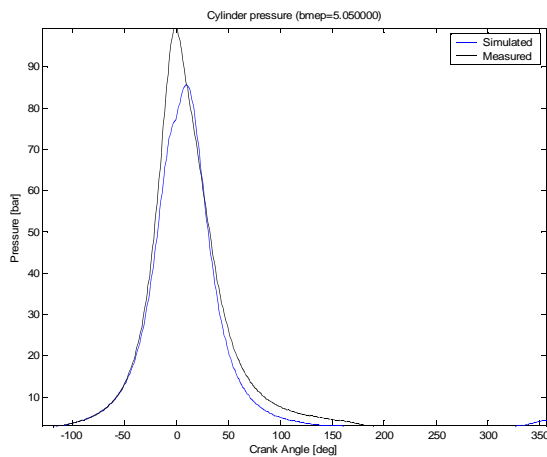
4.5.1 Results

BMEP = 2.02



	Measured value	Simulated value	Relative error [%]
BMEP	2.02	2.026	0.30%
Torque	53	53.18	0.30%
AFR	57.6	65.59	13.87%

BMEP = 5.05

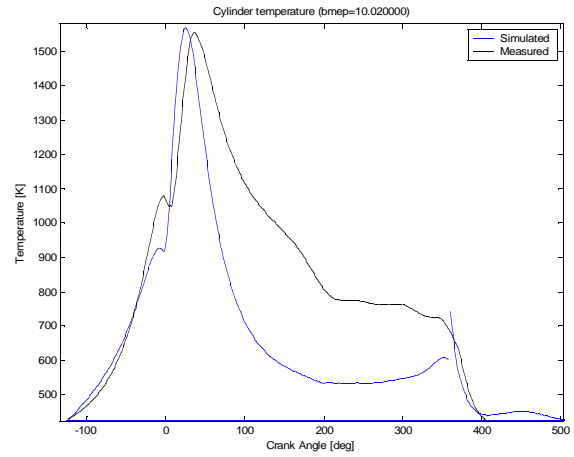
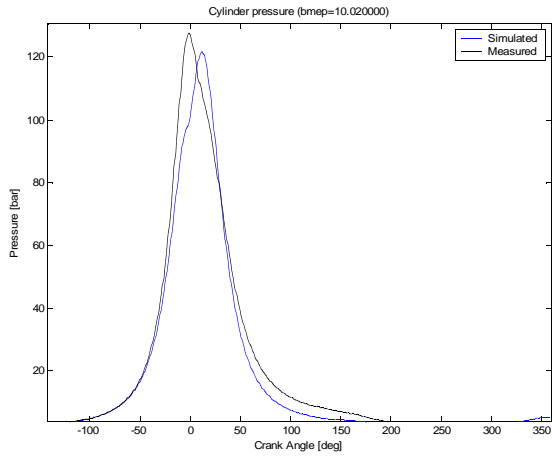


MEASURED DATA
NOT AVAILABLE

	Measured value	Simulated value	Relative error [%]
--	----------------	-----------------	--------------------

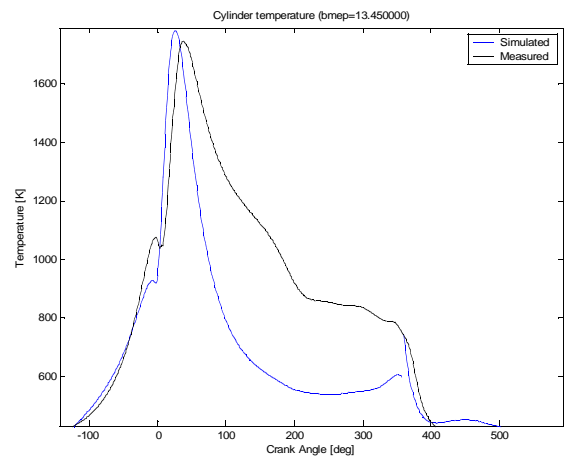
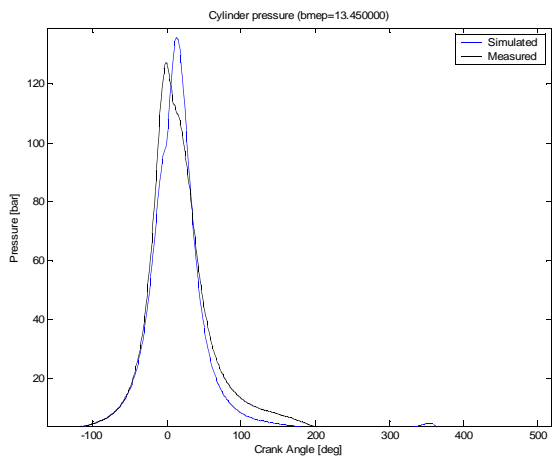
BMEP	5.05	5.178	2.53 %
Torque	132.0	135.9	2.53 %
AFR	41.96	48.28	15.06 %

BMEP = 10.02



	Measured value	Simulated value	Relative error [%]
BMEP	10.02	10.57	5.49%
Torque	262.93	277.4	5.49%
AFR	31.52	35.97	14.12%

BMEP = 13.45



	Measured value	Simulated value	Relative error [%]
BMEP	13.45	13.93	3.57%
Torque	353.53	365.8	3.57%
AFR	24.73	28.16	13.87%

4.5.2 Comparison with other models

The values and figures above show that this model has the maximum error for output torque smaller than 6% at any load conditions. Given amount of input data used and many simplifications made, the model stays in reasonable tolerance all the time. On the other hand the two-zone model described in [8] has the maximum error less than 1% for high loads but their chemical kinetics description covers more reaction than the chemical model used in this thesis. The AFR ratio stays all the time above expected value with a difference of approximately 14%. This means that there is 14% more air entering the cylinder than it should be. This is probably caused by rough approximation of the reference area of intake valve and the empirical setup of the discharge coefficient.

For mean-value models used in Ricardo Prague, the error between simulated and measured data is very low. These models use a huge amount of data maps and almost everything is at least calibrated by the measured data.

To conclude, this model gives a satisfactory description of diesel engine behavior (BMEP, AFR, P_{MAX} , T_{MAX}), but does not give the exact values of pressure and temperature conditions in the cylinder at any time.

5 Control

5.1 Introduction

It was shown in the previous chapters that diesel engine is a very complex system. The primary objective of any engine is to produce mechanical power. However the control problematics gets even more complicated as there are many other requirements that need to be taken in account for the design. The main requirements are [1]:

- The driver's demand for immediate torque response.
- Low fuel consumption
- Good drivability
- Engine must be kept in save operating regime (damage avoided, knocking prevented, catalyst not overheated)
- Emissions limits must be fully met

It is important to stress that unfortunately some of these objectives are contradictory to each other. Another limiting factor is that requirements for emissions and safety issues are set by the legislation and there cannot be any comprise.

5.2 Control objectives

In order to meet all the requirements stated above, it is essential to know which parameters of the engine to control. In this thesis there are two main control goals [15]:

1. Follow the set-point engine torque from the driving cycle. This set-point usually comes from the driver or from cruise control. This goal is key for engine performance. As was said before, the engine torque is proportional to BMEP value [4]. This allows to use the BMEP set-point instead.
2. λ is not allowed to go below certain limit. Otherwise there will be too many pollutants produced. This is key for engine emissions control. The emissions are also controlled through EGR rate.

The control problem is shown below on the Figure 5.1:

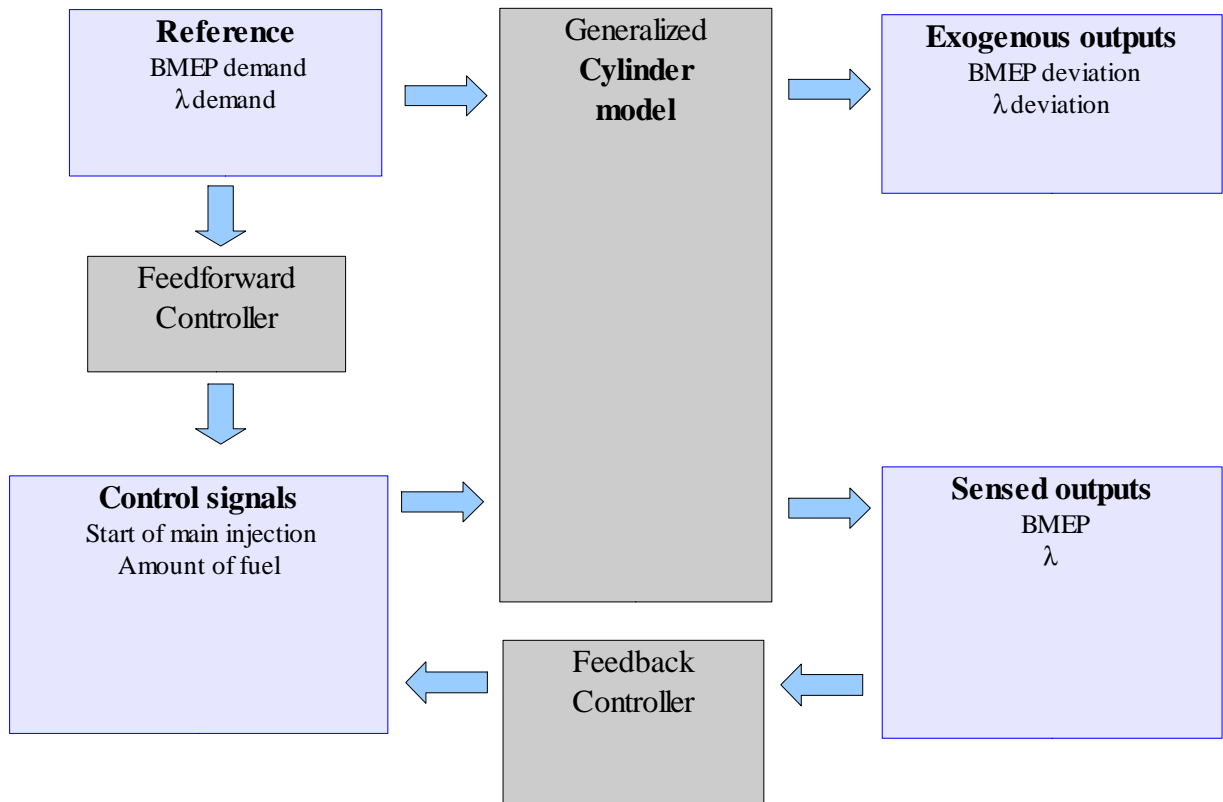


Figure 5.1

5.3 Mathematical description of cylinder behavior

The aim of this chapter is to express the sensed outputs as the functions of control signals.

5.3.1 BMEP

The engine BMEP is a nonlinear function of many variables (fuel mass in cylinder, air/fuel ratio, engine speed, injection timing, EGR rate etc.) [1]. At the same time there are several simplifications allow as to find a reasonable, though non-linear, function for engines BMEP.

