Physical modeling of a heavy-duty engine for test-cycle simulations in Modelica

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Abstract

Engine development for reduced fuel consumption and for the ability to use alternative types of fuels is required as emissions and the limited access of petroleum are issues coupled to diesel engines. Experimental testing is in many cases insufficient and expensive in comparison with simulations for engine development. This thesis is focused on extending a model of a diesel compression-ignition engine for test-cycle simulations with a new cylinder model. The model was developed with Dymola executing code written in the language Modelica and with the external Engine Dynamics Library developed by Modelon AB. There were prior engine models in the Engine Dynamics Library but there was a need for a model extension primarily to get a model able to handle a larger operating range. The result of the thesis is an engine model that can execute simulations with zero speed and fuel injection that the previous models in Engine Dynamics Library could not handle. During model evaluation the model was compared to a calibrated map-based model at a test-cycle where the map-based model could be executed. The model developed during this thesis research was successfully able to capture much of the dynamics of the available map-based model.

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1

Introduction

Why does something exist rather than nothing?

Gottfried Wilhelm von Leibniz

The internal combustion engine was invented during the 19th century. More specifically, the compression-ignition engine, that is in focus of this thesis, was developed by Rudolf Diesel at 1892 [Heywood, 1988]. Combustion engines have played a major role in our society and are used in several areas. One of these areas is vehicle propulsion, especially heavy-duty vehicles. There are some drawbacks with the combustion engine that require future research and development. Drawbacks are the limited assets of petroleum, the cost of fuel and pollution of soot and oxides of nitrogen. Engine simulation is an important part of engine research and development and is used for different applications. The application the modeling in this thesis focuses on is during control design for engines. There are several tools for engine simulation where Modelon AB constructs models using the Modelica language. This thesis focuses on implementation of a compression-ignition diesel engine-model for simulations.

1.1 Problem formulation

Modelon AB had two models for simulations of compression-ignition diesel engines. One of these models was map-based and calibrated with respect to engine data. The other model was derived from physical principles but with large possibilities to be extended to include more aspects with more well-suited models for different mechanisms. The map-based engine model is used in executable simulation-examples in Engine Dynamics Library and was presented in [Nylén, 2015]. The primary limitation with the existing models developed by Modelon AB was that they could not execute when the engine speed or the fuel flow approached zero and therefore could not simulate for complete test-cycles where the engine speed or fuel flow was zero. As the cases with zero speed and fuel flow are important test cases it is crucial for the model to be able to handle these cases. The objective of this thesis was to expand a physical model in order to simulate the World Harmonized Transient Cycle that includes start and stop of the engine [DieselNet, 2007]. The model has furthermore been extended to better capture the engine dynamics and to give the possibility to use the model for analyzing how different additional changes of the engine, that were rejected before, affect the performance of the engine in simulations. One example is the injection timing that has been included in the extended model to capture the behaviour of energy losses depending on injection timing.

1.2 Material

For this project, computer, literature about Dymola and license for Dymola has been borrowed from Modelon AB. Lund University has provided license for MATLAB that has been used for some analysis to the thesis report.

1.3 Thesis Outline

Chapter 2 briefly explains how a compression-ignition engine works. In addition, comments about different approaches to engine modeling are given. The model of the engine that has been obtained in this thesis is described in Chapter 3. The simulation results of the model obtained are presented in Chapter 4. Chapter 5 contains additional comments on the results and the project.

2

Background

> You raise me up, so I can stand on mountains. You raise me up to walk on stormy seas. I am strong when I am on your shoulders. You raise me up to more than I can be.
>
> *Brendan Graham*

2.1 Internal combustion engines

Engine

The workflow of an internal combustion engine is shown in Figure 2.1. Air flows into the engine through a compressor and the air flow can be regulated by adjusting the position of a valve. The inlet air then mixes with recirculated exhaust gas, known as EGR (exhaust gas recirculation). The mixture of air and recirculated exhaust gas flows into the cylinder where it mixes with fuel. In the cylinder the fuel is combusted and the exhaust gases then flows to the exhaust manifold. From the exhaust manifold there is a fraction of the exhaust gas recirculated back to the inlet manifold by the pressure difference between the inlet and exhaust manifolds where the amount of recirculated gas is controlled with a valve. The part of exhaust gas that is not recirculated is released through a turbine and thereafter leaves the engine through the exhaust pipe. The turbine absorbs energy from the exhaust gas flow and uses it to let the compressor wheels' spin and build up a higher pressure in the inlet manifold resulting in a higher mass flow through the cylinder.

The work performed by the engine is obtained from the combustion in the cylinder that increases the pressure inside the cylinder leading to that the piston is accelerated downwards so that the gas inside the cylinder expands. The piston movement drives the crankshaft in a rotating motion. Another result of the combustion process is that heat is released to the engine block and away to the surroundings. For more information about the working principles of a compression-ignition engine, see [Guzzella and Onder, 2009] and [Heywood, 1988].

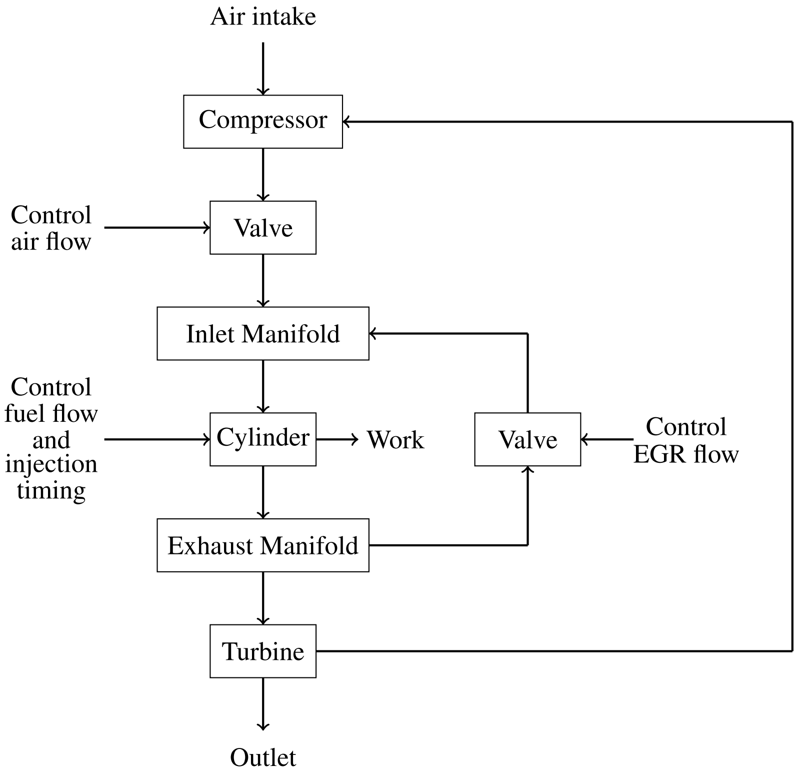

Figure 2.1 Block diagram of an internal combustion engine. The engine is of compression-ignition type and includes a turbocharger and the feature of exhaust gas recirculation.

Cylinder

The main work of this thesis was done updating the cylinder model. The cylinder is where the combustion occurs. The compression-ignition engine receives air drawn into the cylinders where it is compressed so that the temperature rises. Fuel is then injected and explodes shortly after due to the high temperature and pressure of the compressed air. Another way to ignite the fuel is by using an external ignition system that helps to start the combustion. Such engines are referred to as spark-ignited engines. The focus in this thesis is, however, a heavy-duty compression-ignition engine. Compression-ignition engines can be of two-stroke or four-stroke type. In this thesis a four-stroke engine is considered. For one power-generating cycle in a four-stroke engine the crankshaft rotates two revolutions. Description of one power generating cycle is done as follows. Suppose that one cylinder has its piston in the top position where the free cylinder volume is minimal and the exhaust gas has been released. Then the piston moves towards its lower position and draws new air into the cylinder. After the intake stroke the air-intake valves are closed and the piston moves up again, driven by the crankshaft so that the volume decreases which leads to an increase in pressure and temperature. When the piston is close to the top position the fuel is injected and after a short delay it burns so that the pressure and temperature increases even more. The in-cylinder gases then performs work on the piston during its way down to the bottom position. Then the piston moves up again while the exhaust valves are open and so the exhaust gas is released. When the piston is in the top position again there is only a small amount of exhaust gas left in the cylinder and this is where this description of the combustion cycle started.

The engine considered in this thesis is assumed to have six cylinders which is common for heavy-duty diesel engines [AB Volvo, 2018]. For more information about the four-stroke compression-ignition cycle, see [Turesson, 2018] and [Heywood, 1988].

Inputs and outputs

The engine in Figure 2.1 has four inputs that can be controlled. These inputs are the valve controlling the air flow into the engine, the amount of fuel injected, the timing for fuel injection and the valve controlling the EGR. There are additional inputs that can be controlled in a compression-ignition engine, but in this thesis, the engine model is according to the description in this chapter. The outputs for which the engine is controlled are the torque acting on the crankshaft driving the vehicle and efficiency of energy obtained by fuel (to obtain maximum use of the fuel used).

2.2 Modeling of internal combustion engines

When it comes to modeling of systems there are different approaches. Models can be made at different complexity, in the sense of the number of states in the model. Some dynamics of the system are required to be described in an accurate way by the model and some dynamics are less important to capture. The calculations furthermore needs to take reasonably short execution time for many applications. Therefore, the models used are sometimes not the models best capturing the behavior of the system but instead simpler and computationally faster models. One way of making the models simpler is to make all parameters and variables given an average value over an entire cycle of the combustion engine. Such models are called *mean-value engine models* and are often used for control. The opposite, where the parameters and variables change according to the cycle, are called *cycle-resolved models*, usually used for physical engine design. Mean-value engine models are often used, have been studied by many and are the ones that are in focus in this thesis. For further reading about mean-value engine models, see [Eriksson and Nielsen, 2014] and [Guzzella and Onder, 2009].

