Mean Value Engine Model of a Heavy Duty Diesel Engine

Jonas Biteus

Vehicular Systems
Department of Electrical Engineering
Linköpings universitet, SE-581 83 Linköping, Sweden
WWW: www.vehicular.isy.liu.se
E-mail: biteus@isy.liu.se

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Dept. of Electrical Engineering
581 83 Linköping

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Title Mean Value Engine Model of a Heavy Duty Diesel Engine

Författare Jonas Biteus
Author

Sammanfattning
Abstract

A first implementation of a mean value engine model (MVEM) of a Heavy Duty Diesel (HDD) engine is described in this report. Framework and sub models are described. Where applicable ISO standards are followed. Verification against static measurements shows maximum model errors of about 6% for mass flow and inlet/exhaust manifold pressures.

Nyckelord
Keywords mean value engine model, MVEM

Contents

1 Introduction

1.1 Acknowledgments

This work has been supported by *Scania AB, Department of Software and Diagnostics, Sweden.*

1.2 Objectives

The objectives are to construct an accurate model of a HDD engine. The model should be able to predict effects of mechanical and/or control system changes in the engine. Primary the model should be used to test and verify OBD (on board diagnostic) systems.

A framework for the model that supports automatic or semi automatic identification and verification should be implemented.

An object oriented approach should be used to construct the model.

1.3 Simulation Environment

The model is implemented in *Simulink/Matlab.* The model is to a large extent object oriented. In simulink, a component library has been created. From the component library the engine model is implemented.

1.4 Name Convention

Components are abbreviated with a two or three lowercase letters subindex. If several sub indices are required, the second is spelled with a first upper case letter. When sub indices cannot be used, e.g., Matlab code, every subindex first letter is upper case.

Control volumes are named after the component upstream, e.g., control volume fi is after the filter restriction.

In appendix B notations used in the model are described.

1.5 Simulation Structure

The model can be described in state variables and flows. From the states in control volumes *up*, i.e., upstream, and *down*, i.e., downstream, of a component, it is possible to calculate the flow F thru the component. The flow can be divided into flow *in* and *out* of a component. The nominal flow direction defines *up* and *down*.

By having a strict order of “control volume – component – control volume – component” with flows as connectors, an object oriented system is achieved.

Figure 1 shows a schematic overview of the system including components abbreviations.