Two assumptions have to be taken into account to figure out this function:

1. The output BMEP is linearly proportional to the amount of fuel injected and therefore the searched function has to have following form: $f(m_F, \theta_0) = (a_1 \cdot m_F + b_1) \cdot g(\theta_0)$

2. There is an optimal (maximum break torque, MBT) injection angle. This MBT angle is strongly dependent on the engine operating point. For other than MBT the torque drops almost parabolically [1]. Therefore the function $g(\theta_0)$ has the form

$$g(\theta_0) = a_2 \cdot \theta_0^2 + b_2 \cdot \theta_0 + c_2$$

The Figures 5.2 and 5.3 show the simulation results for various injections setups. These injections setups cover most of the operating conditions for a given engine speed 2400rpm. Note that the Figure 5.3 is in relative scale. The validity of assumptions is clearly shown on these figures.

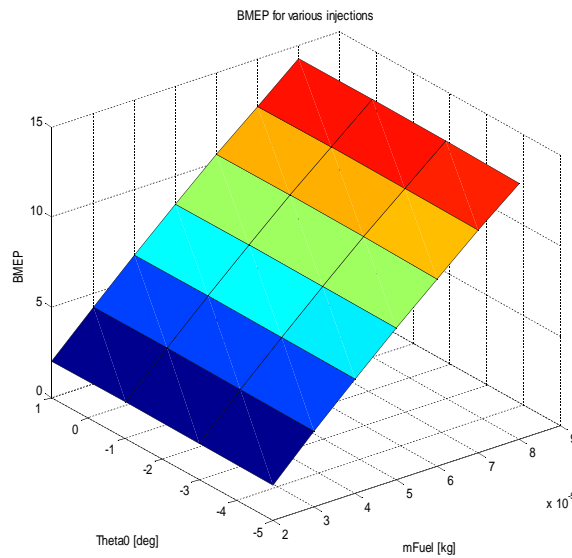


Figure 5.2:

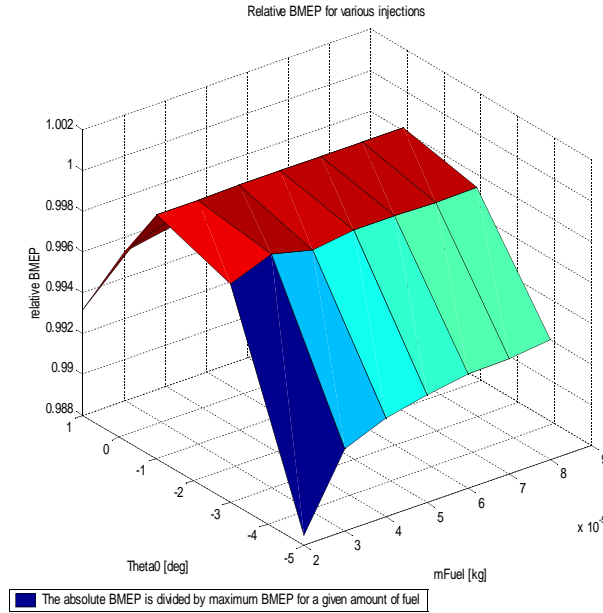


Figure 5.3:

Consequently, the mathematical approximation of output BMEP is given by:

$$BMEP = (a_1 m_F + b_1) \cdot (a_2 \theta_0^2 + b_2 \theta_0 + c_2)$$

Equation 5.1

the following values were found by least square method:

$$BMEP = (1.9857 \cdot 10^5 m_F - 1.7250) \cdot (-0.0007 \theta_0^2 - 0.0020 \theta_0 + 0.9988)$$

The average relative mean error of this approximation is about 3.3% and the maximum error is equal to 12.7%. The maximum power for a given amount fuel is reached with the injection timing of -1.4286 degrees.

5.3.2 1

The λ parameter plays a crucial role in the formation of pollutants. This parameter should not drop under 1.4, otherwise the amount of NO_x, PM and CO particles in the exhaust gases increases rapidly. The λ parameter description was given in the section 4.4.5.3. The objective of our control system is to keep this parameter higher than 1.4 at all operating conditions.

5.4 Torque demand

The demanded torque is defined by the driver through accelerator pedal. In the modern engines, the signal from this pedal is interpreted by ECU as a desired engine torque. The controller structure is shown on the Figure 5.4. The control system contains both: the feedforward controller and the feedback controller. Note that the λ parameter stays all the time above 1.7. Therefore no λ control needs to be applied.

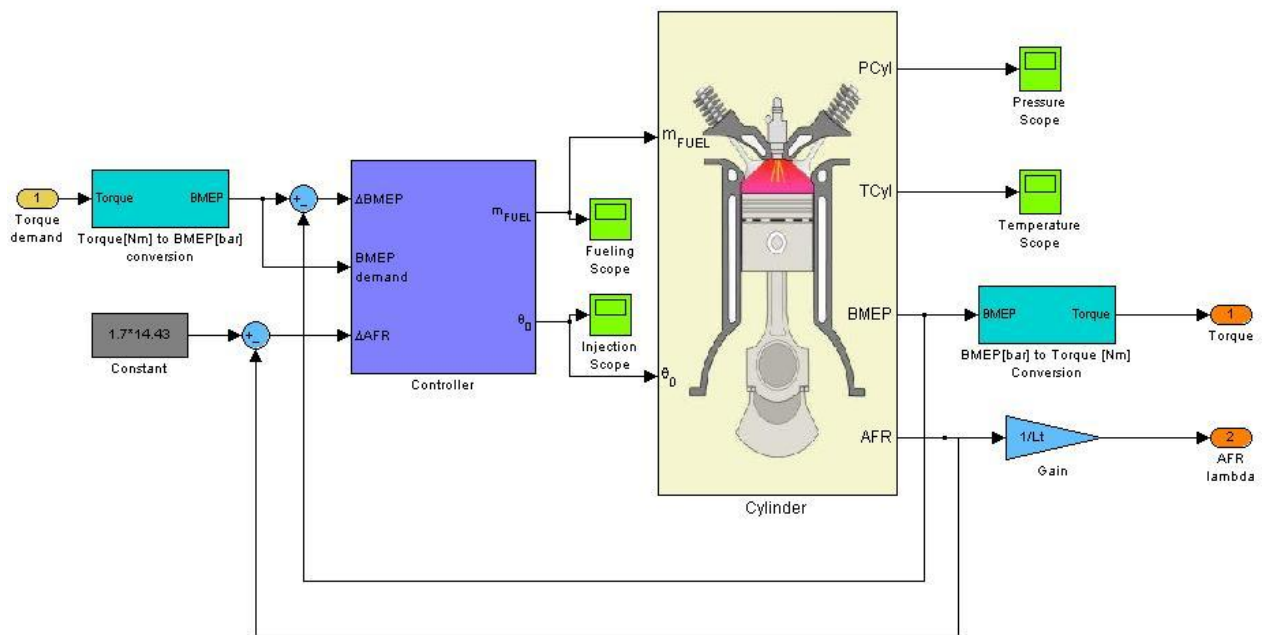


Figure 5.4

5.4.1 Feedforward controller

The Equation 5.1 show the engine performance behavior and one of the objectives is to minimize the fuel consumption. Therefore the angle of maximum output power should be chosen. This angle is equal to -1.4286 degrees before TDC as is shown in section 5.3.1. Therefore the The Equations 5.1 becomes:

$$BMEP = (a \cdot m_F + b) = 1.9861 \cdot 10^5 m_F + 1.2754$$

Using this equation, the amount of injected fuel as a function of desired BMEP is easily derived:

$$m_F = \frac{1}{a} \cdot BMEP_{DEMANDED} - b = \frac{1}{1.9861 \cdot 10^5} - 1.7254$$

The feedforward controller is used to bring the engine output BMEP close to a desired value. The mean error of this control is about 13% and the maximum difference is between desired and actual torque does not exceed 30 Nm. The result of feedforward control for various BMEP demands is shown on the Figure 5.5.

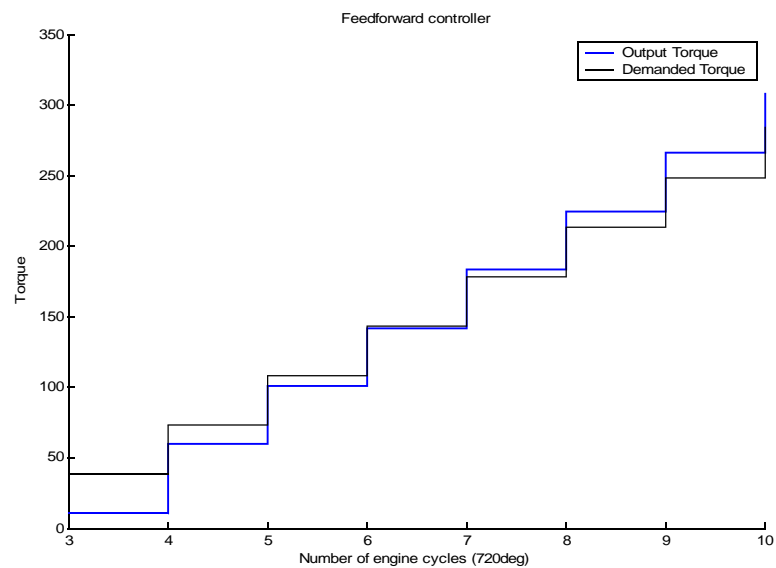


Figure 5.5

As shown on the figure above, the objective for torque demand cannot be met only by feedforward control. This is caused by simplified mathematical description of engine performance. By introducing the feedback controller, the good response for driver's torque should be assured. This feedback control should be robust to the simplifications in mathematical description and also to many approximations in the non-linear Simulink model.

Nevertheless, in the real engines the feedforward control systems are used to control engines performance. This is done through detailed set of injection maps for all operating points of the

engine.

5.4.2 Feedback controller

There is an important feature common for all feedback controllers. All the feedback controllers let the feedforward control to act before they start to regulate. This means that if there is a change in torque demand at any time, the feedback control is applied with a delay of one engine cycle. Only feedforward control is applied during the first cycle. This improves the overall control behavior by limiting the overshoots.

All the controllers were designed in the continuous time domain and then discretized by zero-order hold method with a sample period equal to 1. The implementation in the Simulink was done using the *Enabled Subsystem* block because the control was performed at the beginning of the engine cycle (but the simulation step is equal to one crank-angle degree). This means that the controller was active just once per the engine cycle because it makes no sense to change the amount of injected fuel and injection timing during the cycle.

Note that feedback controllers outputs are not the control inputs to the cylinder model, but only the changes of these control inputs.

5.4.2.1 PI Controller

The PI controller for this control problem is given by:

$$R(s) = \frac{\delta m_F(s)}{\Delta BMEP(s)} = \frac{P \cdot s + I}{s} = \frac{I \cdot \left(\frac{P}{I} + 1 \right)}{s} .$$

The coefficients P and I are chosen with respecting the fueling constraints. The change in the fueling is limited to keep the engine in safe operating regime. The limit for the change of amount of injected fuel is around $6 \cdot 10^{-6}$ kg. Thanks to the feedforward control, we can assume that the maximum torque deviation does not exceed 30 and therefore the $\Delta BMEP < 1.15$. This

brings following constraint: $I < \frac{5 \cdot 10^{-6}}{1.5} \approx 4.35 \cdot 10^{-6}$. The parameter P is deduced by pushing the pole of the close loop to the desired value.

