Institutionen för systemteknik
Department of Electrical Engineering

Examensarbete

Computationally Efficient Model for On-Board Simulation of Heavy Duty Diesel Engines

Examensarbete utfört i Fordonssystem
vid Tekniska högskolan vid Linköpings universitet
av

Per Darnfors & Alfred Johansson

LiTH-ISY-EX--12/4606--SE

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Beräkningseffektiv dieselmotormodell för simulering i inbyggda system

Computationally Efficient Model for On-Board Simulation of Heavy Duty Diesel Engines

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Sammanfattning
Abstract

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The outputs enables gear shift optimization over three variables, torque for vehicle acceleration, fuel consumption for efficiency and exhaust temperature to maintain high efficiency in the exhaust after treatment system.

The engine model captures the low frequent dynamics of the modelled quantities in the closed loop of the engine and it's electronic control system. The model only consists of three states, one for the pressure build up in the intake manifold and two states for modelling the exhaust temperature. The model is compared to measured data from a engine test cell and the mean absolute relative error are lower than 6.8%, 7.8% and 5.8% for brake torque, fuel consumption and exhaust gas temperature respectively. These results are considered good given the simplicity of the model.

Nyckelord
Keywords

real-time engine model,on-board, driveline prediction, AMT, gear selection

Abstract

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Contents

Abbreviations

AFR	Air to Fuel Ratio
AMT	Automatic Mechanical Transmission
BLB-cycle	Volvo intern driving cycle
ECU	Electrical Control Unit
EECU	Engine Electrical Control Unit
EGR	Exhaust Gas Recirculation
EMS	Engine Management System
MARE	Mean Absolute Relative Error
MVEM	Mean Value Engine Models
NARMAX	Non-linear Auto-Regressive Moving Average with Exogenous input
SCR	Selective Catalyst Reduction
SI	Spark Ignited
TECU	Transmission Electrical Control Unit
TMS	Transmission Management System
VGT	Variable Geometry Turbine
WHTC	World Harmonized Transient Cycle

Nomenclature

i	Index
M	Torque
$\dot{m}$	Mass Flow
N	Engine Speed
n	Number
P	Power
p	Pressure
q	Energy per Mass
R	Specific Gas Constant
T	Temperature
V	Volume
λ	Ratio
η	Efficiency
τ	Time Constant

Subscripts

exh	Exhaust
D	Displacement
im	Intake Manifold
LHV	Lower Heat Value
meas	Measurement
vol	Volumetric
r	Revolutions per Cycle
ref	Reference
req	Required
SS	Steady State

Chapter 1

Introduction

For heavy duty truck development reducing fuel consumption is an important task as fuel cost is a major expense for the hauliers. Today AMT's (Automated Mechanical Transmission) are widely used in heavy trucks. The AMT developed at Volvo Truck Technology (I-Shift) has twelve forward gears. When the truck is lightly loaded or unloaded single gear steps would result in very short times between shifts. This results in insufficient acceleration and poor comfort. Other important factors to consider when choosing gear are road inclination and the engine performance. This requires a sophisticated gear shift strategy, i.e. deciding how many gear steps to shift and when to shift. Simulating the vehicles behaviour during shifts to different gears gives a good basis to evaluate which gear to choose and when to shift. An on board simulation environment of the translatory motion of the vehicle provides the possibility to predict the trucks behaviour under the actual circumstances.

In this thesis an engine model is developed, to be used in such a simulation environment, that is able to capture the engines behaviour, regarding torque response, fuel consumption and exhaust gas temperature. The engine modelled is a 13 litre, in line, six cylinder, four stroke, diesel engine with VGT (Variable Geometry Turbine) and EGR (Exhaust Gas Recirculation) including the control system. The engine works in several different modes to achieve the legislated emission standards, for example one mode for best efficiency and one to maintain high exhaust temperature so that the SCR (Selective Catalyst Reduction) can work properly. In these modes the engine performs differently in terms of torque response, efficiency and exhaust temperature.

This thesis is carried out at Volvo Truck Technology in Lundby, Gothenburg at the department of Powertrain Engineering, Control Systems.

1.1 Problem description

The aim of the thesis is to develop a model of the engine to be used for simulating translatory motion of the complete vehicle in order to evaluate gear selection.

The model that is developed is a model over the engine and it's ECU (Electrical Control Unit), i.e. for the closed loop system. The engine performance and response differs between different control modes and the model needs to capture this behaviour. Several times every second a gear selection decision is to be made and every time several gear shifts have to be simulated and compared against each other. The model is to be implemented in the embedded system and this results in a demand on the model to be very computationally efficient.

There are two different ECU's that control the driveline. The EECU (Engine ECU) controls the engine and the TECU (Transmission ECU) controls the AMT. The EECU contains information about how the engine performs and in which mode it is currently working and the TECU has information about gear shifts and needs the result i.e. acceleration and fuel consumption of the vehicle. Redundant data is to be avoided to minimize memory utilization.

1.2 System overview

In diesel engines the fuel is sprayed directly into the combustion chamber and ignited by the heat during the compression see Figure 1.1 for a schematic overview of a diesel engine. During normal operation the AFR (Air to Fuel Ratio)is higher than the stoichiometric AFR. Due to this an almost instantaneous increase in injected fuel is possible, until the stoichiometric AFR is reached. Torque is proportional to injected fuel which means that a instantaneous increase in torque is possible unlike in a spark ignited engine where the AFR is close to the stoichiometric at normal operation points.

Diesel engines produces emissions, mainly NO_x, soot and carbon-oxides, which are controlled by legislations. Two common ways to reduce NO_x are either to lower the combustion flame front temperature by using EGR or changing the characteristics (pressure, timing and number of sprays) of the fuel injection. For more information on diesel engines [1] and [2] are recommended.

The main control modes that are active during normal driving are a combination of how much extra heat in the exhaust gas the engine produce and how much NO_x that is produced in the combustion. The reason to increase heat in the exhaust gas is to control the temperature in the SCR, a cold SCR can not reduce NO_x at the rate needed. Both lower NO_x production and higher exhaust temperature reduces the engines efficiency.

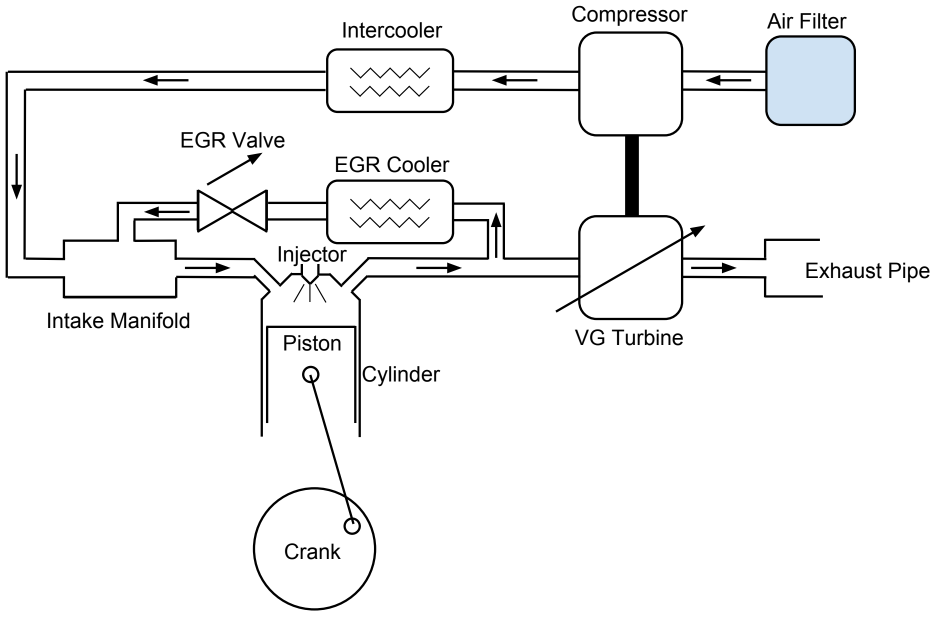

Figure 1.1: Schematic view of a diesel engine

1.3 Related Research

Models for use in control or diagnostic applications describing the engines performance is presented in [1, 3, 4, 5, 6, 7, 8, 9, 10, 11]. In [3] a model based on physical relations and 8 states describes the engines behaviour. Where as in [4] a linear 4 state state-space model is presented. A physical engine model for fixed step execution is presented in [5] and [6]. The book [1] covers some engine modelling and control based on physical models. In [12] a detailed physical model is developed in Simulink for control system evaluation in an off-line environment.

Empirical or data driven models describing diesel engines are presented in [7], where Wiener-Hammerstein models are used and in [9], where NARMAX[1] models are used.

To describe exhaust temperature and emissions during cold start, [8] proposes a simple empirical model with few states. A model that describes the temperature drop in the exhaust system based on physical relations is presented in [11].

A model is presented in [13] that describes in-cycle variations, intended to use in a hardware in the loop system for evaluating control systems in real time. Another model for a hardware in the loop application, where only the mean value is considered, is presented in [14].

[1]Non-linear Auto-Regressive Moving Average with Exogenous input

This far the models cited have described the physical engine and not including the control unit. In [15] a dynamic model of an engine and it's control system is modelled using a neural network approach. A model that is intended to use in an ECU to estimate the output torque is presented in [16].

The models cited above are intended to run at the most at real time in an ECU. The model developed in this thesis is based on [17]. The model is supposed to run much faster than real time and in an ECU. This requires a model that is very computationally efficient.

Chapter 2

Modelling

There are many ways to model a diesel engine. The approaches and ways to model differ due to different restrictions in resources and different requirements in performance and accuracy. All models that are referred to and developed in this thesis are of the type MVEM (Mean Value Engine Model). In MVEM quantities are considered averaged over several cycles and in-cycle variations are neglected. The model presented in this thesis is a model over an engine and it's electronic control system.

Physical mean value modelling is investigated in [1, 3], this type of modelling requires understanding and sub-modelling of each component in the engine e.g. every manifold is modelled with filling and empting and laws of energy conservation. A physical mean value model is not suited for the purpose of this thesis because it's relative computational heavy and the EMS (Engine Management System) controls the engine in such way that dynamic phenomenons in the engine are reduced or changed, it also requires a model of the EECU (or a virtual copy EECU) for the closed loop system.