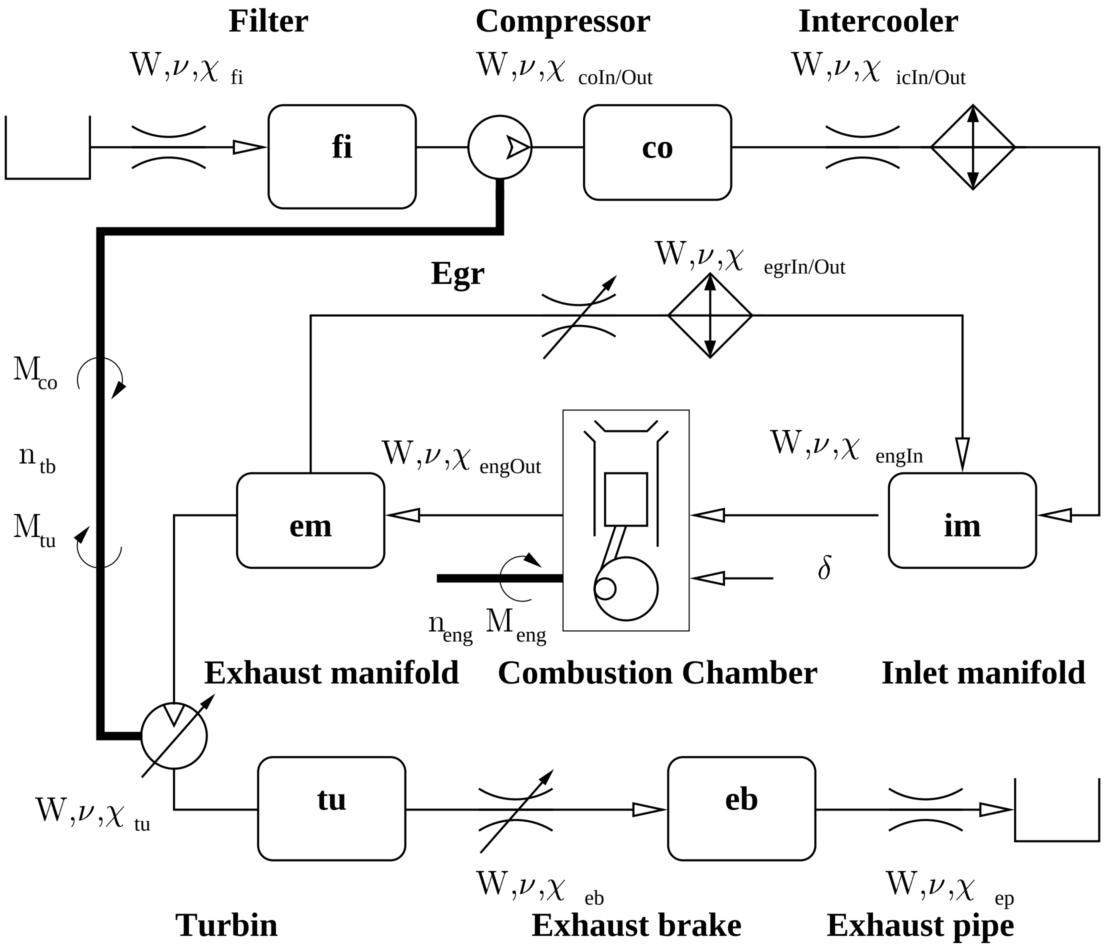

Figure 1: Schematic overview of engine model.

1.6 Reversible vs. Non Reversible

During normal non faulty operation, there is no need for reversible systems. Therefore, it is assumed that all flows are positive. Reversible components are constructed from non reversible with a case system, e.g., if $p_{up} < p_{down}$ in a quadratic restriction $W_{\text{reversible}} = -W_{\text{non reversible}}$ where W_x is mass flow $[kg/s]$.

1.7 References

MVEM:s have been implemented before. Several articles and reports describing the basic ideas are available. Some articles with relevant results are (Guzzella and Amstutz, 1998) for physical modeling and (Müller, 1998) for regression analyze modeling.

At Vehicular Systems at Linköpings universitet, several master's theses that describe MVEM have been published. Most noticeable are (Brugård and Bergström, 1999; Pettersson, 2000), available at `www.fs.isy.liu.se`.

2 Standard Sub Models

In this section the "standard" sub models are described. These models form the model library.

2.1 Control Volume

The control volume is based on energy- and mass-conservation. State variables are energy and mass of gas components. The standard volume is implemented with two components, air and exhaust gas.

The model for a control volume with inlets i and outlets j is

$$
\begin{aligned}
\dot{U} &= \sum_i \nu_i - \sum_j \nu_j - Q && \text{Energy balance} \\
\dot{m}_{air} &= \sum_i W_i(1-\chi_i) - \sum_j W_j(1-\chi_{cv}) && \text{Mass balance air} \\
\dot{m}_{exh} &= \sum_i W_i\chi_i - \sum_j W_j\chi_{cv} && \text{Mass balance exhaust} \\
m &= m_{air} + m_{exh} && \text{Total mass} \\
\chi_{cv} &= \frac{m_{exh}}{m} && \text{Mass fraction} \\
p &= \frac{RU}{Vc_v} && \text{Pressure} \\
T &= \frac{U}{mc_v} && \text{Energy – Temp. relation}
\end{aligned}
$$

$$
\begin{aligned}
\rho &= \frac{p}{TR} && \text{Density} \\
c_v &= c_{vAir}(1-\chi_{cv}) + c_{vExh}\chi_{cv} \\
c_p &= c_{pAir}(1-\chi_{cv}) + c_{pExh}\chi_{cv} \\
\nu_j &= c_p W_j T \quad \forall j && \text{Energy flow out} \\
Q &= f(\cdot) && \text{Heat losses.}
\end{aligned}
$$

The heat losses can be static or dynamic and be positive (energy decreases) or negative (energy increases).