The transfer function of the system can be estimated as a simple gain:

$$G(s) = \frac{BMEP(s)}{m_F(s)} = k \approx 1.9861 \cdot 10^5$$

The closed loop transfer function with PI controller:

$$T(s) = \frac{K(s) \cdot G(s)}{1 + K(s) \cdot G(s)} = \frac{Ps + I}{\left(\frac{1}{k} + P\right) \cdot s + I}$$

If the pole of this transfer function is placed to -0.5 and $I = 4.35 \cdot 10^{-6}$ then the controller coefficients are $P = 3.6650 \cdot 10^{-6}$; $I = 4.35 \cdot 10^{-6}$. Other PI controllers were tried empirically. However the PI controller achieved the best results. PI controllers simulation results are shown in the Figure 5.6.

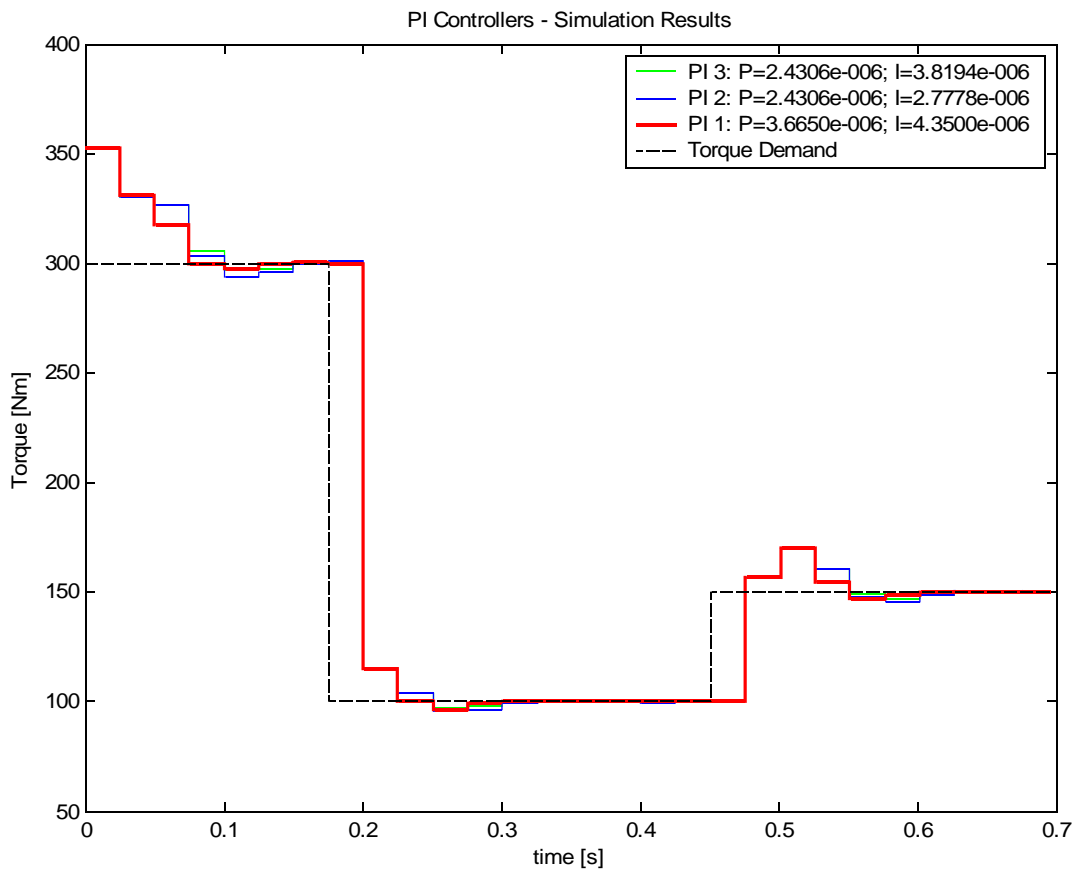


Figure 5.6

This controller works well and no further search for better controller is needed. However there are some features of this controller that could be still improved. The Figure 5.7 shows the sensitivity functions of this controller.

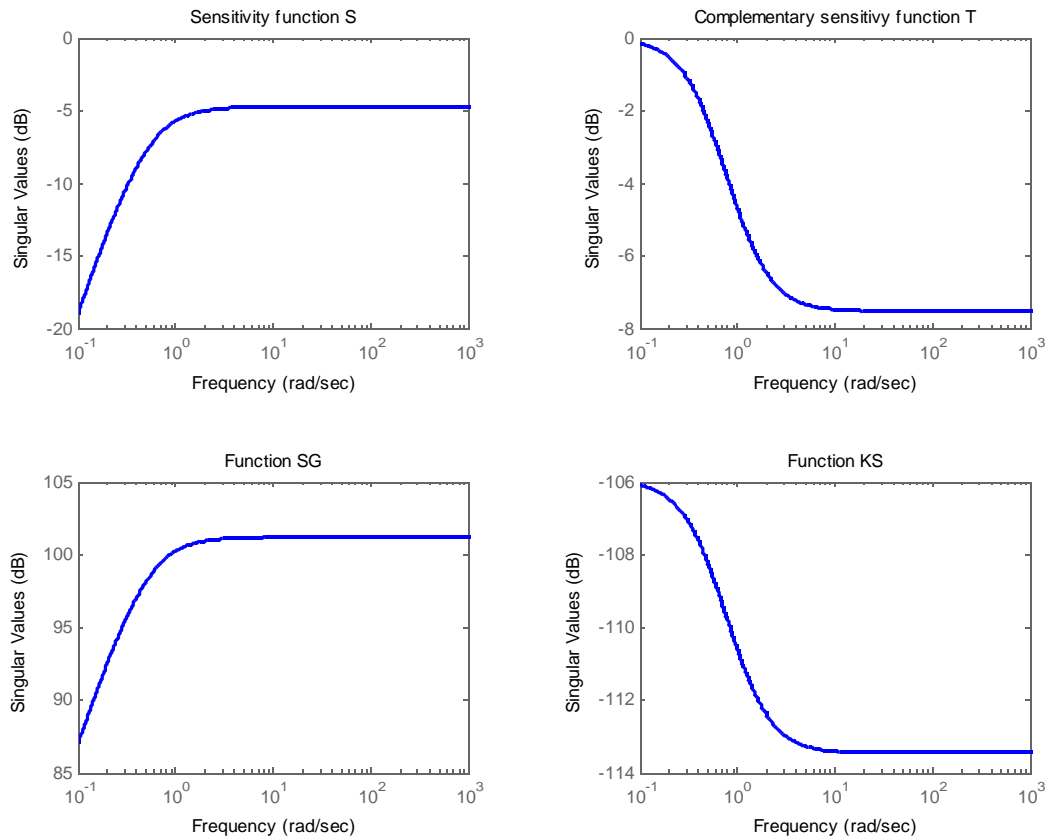


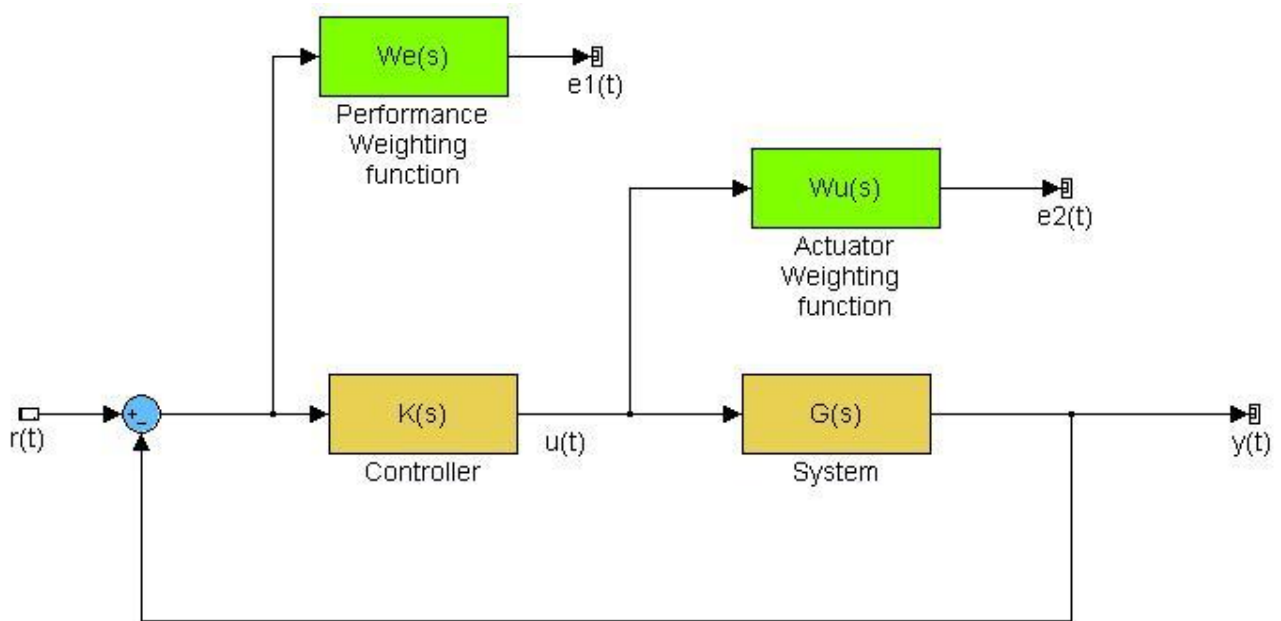
Figure 5.7:

Generally, it is required to reject the disturbance effects in low frequencies and to minimize the noise effects at high frequencies. This makes the S and SG sensitivity functions to be small in low frequencies in order to reduce the load disturbance effects on the controlled output. Also the T and KS sensitivity functions has to be small at high frequencies to reduce the effects of measurements noises on the controlled output and control input [13]. As shown on the Figure 5.7, the sensitivity function S, T and KS of this PI controller respects the requirements on these functions. On the other hand the sensitivity function SG is not small in low frequencies. This

might be improved by controller designed with loop-shaping H_∞ method.