Empirical mean value models are also commonly used to describe diesel engines. In [15] a neural network approach is used to model brake torque and fuel consumption in a SI (Spark Ignited) engine. In [15] a model with 69 states for brake torque is suggested with sigmoid transfer functions connecting them. This approach is not suited for an on-board model due to the complicated functions between states and the large number of states.

The approach that is used in this thesis is inspired from [17] that uses engine speed and desired torque as inputs to calculate set-points in inlet air mass flow. The set-point is filtered though a first order system with varying time constant and then used to limit the output torque. The output torque is also filtered though a first order system with varying time constant. An overview of the model presented in [17] can be seen in Figure 2.1.

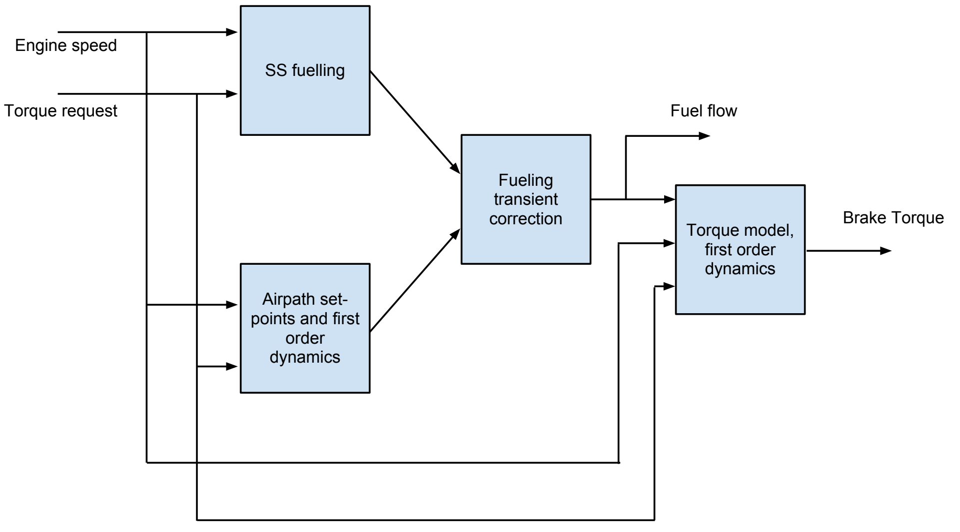

Figure 2.1: Overview of the model presented in [17].

The EMS uses mapped set-points for requested indicated torque based on pedal position and engine speed. The engine model relies on the value of the EMS torque set-point and uses this value as output at steady state operation. The model consists of different restrictions that limits the output torque from the requested. These different restrictions are derived from e.g. dynamics in boost pressure build up and engine control strategies. Often the EMS has knowledge about the engines physical limitations so even if the limitation derives from a physical restriction the model can use information from the EMS to explain the engines physical behaviour.

Due to this approach, different types of models are used for different types of restrictions depending on if it's the physical engine or the controller that limits the engines performance. The sub-models are:

- Air Flow Dynamics, estimates air flow into the combustion chamber.
- Torque Set Point, calculates the torque demand from the driver.
- Smoke Limiter, limits the torque output (fuel flow) to avoid smoke in transients.
- Fuel Flow, calculates the fuel flow.
- Brake Torque, subtract losses from indicated torque.
- Exhaust Temperature, estimates the exhaust temperature after the turbine.

Details of the different sub models are presented in Section 2.1-2.6 and an overview over the created engine model is seen in 2.2.

Further in this rapport MARE (Mean Absolute Relative Error) and WHTC (World Harmonized Transient Cycle) are typically used to evaluate the model. The definition of MARE can be seen in Equation (2.1) where M is the number of samples, y is the modelled quantity and y_{meas} measured quantity. The WHTC is a global accepted transient driving cycle used for emission evaluation and is defined according to [18].

$$MARE = \frac{1}{N}\sum_{i=1}^{M}\frac{|y(i)-y_{meas}(i)|}{\frac{1}{M}\sum_{j=1}^{M}|y_{meas}(j)|} \tag{2.1}$$

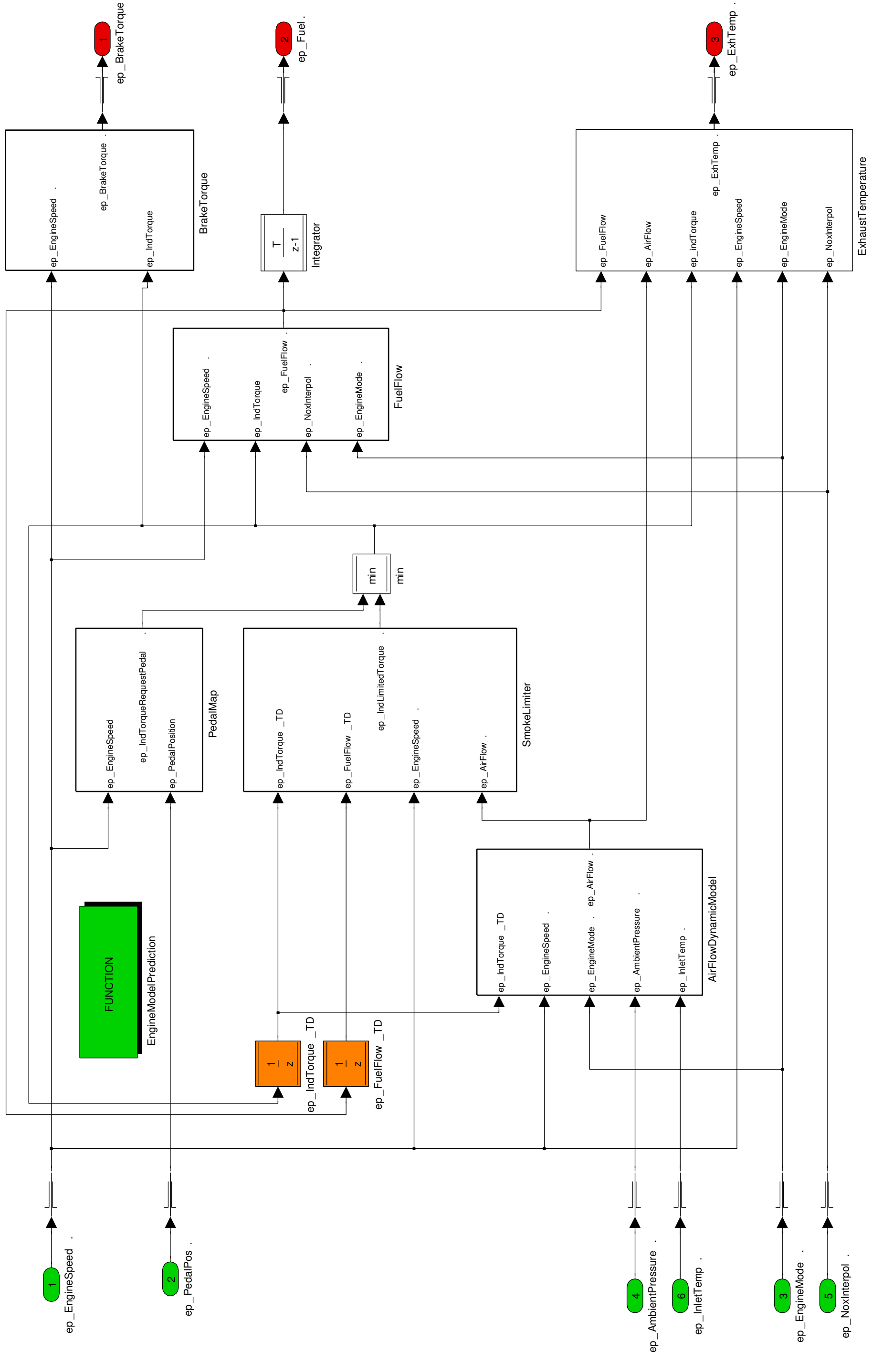

Figure 2.2: Simulink scheme of Engine Model.

2.1 Air Flow

This submodel estimates the air mass flow into the cylinders which is needed as input to the smoke limiter model (Section 2.3), in- and outputs to the sub-model are stated in Table 2.1. In the engine investigated an EGR and VGT system is used. With this configuration the air path consists of an air filter, compressor, intercooler, intake manifold, combustion chambers, exhaust manifold, VGT, and EGR valve, Figure 1.1.

Table 2.1: In and out-ports of sub-model Air Dynamics

Quantity	In/Out	Unit	Dependency
Engine Speed	In	RPM	-
Torque Demand	In	Nm	-
Engine Mode	In	-	-
Ambient Pressure	In	kPa	-
Inlet Temperature	In	°C	-
Air Mass Flow	Out	kg/s	Torque Limiter

2.1.1 Approaches

A common way to model the air path in academic papers and research is to use physical mean value modelling and use both filling and emptying and laws of energy conservation for each control volume. This method is used in [3] with good results for a diesel engine with VGT and EGR. The drawbacks with this type of modelling are that the model is heavy to compute [14] and requires good understanding of each component in the engine. They are also not suited for the model in this thesis due to the complexity of the controller that is needed to control the model, which would result in a computational heavy model of air flow and a computational heavy model over the control system.

Apart from physical mean value modelling, purely mathematical tools, such as system identification, are commonly used for modelling. The air path of a diesel engine is highly nonlinear so linear models can not describe the air flow in all operating points in a sufficient way. An NARMAX-model is suggested in [9] for modelling of the air path dynamics of a diesel engine with VGT and EGR using eight states. This model is a model over the engine but the same method could be used for the closed-loop system of the engine and EMS. The drawback with a mathematical model over both engine and EMS is that they have to be identified from data, i.e. every time the EMS is updated with new software for air control new tests have to be performed and new parameters have to be identified.

A simple model of the air dynamics is presented in [17] where the air flow dynamics are modelled as a first order system with a static gain of one and varying time constant, one time constant for increasing torque demand and another if there is a non-increase in torque demand. The input to the first order system is the steady state air flow which is estimated from the volumetric efficiency and the density of inlet air which result in Equation (2.2), where $\eta_{vol}(p, N)$ is according to Equation (2.3) and T is a initiated value.

$$\dot{m}_{air,ss} = \eta_{vol}(p, N)\frac{pV_D N}{n_r RT} \tag{2.2}$$

$$\eta_{vol}(p, N) = \frac{k_1(N)}{p_{ss}} + k_2(N) \tag{2.3}$$

2.1.2 Implementation

The model that is chosen for the air dynamics is a first order system for inlet pressure with varying time constant. The relationship between pressure and air flow is estimated using volumetric efficiency and the density of the inlet air. This type of model is easy to calibrate and can reuse data from the EMS i.e. static set-points for inlet pressure and volumetric efficiency. A overview of the sub-model can be seen in Figure 2.3-2.5.