Models can, moreover, be map-based or physics-based as described in [Eriksson and Nielsen, 2014]. Physics-based modeling means that there are equations that needs to be evaluated to obtain the value of a variable from the values of parameters and other variables. In map-based modeling the equations are replaced with a map, sometimes called look-up table, so that the value of variables are given by reading values of some other known variables that are then inserted to a map that yields the value for the modeled variable. There can be a mix of physics-based and map-based variables in the same model. Maps are usually obtained direct from measured system data. Calibration of maps is simple and gives good accuracy for a certain component (if calibrating for many points) but it can consume much time. Maps are furthermore easy to use in implementation and are computationally efficient. One drawback with maps is that if the engine is redesigned they may have to be re-calibrated as extrapolation with maps often gives weak results and re-calibration takes time. Maps are often bad when there are differences in the component due to production or if components are changed. The execution time for evaluating the equations in physics-based models can in some cases be a problem, for instance, if used online in a controller. If the execution time for a physics-based model is an issue and the equations take long time to solve, physics-based models can be used offline to calculate values to a map that then can be used online to (potentially) give the value of the searched variable faster. The benefits of extrapolation and capturing component differences due to production are sometimes wanted and therefore the physics-based model was developed in this thesis.

2.3 Software

Modelica was used in this thesis for engine modeling and simulation. Modelica is an open-source modeling language developed by the Modelica Association since 1996 [Otter, 2013]. Modelica is an object-oriented language. The program Dymola was used as a simulation environment, wherein the Modelica language was used. The engine model uses many of the features from the Modelon Base Library and the Engine Dynamics Library that are both developed by Modelon AB [Modelon AB, 2018]. One advantage with the models developed in Modelica is that one part in a model, can be exchanged for another one if they have the same connections. This means that if one engine is modeled then one part can be exchanged. The old model can be reused with only the part exchanged needed to be exchanged in the model to be simulated. For instance, in the engine model shown in Figure 2.3 the cylinder component could be exchanged for another one with the same connections. This is useful as smaller parts can get their parameters identified and then put in the larger model as there can be much work needed to identify all parameters at once for large models [Eriksson and Nielsen, 2014]. The engine models in the Engine Dynamics Library are of mean-value type and therefore are suitable for engine optimization and evaluation of control strategies [Dahl and Andersson, 2012].

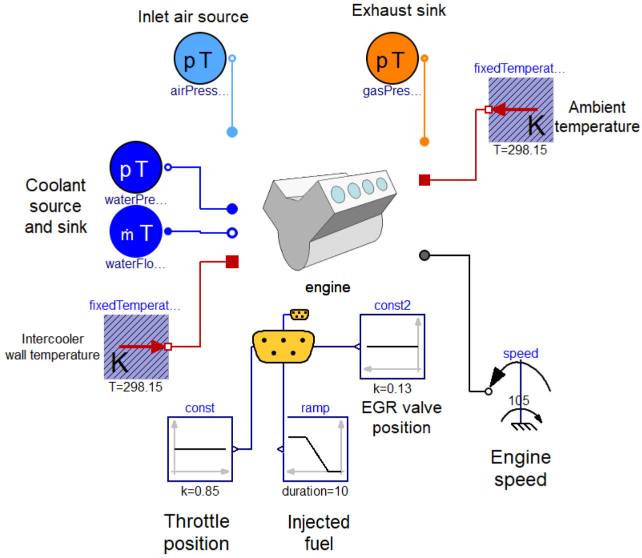

Figure 2.2 Experiment model for the original map-based engine taken from the Engine Dynamics Library.

Modelica is sometimes used for rather large models, like the combustion engine system described in this thesis with thousands of variables. Therefore, it is important to have a suitable model structure. Usually there is a test class that is to be executed to run a test and that consists of several components that are their own classes. Every component can then consist of more sub-components. There can be parameters declared in a parent class that are inherited in a child class. It is possible to replace a sub-class used in a component if properly implemented by specifying that when the component is included. Another possibility is to make a test bench for a component where other parts with desired connectors are specified and where the component can be tested. Test benches can furthermore be used for model-parameter calibration.

In Modelica, a common model structure is that there are models that can be simulated and these are usually called *experiments*. One experiment that is implemented in the Engine Dynamics Library is shown in Figure 2.2. In an experiment model, there can be several components that act as boundary conditions to one or several components that symbolize the system to be modeled. The experiment in Figure 2.2 has several components describing boundary conditions for temperature, pressure, speed and control signals that are all connected to the engine component. The control inputs are the throttle position for the valve for the air intake, the flow

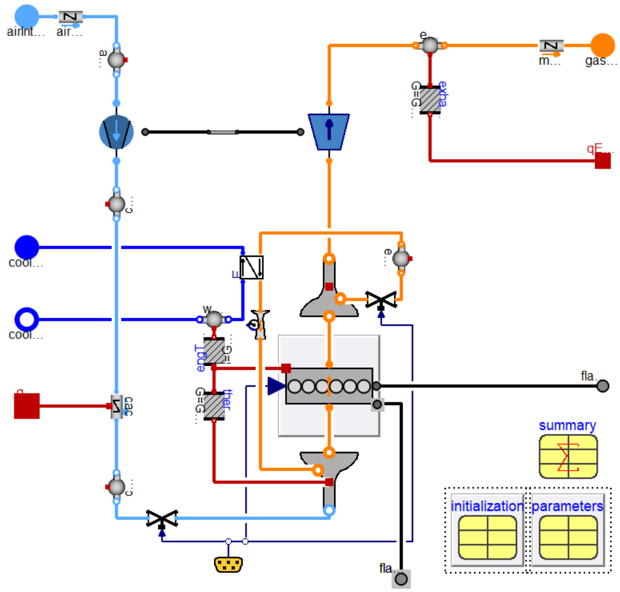

Figure 2.3 Engine model for the original map-based engine taken from the Engine Dynamics Library. The air comes in through the light blue connector in the upper left corner. The air goes through the compressor component that is connected to the turbine with a shaft. There is some heat exchanged between the air and the tubes, where the red lines indicate heat flow. There is a valve for the air intake before the air reaches the inlet manifold where the light blue line ends. The inlet manifold has some heat exchanged and some EGR-gas incoming from left and out to the cylinder (up) goes the mixed air and EGR-gas. The cylinder has fuel flow as an input and has moreover heat exchanged both to the inlet manifold and to the cooling system. The cylinder is connected with a rotating shaft that gets a speed input stating the speed of the crankshaft. Exhaust gas flows out from the cylinder (upwards) to the exhaust manifold where some part of the gas flows through a valve and heat exchange-component back to the inlet manifold. From the exhaust manifold another part of the gas is flowing out upwards through the turbine and out to the connector in the upper right corner.

of fuel injected and the EGR-valve position.

The engine component cannot be simulated alone as there are too many unknown parameters to give a solution to the system of equations for all variables when the boundary connectors are not connected to appropriate components. The engine component is shown in Figure 2.3. The engine component contains some connectors that are taking values from the components connected in the model where the engine component is used. Then, the engine component is built up by

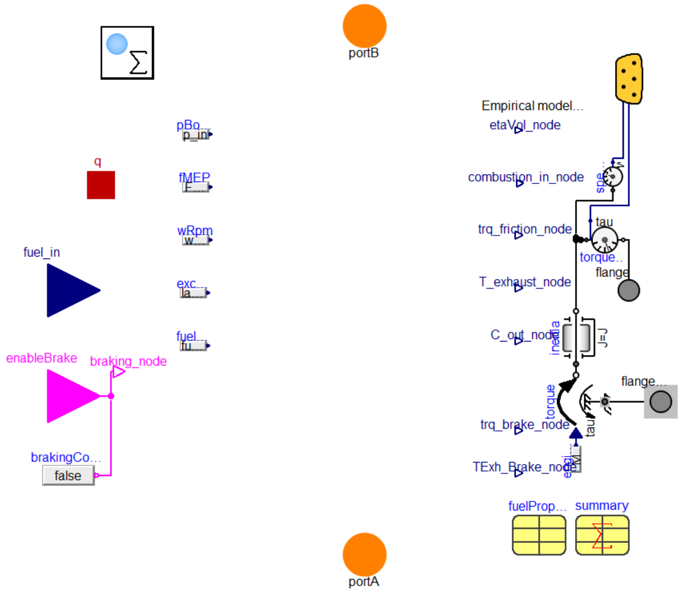

Figure 2.4 Cylinder template with the basic structure of the cylinder components. The template has several connections to be connected and that do not need to be specified in each cylinder model. There are some records specifying fuel properties and some basic variable properties. There are some variables obtaining values from the inputs to the connectors in the right side (for instance etaVol_node in upper right corner should get a input specifying the volumetric efficiency that will be discussed more in Chapter 3.

several sub-components symbolizing compressor, turbine, cylinder etc. The tubes where gas flows are modeled by small volume components and small flow components. Some parameters are collected in data records placed in the lower right corner of Figure 2.3. The two parts where most effort has been made during this thesis are the cylinder and the turbine. The cylinder has connectors for heat exchange, input stating fuel flow, shaft connection, and gas connections for input and output gas. The turbine has input gas connection, output gas connection and one connection to a rotating shaft.