5.4.2.2 H_∞ Controller

This method was introduced by Graham and Doyle [16]. There is one main advantage of H_∞ method. The plant modelling errors as well as the disturbance and noise effects can be specified in the frequency domain [13]. The performance specifications usually concerns at least two sensitivity functions. The requirements for these sensitivity functions are shown above in the PI controller section. The sensitivity functions S and KS need to be taken into account in order to satisfy the tracking objective as well as the actuator constraints. These performance specifications can be given as some weights on the controlled as follows [13]:



where the weights W_e and W_u are given in this particular case by:

$$W_e = \frac{\frac{s}{M_s} + \omega_B}{s + \omega_B \epsilon} = \frac{\frac{s}{2} + 1}{s + 1 \cdot 0.01} \quad \text{and} \quad W_u = M_u = \frac{4.35 \cdot 10^{-6}}{13.45}$$

The solution of H_∞ control problem can be found using the Riccati approach or the LMI approach. The Matlab command *hinfsyn* uses the Riccati approach. The following solution for H_∞ control problem was found:

$$R_{H_\infty}(s) = \frac{5.194 \cdot 10^{-9} s + 1.967 \cdot 10^{-6}}{s + 0.01}$$

The sensitivity function Figure 5.8 shows that no improvement was achieved with H_∞ controller. This controller is even more sensitive to input disturbances than the PI controller.

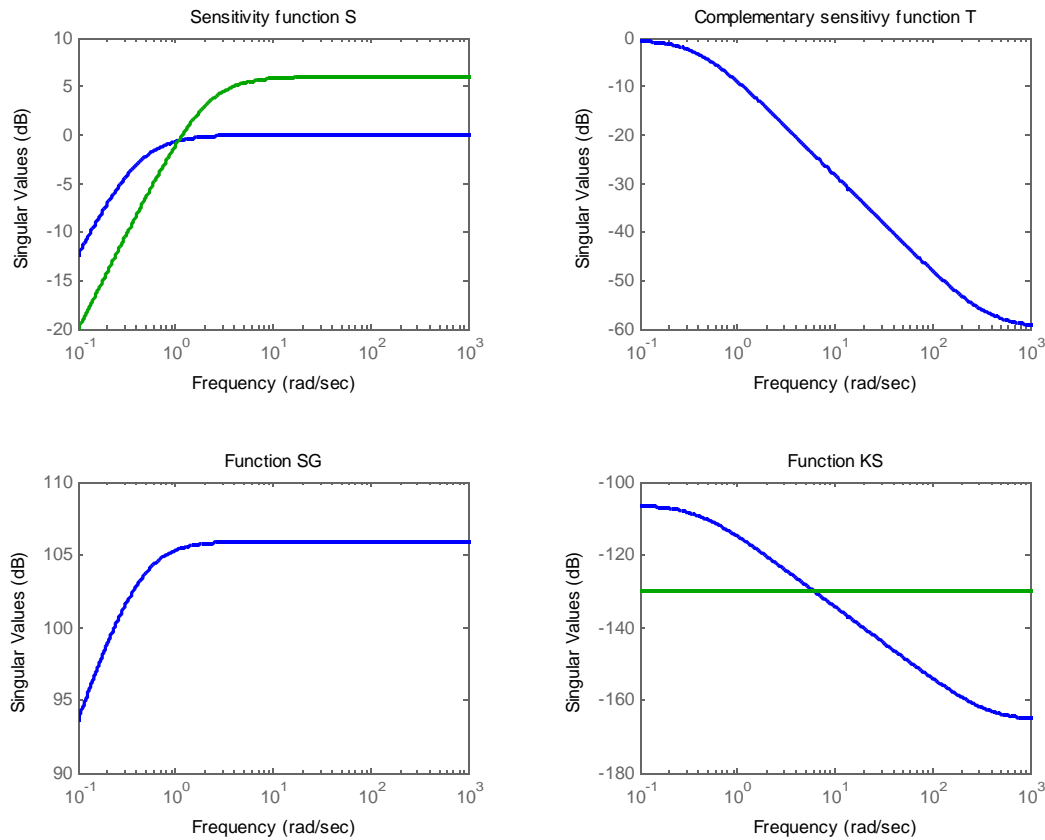


Figure 5.8

The simulation results for this controller are shown in the next section of this chapter in the Figure 5.9.

The H_∞ method found a reasonable controller that could be without any doubt used for controlling the diesel engine process. However the H_∞ controller is more suitable for the applications with more controlled outputs. The H_∞ approach is often used for control of λ parameter when exhaust gas recirculation and variable geometry turbocharger control mechanisms are applied. The results obtained with this controller for torque control were not

worth the effort.

5.4.3 Simulation results

The Figure 5.9 shows the simulation results for all the essential controllers. The controller currently by Ricardo Prague for simulation purposes is a PI controller. All three controllers perform well in the control of non-linear cylinder model. As shown below the best performance was achieved with the PI controller.

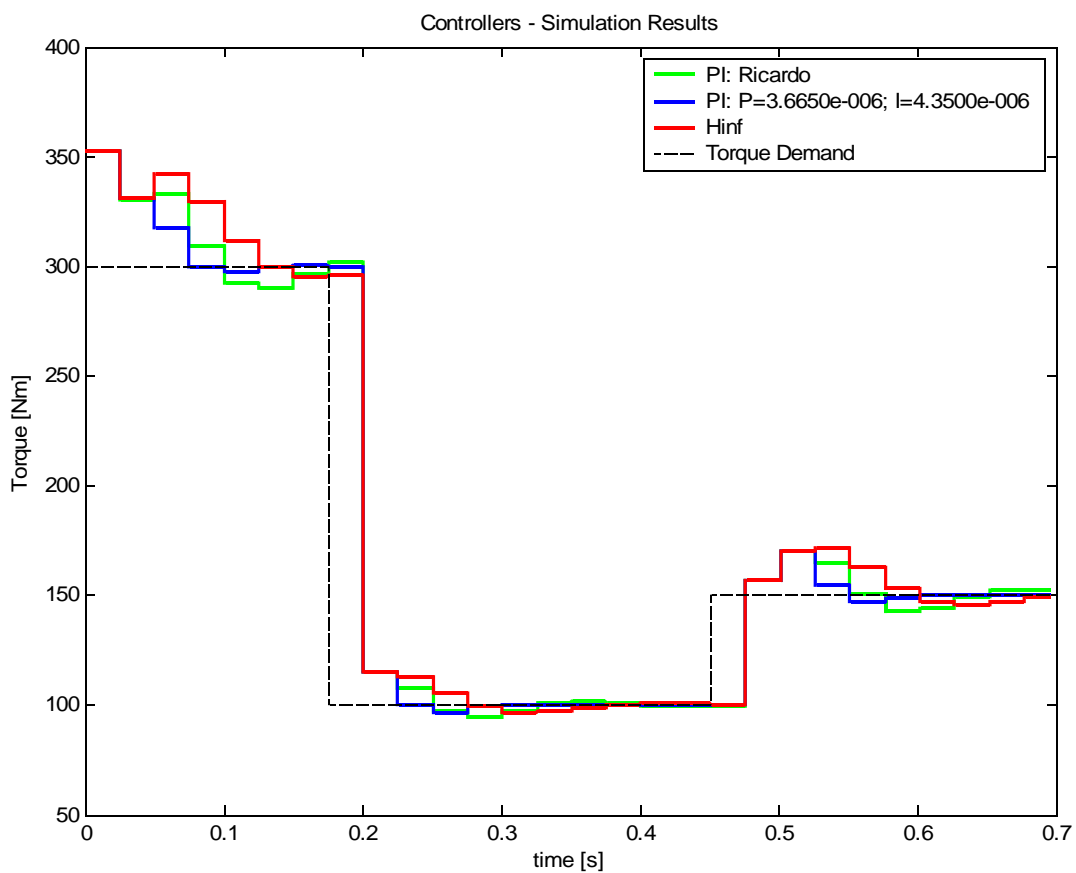


Figure 5.9

6 Conclusion

This thesis has achieved the following:

1. The key processes of the diesel engine cylinder were explained and studied in great detail. This includes the description of cylinders thermodynamics, geometry and mass flows, combustion process and the engine performance.
2. The thermodynamical cylinder model has been derived and implemented in *Matlab-Simulink*. This model allows to simulate the cylinder pressure and temperature development in crank-angle domain. The observation and control of engine performance parameters is also possible. The validation was done through the output torque (maximum error is less than 4%) and the air/fuel ratio (maximum error is less than 15%). Although there were many simplifications in the modelling of the diesel engine processes the model gives satisfactory results.
3. The torque control problem was defined and several controllers were suggested and designed. The performance of PI and H_∞ controllers was compared with non-linear Simulink model. Unfortunately, there was no opportunity to try it on a real engine. However, the best PI controller has given a better results than a controller currently used by Ricardo Prague for simulation purposes.

7 References

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8 Appendix

8.1 A. Engine parameters

<i>Parameter</i>	<i>Value</i>
Displacement volume	3.3l
Bore	100mm
Stroke	100mm
Compression ration	17.5
Number of cylinders	4
Intake valve diameter	40.4mm
Exhaust valve diameter	36.8mm

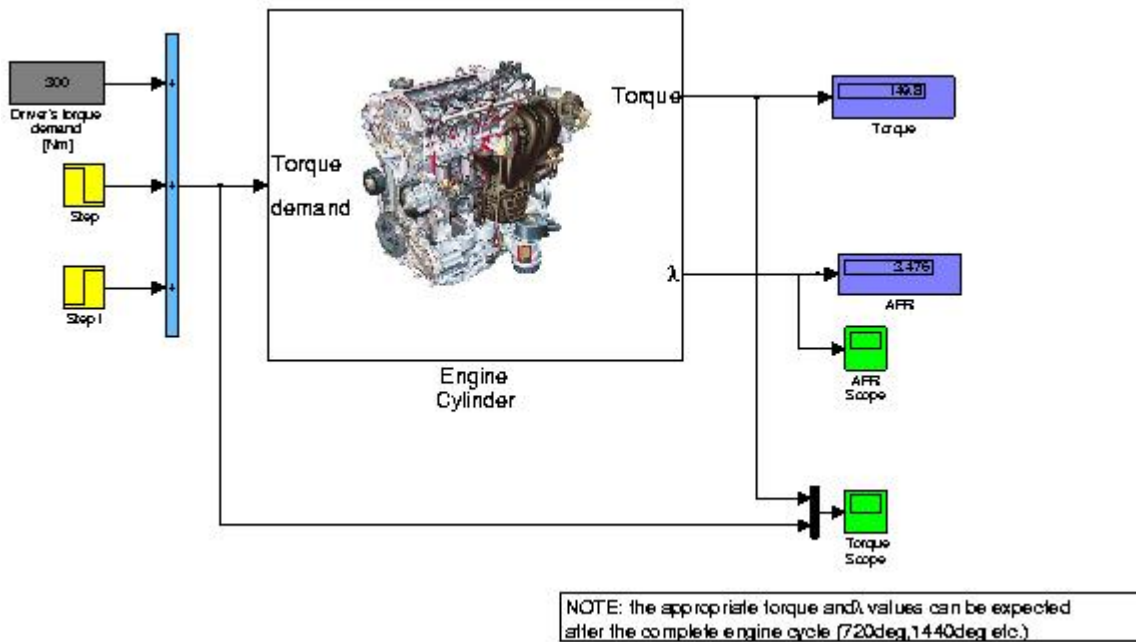
8.2 B. Cylinder model documentation

The theoretical background for the model can be found in the main part of this thesis. Please refer to previous chapters for further information.