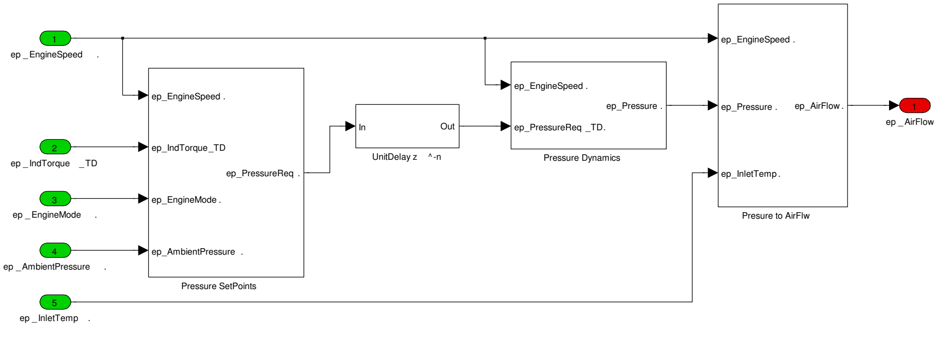

Figure 2.3: Simulink scheme of the sub-model Air Dynamics. Consists of three sub-models, inlet pressure set-point, inlet pressure dynamics and air mass flow.

The set-points for static pressure are modelled as mapped values with engine speed, torque request, ambient pressure and the operating mode as inputs, see Equation (2.4).

$$p_{ref} = f_{p_{ref}}(N, M_{req}, p_{amb}, Mode) \tag{2.4}$$

The dynamics in inlet pressure are modelled according to Equation (2.5). The time constant is a map-function of engine speed and the sign of the difference between dynamic pressure and the pressure set-point, Equation (2.6). There are

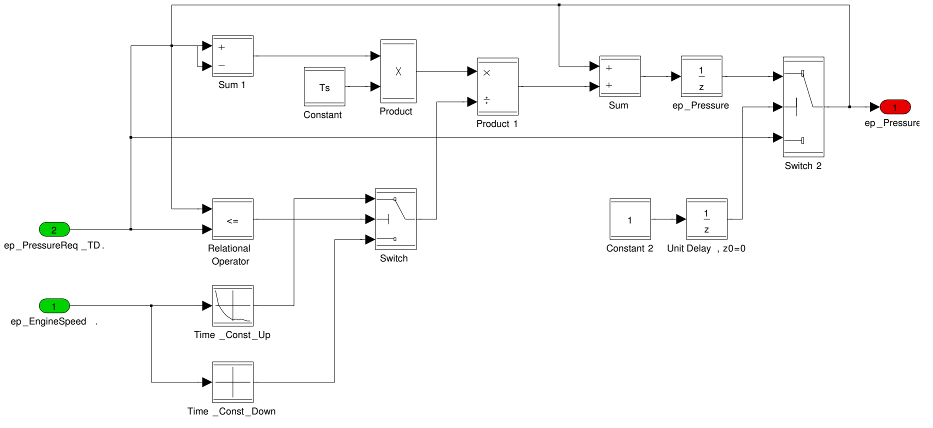

Figure 2.4: Simulink scheme of sub-model Pressure Dynamics, a first order system with varying time constant, corresponding to Equation (2.7).

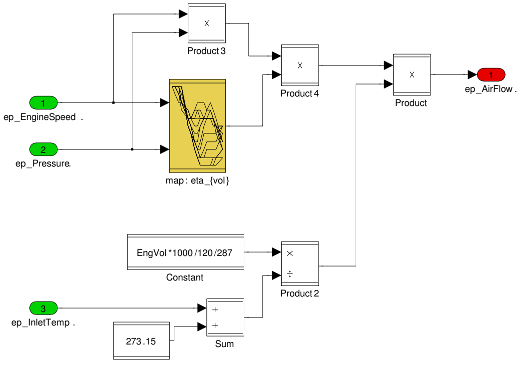

Figure 2.5: Simulink scheme of sub-model Pressure to AirFlw, corresponding to Equation (2.8). Uses the density of the inlet air, the swept volume and a efficiency map to estimate air mass flow in to the cylinder.

two main reasons to have different time constants for pressure increase and non-increase. The first is that there are different contributing inertias. In pressure build up there are two major contributing inertias , inlet manifold filling and the turbo lag. In pressure decrease only the inlet manifold emptying is contributing to the main dynamics. The second reason is that the control system works differently in pressure build up and pressure decrease. The reason the time constant varies with engine speed are the non-linearities in the air path. The input (τ) to the dynamics is time delayed due to the time delay in the closed loop system. The model is implemented in discrete time and in Equation (2.7) the discrete dynamic difference equation is presented, t and k are the current time and t_0 and m are the amount the input is delayed in seconds and samples.

$$\tau_p \dot{p}(t) = p_{ref}(t - t_0) - p(t) \tag{2.5}$$

$$\tau_p = f_{\lambda_p}(N, sgn(p_{ref} - p)) \tag{2.6}$$

$$p(k+1) = (1 - \frac{T_S}{\tau})p(k) + \frac{T_S}{\tau} p_{ref}(k - m) \tag{2.7}$$

Volumetric efficiency has been modelled in several ways during the history of engine modelling, polynomials in engine speed and inlet pressure are widely suggested [1]. Instead of a polynomial, a map with engine speed and inlet pressure as inputs is chosen due to better accuracy for this specific engine. The volumetric efficiency together with the density of the air gives the airflow in to the cylinder according to Equation (2.8). The volumetric efficiency $\eta_{vol,air}$ does not consider the total mass flow but only the fresh air mass flow (not EGR) that flows into the cylinder at given engine speed and inlet pressure in steady state.

$$\dot{m}_{air} = \eta_{vol,air}(p, N) \frac{V_D N p}{n_r R T_{im}} \tag{2.8}$$

2.1.3 Calibration

Static functions are estimated in least square sense from measurements in steady-state.

The time constant for pressure build-up and decrease is estimated from measurements at constant engine speed where steps up and down in requested torque are performed. This is done at several different engine speeds. At every engine speed one time constant for pressure increase and one for pressure decrease is estimated, through minimizing the mean square error according to Equation (2.9). The delay on p_{ref} is tuned by hand.

$$\hat{\tau}_i = arg\, \min_{\tau_i} \frac{1}{M} \sum_{i=1}^{M} (p(i, \tau_i) - p_{meas}(i))^2 \tag{2.9}$$

2.1.4 Validation

Modelled and measured inlet pressure and air mass flow are compared for two different time intervals in the WHTC in Figure 2.6-2.7. The simulation time interval is the same as the simulation time interval where the model is intended to be used. The MARE for the pressure is 3.5% and for the air mass flow 3.2%.

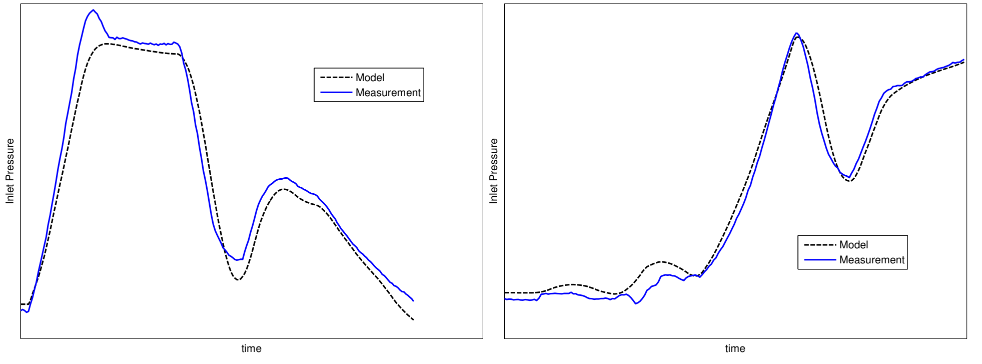

Figure 2.6: Comparison between measured and modelled inlet pressure in two different time intervals. The MARE is 3.5%.

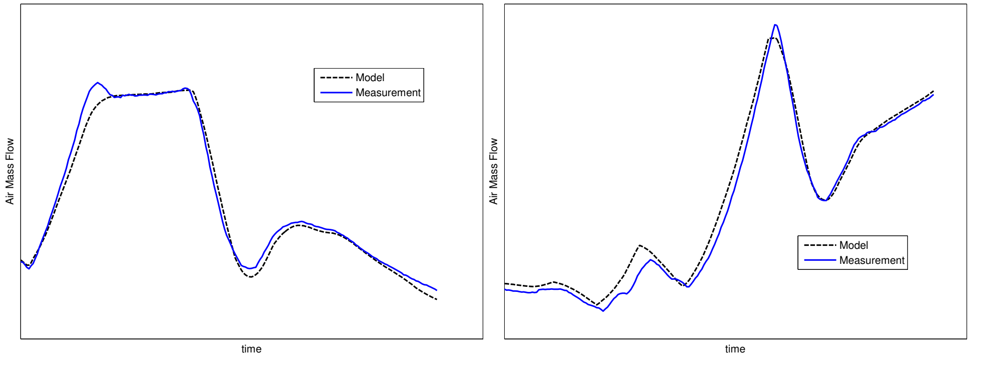

Figure 2.7: Comparison between measured and modelled air mass flow in two different time intervals. The MARE is 3.2%.

2.2 Torque Set Point

The torque set-point for the engine is the input from the driver through the pedal. How the pedal position is translated into torque request is a major factor in drive-ability and how the driver experience the vehicle. This model uses the same map as the EMS to translate pedal position into requested torque. In and outputs from the sub-model are seen in Table 2.2.

Table 2.2: In and out-ports of sub-model Static Fuel

Port	In/Out	Unit	Dependency
Engine Speed	In	RPM	-
Pedal Position	In	%	-
Torque Demand	Out	Nm	-

2.2.1 Implementation

The pedal position to torque request model is implemented as mapped values with pedal position and engine speed as inputs.