Another important concept in Modelica is model *templates*. Model templates are used as a collection of basic properties of components. One example is the base class for the cylinder that was used to get the same basic structure of all cylinder components. The template includes many common variables and specifies some relationships, for instance, what variable specifies the torque affecting the crankshaft. The template for the cylinder is shown in Figure 2.4. Templates are usually used by a component class that extends the template class and can give values to the free variables of the template by use of maps or equations. In this thesis, the cylinder model was built up with a structure as close to the cylinder models available in the Engine Dynamics Library as possible to make it easy to change between different cylinder models in an engine model and to have many properties collected in a cylinder template made that smooth.

3

Modeling

> A lot of people in our industry haven't had very diverse experiences. So they don't have enough dots to connect, and they end up with very linear solutions without a broad perspective on the problem. The broader one's understanding of the human experience, the better design we will have.
>
> *Steve Jobs*

3.1 Cylinder

The cylinder-output variables considered in this thesis are the indicated gross efficiency η_{ig}, the exhaust temperature T_{eo}, the torque output affecting the crankshaft M_{e} and the output mass flow $\dot{m}_{\text{out}}$; η_{ig} is modeled to be used for computing M_{e}. The outputs that affect the performance of the other parts of the engine are M_{e}, $\dot{m}_{\text{out}}$ and T_{eo} which makes it necessary to model these variables. These variables depend on cylinder boundary conditions such as pressure in the inlet manifold p_{in}, pressure in the exhaust manifold p_{out}, temperature in the inlet manifold T_{in}, relative air-to-fuel ratio λ, speed of crankshaft w, fuel flow $\dot{m}_{\text{f}}$ and the angle of start of injection of the fuel for each cycle θ_{SOI}. A schematic picture of inputs and outputs for the cylinder model and the states that are used for calculations in the cylinder model is shown in Figure 3.1.

There are several variables that need to be modeled to compute the output variables considered. There are different models for these variables to approximate the real processes and from these models there have been made choices described in this section.

First, the volumetric efficiency, denoted η_{vol}, was modeled. The volumetric efficiency represents how much of the combusted exhaust gas that is replaced by new air during the cycle. The model for volumetric efficiency is taken from [Eriksson

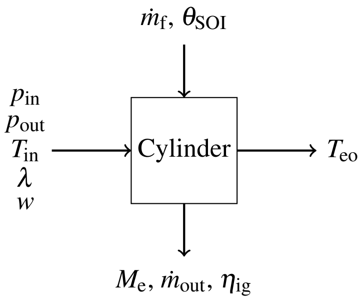

Figure 3.1 Cylinder model with inputs and outputs. The inputs and outputs in vertical direction symbolize physical variables and the horizontal direction has states that does not symbolize a flow directly.

and Nielsen, 2014, p. 160] and is given by

$$\eta_{vol}(p_{im}, p_{em}) = C_{\eta_{vol}} \frac{r_c - \left(\frac{p_{em}}{p_{im}}\right)^{1/\gamma}}{r_c - 1} \tag{3.1}$$

Here, $r_c = (V_d + V_c)/V_c$ is the compression ratio with V_d denoting displacement volume and V_c denoting minimum cylinder volume (clearance volume), $\gamma = c_p/c_v$ is specific heat ratio and $C_{\eta_{vol}} = 0.95$ is a calibration parameter that is usually a function of engine speed.

It is necessary to describe the in-cylinder conditions prior to combustion in order to describe the combustion and specifically the ignition delay, well. The volume of the cylinder for the crank angle when the combustion approximately occurs (the combustion is quite fast and therefore it is approximated to take zero time here) can be calculated from the angle for where the combustion occurs according to [Heywood, 1988, p. 44]

$$V = V_C + \frac{V_C}{2}(r_c - 1)\left(R_r + 1 - \cos(\theta) - \sqrt{R_r^2 - \sin^2(\theta)}\right) \tag{3.2}$$

Here, R_r is the ratio of the connecting-rod length to the crank radius and is here chosen to $R_r = 4$ ([Heywood, 1988, p. 45] has $R_r = 3.5$ in one example). The in-cylinder pressure p was modeled assuming an adiabatic process during compression (assuming no heat exchange with the cylinder walls) the following equation can be used according to [Eriksson and Nielsen, 2014, p. 88]

$$pV^{\gamma} = C \tag{3.3}$$

Here, C is a constant value. So, the constant value can be obtained from taking p_{in} (inlet pressure), $V = V_C + V_D$ and the value of γ i.e. ratio of specific heat and then the volume of the start of injection angle can be used to calculate the pressure of when the fuel is injected with Equation (3.3). The pressure of injection time is then used to calculate the ignition delay according to Equation (3.6). So, the pressure at the time of combustion p_{CT} can be calculated as

$$p_{\text{CT}} = p_0 \left(\frac{V_{\text{C}} + V_{\text{D}}}{V_{\text{CT}}} \right)^{\gamma} \tag{3.4}$$

where p_0 is the pressure at the inlet manifold that is approximately the pressure when the air is flowing in to the cylinder when the cylinder is open against the inlet manifold and where V_{CT} is the volume in the cylinder at the combustion time (depending on the angle of the crankshaft for when the combustion occurs). Here the assumption of that the cylinder starts compressing the new air from the bottom position of the piston is made. This is seldom the case as the compression can sometimes start when the piston has started to move up. Then $V_{\text{C}} + V_{\text{D}}$ should be replaced with the volume when compression starts. If the angular deviation of crankshaft for this time from the bottom position is known this could be used to calculate the cylinder volume and replace $V_{\text{C}} + V_{\text{D}}$. From this equation the equation for the temperature can be calculated with the use of the ideal gas law ($pV = nRT$) and yields

$$T_{\text{CT}} = T_0 \left(\frac{V_{\text{C}} + V_{\text{D}}}{V_{\text{CT}}} \right)^{\gamma - 1} \tag{3.5}$$

where T_{CT} is the temperature at the time of combustion and T_0 is the temperature from the inlet manifold that becomes the temperature of the gas inside the cylinder for a short time when the gas in the cylinder is exchanged. Here again the assumption of compression starting at bottom piston position is done and the volume $V_{\text{C}} + V_{\text{D}}$ should be changed if this is not the case.

It takes some time between the fuel is injected into the cylinder and the combustion starts. This is called ignition delay, denoted τ_{id}, and is modeled with the Arrhenius equation according to [Heywood, 1988, p. 543–545] as

$$\tau_{id} = A p_{\text{CT}}^{-n} e^{E_{\text{a}}/RT_{\text{CT}}} \tag{3.6}$$

Here, E_{a} is an activation energy for fuel, R is the universal gas constant, A and n are constants dependent on fuel with parameter values given in [Heywood, 1988, p. 543–545]. The ignition delay would get a more accurate value if using cycle-resolved calculations so that the pressure and temperature would be functions depending on time. But to avoid these extra equations in order to have a computationally faster mean-value model, the pressure and temperature values used are the values in the injection time. Injection typically occurs prior to top-dead-center and thus the piston is moving towards the position where the cylinder volume is minimal. The piston moves up to the top position and then down again so the cylinder volume is decreasing first and then increasing again. The implementation in this thesis has start of injection at top dead center, when the piston is in top position and the cylinder volume is minimal. The ignition delay is then calculated with the pressure and temperature taken from this point. As the pressure and temperature are higher in the top dead center then at the rest of the cycle this gives a shorter ignition delay then what could be expected for the real process. It is however expected to give better results to have a model yielding a shorter ignition delay than in the actual case then to have a model that does not include the ignition delay at all.

The crank angle of combustion timing θ_{CT} is given when ignition delay is converted to crank-angle degrees as $\theta_{\mathrm{id}} = v_{\mathrm{e}} \cdot \tau_{\mathrm{id}}$ where v_{e} is the velocity of the engine and if denoting the angle where the start of injection occurs θ_{SOI} and then the angle where the combustion approximately appears is given as

$$\theta_{\mathrm{CT}} = \theta_{\mathrm{SOI}} + \theta_{\mathrm{id}} \tag{3.7}$$

The combustion takes some time so the crankshaft can possibly move during the combustion but if the angle moved is small then the notation for the angle of where the combustion (approximately) appears is θ_{CT}. The optimal combustion timing is located shortly after the top-dead-center position so the volume is small and thus the pressure is high and then the energy from the fuel is used in an efficient way to apply a force on the crankshaft. There are, however, higher heat losses when the combustion occurs close to top-dead-center and the optimal combustion timing is thus a trade-off with respect to these aspects. The optimal angle is denoted θ_{opt}. A model for the efficiency of energy obtained due to combustion angle is denoted η_{ign} and given by Equation (3.8) [Eriksson and Nielsen, 2014, p. 190].

$$\eta_{\mathrm{ign}} = 1 - C_{\mathrm{ign},2} \cdot \left(\frac{\theta_{\mathrm{CT}} - \theta_{\mathrm{opt}}}{100} \right)^2 \tag{3.8}$$

Here, $C_{\mathrm{ign},2} = 4.316$ to give an example and was identified for data from a spark ignition engine [Eriksson and Nielsen, 2014, p. 190]. Heat losses are neglected in the model for ignition timing efficiency. The parameter for the model of the ignition timing efficiency needs to be identified for the engine modeled as the model depends on this. The model for ignition timing, however, has the same structure as the model in Equation 3.8, that with the given parameter value gives the plot shown in Figure 3.2.