2.2 Heat Exchanger

The heat exchanger decreases the energy flow. The model is

$$
\begin{aligned}
T_{out} &= T_{in} - \eta(T_{in} - T_{surr}) \\
\nu_{out} &= c_{pUp} W T_{out},
\end{aligned}
$$

where η is the efficiency of the heat exchanger and T_{surr} the surrounding temperature.

2.3 Variable Restriction

The variable restriction uses the pressure ratio $\frac{p_{down}}{p_{up}}$ and mapped area $A(u)$ to evaluate mass flow. The model is

$$
\begin{aligned}
W &= A(u)\frac{p_{up}}{\sqrt{T_{up}R_{up}}}\Psi(\frac{p_{down}}{p_{up}}, \gamma_{up}) \\
\Psi(\Pi,\gamma) &= \begin{cases} \sqrt{\frac{2\gamma}{\gamma-1}\left(\Pi^{\frac{2}{\gamma}} - \Pi^{\frac{\gamma+1}{\gamma}}\right)} & \text{if } \Pi \geq \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}} \\ \sqrt{\gamma\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{\gamma-1}}} & \text{else,} \end{cases}
\end{aligned}
$$

where u is the control signal.

2.4 Quadratic Restriction

The quadratic restriction is based on a quadratic relationship between pressure drop and mass flow. It is a simplified version of the variable restriction. Restrictor constant k_{res} depends on area, wallfriction etc. The model is

$$
\begin{aligned}
p_{up} - p_{down} &= \frac{k_{res}W^2}{\rho_{up}} \\
\rho_{up} &= \frac{p_{up}}{R_{up}T_{up}}.
\end{aligned}
$$

3 Specific Sub Models

The specific sub models used in the engine model are described in this section. For the control volumes, in- and outlet symbols are defined.

3.1 Filter

The filter model should describe the pressure losses inflicted by the air filter.

Restriction

The filter is modeled as a standard restriction with constant k_{fi}.

Filter Control Volume

$$\begin{aligned} IN &: [\nu, W, \chi]^T_{fi} \\ OUT &: [\nu, W, \chi]^T_{coIn} \\ Q_{fi} &= 0. \end{aligned}$$

3.2 Turbo

The turbo is divided into three parts; compressor, turbine and turboshaft.

Two different turbo models have been implemented. The first uses physical relations and compressor and turbine efficiency and mass flow maps from the manufacturer. The turbo speed is considered a state variable. However, due to inaccuracy in maps and disturbances, it was not possible to achieve correct static turbo speeds.

The second turbo model is based on the same physical relations but developed with regression analysis. In this model the turbo speed is considered an input.

To be able to use the compressor and turbine maps the pressure ratios and turbo speed have to be normalized. Normalization is implemented in the simulation library. For more information about the normalization see (Mårberg, 1999).

Compressor – Model I

The model is based on thermodynamic energy transformation described in (Guzzella and Amstutz, 1998). Compressor efficiency and mass flow are taken from manufacturer data that have been inter- and extrapolated with GT-Power and algorithms developed by *Scania* (Mårberg, 1999).