2.3 Smoke Limiter

In steady state operating conditions a diesel engine runs with a higher AFR than the stoichiometric AFR. In load transients the fuel flow is increased and because the turbo can not instantly provide the amount of air needed, due to inertia in the turbo and air, the AFR decreases and in some cases to a level where the engine produces soot. To avoid soot production the control system limits the injected fuel to keep the AFR over a predefined limit, typically just above the stoichiometric AFR. In this chapter such a model is presented. The in- and out-ports are stated in Table 2.3.

Table 2.3: In and out-ports of sub-model Smoke Limiter

Port	In/Out	Unit	Dependency
Engine Speed	In	RPM	-
Old Torque Demand	In	Nm	-
Air Flow	In	kg/s	-
Old Fuel Flow	In	kg/s	-
Limited Torque Demand	Out	Nm	Steady State Fuel Flow

2.3.1 Implementation

This part of the model only consists of simplified EMS software. An overview of the sub-model is presented in Figure 2.8.

The model over the smoke limiter uses a lookup table to determine the lowest allowed AFR based on the engine speed and the previous torque value. The actual airflow is then divided by this value to get the maximal allowed fuel flow. The previous torque is divided by the previous fuel flow to get an approximate torque to fuel ratio. This ratio is then multiplied with the maximum allowed fuel flow to get the maximum allowed torque.

2.3.2 Validation

The smoke limiter sub model is validated by feeding it with measured signals of air flow, fuel flow, engine speed and indicated torque where the fuel flow and indicated torque are delayed one sample to simulate the previous value. The result of a part of a WHTC is shown in Figure 2.9 and it shows that the smoke limiter works as expected.

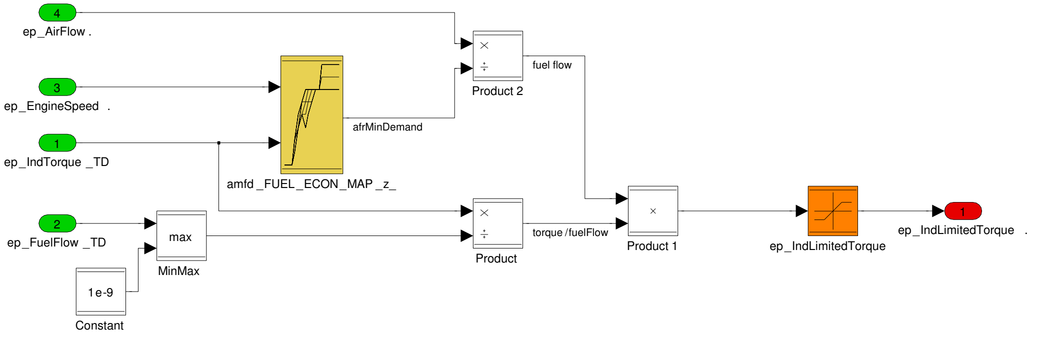

Figure 2.8: Simulink scheme of SmokeLimiter. Limits the desired torque, considering the current air-flow, so the engine doesn't produce as much soot in the exhausts.

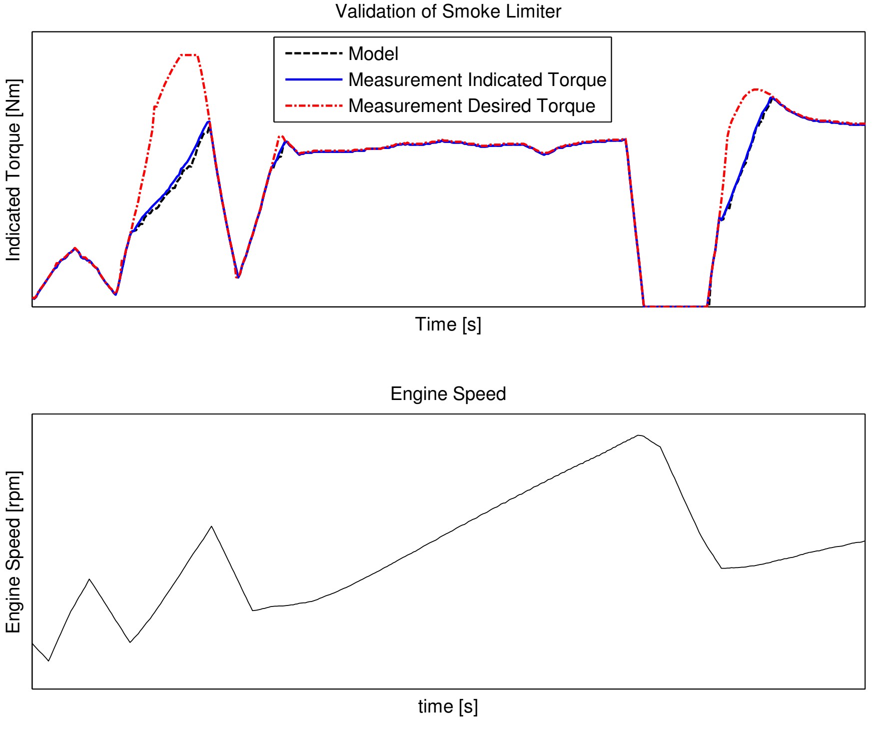

Figure 2.9: Comparison between measured and modelled indicated torque. Desired torque is what the driver demands with the pedal. In heavy load transients the desired torque is limited to keep the soot production down.

2.4 Fuel Flow

To know how efficient the engine is running the engine model needs to calculate the fuel consumption. The fuel injection system is of common-rail type and basically consists of a chamber containing fuel under high pressure which can be sprayed in to the cylinder at a desired time and flow rate. In this section a model over the fuel flow is presented. The in- and out ports are stated in Table 2.4.

Table 2.4: In and out-ports of the fuel flow sub-model

Port	In/Out	Unit	Dependency
Engine Speed	In	RPM	-
Torque Demand	In	Nm	-
Engine Mode	In	-	-
Power Demand	In	RPM·Nm	-
Inlet Temperature	In	°C	-
Fuel Flow	Out	Kg/s	Smoke Limiter

2.4.1 Implementation

The dynamics in the fuel injection system are very fast (the fuel arrives in the cylinders in the same cycle as the control signal is sent from the EECU) and are therefore neglected. Even the torque response from the injected fuel is fast enough to be neglected as the burned fuel in a certain cycle delivers the desired force on the piston the same cycle it is injected.

The actual fuel flow is approximated with the, by the EMS, requested fuel flow. This isolates the model to describe the behaviour of the EMS concerning fuel flow. To use the same calculations as the EMS would be too time consuming because it uses inputs from many sensors and the function is relatively demanding to compute. The timing, i.e. when and how the fuel is injected, is also calculated at the same time as the fuel flow in the EMS. In the model the timing itself is not interesting but a change in timing affects the behaviour of the engine. To get more heat in the exhaust gases (engine in heat mode described in Section 1.2) the injection timing is changed from it's optimum. This causes the efficiency to decrease and therefore the amount of injected fuel has to increase for the engine to produce the same torque as before. The timing is also changed to get the engine to produce a different amount of NO_x.

Different ways of controlling fuel injection electronically has been done. One common way is to use mapped values depending on operating point [17]. In this model the fuel flow is modelled as a linear function of the requested power, requested torque and actual engine speed as seen in Equation (2.10). The corresponding Simulink implementation is seen in Figure 2.10 and 2.11. This way is chosen because the number of function parameters is far less than all the mapped values required which will save memory utilisation. One advantage with the map-based solution is that the model will cover a larger span of operating points accurately. For this model the fuel flow at engine speed near idle and maximum and engine torque near zero are not very important as the model is used mostly in normal driving conditions.

$$\dot{m}_{fuel} = \Theta_1 P + \Theta_2 N + \Theta_3 T + \Theta_4 \tag{2.10}$$

Where the engine power has the largest impact on the fuel flow model, the engine speed and torque makes smaller adjustments. Several possible functions are tested to see which one gives the best result but the function in Equation (2.10) is shown to be the best concerning computation time and ease of finding parameters.

The parameters in Equation (2.10) differs when the engine runs in different modes and produce different amount of NO_x. Therefore different parameters are estimated for the different modes and for 100% and 0% NO_x production for each mode. This gives six sets of function parameters. When using the model for simulation the model selects the parameters for the actual mode and interpolates the parameters according to NO_x production and keeps the parameters the same throughout the whole simulation.

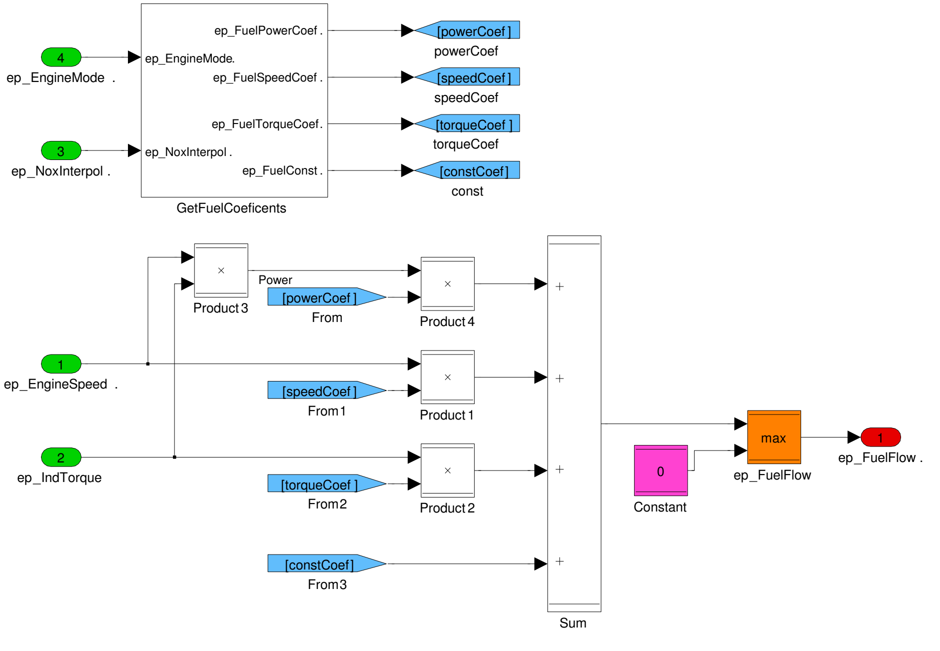

Figure 2.10: Simulink scheme of FuelFlow, The fuel flow is modelled as a linear function of indicated power, indicated torque and engine speed, equation (2.10).