The efficiency of ignition angle is then used to calculate the efficiency for how much energy it is possible to get out of the fuel. This efficiency is named indicated gross efficiency and denoted η_{ig} given by the following function [Eriksson and Nielsen, 2014, p. 187–188]

$$\eta_{\mathrm{ig}} = (1 - \frac{1}{r_{\mathrm{c}}^{\gamma-1}}) \cdot \min(1, \lambda) \cdot \eta_{\mathrm{ign}} \cdot \eta_{\mathrm{ig,ch}} \tag{3.9}$$

Here, λ is the relative air-to-fuel ratio, η_{ign} is the efficiency of energy obtained depending on combustion timing and $\eta_{\mathrm{ig,ch}} = 0.6894$ includes the remaining losses,

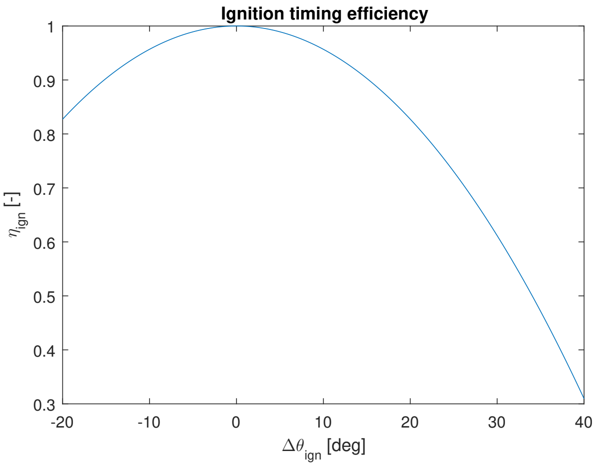

Figure 3.2 Ignition timing efficiency as a function of ignition angle difference from optimal angle. Heat losses are neglected in the model for ignition timing efficiency.

mainly heat losses, and is given same value as used in a MATLAB Simulink model [Wahlström and Eriksson, 2013] which is described in [Wahlström and Eriksson, 2011].

The gross indicated work $W_{\mathrm{i,g}}$ is the work from the fuel that affects the crankshaft and is given by [Eriksson and Nielsen, 2014, p. 187]

$$W_{\mathrm{i,g}} = m_{\mathrm{f,cycle}} Q_{\mathrm{LHV}} \eta_{\mathrm{ig}} \tag{3.10}$$

Here, $m_{\mathrm{f,cycle}}$ is mass of fuel used during one entire power generating cycle in the motor ($m_{\mathrm{f,cycle}} = \dot{m}_{\mathrm{f}} \cdot (2\pi n_r)/v_{\mathrm{angle}}$), Q_{LHV} is a constant depending on fuel (lower heating value).

There is friction affecting the system and therefore the total friction work is computed and the model for this is taken from [Heywood, 1988, p. 724]

$$W_{\mathrm{tf}} = \left(C_1 + 48\frac{N}{1000} + 0.4S_{\mathrm{p}}^2\right)/1000 \tag{3.11}$$

Here, $C_1 = 75000$ and S_{p} is the mean speed of the piston [m/s] calculated by $S_{\mathrm{p}} = N \cdot 2 \cdot L_{\mathrm{pm}}/60$ where N is the engine speed in revelations per minute and $L_{\mathrm{pm}} = 0.142$ is the distance the piston can move (so for one revelation it moves 2 times the distance L_{pm}); the parameter values were chosen according to [Heywood, 1988, p. 724].

The pumping work was modeled according to [Eriksson and Nielsen, 2014, p. 190] as

$$W_{\mathrm{i,p}} = V_{\mathrm{D}}(p_{\mathrm{em}} - p_{\mathrm{im}}) \tag{3.12}$$

Then the torque of the engine affecting the crankshaft M_e can be calculated according to [Eriksson and Nielsen, 2014, p. 187] as

$$M_e = \frac{W_e}{n_r 2\pi} = \frac{W_{i,g} - W_{i,p} - W_{fr}}{n_r 2\pi} \tag{3.13}$$

Here, $W_{i,g}$ is the indicated gross work, $W_{i,p}$ is the pumping work and W_{fr} is the friction work; n_r is the number of crank revelations in the complete power generation cycle, i.e., 2 for four-stroke engines as in this case.

The exhaust-gas temperature, T_{eo}, is obtained from the following equation [Eriksson and Nielsen, 2014, p. 194]

$$T_{eo} = \frac{1}{(\dot{m}_{ac} + \dot{m}_f)c_p} \Big(\dot{m}_{ac} c_p T_{im} + \dot{m}_f c_p T_f + \dot{m}_f Q_{LHV} \eta_\lambda - x_e \dot{m}_f h_{fg} - \dot{m}_f Q_{LHV} \eta_{ig} - \dot{Q}_{ht} \Big) \tag{3.14}$$

Here, mass flow through engine $\dot{m}_e = \dot{m}_{ac} + \dot{m}_f$ equals mass flows of air and fuel added. The variable T_{im} is temperature of air from inlet manifold; T_f is temperature of fuel injected to cylinder; c_p is constant for specific heat at constant pressure; Q_{LHV} is lower heating value and a constant depending on fuel. The efficiency η_λ comes up due to that it is not always possible to get all the energy out from the fuel due to that there is not enough air in the cylinder for all the injected fuel to be combusted and is here modeled as $\min(1, \lambda)$ where λ is the relative air-to-fuel ratio. The energy loss in evaporation of fuel is denoted $x_e \dot{m}_f h_{fg}$. For diesel the vaporization enthalpy is $270\,000\,\mathrm{J\,kg^{-1}}$ so the work loss due to vaporization is 270000 Joule per kilogram of fuel vaporized, i.e., $x_e \dot{m}_f h_{fg} = 270000 \dot{m}_f$ [Heywood, 1988, p. 915].

$$\dot{m}_f Q_{LHV} \eta_{ig} + \dot{Q}_{ht} = W_{i,g} \tag{3.15}$$

is the gross indicated work as the two terms symbolize energy obtained from fuel and heat losses respectively and are both included in the formula for indicated gross work that includes the factor of indicated gross efficiency.

This cylinder model was also extended with expanded calculations for mass flow and air fraction to include the limitation of the amount of fuel combusted due to the limited amount of oxygen in cylinder.

3.2 Turbine

For the turbine, there was a simple physical model implemented in the Engine Dynamics Library. That model could not handle several special cases occuring in the test-cycles and was therefore adjusted. The mass-flow model that was implemented according to [Guzzella and Onder, 2009, p. 52] was

$$\dot{m} = \frac{p_{in}}{\sqrt{T_{in}}} \cdot C_t \cdot \sqrt{\left| 1 - \left(\frac{p_{out}}{p_{in}} \right)^{k_t} \right|} \tag{3.16}$$

where $\dot{m}$ denotes mass flow, p_{in} is the pressure before the turbine, p_{out} is the pressure after the turbine, T_{in} is the temperature of the gas before the turbine, $k_t = 2.2$ is a constant chosen according to [Guzzella and Onder, 2009, p.52] and C_t is a constant obtained by tuning. The special case if the pressure before the turbine is lower then after is especially handled by setting the mass flow to zero in that case. The absolute value is not included in the model in [Guzzella and Onder, 2009] but inserted to ensure that the square root will not give imaginary values, even though that should not be possible with the current handling for the pressure difference that sets the mass flow to zero for $p_{\text{in}}/p_{\text{out}} \leq 1$.

There is also an efficiency stating how much of the potential energy the turbine performs at the shaft connected to the compressor. This efficiency is here called isentropic efficiency. The model for isentropic efficiency is taken from the simple model in the Engine Dynamics Library and it has a tuning parameter that symbolizes the radius of the turbine wheel here denoted r_{t}.

3.3 World Harmonized Transient Cycle

The model was tested on the World Harmonized Transient Cycle (WHTC) that has trajectories for desired speed and torque to be tested on as seen in Figure 3.3. To simulate the engine models on the WHTC test-cycle there was a reference generator implemented.

Reference generator

The reference generator reads values specifying reference trajectories from files and translates the signals to be compatible with the engine models. The speed trajectory is read and put as a boundary condition to the model. The torque trajectory needs to be translated in the reference generator to give a value of how much fuel that needs to be injected for this torque to occur as that is the variable that can be controlled. This has been done in this thesis with the following formula derived from [Lanský, 2008, p. 21–31]

$$\dot{m}_{\text{f}} = a \cdot \left(\frac{\pi \cdot n_{\text{r}}}{i_{\text{v}} \cdot i_{\text{p}} \cdot V_{\text{d}}} \right) M_{\text{e}} + b \tag{3.17}$$

Here, a is a parameter that should be identified for the engine and has been chosen to 0.000005 to get a fuel flow that is approximately as large as for an old example with a constant velocity that were implemented before this project started in the Engine Dynamics Library. b was chosen to 0.0001. $i_{\text{v}} = 4$ is the number of cylinders, $i_{\text{p}} = 1$ is the number of pistons per cylinder and $n_{\text{r}} = 2$ is the number of strokes per cycle.

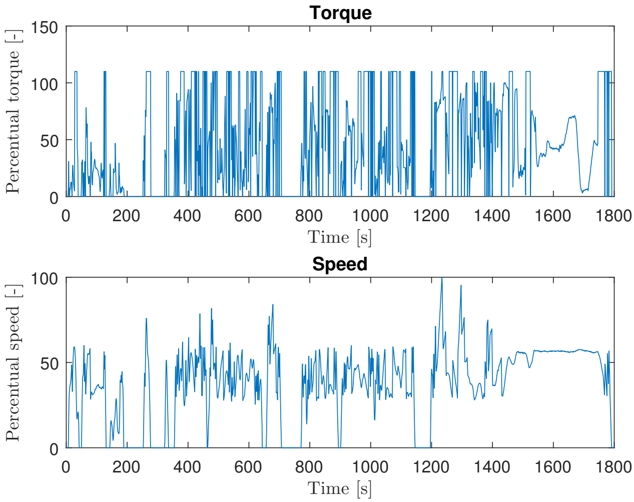

Figure 3.3 World Harmonized Transient Cycle with specifications for torque and speed out of maximum for the engine tested. The torque reaches the value 110 sometimes during the cycle which is used to denote that the engine is to operate in open rack for these parts. This has been treated by replacing all values where the engine operates in open rack with the torque value 0 before the test-cycle was used.