The model for the compressor is

$$W = f_w(\frac{p_{up}}{p_{down}}, n_{tb}) \tag{1a}$$

$$T = T_{up}(1 + \frac{1}{\eta}(\mu - 1) \tag{1b}$$

$$M = \frac{W c_{pUp} T_{up}}{\eta n_{tb}}(\mu - 1)$$

$$\eta = f_e(\frac{p_{up}}{p_{down}}, n_{tb})$$

$$\mu = \left(\frac{p_{down}}{p_{up}}\right)^{\frac{\gamma_{up}-1}{\gamma_{up}}}$$

$$\nu_{out} = c_{pUp} W T.$$

Compressor – Model II

A second model based on thermodynamic energy transformation but developed with regression analysis has been implemented. The model is based on relations described in (Müller, 1998) and implemented on a turbo charged SI engine in (Brugård and Bergström, 1999; Pettersson, 2000).

The model for mass flow is

$$W = k_1(1 - \frac{p_{up}}{p_{down}}) + k_2 n_{tb}\sqrt{1 - \frac{p_{up}}{p_{down}}} + k_3 n_{tb}\sqrt[4]{1 - \frac{p_{up}}{p_{down}}} + k_4 n_{tb},$$

where $k_{1,2,3,4}$ are constants. The model for the temperature out is

$$T_{out} = s_1 W^2 + s_2 W n_{tb} + s_3 n_{tb}^2 + s_4 T_{up},$$

where $s_{1,2,3,4}$ are constants. Note that these models depends on the same variables as model equation (1a) and (1b).

Compressor Control Volume

$$IN : [\nu, W, \chi]^T_{coOut}$$
$$OUT : [\nu, W, \chi]^T_{icIn}$$
$$Q_{co} = 0.$$

Turbine – Model I

A model based on thermodynamic energy transformation as described in (Guzzella and Amstutz, 1998) has been implemented.

Compressor efficiency and mass flow are taken from manufacturer data that have been inter- and extrapolated with GT-Power and algorithms developed by *Scania* (Mårberg, 1999).

The turbine model is

$$\begin{aligned}
W &= f_w(\frac{p_{up}}{p_{down}}, n_{tb}) \\
T &= T_{up}(1 - \eta(\mu - 1)) \\
M &= \frac{W c_{pUp} T_{in} \eta}{n_{tu}}(1 - \mu) \\
\eta &= f_e(\frac{p_{up}}{p_{down}}, n_{tb}) \\
\mu &= \left(\frac{p_{down}}{p_{up}}\right)^{\frac{\gamma_{up}-1}{\gamma_{up}}} \\
\nu_{out} &= c_{pUp} W T.
\end{aligned}$$

Turbine – Model II

The second model is based on thermodynamic energy transformation but developed with regression analysis. The model is based on relations described in (Müller, 1998) and implemented on a turbo charged SI engine in (Brugård and Bergström, 1999; Pettersson, 2000). It should be noted that this model is based on similar assumptions as the algorithms developed in (Mårberg, 1999).

The model for mass flow is

$$W = \frac{p_{down}}{\sqrt{p_{up}} 2 t_1}\left[-t_2 + \sqrt{t_2^2 + 4 t_1\left(\frac{p_{up}}{p_{down}} - t_3\right)}\right],$$

where $t_{1,2,3}$ are constants, found from a least square fit of

$$\frac{p_{up}}{p_{down}} = t_1\left(\frac{W\sqrt{T_{up}}}{p_{down}}\right)^2 + t_2\frac{W\sqrt{T_{up}}}{p_{down}} + t_3,$$

w.r.t. $t_{1,2,3}$.

Due to lack of sensors in the exhaust system, T_{tuOut} is not modeled. However, this is not very important because of the low restrictions down stream the turbine. Note that if an exhaust brake is used, this assumption does not hold.

Turbine Volume

$$\begin{aligned}
IN &: [\nu, W, \chi]^T_{tuOut} \\
OUT &: [\nu, W, \chi]^T_{eb} \\
Q_{tu} &= 0.
\end{aligned}$$

Turboshaft

Friction is assumed to be included in the turbine efficiency mapping. The model for the turboshaft is

$$J_{tb}\frac{dn_{tb}}{dt}\frac{2\pi}{60} = M_{tu} - M_{co} - M_{fr},$$

where M_{fr} is assumed zero.