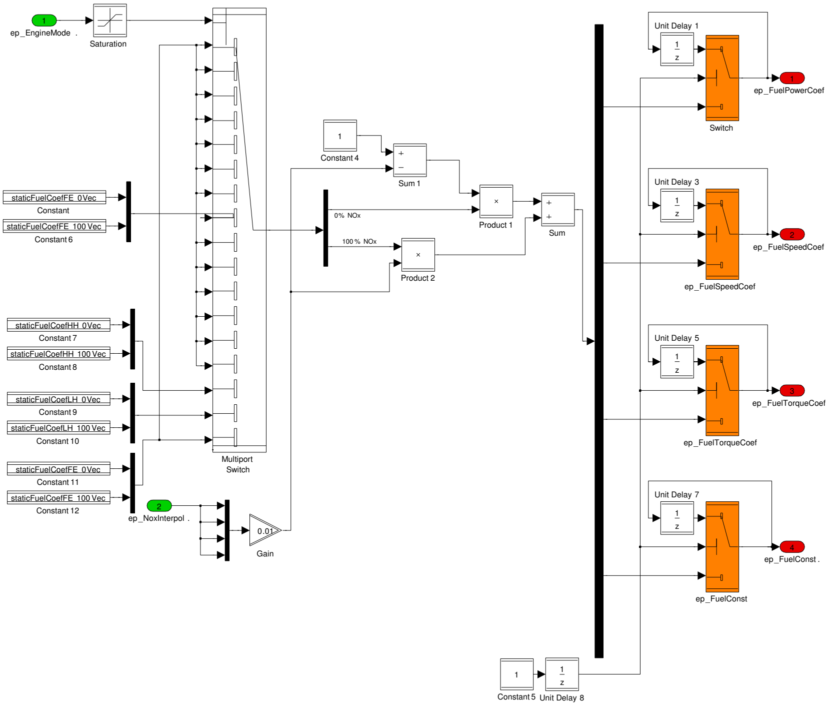

Figure 2.11: Simulink scheme of the sub-model GetFuelCoefficients in Figure 2.10. Initializes the model coefficients depending on NO_x production and engine mode.

2.4.2 Calibration

To find the parameters Θ in Equation (2.10) the engine is run at several different operating points for different modes and with different NO_x production. The fuel flow is measured at these operating points. The cost function to minimize is seen in Equation (2.11).

$$V_M(\Theta) = \frac{1}{M}\sum_{i=1}^{M}(\dot{m}_{fuel}(i|\Theta) - \dot{m}_{fuel,meas}(i))^2 \tag{2.11}$$

Where M is the number of measurements and i is operation point. The measured data for calibration are steady state measurements from an engine test cell.

2.4.3 Validation

The fuel flow model is validated by feeding it with measured signals of engine speed, indicated torque, mode and NO_x production and compare the output fuel flow against the measured.

Figure 2.12 shows measured and modelled fuel flow during steady state operation where the engine is running in one mode and running as economical as possible thereby as cold as possible, producing maximal NO_x. For this comparison the model gave a MARE of 2.5% and a maximal relative error of 5.7%. However, when comparing the model with measurements from a transient cycle (WHTC) the result is not as good. The MARE and max absolute relative error in this case is 5.9% and 38% respectively. A plot over the result can be seen in Figure 2.13. These two measurements are taken at different occasions and during the development phase of the engine and the EMS so there could be differences in hardware and small adjustments in EMS software. This can be one reason why they differ. In the WHTC the engine runs much more at idle and deceleration which is not modelled and is another reason why the result is not as good as for the steady state measurement.

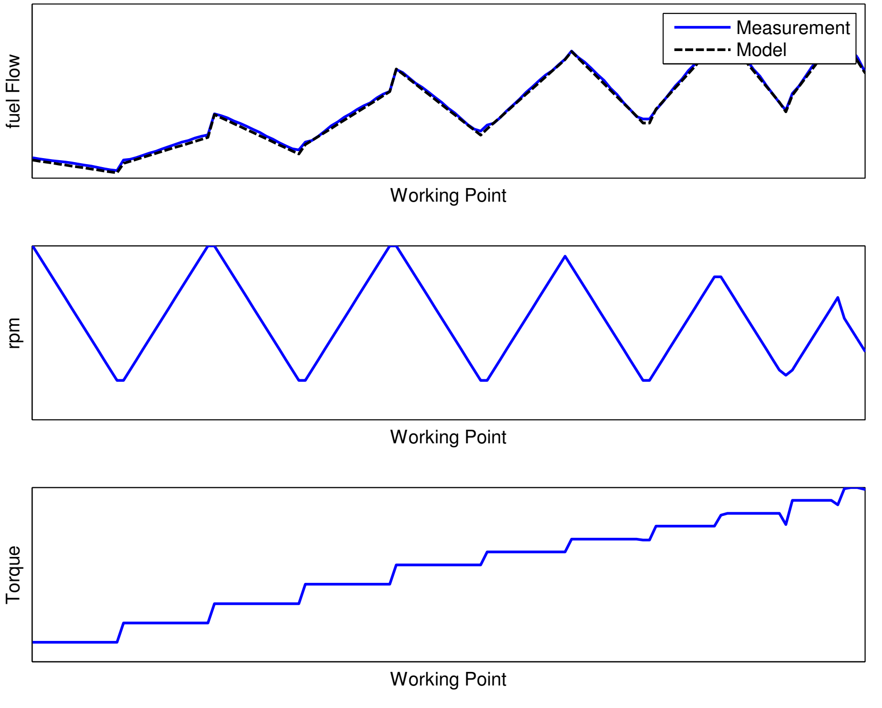

Figure 2.12: Comparison between measured and modelled fuel flow at steady state operation. The model is more accurate at higher torque where the engine works at accelerations.

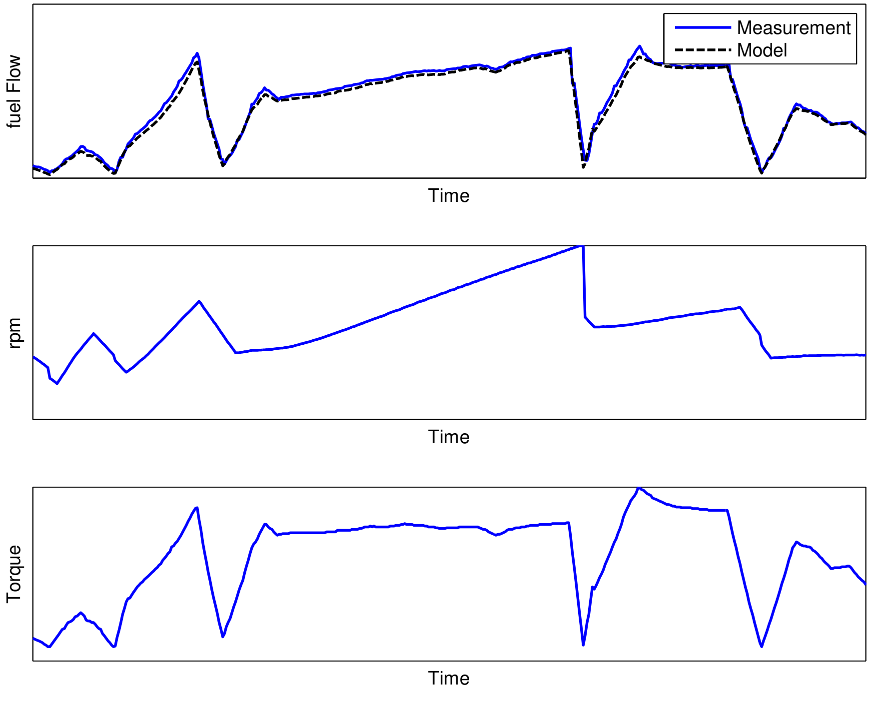

Figure 2.13: Comparison between measured and modelled fuel flow at transient operation, in a part of the WHTC.

2.5 Brake Torque

To get the brake torque from the indicated torque the mechanical losses in the engine has to be subtracted. The mechanical losses consists of auxiliary devices, e.g. climate control and cooling fan, and friction in the engine. It is common to include pumping losses when modelling SI engines but in diesel engines they can be neglected due to absence of a throttle [1]. The in- and out-ports to the sub-model can be seen in Table 2.5.

Table 2.5: In and out-ports of sub-model Brake Torque

Port	In/Out	Unit	Dependency
Engine Speed	In	RPM	-
Indicated Torque	In	Nm	-
Brake Torque	Out	Nm	-

2.5.1 Implementation

An easy way to model these losses is with a second degree polynomial suggested in [1]. In this model mapped values are used for the torque losses, as a function of engine speed as seen in Equation (2.12) and Figure 2.14.

$$M_{brake} = M_{ind} - M_{loss}(N) \tag{2.12}$$

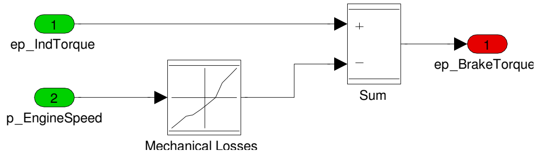

Figure 2.14: Simulink scheme of sub-model Brake Torque, corresponds to Equation (2.12). Subtracts losses from indicated torque and returns brake torque.

2.5.2 Calibration

Calibration is done in steady state measurements where the indicated torque is logged (from the EECU) and the brake torque is measured. This is done at several different engine speeds and for every engine speed a value of M_{loss} is estimated using least square error minimization.

2.5.3 Validation

The torque loss model is validated against measured data and can be seen in Figure 2.15, the MARE is 8.8%.