4

Results

> Then you will know the truth, and the truth will set you free.
>
> John 8:32

Both the models implemented during this thesis and the models available in the Engine Dynamics Library have been simulated on experiments with the same boundary conditions. The simulation results have then been compared in order to evaluate different models.

4.1 Model comparison

Model parameters

An efficient way to evaluate the models would have been to perform optimization with respect to model parameters. Methods for system identification are described in, for instance [Johansson, 1993]. This was, however, not included in this thesis and instead many parameters have been taken from previous literature, as described in Chapter 3. For the turbine two parameters were manually tuned, done using a part of the test-cycle WHTC (World Harmonized Transient Cycle) where all models could execute.

Testing conditions

WHTC contains trajectories for torque and speed in percent of maximum. The engine models all have speed and fuel flow as inputs. To run the model with the WHTC test-cycle the speed and fuel flow need to be calculated from the torque and speed trajectories in percent. The speed trajectory that was in percent was used as input to the engine stating the crankshaft velocity in radians per second. So, for 100 percent speed in reference the speed is set to 955 revolutions per minute. This could be changed if the speed reference would be multiplied with a scaling factor so that the speed trajectory would be different but in this test, it was used without scaling.

Table 4.1 Abbreviation of the different engine models evaluated and specification of their cylinder and turbine models. 'EDL' stands for Engine Dynamics Library and means that this model is included in the Engine Dynamics Library. 'Extended' means that the model was extended during this thesis. For the turbine models 'Map-based' means that the turbine model used is the map-based model taken from Engine Dynamics Library. The turbine model called 'Physical' is a model extended during this thesis from a physical model in the Engine Dynamics Library.

Abbreviation	Cylinder model	Turbine model
MEDL	Map-based EDL	Map-based
PEDL	Physical EDL	Map-based
ECMT	Physical extended	Map-based
ECET	Physical extended	Physical

For the turbine component, there was a test bench created where both the map-based turbine model and the extended physical model were tested and the results compared. One test setup created to evaluate the engine model was simulated with four different engine models in order to be compared against each other. These models were simulated on one part of the WHTC test-cycle where they all could operate. The models tested are referred to with abbreviations presented in Table 4.1. The main resulting model of this thesis is the engine model with abbreviation 'ECET' where both the cylinder and turbine components that have been extended during this thesis have been used. Engine model in this case indicates the entire engine model as shown with one implementation in Figure 2.3 with the workflow as shown in Figure 2.1.

4.2 Turbine evaluation

The turbine was tested on a simple experiment model with a compressor connected and gas inputs and outputs to the turbine and compressor stating pressure and temperature. The pressure of gas before the turbine was adjusted according to a ramp and all other gas pressures and temperatures were constant. The extended version of the physical turbine model based on the physical turbine model in *Engine Dynamics Library* was compared with the map-based turbine-model from the *Engine Dynamics Library*. The test bench for the turbine is shown in Figure 4.1.

The test bench for the turbine contains a turbine component, a compressor component, a shaft connecting the turbine with the compressor, air sources with constant pressure and temperature connected to the compressor and gas sources with constant temperature and stated pressure connected to the turbine. The outlet gas source from the turbine had constant pressure at 1.01 bar and the inlet gas source had a ramp for pressure. The pressure at the inlet source was during the first 50 s constant at 1 bar and then increasing during time 50 s to 60 s with constant speed and then staying at 2 bar for the remaining time, i.e., time 60 s to 100 s. The pressure ramp together with

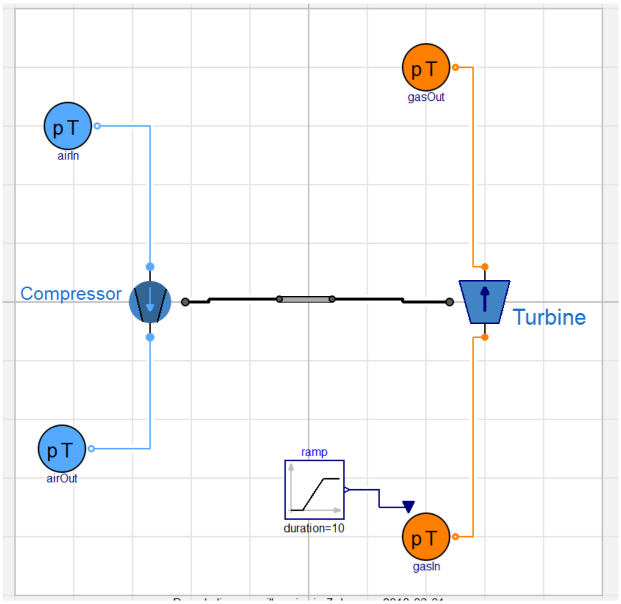

Figure 4.1 Structure of the executable turbine test bench shown in the diagram view in Dymola. The test bench consists of a turbine, a compressor, a rotating shaft connecting turbine with compressor and several gas source components specifying boundary conditions for the turbomachinery. The pressure specification in the gas source component for inlet gas to turbine is changing during the simulation according to a ramp, specified by a separate ramp component in the test bench.

the turbine mass flow for the map-based model together with the extended physical model, modified during this thesis, are shown in Figure 4.2.

The model parameters were primarily tuned on the complete engine test on the experiment with a small part of the World Harmonized Transient Cycle so the turbine test was only used for model evaluation. Figure 4.2 shows that during the first part of the experiment the map-based model has a positive mass flow even though the pressure is higher after the turbine then before. That is considered as bad behavior as the mass flow should go from the side with higher pressure to the side with lower pressure. When the turbine inlet pressure increases as seen in Figure 4.2 both turbine models obtain increasing mass flow. However, the map-based model has decreasing mass flow after a while but the physical model has constant flow after the pressure ramp. Some errors may be expected as the model parameters was not optimized but when the physical model does not capture a dynamic from the map-based model it is concerning. In the other tested test-cycles the pressure was either changing or staying for a limited time in resting position. Thus, this has probably not been a problem when tested at the World Harmonized Transient Cycle. How-

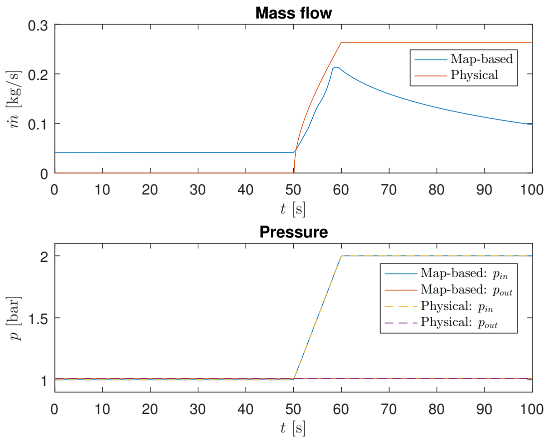

Figure 4.2 Mass flow through turbine and pressure before and after the turbine for the map-based model and the extended physical model during the tests in the test bench experiment. The parameter values for the extended physical model used in this experiment were $C_\mathrm{t} = 2.58$ (described in Equation 3.16) and $r_\mathrm{t} = 0.0375$ (denotes the radius of the turbine wheel).

ever, the physical turbine model seems to have large possibilities to be improved. One variable that could be of interest for improvement of the model for mass flow could be the speed of the turbo shaft.

4.3 Engine evaluation

Four engine models, included in Table 4.1, were simulated where the boundary conditions for fuel and speed were the same, taken from one part of the WHTC test-cycle to be compared so that the engine models could be evaluated. The models have primarily been evaluated against the map-based model. This makes it impossible to evaluate some of the behavior of the model developed during this thesis, for instance the ignition delay. Some parts are however possible to evaluate against the map-based model and this has been done as seen in this section. During the model implementation and tuning there were several other aspects considered but here only the most important ones are shown to evaluate the model. The idea behind the implementation is, however, that the models developed during this thesis are compared with the models included in Engine Dynamics Library on test-cycles that they all can run. In this part, model parameters can be tuned to give a good model behavior. The extended physical models can be executed with other test-cycles that the models from Engine Dynamics Library cannot execute to analyze the behavior of the plant for these examples.

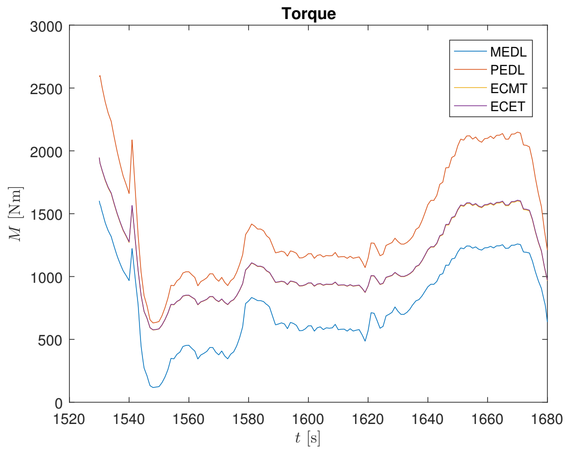

Figure 4.3 Torque for the different models. The model 'ECMT' and 'ECET' follow each other so closely that it is hard to see the underlying one.