3.3 Intercooler

The intercooler model should describe the heat exchange and restriction in the intercooler.

Restriction

Modeled as quadratic restriction with constant k_{ic}.

Heat Exchanger

Modeled as a heat exchanger with constant η_{ic}. It is assumed that T_{icSurr} is constant.

3.4 Inlet Manifold

The inlet manifold collects gases from the intercooler and the EGR (exhaust gas recycling) system. The walls of the inlet manifold have a relatively high temperature which will lead to negative heat losses. In this first implementation the heat losses are assumed zero.

Control Volume

$$\begin{aligned} IN &: [\nu, W, \chi]^T_{icOut}, [\nu, W, \chi]^T_{egrOut} \\ OUT &: [\nu, W, \chi]^T_{engIn} \\ Q_{im} &= 0. \end{aligned}$$

3.5 Combustion Chamber

The combustion chamber models the mean value of the in cylinder combustion. Most notable is that fuel is added, temperature increased, and the amount of exhaust gas increased.

The theoretic volumetric efficiency is modeled with η_{volEm}.

The model for the combustion chamber is

$$
\begin{aligned}
W_{out} &= W_{in} + W_{fuel} \\
W_{in} &= \frac{\rho_{up} V_d n_{cyl} N_{eng}}{2*60} \eta_{volEm} \eta_{vol} \\
W_{fuel} &= \frac{\delta n_{cyl}}{2*60} N_{eng} \\
W_{air} &= W_{in}(1 - \chi_{up}) \\
\lambda &= \frac{W_{air}/W_{fuel}}{(A/F)_s} \\
T &= T_{up} + f_{temp}(\delta, W_{engIn}) \\
\chi_{out} &= \begin{cases} 1 & \text{if } \lambda < 1 \\ \lambda^{-1} & \text{else} \end{cases} \\
\nu_{out} &= c_p T W_{out} \\
\rho_{up} &= \frac{p_{up}}{R_{up} T_{up}} \\
\eta_{volEm} &= \frac{r_c}{r_c - 1} - \frac{1}{\gamma_{up}(r_c - 1)} \left(\frac{p_{down}}{p_{up}} + \gamma_{up} - 1 \right) \\
\eta_{vol} &= f_{vol}(N_{eng}, \frac{p_{down}}{p_{up}}) \\
c_p &= c_{pAir}(1 - \chi_{out}) + c_{pExh}\chi_{out} \\
c_v &= c_{vAir}(1 - \chi_{out}) + c_{vExh}\chi_{out}.
\end{aligned}
$$

Note, in the first implementation of the model, it is assumed that $\chi_{out} = 1$.

3.6 Exhaust Manifold

The exhaust manifold divides gases to turbine and EGR system. The walls of the exhaust manifold have a relative low temperature which will lead to heat losses. In this first implementation the heat losses are assumed zero. A regression model for the energy losses in the exhaust manifold is described in (Müller, 1998). However, this model has not been implemented.

$$
\begin{aligned}
IN &: [\nu, W, \chi]^T_{engOut} \\
OUT &: [\nu, W, \chi]^T_{tuIn}, [\nu, W, \chi]^T_{egrIn} \\
Q_{em} &= 0.
\end{aligned}
$$

3.7 EGR System

The model of the EGR system has not been validated. To be able to validate the system, more measurements are needed, see Section 6. In this section two different types of EGR systems are considered. Both uses a EGR variable restriction to restrict EGR flow and EGR cooler to gain a high density EGR flow. Additional pressure drop is created with a throttle or a venturi.