8.2.1 Cylinder model

The initialization of the model is done using following *cylinder_model_init.m* function. The source code is given in the code part of this documentation.

Simulink schematics

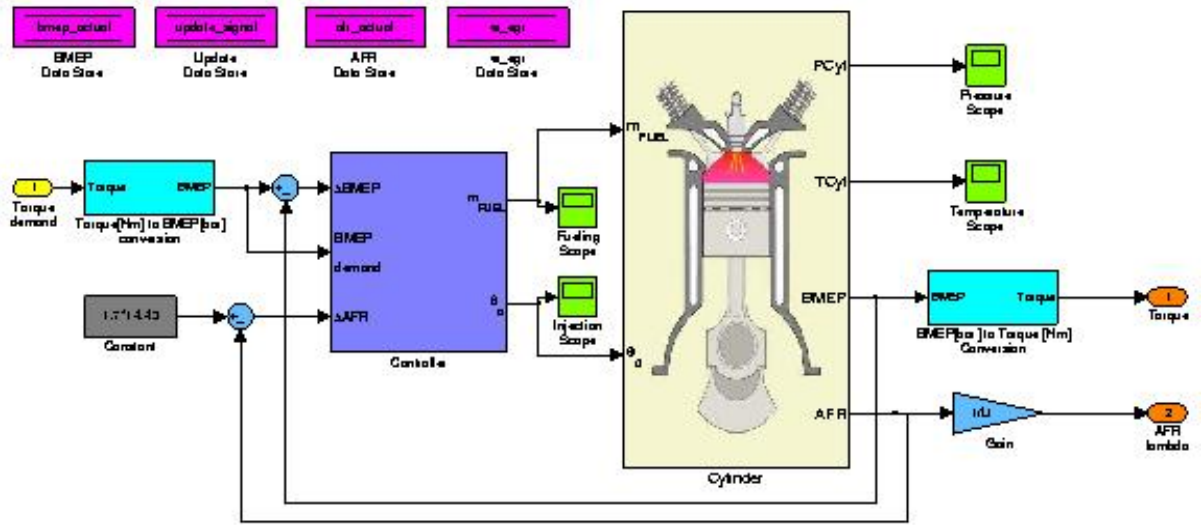


8.2.2 Engine Cylinder

<i>Input</i>	<i>Units</i>	<i>Range</i>	<i>Comments</i>
Torque demand	[Nm]	{40, 370}	The driver's demand for engine torque

<i>Output</i>	<i>Units</i>	<i>Range</i>	<i>Comments</i>
Torque	[Nm]	{40, 370}	The actual output torque

Simulink schematics



8.2.3 Controller

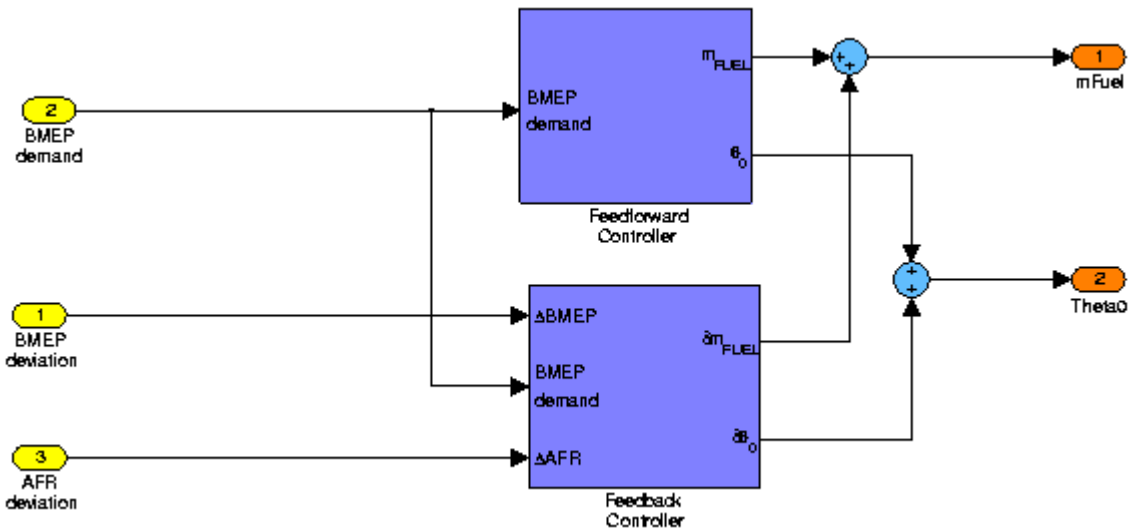
Input	Units	Range	Comments
Δ BMEP	[-]	{0, 40}	BMEP deviation
BMEP demand	[Nm]	{1, 13.45}	The driver's demand for engine BMEP (recalculated from torque)
Δ AFR	[-]		The AFR deviation

Output	Units	Range	Comments
m_{FUEL}	[kg]	{0.5E-5, 8E-5}	Amount of injected in the cylinder
θ_0	[deg]	{-30, 10}	The injection timing

Notes

The Δ AFR is not used by the controller. The future controllers could use it to control the pollutant formation process.

Simulink schematics

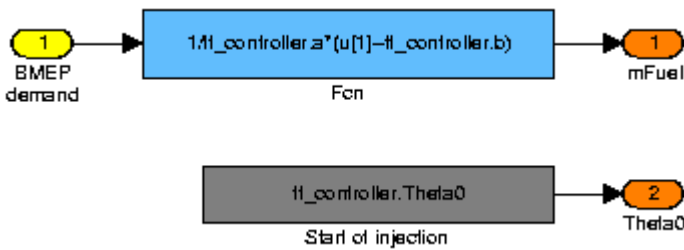


8.2.4 Feedforward controller

Input	Units	Range	Comments
BMEP demand	[Nm]	{1, 13.45}	The driver's demand for engine BMEP (recalculated from torque)

Output	Units	Range	Comments
m _{FUEL}	[kg]	{0.5E-5, 8E-5}	Amount of injected in the cylinder
θ ₀	[deg]	{-30, 10}	The injection timing

Simulink schematics



8.2.5 Feedback Controller

Input	Units	Range	Comments
ΔBMEP	[-]	{0, 40}	BMEP deviation

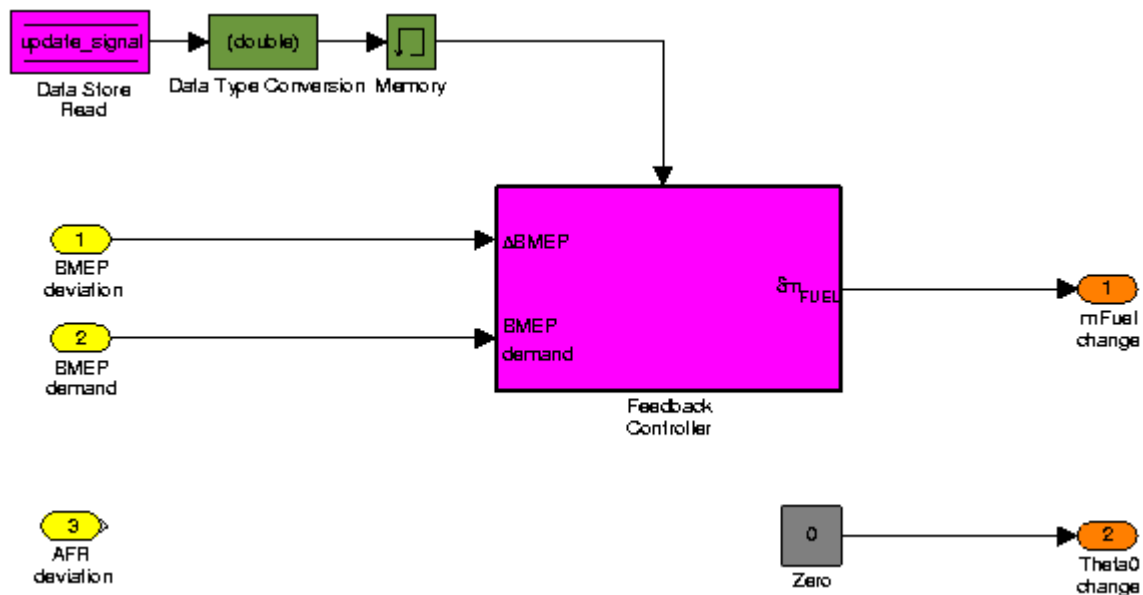
BMEP demand	[Nm]	{1, 13.45}	The driver's demand for engine BMEP (recalculated from torque)
Δ AFR	[-]		The AFR deviation

Output	Units	Range	Comments
δm_{FUEL}	[kg]	{0, 1E-5}	The change in the amount of injected in the cylinder
$\delta\theta_0$	[deg]	{-5, 5}	The change in the injection timing

Notes

The feedback controller is activated just at the beginning of the engine cycle.

Simulink schematics



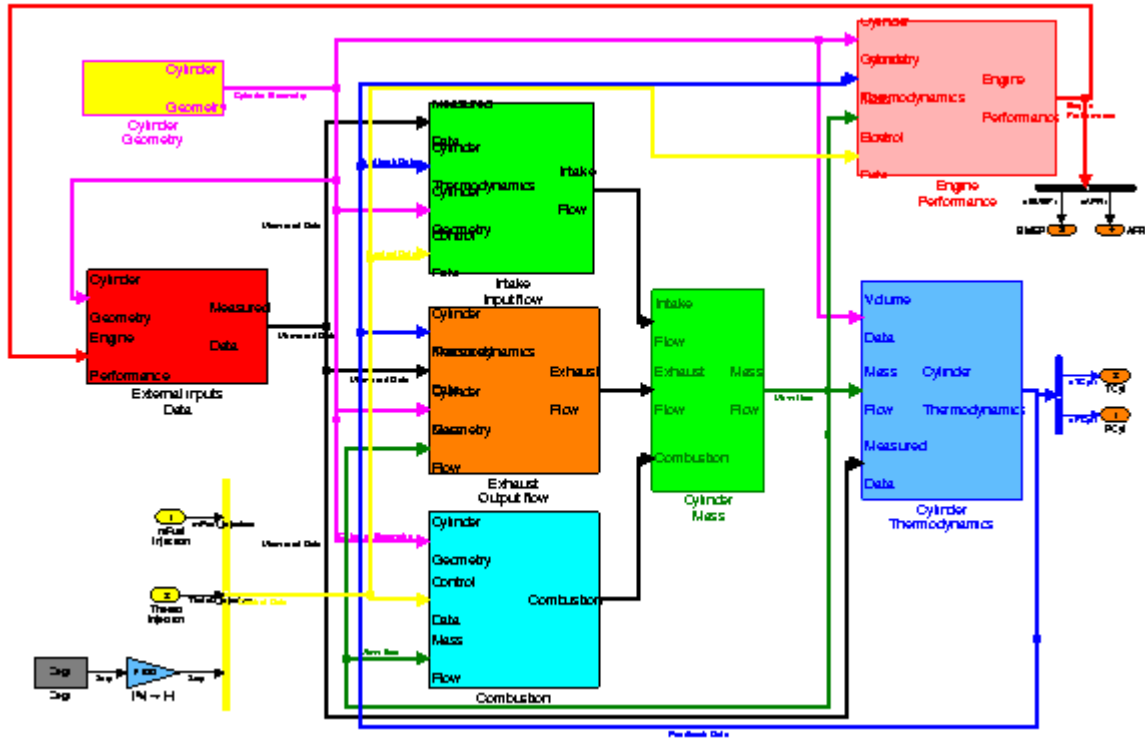
8.2.6 Cylinder

Input	Units	Range	Comments
m_{FUEL}	[kg]	{0.5E-5, 8E-5}	Amount of injected in the cylinder
θ_0	[deg]	{-30, 10}	The injection timing