Figure 4.3 shows that the physical cylinder models capture the behavior of the torque for the map-based model but that there seems to be a stationary error. As the ECMT and ECET models are closer to the map-based model than the PEDL model together with the fact that the behavior is captured, the torque of the extended physical cylinder-model is considered good in this sense. The errors when comparing the engine models with physical cylinder-model with the engine including the map-based cylinder-model are shown in Figure 4.4.

The pressure in the inlet manifold is shown in Figure 4.5 and here the engine models with extended physical cylinder-model seems to capture the behavior and generally be closer to the map-based model than the PEDL model.

Considering the simulation values of the temperature of the inlet manifold in Figure 4.6 there again seems to be a stationary error but this time larger than for the PEDL model. One reason for this could be that the PEDL model may have its parameters, better tuned or potentially optimized for this variable. The dynamics of the map-based model are however not seemed to be better captured by the model PEDL then by the models with extended physical cylinder-models.

The relative air-to-fuel ratio is shown in Figure 4.7. Once again the own model is considered better then PEDL as the errors when compared to the map-based model are generally smaller but here something strange happens in the own models somewhere between 1640 and 1650 in time.

Figure 4.8 shows the indicated gross efficiency for the different models. Here the dynamics of the map-based model are inadequately captured in all the physical cylinder models as they have much smaller variations than the model with the map-based cylinder-model. However, the engine models with the extended physical

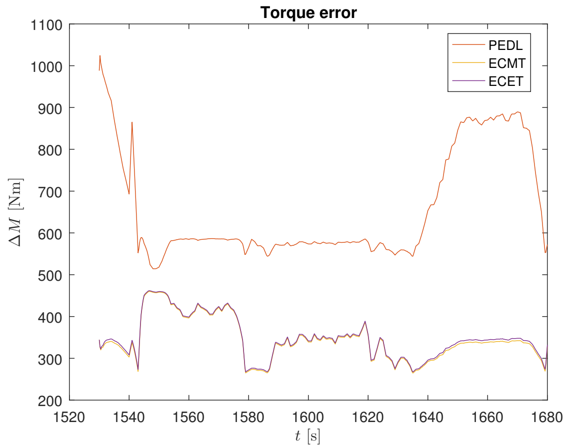

Figure 4.4 Torque error for the different models when compared to the map-based model. The root-mean-square error when compared to the map-based model is 671, 349 and 352 respectively for the models PEDL, ECMT and ECET.

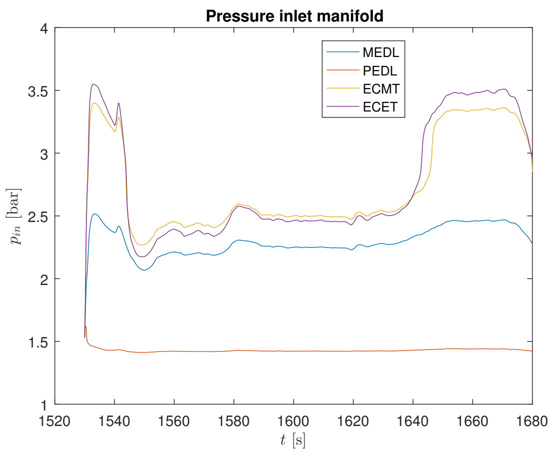

Figure 4.5 Pressure in inlet manifold for the different models. The engine models with extended physical cylinder-model seems to capture some of the dynamics of the engine but with errors while the PEDL model does not capture the dynamics of the map-based model well.

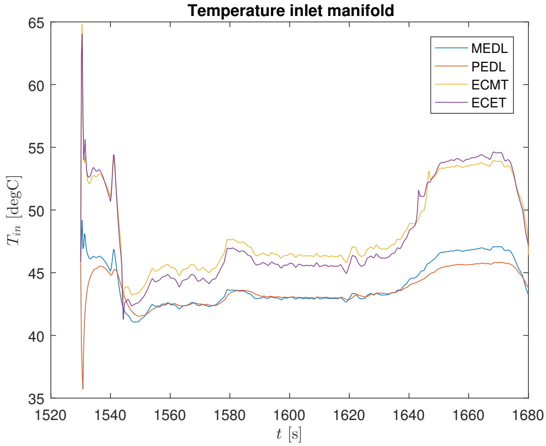

Figure 4.6 Temperature in inlet manifold for the different models. The PEDL model is closer to the MEDL model much of the time. The dynamical behavior of MEDL seems to be better captured by ECMT and ECET than by PEDL.

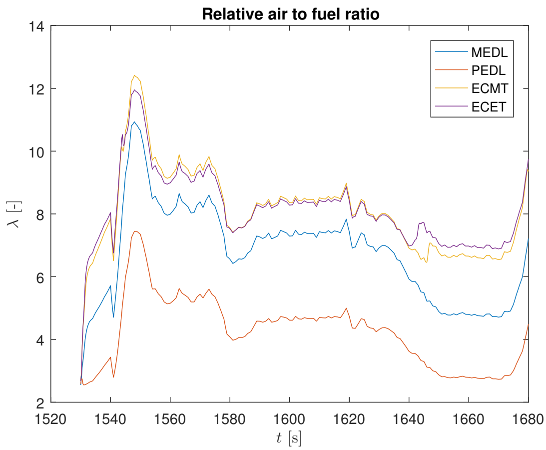

Figure 4.7 Relative air to fuel ratio for the different models. All models seems to have similar dynamical behavior. ECMT and ECET are closer than PEDL to the model MEDL during large parts of the simulation. ECMT and ECET also have one dynamic that is not present in the other models between 1640 and 1660 in time.

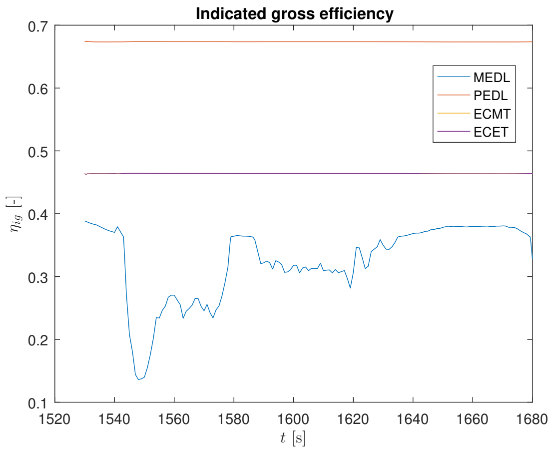

Figure 4.8 Indicated gross efficiency for the cylinder in the different models. The engine models with physical cylinder models seems to have issues capturing the map-based cylinder model dynamics. ECMT and ECET are so close that it is hard to see the graph of the ECMT model plotted under the graph of the ECET model. The ECMT and ECET models gives values closer to the map-based model than the PEDL model does.

cylinder-model are closer to the map-based model and are therefore considered as better than the PEDL model.

For the pressure in the outlet manifold the extended cylinder model is considered better than the model PEDL and capturing the dynamics of the map-based model fairly according to Figure 4.9.

In Figure 4.10 the extended cylinder model is seemed to capture the dynamics of the map-based model in a much better way then by the model PEDL. The errors for the ECMT and ECET models are smaller then for the PEDL model during the major part of the simulation time. The errors between the engine model with map-based cylinder model and the other engine models are presented in Figure 4.11.

The mass flow through the turbine is plotted in Figure 4.12. One interesting thing is that the PEDL model is neither capturing the map-based model dynamics adequately nor has small errors when compare to the map-based model even though they have the same turbine model. This shows how much the cylinder model affects the entire engine system. The engine model with the extended physical cylinder-model captures the map-based dynamics but with some errors again. When comparing the extended physical model with the map-based turbine model (ECMT) with the ECET model with the extended physical turbine model the only difference is the turbine model and this yields a different model-behavior. However, both models with extended physical cylinder model are capturing the map-based dynamical

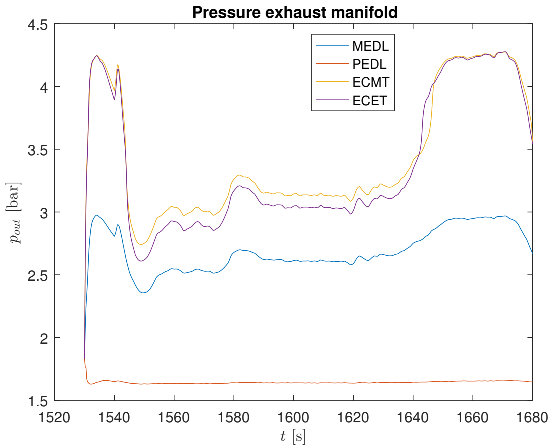

Figure 4.9 Pressure in the exhaust manifold for the different models. The models ECMT and ECET captures some of the dynamics in the MEDL model but with errors while the PEDL model does not capture the dynamics of the MEDL model.