EGR Variable Restriction

The EGR valve is modeled as a variable restriction. In the model it is assumed that EGR valve angle, α_{egr}, can be predicted from EGR control signal, u_{egr}. In the fault free case this is correct, since pneumatic actuators are designed to be linear, i.e., $\alpha_{egr} = f(l_{egr})$ and $l_{egr} \propto u_{egr}$. However, in some cases hysterics can cause large deviation from linearity. In Appendix A the linearity of the actuator is analyzed.

EGR Cooler

The EGR cooler is modeled as a standard heat exchanger. The constant is η_{egr} and it is assumed that $T_{egrSurr}$ is constant.

EGR throttle

The EGR throttle is used to maintain a positive pressure drop over the EGR. The throttle is positioned between the intercooler and inlet manifold.

The throttle is modeled as a variable restriction.

Venturi

The venturi system is used to maintain a positive pressure drop over the EGR. The EGR flow is added to the main flow at the minimum area of the venturi. At this point the pressure is at minimum. After this point a diffusor is used to recover the pressure.

It has not been possible to find a working model for the venturi system. Following is a suggestion of a venturi model used to model $p_{downEgr}$, i.e., the pressure downstream the EGR variable restriction, used to predict W_{egr} with the model for the EGR variable restriction. See Fig. 2 for a schematic overview of the venturi system with EGR.

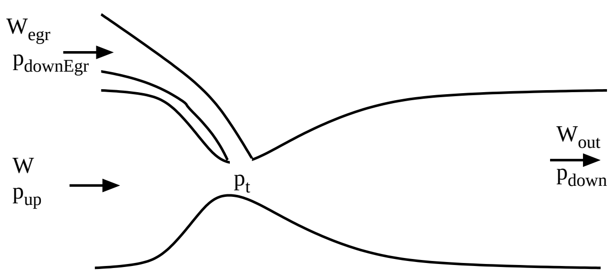

Figure 2: Schematic overview of a venturi system.

If non choked flow is assumed, a venturi can be modeled as

$$W = A\frac{p_{up}}{\sqrt{T_{up}R_{up}}}\Psi[\frac{p_t}{p_{up}}, \gamma_{up}] \tag{2}$$

$$\Psi[\Pi, \gamma] = \sqrt{\frac{2\gamma}{\gamma - 1}\left(\Pi^{\frac{2}{\gamma}} - \Pi^{\frac{\gamma+1}{\gamma}}\right)}.$$

If it is assumed that the venturi can be model as a quadratic restriction, W can be calculated

$$W = \sqrt{\rho\frac{p_{up} - p_{down}}{k}}.$$

Further assume that $W_{egr} << W$, then (2) can be solved numerically for p_t

$$p_t = f(p_{up}, W_{egr}, T_{up}).$$

Now p_t can be used as $p_{downEgr}$ and a model for the venturi-EGR system has been achieved.

3.8 Exhaust Brake

Exhaust Brake Variable Restriction

The exhaust brake has not been implemented but can be modeled as a variable restriction. For a good implementation it is necessary to model the temperature drop over the turbine.

Exhaust Brake Volume

$$\begin{aligned} IN &: [\nu, W, \chi]_{eb}^T \\ OUT &: [\nu, W, \chi]_{ep}^T \\ Q_{eb} &= 0. \end{aligned}$$

3.9 Exhaust Pipe

Exhaust Pipe Restriction

The exhaust pipe is modeled as a quadratic restriction with constant k_{ep}.

4 Identification

To identify the sub models least square optimization of has been used. The data used is from the static mapping of a HDD engine without EGR.

5 Validation

The validation has only been performed against static data, due to the problems with accurate EGR models. The data used is from the static mapping of a HDD engine without EGR. Figure 3 shows the static points used for simulation and identification. Simulation of the points marked with "o" failed due to simulation problems.

In these simulations the second turbo model has been used. Note that in data to this model are N_{eng}, n_{tb}, and δ.

5.1 Static Validation

Simulations have been performed for the major part of the stationary points. Figure 4, 6, and 8 shows simulated and reference values for W_{fi}, p_{im}, and p_{em} in static points. Relative error is shown in Fig. 5, 7, and 9. As can be seen in the figures, the maximum relative error is about 6%.