Output	Units	Range	Comments
P_{CYL}	[Pa]	{1E5, 150E5}	The cylinder pressure
T_{CYL}	[K]	{300, 3000}	The cylinder temperature
AFR	[-]	{20,80}	Air/fuel ratio

BMEP	[bar]	{1, 13.45}	Output BMEP
------	-------	------------	-------------

Simulink schematics



8.2.7 Cylinder geometry

<i>Input</i>	<i>Units</i>	<i>Range</i>	<i>Comments</i>

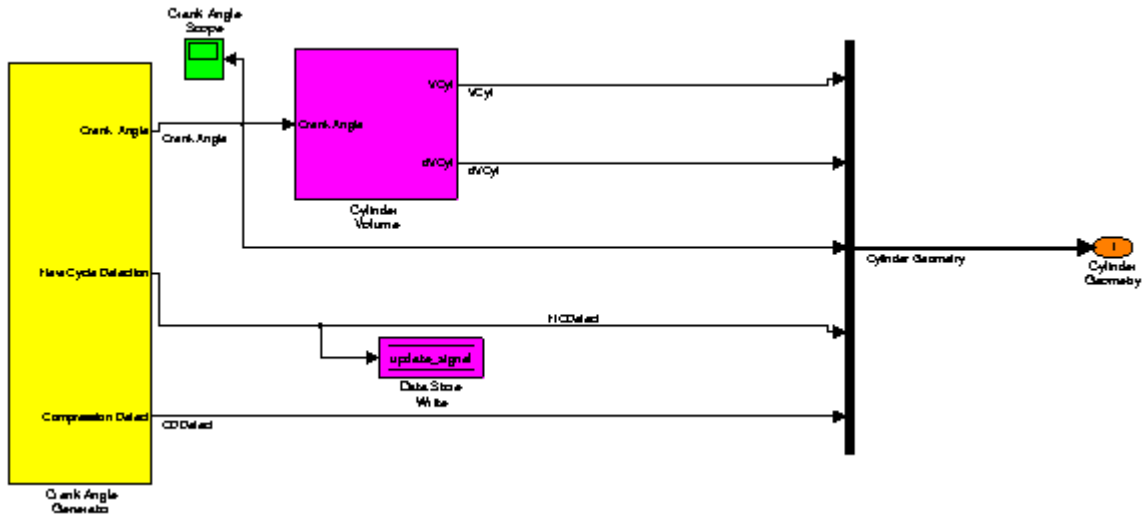
<i>Output</i>	<i>Units</i>	<i>Range</i>	<i>Comments</i>
Cylinder Geometry	[bus]	{1E5, 150E5}	This bus contains following data: cylinder volume, the change of cylinder volume, crank angle, new cycle detection, the intake valve closed detection

Notes

The TDC has the value of 0° and the range is designed in such a way that the real measured data can be easily used in the model. The simulation always starts with Intake phase. The crank

angle is used as an independent variable to synchronize all the parts of the model. The change of one crank angle degree also correspond to the time change of 1s in the time domain.

Simulink schematics



8.2.8 External inputs

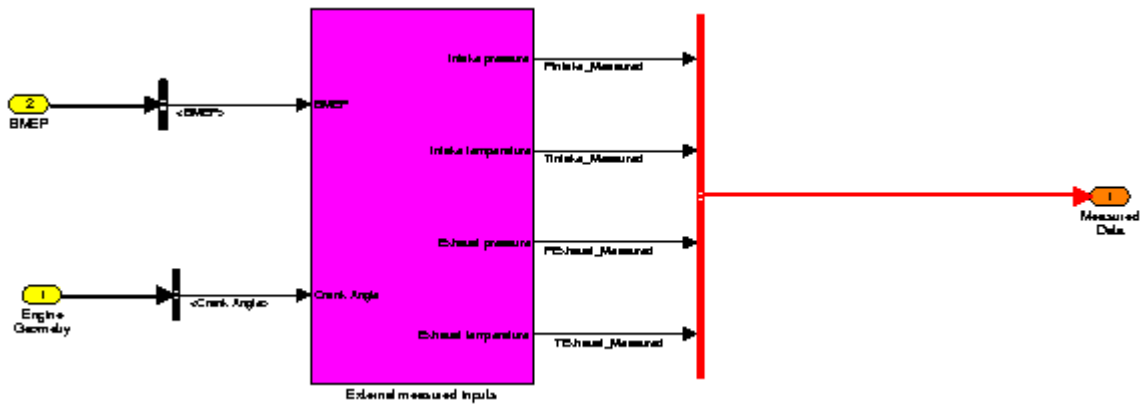
Input	Units	Range	Comments
Cylinder geometry	[bus]	-	
Engine performance	[bus]	-	

Output	Units	Range	Comments
Measured data	[bus]	-	

Notes

This block uses the function *interpolateData.m* to find the appropriate outputs. The source code is given in the code part of this documentation.

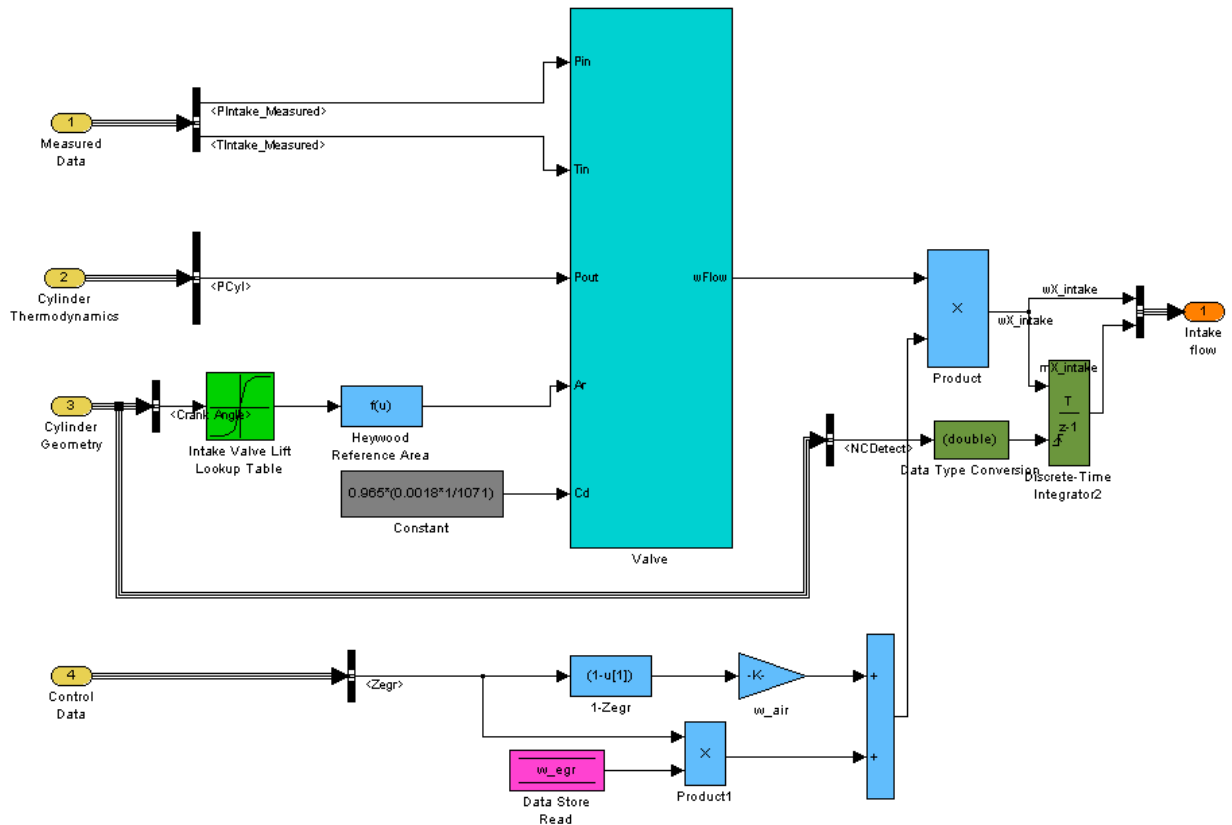
Simulink schematics



8.2.9 Intake flow

<i>Input</i>	<i>Units</i>	<i>Range</i>	<i>Comments</i>
Cylinder geometry	[bus]	-	
Measured data	[bus]	-	
Cylinder thermodynamics	[bus]	-	
Control data	[bus]	-	

<i>Output</i>	<i>Units</i>	<i>Range</i>	<i>Comments</i>
Intake flow	[bus]	-	This bus contains the air mass flow in the cylinder and the total mass taken in the cylinder during given engine cycle.



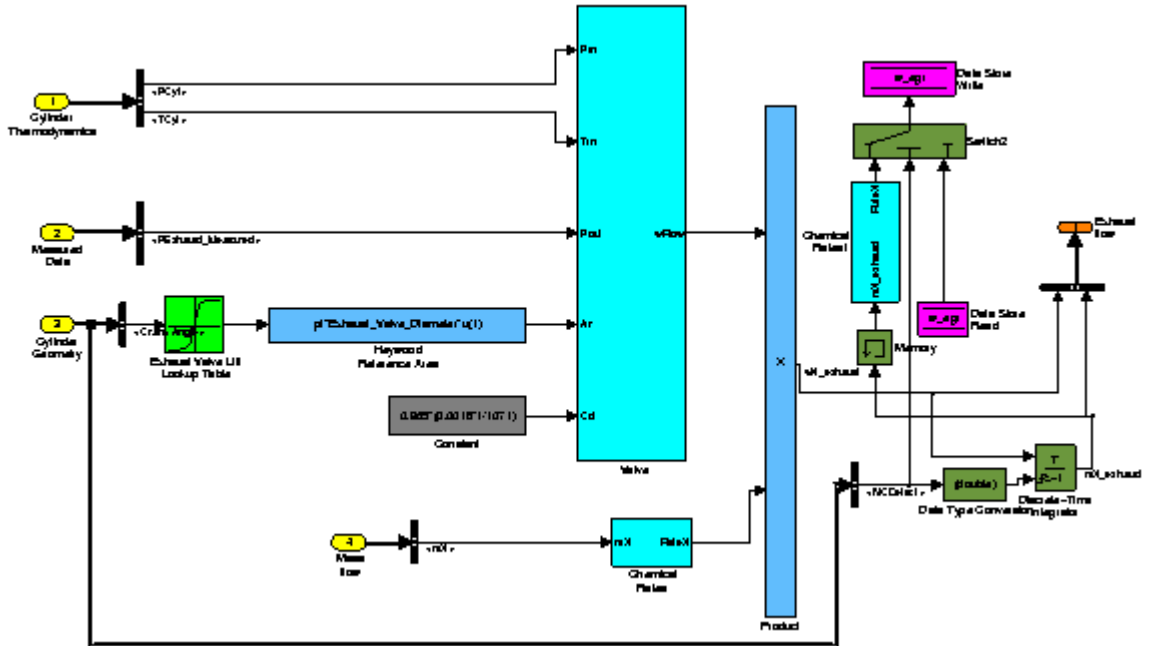
Simulink schematics

8.2.10 Exhaust flow

<i>Input</i>	<i>Units</i>	<i>Range</i>	<i>Comments</i>
Cylinder geometry	[bus]	-	
Measured data	[bus]	-	
Cylinder thermodynamics	[bus]	-	
Mass flow	[bus]	-	

<i>Output</i>	<i>Units</i>	<i>Range</i>	<i>Comments</i>
Exhaust flow	[bus]	-	This bus contains the exhaust gases flow in the exhaust manifold and the total mass of exhaust gases produced during given engine cycle.