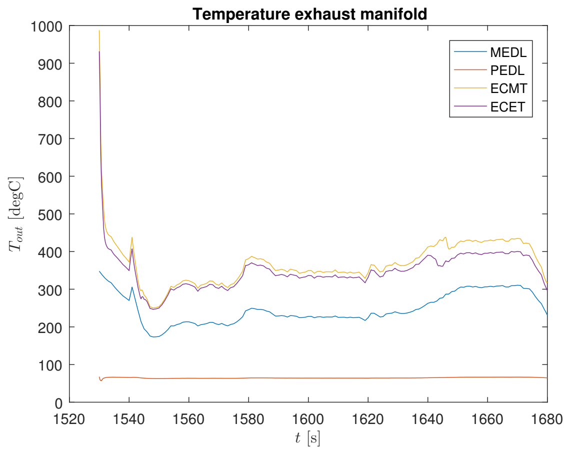

Figure 4.10 Temperature in the exhaust manifold for the different models. The ECMT and ECET models capture some of the dynamics of the MEDL better than the PEDL model captures the dynamics of the MEDL model.

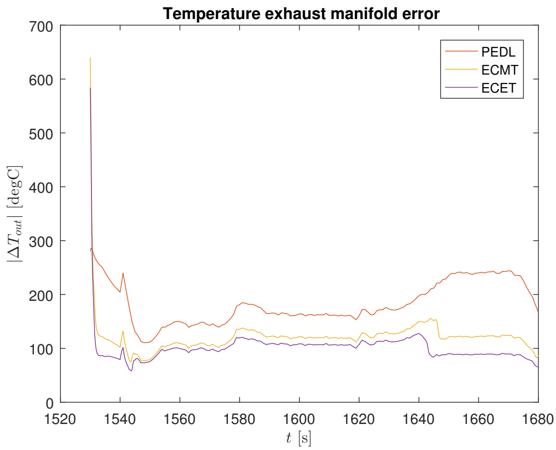

Figure 4.11 Absolute value of error of temperature in the exhaust manifold for the different models when compared to the map-based model. The root-mean-square error when compared to the map-based model is 188, 125 and 105 respectively for the models PEDL, ECMT and ECET.

behavior but with errors present.

Figure 4.13 shows the isentropic efficiency for the turbine. The isentropic efficiency states how much of the potential work that the turbine could perform that actually acts on the rotating shaft connected to the compressor. In this case PEDL seems to be closer to the map-based then ECMT and ECET during much of the simulation but it is close to constant and for another test case it could possibly be worse. The ECMT model captures the dynamics of the map-based model better but still errors clearly exist. The model ECET with extended physical turbine model captures the dynamics of the map-based poorly. Possibly the parameters in the physical turbine model could be tuned more accurate.

The MEDL and ECET models have some simulation results collected in Figure 4.14 to understand the coupling of the variables. The figure includes plots for torque acting on crankshaft, fuel flow, speed of crankshaft, friction torque acting on crankshaft, indicated gross efficiency for the cylinder, temperature of the exhausts coming out from the cylinder, total (brake) efficiency and the pressures before and after the cylinder.

4.4 Model testing for other test-cycles

The obtained ECET model then was tested on other data sets to check the functionality. This section contains two such simulation results where one test had constant speed and a ramp for fuel injection while the other test was the whole WHTC test-

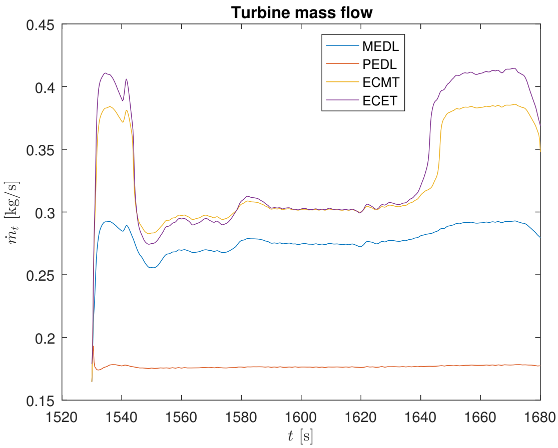

Figure 4.12 Mass flow through the turbine for the different models. The ECMT and ECET models capture some of the behavior of the MEDL model but with errors. The model PEDL has problems capturing the dynamics of the MEDL model.

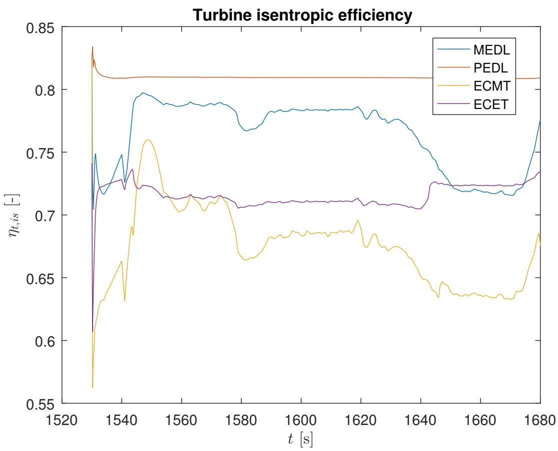

Figure 4.13 Isentropic efficiency for the turbine in the different models. The PEDL model seems to not capture the dynamics of the MEDL model. The ECMT model captures some of the dynamics of the MEDL model but with errors. The ECET model has the extended physical turbine model and it does not seem to capture the dynamics of the MEDL model well.

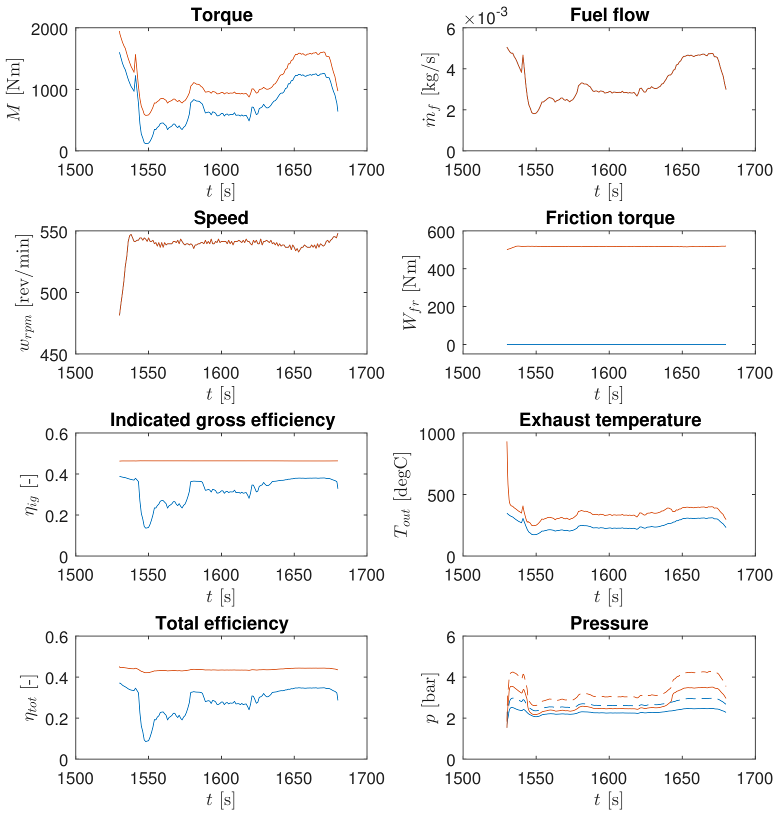

Figure 4.14 Simulation results for the MEDL and ECET engine models for the test at a small part of WHTC. The MEDL model is plotted with blue curves and the ECET model with red curves. The pressure plot has input pressure to cylinder plotted with solid line and output pressure from cylinder plotted with dashed line.

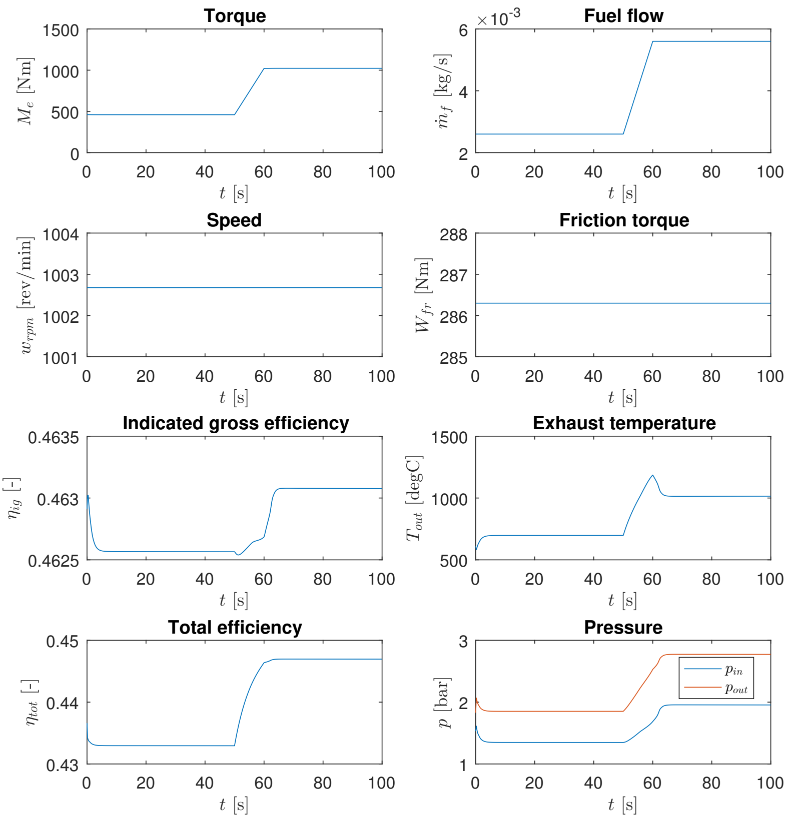

Figure 4.15 Simulation results for the ECET engine model for the test with constant speed and with a ramp in fuel injection.

cycle. Simulation results for a test with constant speed and a ramp in fuel injection are presented in Figure 4.15. The friction is constant as it is modeled with dependency only on speed. The pressure out from the cylinder is larger than the pressure in to the cylinder.