5.2 Dynamic Simulation

The model has *not* been validated against dynamic measurements.

Figure 10, 11 and 12 shows step responses for the model. Note that these figures are only presented to show that the system works for dynamic references. The references are N_{eng}, δ, and n_{tb}. To obtain stability, N_{eng} and n_{tb} are low pass filtered.

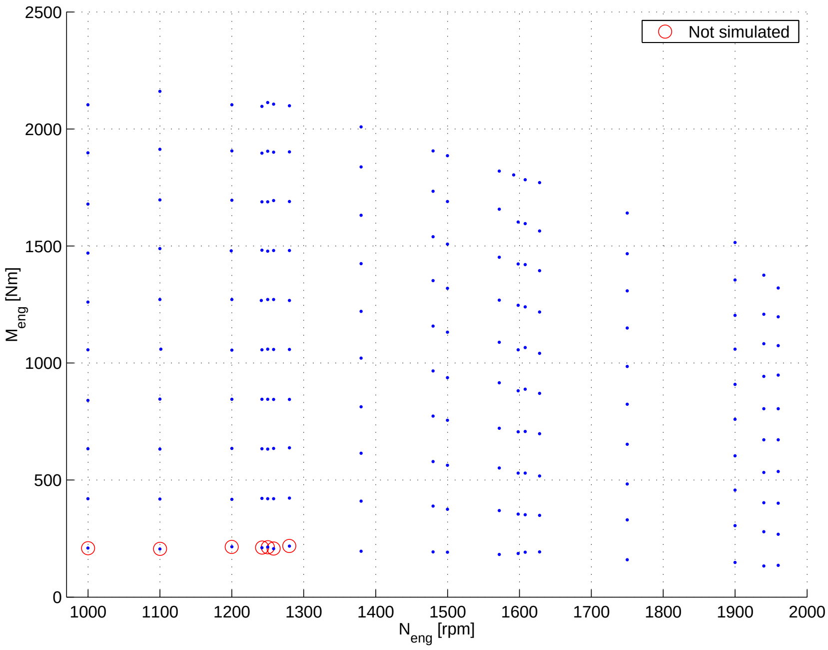

Figure 3: Static measurements points in N_{eng} and M_{eng}.

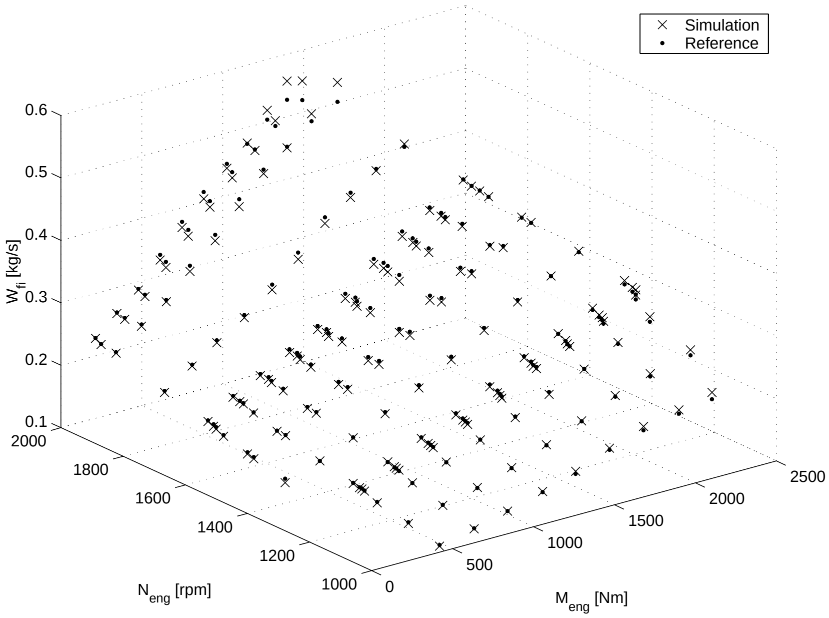

Figure 4: Measured and simulated W_{fi}.