Simulink schematics

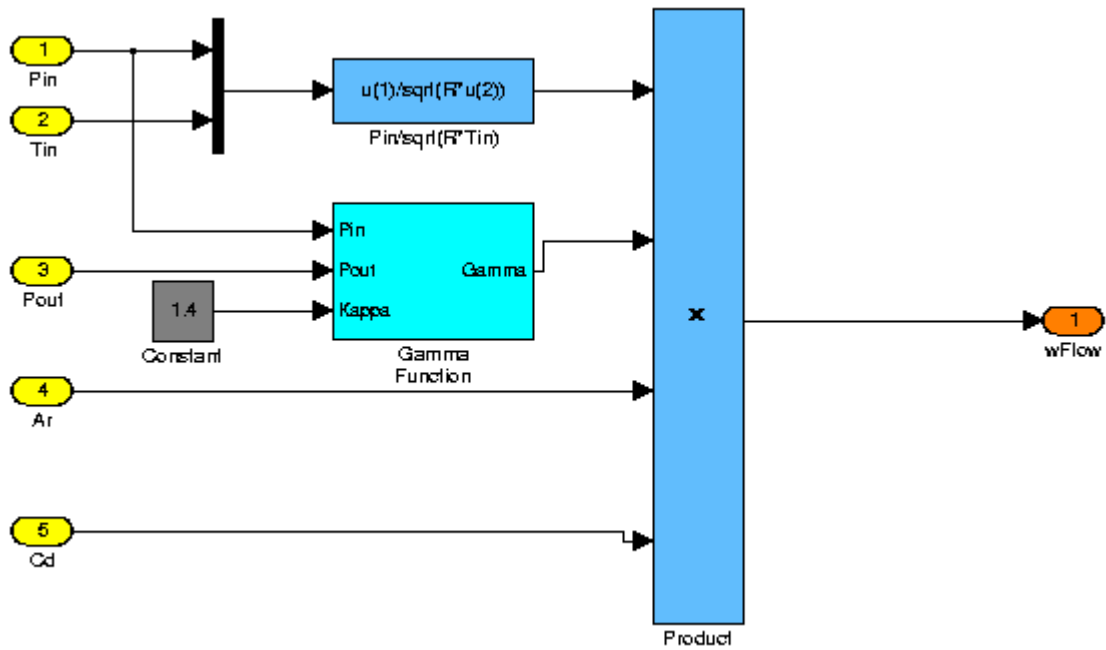


8.2.11 Valve

Input	Units	Range	Comments
Pin	[Pa]	-	
Tin	[K]	-	
Pout	[Pa]	-	
Ar	[m ²]	-	The reference area of the valve
Cd	[-]	-	The discharge coefficient

Output	Units	Range	Comments
wFlow	[kg]	-	Flow through valve

Simulink schematics

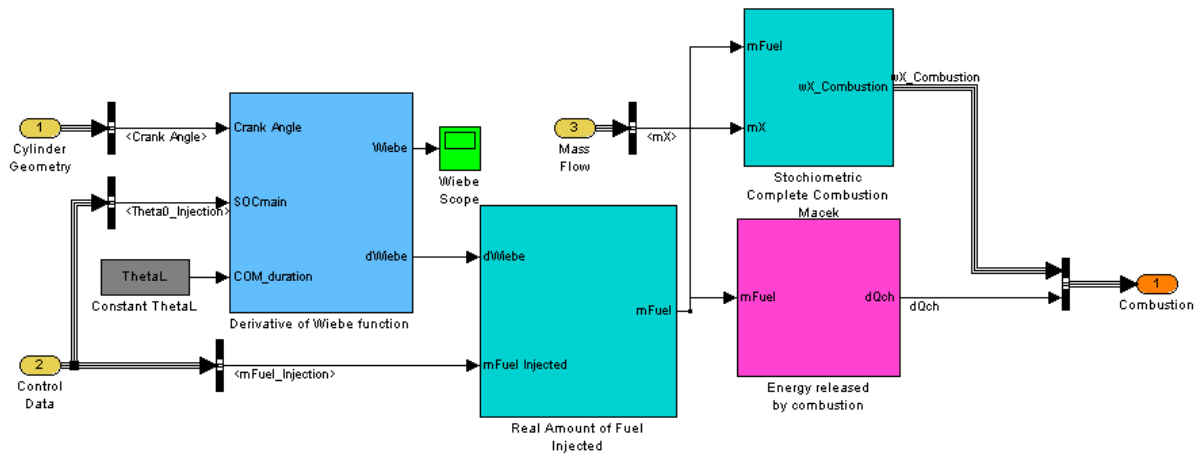


8.2.12 Combustion

<i>Input</i>	<i>Units</i>	<i>Range</i>	<i>Comments</i>
Cylinder geometry	[bus]	-	
Control data	[bus]	-	
Mass flow	[bus]	-	

<i>Output</i>	<i>Units</i>	<i>Range</i>	<i>Comments</i>
Combustion	[bus]	-	This bus contains the changes in the chemical compounds in the cylinder due to the combustion and the heat released calculated by standard heat release model.

Simulink schematics

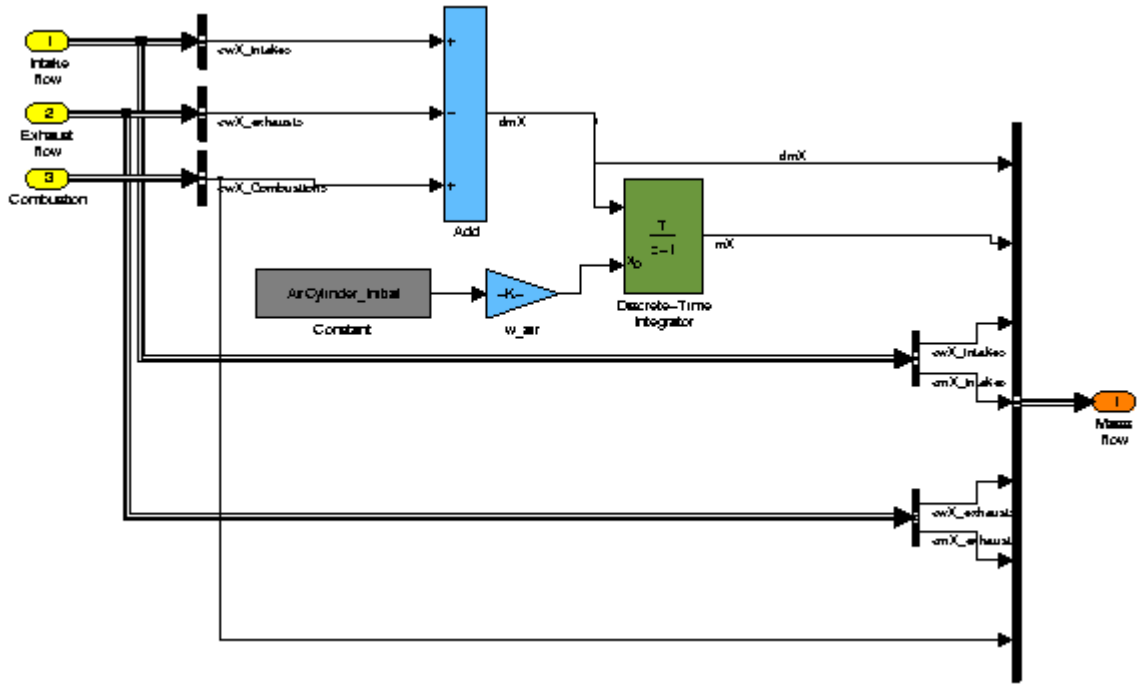


8.2.13 Cylinder Mass

<i>Input</i>	<i>Units</i>	<i>Range</i>	<i>Comments</i>
Intake flow	[bus]	-	
Exhaust flow	[bus]	-	
Combustion	[bus]	-	

<i>Output</i>	<i>Units</i>	<i>Range</i>	<i>Comments</i>
Mass flow	[bus]	-	This bus contains the information about all the mass flows as well as the total mass of chemical compounds present in the cylinder.