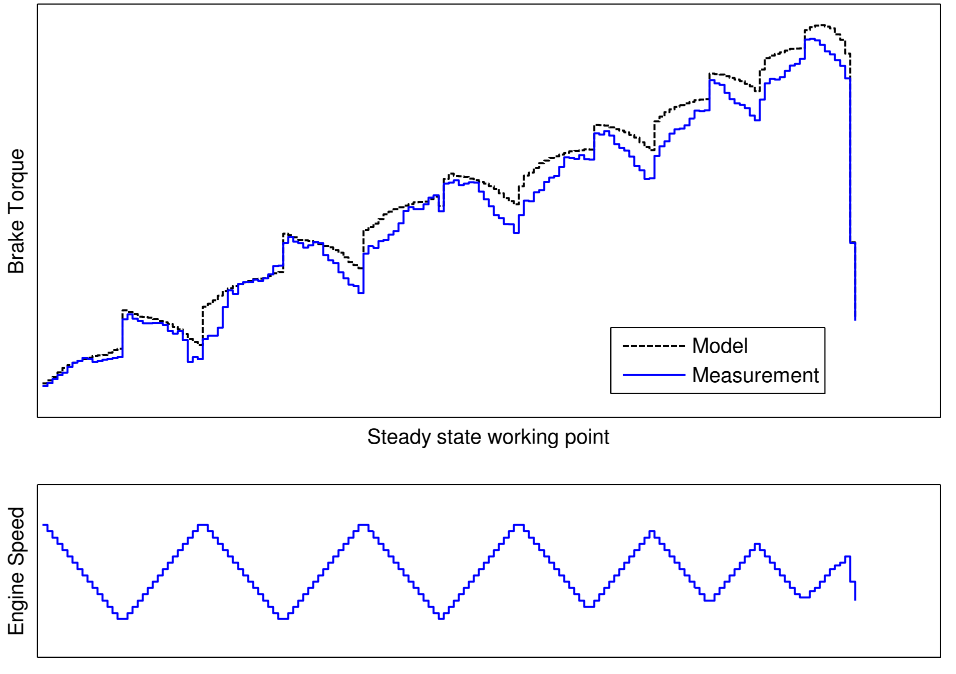

Figure 2.15: Modelled and measured brake torque

2.6 Exhaust Temperature

All of the energy from the injected fuel that isn't used to push the piston down in the cylinder will end up as heat that is transported from the cylinder in different ways. Some of the heat will pass through the cylinder head and walls and be transported away through the cooling water, engine oil or directly to the surrounding air. The rest of the heat will leave the combustion chamber through the exhaust gases. The in- and out ports of the model are stated in Table 2.6.

Table 2.6: In and out-ports of the exhaust temperature sub-model

Port	In/Out	Unit	Dependency
Fuel Mass Flow	In	kg/s	-
Indicated Power	In	$RPM \cdot Nm$	-
Air Mass Flow	In	kg/s	-
Indicated Torque	In	Nm	-
NO_x production	In	%	-
Engine Mode	In	-	-
Inlet Temperature	In	°C	-
Exhaust Temperature	Out	°C	-

2.6.1 Implementation

In [10] the exhaust temperature is modelled using power balance of the combustion chamber to calculate how much the intake air temperature rises in a spark ignited engine. This power balance is seen in the first law of thermodynamics, Equation (2.13). The heat losses, P_{heat}, is in [10] an empirical function of air and fuel mass flow, spark timing and engine speed. This is the only term, except the exhaust temperature, T_{exh}, that isn't measured.

In [8] the exhaust temperature is modelled as a sum of three first order dynamic systems with engine speed, AFR and spark timing as inputs for each respective system. Both these two models are for spark ignited engines but the approaches are interesting for diesel engines as well. The injection timing affects the combustion almost like the spark timing. The concepts used in these two models are interesting because they are relatively easy to calculate and optimize.

In this thesis the exhaust temperature is modelled almost the same way as the air flow in section 2.1 i.e. with one static model that returns a steady state value which is filtered through a dynamic system. This approach is chosen before the one in [8] to reduce the number of parameters.

The first law of thermodynamics [10] seen in Equation (2.13) is used to describe the power balance in the combustion chamber. The known variables are injected fuel and the power output from the engine. Left to determine is how much power that leaves the engine in form of heat losses through the cylinder wall and head, P_{heat}, and the temperature of the air entering the cylinders, T_{inl}. These two terms are not explicitly modelled. The steady state exhaust temperature, $T_{exh,SS}$, is modelled as a linear function of fuel mass flow, $\dot{m}_{fuel}$, indicated power, P_{ind}, air mass flow, $\dot{m}_{air}$ and a mapped function of indicated torque and engine speed, W, seen in Equation (2.14). The Simulink implementation is seen in Figure 2.16 where output from the map called exhMap is W.

$$\dot{m}_{air}c_p(T_{inl} - T_{exh}) + \dot{m}_{fuel}q_{LHV} - P_{eng} - P_{heat} = 0 \tag{2.13}$$

$$T_{exh,SS} = \Theta_1\dot{m}_{fuel} + \Theta_2 P_{ind} + \Theta_3\dot{m}_{air} + \Theta_4 W + \Theta_5 \tag{2.14}$$

The coefficients in Equation (2.14) differs when the engine runs in normal or heat mode or produce different amount of NO_x. Therefore different coefficients are estimated for the different modes and for 100% and 0% NO_x production. This gives four sets of function coefficients. When using the model for simulation the model selects the coefficients for the actual mode and interpolates them according to NO_x production. These are kept the same throughout the whole simulation. The simulink model of the parameter selection is seen in Figure 2.17.
Equations (2.15)-(2.17) describes the state space model handling the dynamics of the exhaust temperature and are the equations implemented in the Simulink sub-model in Figure 2.18. $T_{exh,SS}$ is the steady state exhaust temperature calculated in Equation (2.14) and is the input to the dynamic sub-model. $S_{exh,1}$ and $S_{exh,2}$ are the two states that sums together, with the ratio λ, to the exhaust temperature in Equation (2.17)

$$S_{exh_1}(k+1) = (1 - \frac{T_s}{\tau_1})S_{exh_1}(k) + \frac{T_s}{\tau_1}T_{exh,SS}(k) \tag{2.15}$$

$$S_{exh_2}(k+1) = (1 - \frac{T_s}{\tau_2})S_{exh_2}(k) + \frac{T_s}{\tau_2}T_{exh,SS}(k) \tag{2.16}$$

$$T_{exh}(k) = \lambda S_{exh_1}(k) + (1-\lambda)S_{exh_2}(k) \tag{2.17}$$

2.6.2 Calibration

To estimate the coefficients Θ in Equation (2.14) the engine is run at steady state at several different operating points for different modes and with different NO_x production. The exhaust temperature and the input signals for the model are measured at these operating points. The least square cost function to minimize is seen in Equation (2.18), where M is the number of measurements and i is the operation point.

$$V_M(\Theta) = \frac{1}{M}\sum_{i=1}^{M}(T_{exh,meas}(i) - T_{exh}(i|\Theta))^2 \tag{2.18}$$

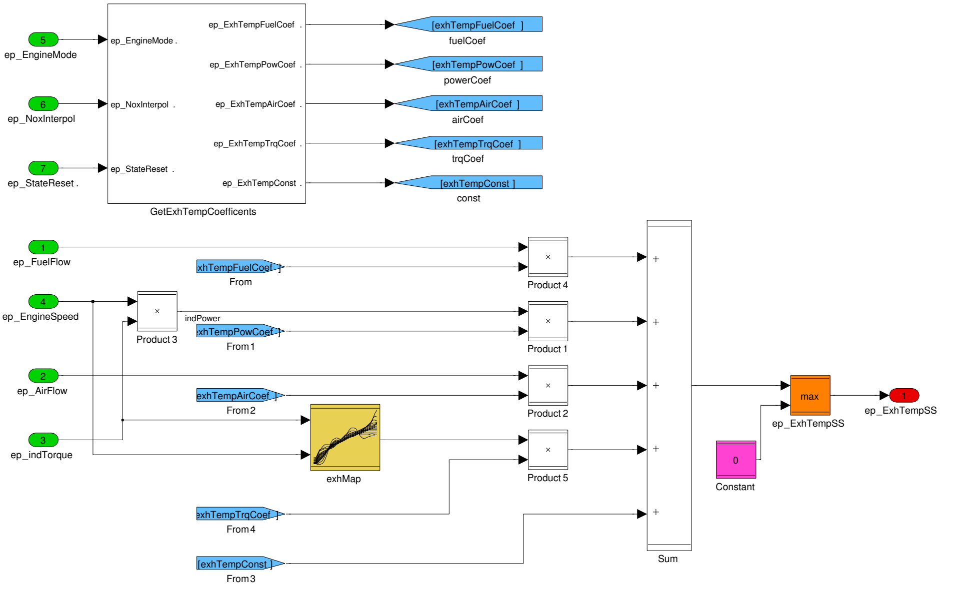

Figure 2.16: Simulink scheme of the steady state exhaust temperature, implementation of Equation (2.14). The output is input to the Simulink scheme in Figure 2.18.

The parameters to determine in the dynamic part of the model, Figure 2.18, is the time constants, τ_1 and τ_2 for both first order systems and the ratio between their outputs. The same criterion as in (2.18) is used for optimizing the dynamic model.