The ECET model was executed on the whole WHTC and the result from this simulation is presented in Figure 4.16. The torque obtained some high peaks that are hard to understand but is during much of the time on values below 2500 as is usual for engines of this type. The friction and indicated gross efficiency seems to be in the same range as during the other tests which is good. The exhaust temperature flowing out from the cylinder has some high peaks but seems to be reasonable much of the time. The high temperatures are hard to understand but with parameters calibrated in order to capture the dynamics of the map-based model the exhaust temperature could potentially get decreased, at least in the high peaks. The total efficiency drops below zero at some time points. This means that the friction and pressure performs a larger work than the combusted fuel at the crankshaft. This means that the crankshaft obtain a total force from the engine that is breaking the shaft in the rotation. If the crankshaft would rotate slower, then, the friction would be smaller (according to the used friction model) and then there is a possibility that the total torque on the crankshaft would be driving the shaft instead of breaking it.

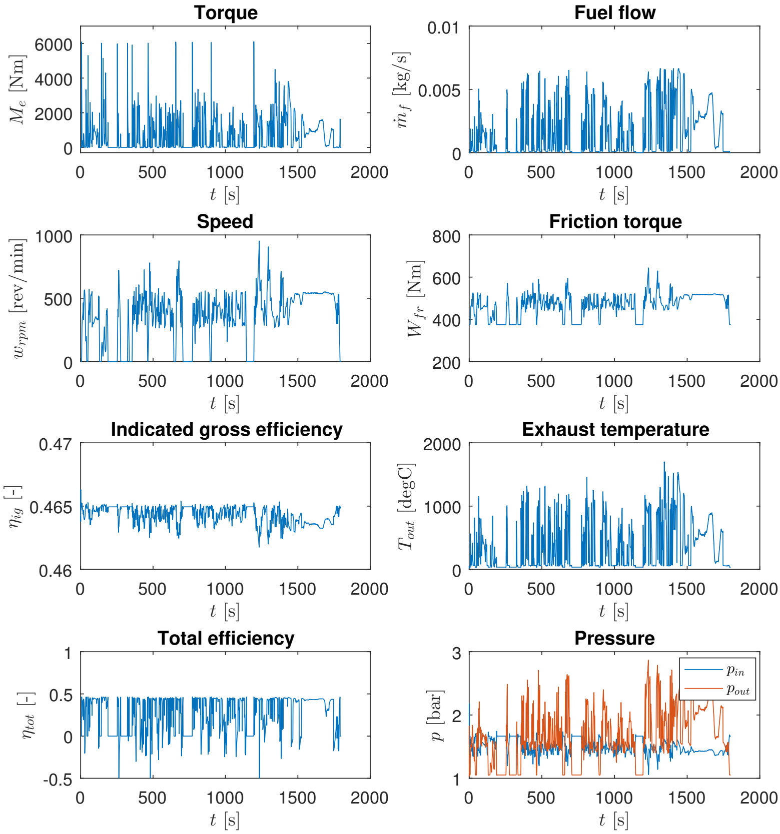

Figure 4.16 Simulation results for the ECET engine model for the test at the whole test cycle WHTC.

The DAE has 2317 scalar unknowns and 2317 scalar equations.

Figure 4.17 The test experiment for the final engine model gives after translation the given output in the translation log that indicates the large scale of the model measured in number of unknown variables that should be calculated. DAE is an abbreviation of *Differential-Algebraic Equations*.

5

Discussion

> Next time that someone tells you that something is true, why not say to them: 'What kind of evidence is there for that?' And if they can't give you a good answer, I hope you'll think very carefully before you believe a word they say.
>
> *Richard Dawkins FRS*

5.1 Evaluation of models

The models were executed with respect to a small part, regarding simulation time, of the World Harmonized Transient Cycle in Section 4.3. Some general behavior seen in Section 4.3 is that the ECET model capture some behavior that the PEDL model does not capture. This was expected as the extended physical cylinder model is more advanced then the simple physical cylinder model taken from the Engine Dynamics Library as it takes more dependencies into account during computations of the cylinder outputs. It is possible to see that the physical cylinder models do not give the same results as the map-based does and the extended physical turbine model gives slightly different behavior when compared to the map-based one. It is not always possible for a physics-based model with limited number of variables to imitate a map-based model perfectly as there can be effects that the physics-based model has not been modeled for and other disturbances that were present during calibration of the map-based model but a good physics-based model should get small errors when compared to the map-based model. To give small errors all parameters in the physics-based model should have values close to the ones considered when calibrating the map for the map-based model. To obtain good values for parameters there could be system identification performed, but this is outside the scope of this thesis. Instead parameters have been chosen to values found in different literature sources to give the possibility to evaluate the model even though this was expected to give a slightly different behavior when comparing the physics-based model with a map-based model.

The ECET model was developed to handle experiments with low speed and small fuel injection. One of the experiments testing that abilities of the ECET engine model was the test at the whole WHTC test-cycle as shown in Figure 4.16. The WHTC test was not able to be simulated with the MEDL and PEDL models. The developed model can handle both small speed and fuel injection and runs the whole WHTC test-cycle which was the objective of the thesis.

5.2 Conclusion

During this thesis a new cylinder model was developed and a turbine model was reconstructed so that when they are used in the engine model included in *Engine Dynamics Library* they can handle trajectories with zero-fuel flow and zero speed. To be able to analyze the model more accurately there is a need for finding better values on model parameters with the use of some method from system identification, as a suggestion. Then hopefully the extended physical model would be a good approximation of the available map-based model on the domain where both models can operate. With the current parameters, much of the dynamics of the system is already captured. This together with the fact that the new model has a larger domain and, thus, can operate on trajectories where the fuel flow and speed go down to zero makes the project to be regarded as successful.

5.3 Future work

The extended physical engine model is to be used after parameter identification. Then it should be possible to better analyze its performance. The model is developed to simulate how a real process operates and therefore it would be interesting to compare it with real experimental data. One thing that was not included in the map-based model was how the injection timing affects the result and therefore this was not possible to evaluate for the cylinder-model implemented during this thesis but would be interesting to check against real data. Accessible real data can in many cases be protected in order to not help other competitors. If real data would be obtained for several engines it would be interesting to evaluate the strength of the model-based model in comparison with the map-based model when considering differences of different engines due to production conditions. One more thing that would be interesting to check against real data would be how well the extended physical model can capture the behaviour if one part in the engine is exchanged for another part. When good enough model parameters are found, the model could be used for control design purposes.

Notation

c_p	Specific heat value at constant pressure
c_v	Specific heat value at constant volume
E_a	Activation energy for fuel
L_{pm}	Piston displacement distance
M_e	Engine torque
$\dot{m}$	Mass flow
$\dot{m}_{ac}$	Gas mass flow into cylinder
$\dot{m}_e$	Engine mass flow
$\dot{m}_f$	Fuel mass flow
$m_{f,cycle}$	Fuel used during one cycle
N	Revolutions per minute
n_r	Number of crank revolutions per cycle
p	Pressure
p_{CT}	Pressure at combustion timing
p_{in}	Pressure before component
p_{out}	Pressure after component
$\dot{Q}_{ht}$	Heat transfer
Q_{LHV}	Lower heating value
R	Universal gas constant
R_r	Ratio of connecting-rod length to crank radius
r_c	Compression ratio
S_p	Piston speed
T_{CT}	Temperature inside cylinder at combustion timing
T_{eo}	Temperature of exhaust outlet from cylinder
T_f	Temperature of fuel
T_{in}	Temperature in inlet manifold
V	Cylinder volume
V_c	Cylinder clearance volume

V_{C}	Cylinder total clearance volume
V_{d}	Cylinder displacement volume
V_{D}	Cylinder total displacement volume
w	Engine speed
W	Work
W_{e}	Engine work
W_{fr}	Friction work
$W_{\mathrm{i,g}}$	Indicated gross work
$W_{\mathrm{i,p}}$	Pumping work
γ	Specific heat ratio
η_{ig}	Indicated gross efficiency
$\eta_{\mathrm{ig,ch}}$	Ignition chamber efficiency
η_{ign}	Ignition efficiency
η_{tot}	Total (brake) efficiency
η_{vol}	Volumetric efficiency
θ_{CT}	Crank angle at combustion timing
θ_{id}	Crank angle moved during ignition delay
θ_{opt}	Optimal crank angle for combustion
θ_{SOI}	Crank angle at start of injection
λ	Relative air-to-fuel ratio
τ_{id}	Ignition delay

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Title and subtitle
Physical modeling of a heavy-duty engine for test-cycle simulations in Modelica

Abstract

Engine development for reduced fuel consumption and for the ability to use alternative types of fuels is required as emissions and the limited access of petroleum are issues coupled to diesel engines. Experimental testing is in many cases insufficient and expensive in comparison with simulations for engine development. This thesis is focused on extending a model of a diesel compression-ignition engine for test-cycle simulations with a new cylinder model. The model was developed with Dymola executing code written in the language Modelica and with the external Engine Dynamics Library developed by Modelon AB. There were prior engine models in the Engine Dynamics Library but there was a need for a model extension primarily to get a model able to handle a larger operating range. The result of the thesis is an engine model that can execute simulations with zero speed and fuel injection that the previous models in Engine Dynamics Library could not handle. During model evaluation the model was compared to a calibrated map-based model at a test-cycle where the map-based model could be executed. The model developed during this thesis research was successfully able to capture much of the dynamics of the available map-based model.

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