Simulink schematics

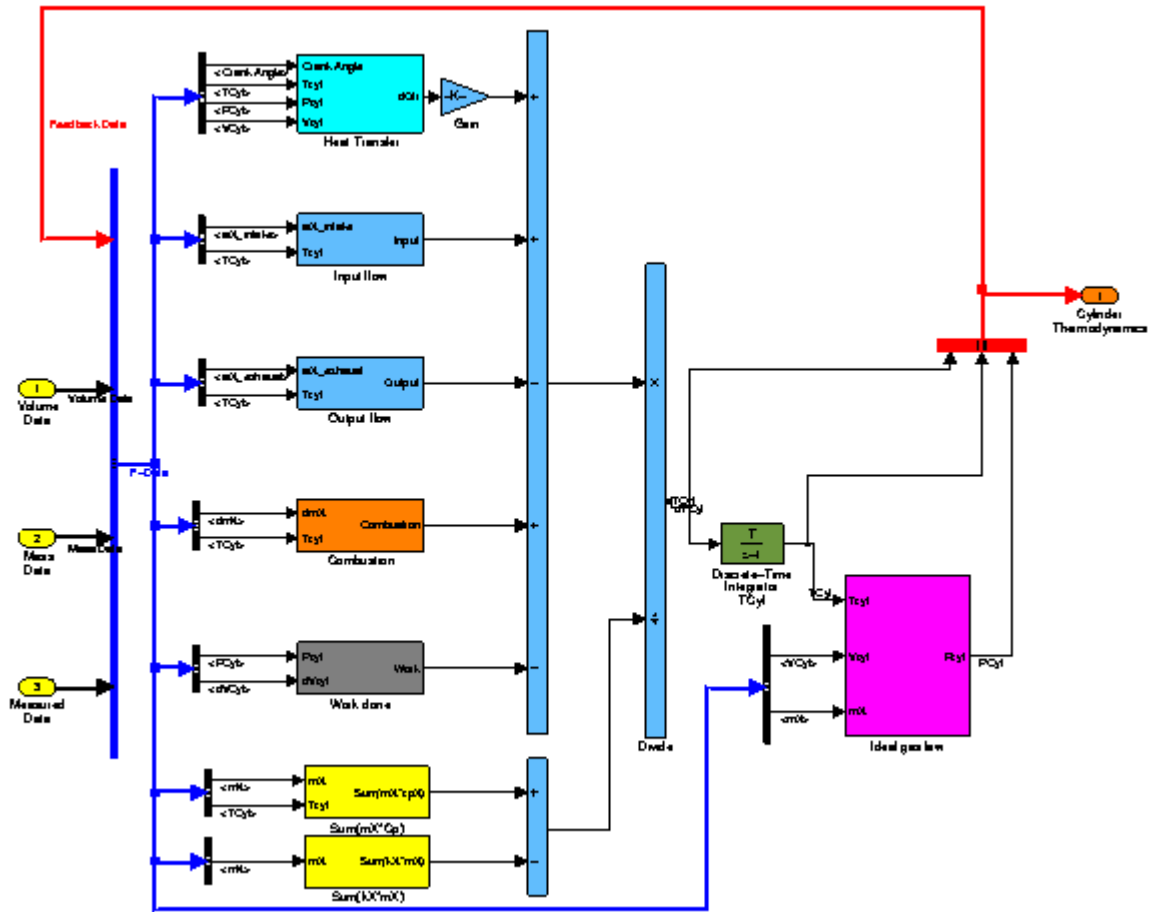


8.2.14 Cylinder Thermodynamics

<i>Input</i>	<i>Units</i>	<i>Range</i>	<i>Comments</i>
Volume data	[bus]	-	
Mass flow	[bus]	-	

<i>Output</i>	<i>Units</i>	<i>Range</i>	<i>Comments</i>
Cylinder Thermodynamics	[bus]	-	This bus contains the current cylinder pressure and temperature condition.

Simulink schematics



8.2.15 Engine performance

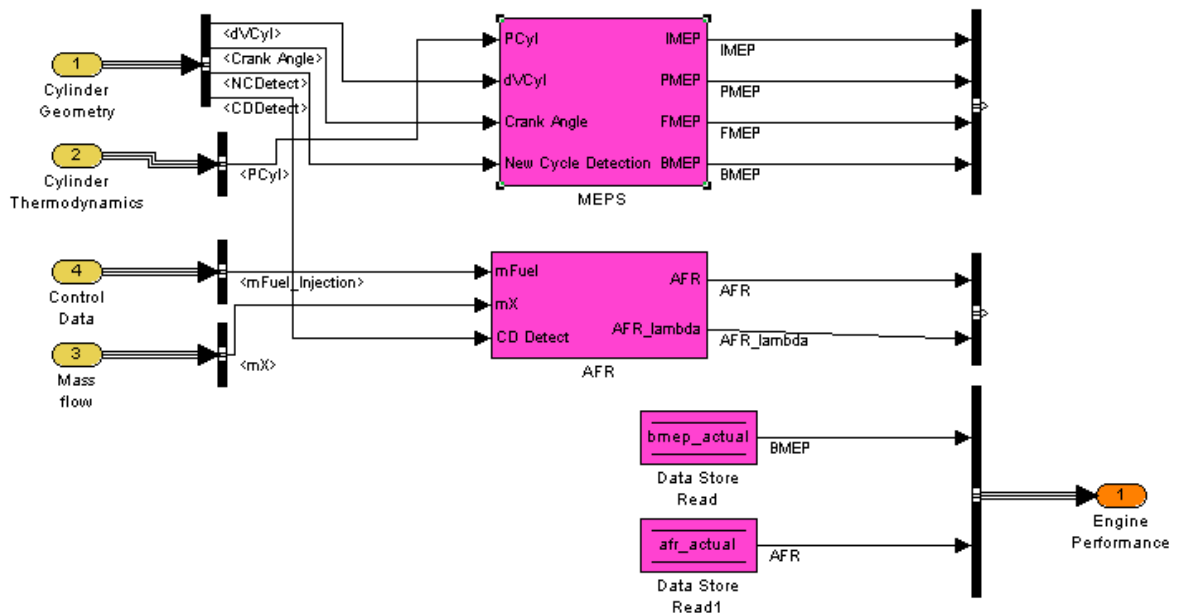
<i>Input</i>	<i>Units</i>	<i>Range</i>	<i>Comments</i>
Cylinder geometry	[bus]	-	
Cylinder Thermodynamics	[bus]	-	
Mass flow	[bus]	-	
Control data	[bus]	-	

Output	Units	Range	Comments
Engine performance	[bus]	-	This bus contains the AFR and BMEP

Notes

The AFR and BMEP values are saved once per engine cycle to appropriate Simulink variables.

Simulink schematics



8.2.16 cylinder_model_init.m

```
% Cylinder model initialization
```

```
warning off;  
clear all;  
loadmodeldata;
```

```
%General constants
```

```
R = 8.313;
```

```
%chemical vectors has the form chemical_vector = [O2 N2 Ar CO2 H2O]
```


%Engine properties

Engine_N = 2400; %[rpm]

Engine_Bore = 0.1; %[m]

Engine_Stroke = 0.105; %[m]

Engine_Rod = 0.185; %[m]

Intake_Valve_Diameter = 0.0404; %[m]

Exhaust_Valve_Diameter = 0.0368; %[m]

Engine_CompressionRatio = 17.5;

Engine_DisplacementVolume = pi*Engine_Bore^2/4*Engine_Stroke;

Engine_ClearenceVolume = Engine_DisplacementVolume/(Engine_CompressionRatio - 1);

MaximumCylinderVolume = Engine_DisplacementVolume + Engine_ClearenceVolume;

%Air properties

w_air_O2 = 23.16/100; %the ratio of oxygen in the air

w_air_N2 = 75.51/100; %the ratio of nitrogen in the air

w_air_Ar = 0.93/100; %the ratio of argon in the air

w_air = [w_air_O2 w_air_N2 w_air_Ar 0 0];

%Diesel fuel properties

w_fuel_C = 86/100; %the ratio of carbon in the fuel

w_fuel_H = 14/100; %the ratio of hydrogen in the fuel

w_fuel_S = 0.05/100; %the ratio of sulfid in the fuel

fuel_LHV = 42E6; %lower heating value of the diesel fuel [J/kg]

fuel_C = 12.5; %number of carbon atoms in the diesel fuel

fuel_H = 22.5; %number of hydrogen atoms in the diesel fuel

Lt = 14.43; %stoichiometric ratio for the diesel fuel (amount of air [kg] needed to burn 1kg of diesel fuel)

%Initial conditions

TCylinderInitial = 743.512;

PCylinderInitial = 291660;

PIntakeInitial = 2.5997e+005;

TIntakeInitial = 338.02;

PExhaustInitial = 3.0864e+005;

TExhaustInitial = 876.68;

AirCylinder_Initial = 0.0018*Engine_ClearenceVolume/MaximumCylinderVolume;

w_egr_exhaust = [0 0.7228 0 0.1981 0.0790];

w_egr = w_egr_exhaust;

```

%Individual gas constant
%Molar masses [kg/mol]
M_O2 = 31.9988/1000;
M_N2 = 28.0134/1000;
M_Ar = 39.948 /1000;
M_CO2 = 44.0098/1000;
M_H2O = 18.0152/1000;
M_C = 12/1000;
M_H2 = 2.016 /1000;

%specific gas constant %[J/kg/K]
r_O2 = 259.82;
r_N2 = 293.73;
r_Ar = 29.11;
r_CO2 = 188.96;
r_H2O = 461.91;
r_all = [r_O2; r_N2; r_Ar; r_H2O; r_CO2];
k_all = r_all;

%Simulation parameters
TSampleTime = 1;

%Initial values for control inputs, if needed
try,
    mFuel = evalin('base','mFuel');
    Theta0 = evalin('base','Theta0');
    Zegr = evalin('base','Zegr');
    bmep = evalin('base','bmep');
catch,
    assignin('base','mFuel',7.1979e-005);
    assignin('base','Theta0',-3.3228);
    assignin('base','Zegr',11);
    assignin('base','bmep',13.45);
end

```

8.2.17 loadmodedata.m

```

%Load data
%Model

```

```

load data/enginedata
load data/chemicaldata
load data/valvesdata
load data/combustiondata
load data/cylinderdata
load data/intakedata
load data/exhaustdata
load data/crankangle
load data/vibe
load data/motoreddata
load data/engineexternaldata

%Controller
load data/ff_controller
load data/fb_controller
load data/hinf_controller
load data/pi_controller

```

8.2.18 interpolateData.m

```

function out = interpolateData(in)
% Interpolates intake and exhaust manifold temperatures and pressures from
% the know operating points (different loads) for any chosen operating
% point. This function is used by cylinder_model

%Inputs
bmep = in(1); %Operating point
CrankAngle = in(2); %Given crank shaft angle
EngineExternalData = evalin('base','EngineExternalData');
%Other
method = 'linear';
bmeps = EngineExternalData.bmep;
intake_pressures = [];
intake_temperatures = [];
exhaust_pressures = [];
exhaust_temperature = [];
for i=1:EngineExternalData.ndata,
%Intake manifold pressure
intake_pressures(i)=
interp1(EngineExternalData.data{i}.Intake_Pressure_Crank,EngineExternalData.data{i}.Intake_Pressure,CrankAngle,method,'extrap');

```

```

%Intake manifold temperature
intake_temperatures(i)=
interp1(EngineExternalData.data{i}.Intake_Temperature_Crank,EngineExternalData.data{i}.Intake_
Temperature,CrankAngle,method,'extrap');
%Exhaust manifold pressure
exhaust_pressures(i)=
interp1(EngineExternalData.data{i}.Exhaust_Pressure_Crank,EngineExternalData.data{i}.Exhaust_
Pressure,CrankAngle,method,'extrap');
%Exhaust manifold temperature
exhaust_temperatures(i)=
interp1(EngineExternalData.data{i}.Exhaust_Temperature_Crank,EngineExternalData.data{i}.Exha
ust_Temperature,CrankAngle,method,'extrap');
end
out = [];
out(1) = interp1(bmeps,intake_pressures,bmep,method,'extrap'); %intake pressure
out(2) = interp1(bmeps,intake_temperatures,bmep,method,'extrap'); %intake temperature
out(3) = interp1(bmeps,exhaust_pressures,bmep,method,'extrap'); %exhaust pressure
out(4) = interp1(bmeps,exhaust_temperatures,bmep,method,'extrap'); %exhaust temperature
return;

```