2.6.3 Validation

The exhaust temperature model is validated by feeding the input of the exhaust temperature Simulink model with measured signals and compare the output from the model with measured exhaust temperature. Figure 2.19 shows the simulation result compared to measured exhaust temperature. Figure 2.20 shows the simulation result compared to measured steady state exhaust temperature for one mode. When simulating the model for six different mode configurations the total result is an MARE of 5.4%. For this validation the dynamics of the model are not activated because the measurement points are not time dependent (steady state engine run). This validation is done to see if the errors derives from the steady state model or the dynamic model afterwards. To validate the dynamic model the WHTC is used and the model is run at 10 Hz. The cycle is divided into 20 second simulation intervals. The absolute difference between the last measured and simulated value for every simulation is considered. The maximum difference is 38 degrees Celsius

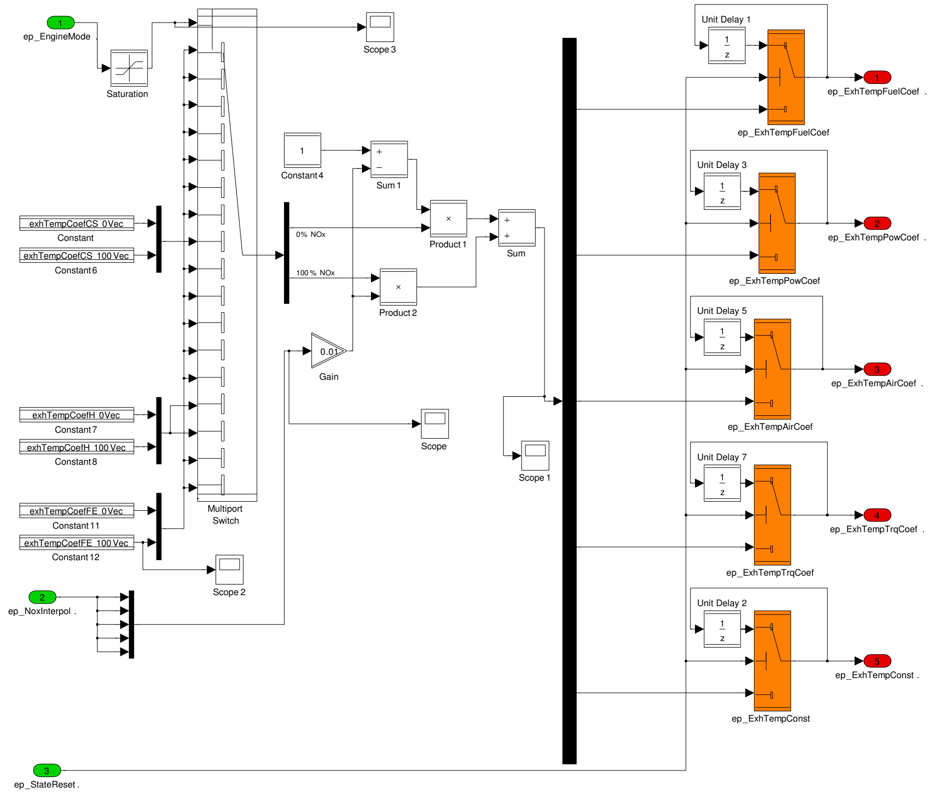

Figure 2.17: Simulink scheme of the block GetExhTempCoefficients in Figure 2.16. Initializes the model coefficients depending on NO_x production and engine mode.

and the mean difference for all simulations is 22 degrees Celsius. The MARE for the whole cycle is 5.8%. If this result is good enough depends on the application which at this time is not specified in detail.

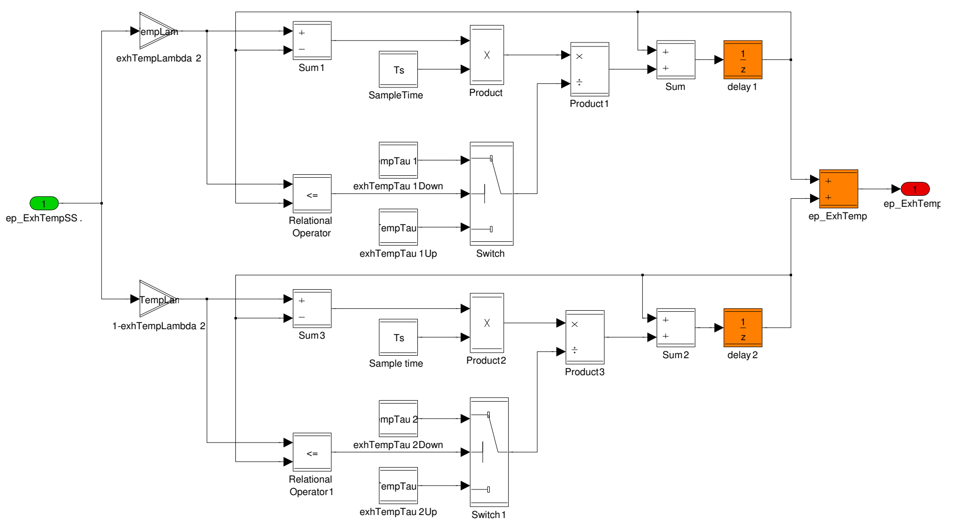

Figure 2.18: Simulink scheme of the dynamic exhaust temperature, implementation of Equation (2.15)-(2.17). A sum of two first order systems that filters the steady state exhaust temperature from the Simulink scheme in Figure 2.16.

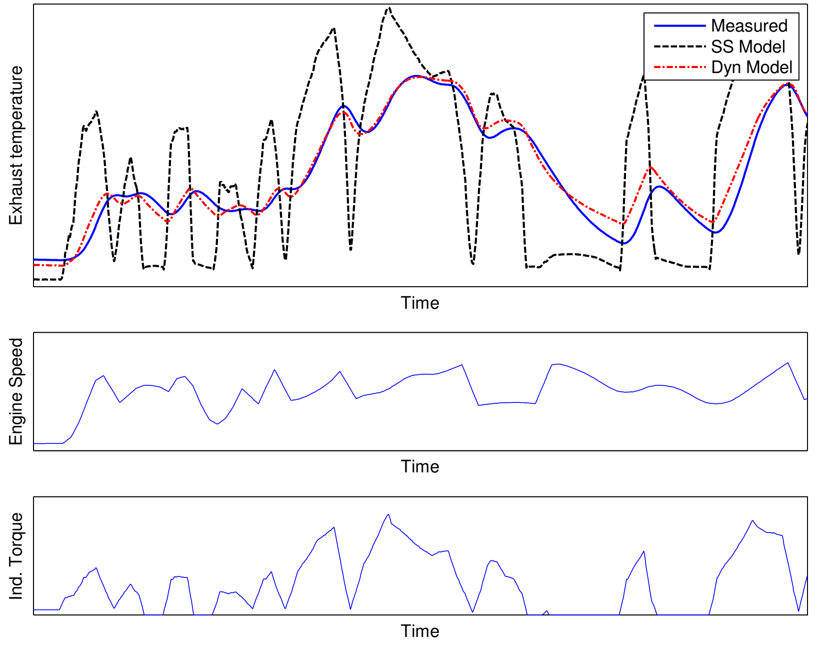

Figure 2.19: Simulated compared to measured exhaust temperature

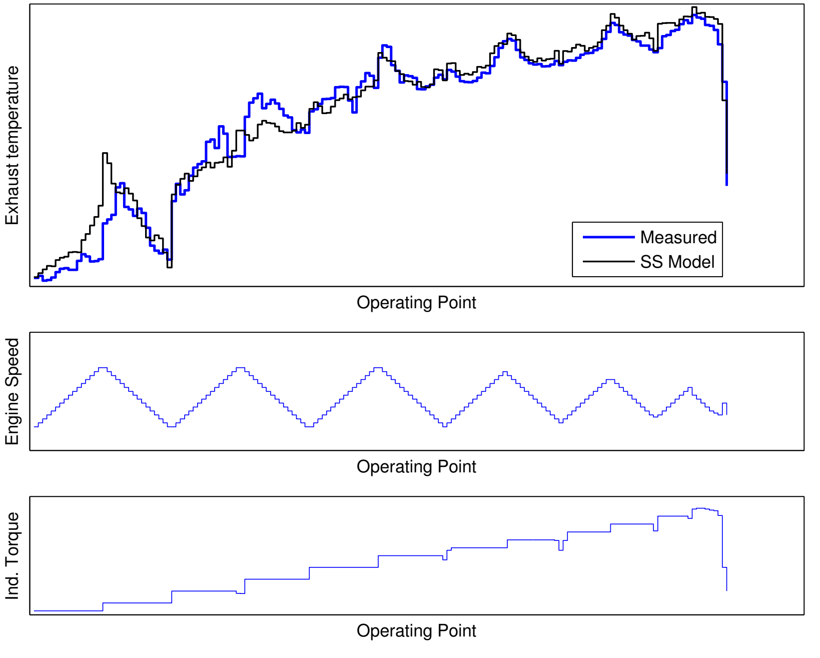

Figure 2.20: Simulated compared to measured steady state exhaust temperature.

Chapter 3

Model verification and results

In this chapter the total model is evaluated with the respect to brake torque and fuel consumption. For verification of exhaust temperature see Section 2.6. For validation the model is run and compared to measurements from an engine test cell. The inputs to the model are taken from the same test run. The measurements used for validation are independent from the measurements used for estimating the model. All simulations are over the same time span.

3.1 Brake Torque

In Figure 3.1-3.2 the result from four simulations compared to measurement of torque transients can be seen. In Figure 3.1 the AFR does not reach the limit in the smoke limiter, neither during data collection in the real system nor during simulation. In Figure 3.2 there are greater steps in pedal position and the AFR limit is reached both during data collection and simulation. The torque measurements are not corrected for rig inertia i.e. when the engine accelerates the rig measures a lower output torque than the actual. The MARE are 6.3, 4.6, 6.8 and 4.2% for simulations in 3.1(a) - (b) - 3.2(a) - (b) respectively.

To evaluate the torque model comparison with only measurements are not sufficient. In which context the model should be used and the external uncertainties that affects the model has to be taken in to consideration. Assume that an uncertainty in the estimated weight of the vehicle of 10% is translated to an uncertainty of inertia at the flywheel. The difference in torque that would be needed to accelerate the engine, see Figure 3.3, would be in the same range as the error between modelled and measured brake torque.

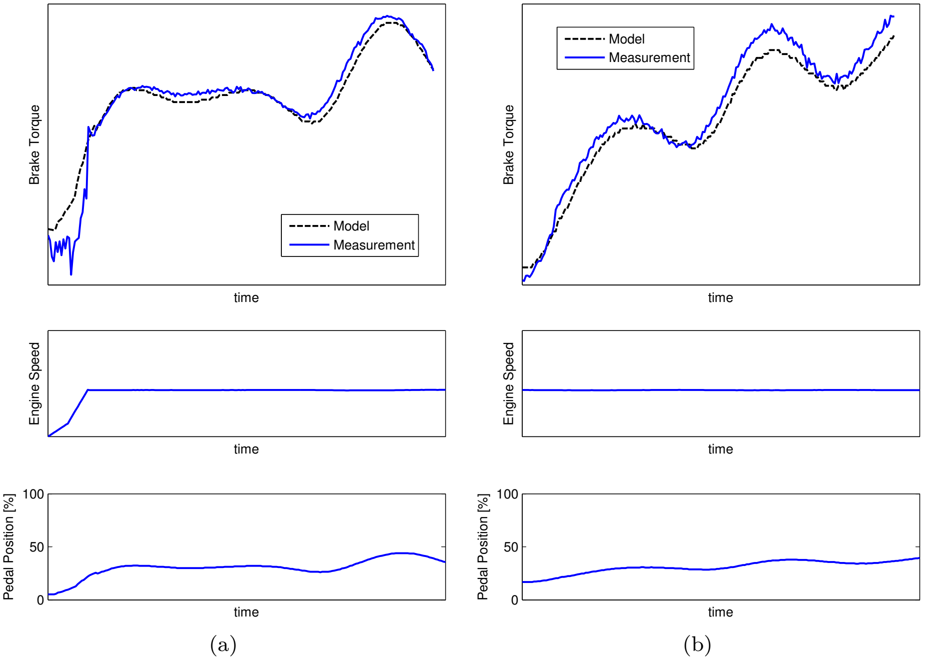

Figure 3.1: Brake torque transients in the BLB-cycle where the smoke limiter is active during the transients

3.2 Fuel Consumption

In Figure 3.4 the fuel consumption from two simulations can be seen, in Figure 3.4a and 3.4b the relative error is 7.8% respectively 1.0%. The fuel consumption model's purposes is to be able to evaluate the fuel efficiency in different gear shifts. It is important that the mutual order and ratio of fuel efficiency between different gears shifts are correct not the absolute value. There has been no data available to do this comparison in a good way, further testing must be performed to validate this.

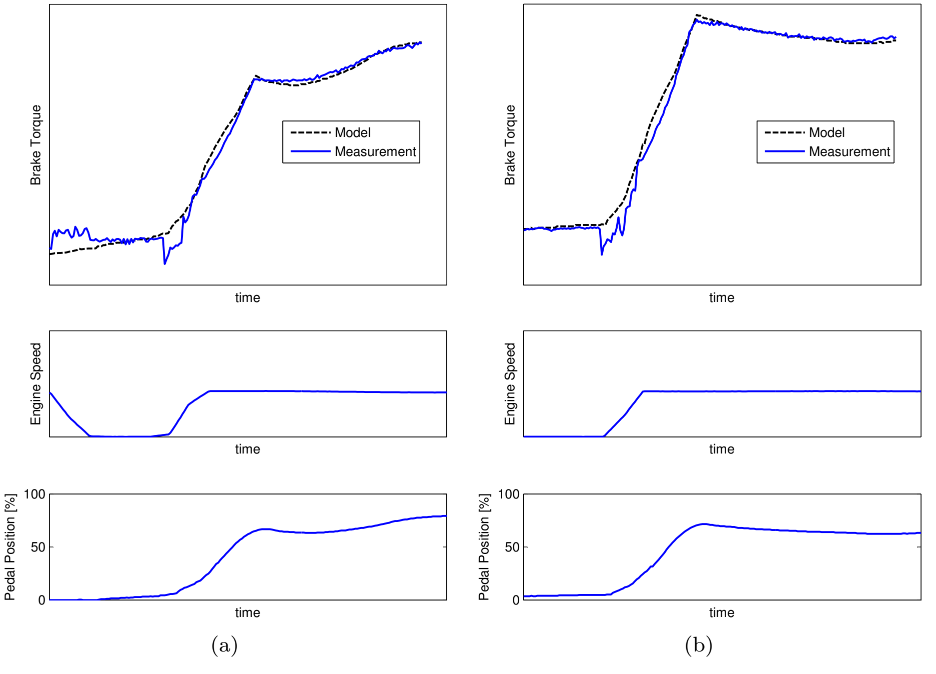

Figure 3.2: Brake torque transients in the BLB-cycle where the smoke limiter is active during the transients

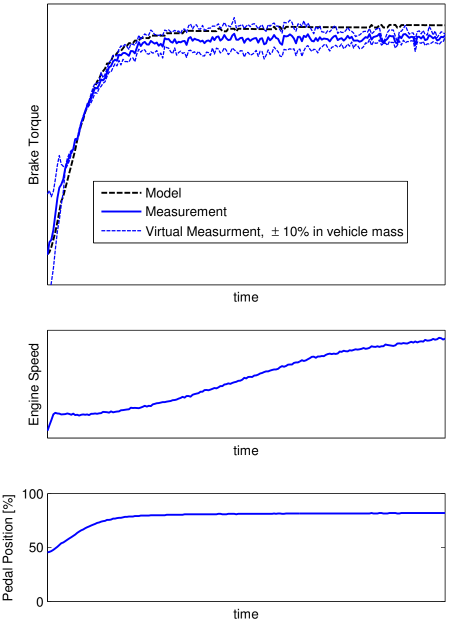

Figure 3.3: Shows how ±10% in mass affects the brake torque with given acceleration.

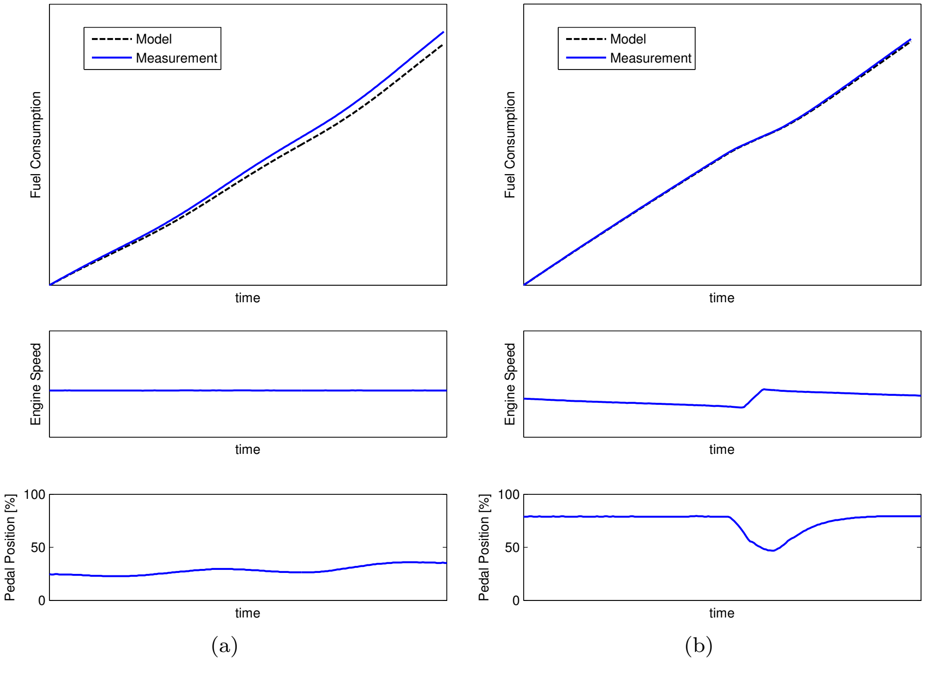

Figure 3.4: Fuel consumption in two sections of the BLB-cycle

Chapter 4

Summary and Conclusion

A computationally efficient mean value model of a Volvo 13 litre, six cylinder diesel engine and it's electronic control system is developed for use in a simulation environment on-board a vehicle. The properties modelled are torque, exhaust temperature and fuel consumption based on engine speed and pedal position. In this chapter a summary of the model, conclusions and proposed future work is presented.

4.1 Summary

The model contains sub-models that can be calibrated in a systematic way. Each sub-model has a clear purpose and is derived from components in the real system. The sub-models that are modelled are presented below.

Air Flow

The air flow model (see Section 2.1) consists of three parts. The first part is steady state inlet pressure based on indicated torque and engine speed. The second part is the the dynamics of the inlet pressure. The air mass flow in to the cylinders based on volumetric efficiency and the density of the inlet air is the third part. The sub-model estimates the inlet pressure and air mass flow with a mean absolute relative error of 3.5% and 3.2%.

Smoke Limiter

The smoke limiter (see Section 2.3) is simplified control software that limits the amount of injected fuel during torque transients to avoid smoke in exhaust gases.

Fuel Flow

The fuel flow model (see Section 2.4) consists of a polynomial with engine speed, indicated torque and power as variables. The brake torque model subtracts the mechanical losses in the engine from the indicated torque. When the sub-model is compared to measurements from a transient cycle the mean absolute relative error is 5.9%.

Exhaust Temperature

The exhaust temperature model (see Section 2.6) consists of two parts, a steady state part and a dynamic add-on. The steady state model consists of a map and a polynomial. The dynamic model uses the steady state value as input and filters it through two parallel first order systems. The mean absolute relative error of the exhaust temperature model compared to measurements in a transient cycle is 5.8%.

4.2 Conclusion

The model performs well consider the computational efficiency. The brake torque, fuel consumption and exhaust gas temperature have a mean absolute relative error lower than 6.8%, 7.8% and 5.8% respectively. The three outputs from the model describes the modelled system well enough to continue with testing in the embedded system.

4.3 Future Work

During the master thesis some interesting modelling and evaluation was discovered but the resources were limited. Below some interesting suggestions are discussed for future work to improve the model.

Turbo dynamics at high altitudes

The turbo dynamics changes when the engine drives in high altitude due to reduced density in the ambient air. The pressure build up gets slower at high altitudes and this affects the torque response in a negative way. Due to absence of high altitude data with the engine used for modelling no implementation and evaluation on this has been done.

EGR

The air mass flow (as function of pressure and engine speed) is modelled from steady state measurement in intake air flow and not from the total amount of gas that enters the combustion chamber. How large amount of the gases that goes in to the cylinder that is EGR is hard to estimate. A model over the EGR control system and how much EGR that enters the intake manifold would greatly improve the air mass flow estimate in very sharp transients which would give a better AFR estimate, which in turn would provide a better torque estimate.

NO$_x$

It would be interesting to model the amount of NO_x the engine produces to be able to select gears that minimizes NO_x-production when the SCR is cold.

Exhaust Temperature

It would be interesting to try a solution where a sum of several first order systems with different signals is used as inputs. The input signals could be all of the inputs to the sum block in the steady state exhaust temperature Simulink model (Figure 2.16). An alternative or complement to this could be to make the time constants in the current model dependent of some variable, e.g. engine speed. It would also be interesting to investigate the physical relations that affects the exhaust temperature more in detail. This to be able to implement the non-linearities better and to see if some other quantity is needed to be modelled in order to describe the exhaust temperature accurately.

Fuel Flow

To improve the fuel flow accuracy the fuel flow could be mapped in a lookup table for the operating points and the different modes. An interesting alternative to different lookup tables for different modes could maybe be to have a function based on mode and NO_x-production that compensates a base lookup table to save memory utilization.

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