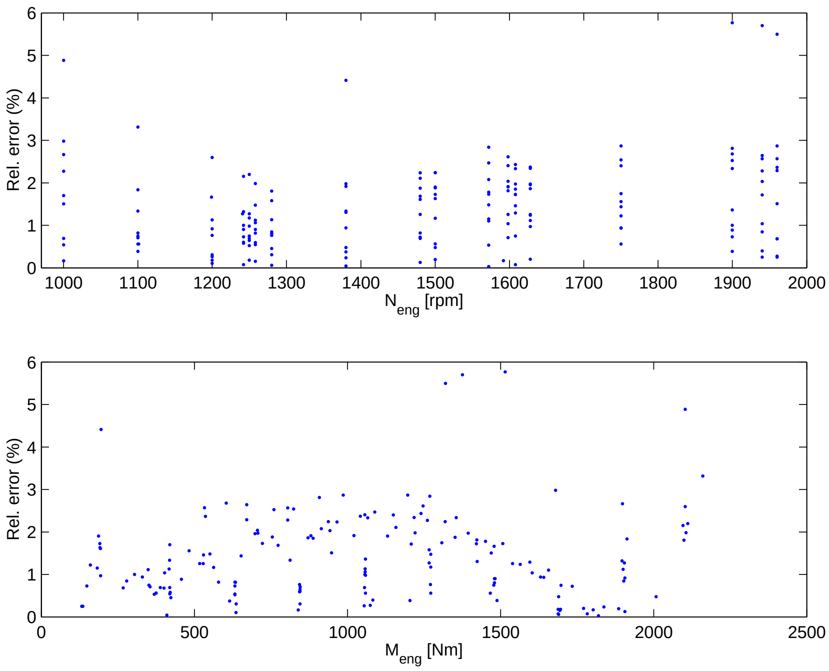

Figure 5: Relative error for simulated W_{fi} sorted w.r.t. N_{eng} and M_{eng} respectively.

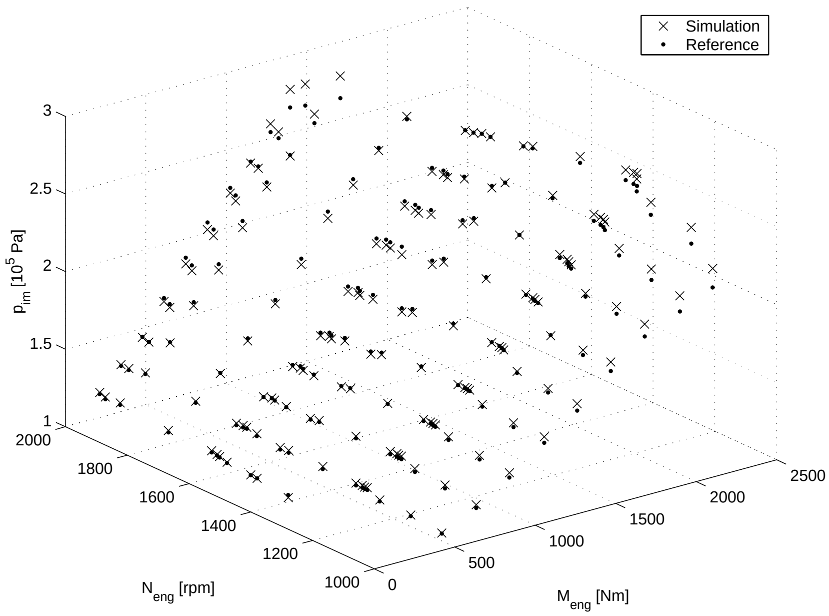

Figure 6: Measured and simulated p_{im}.

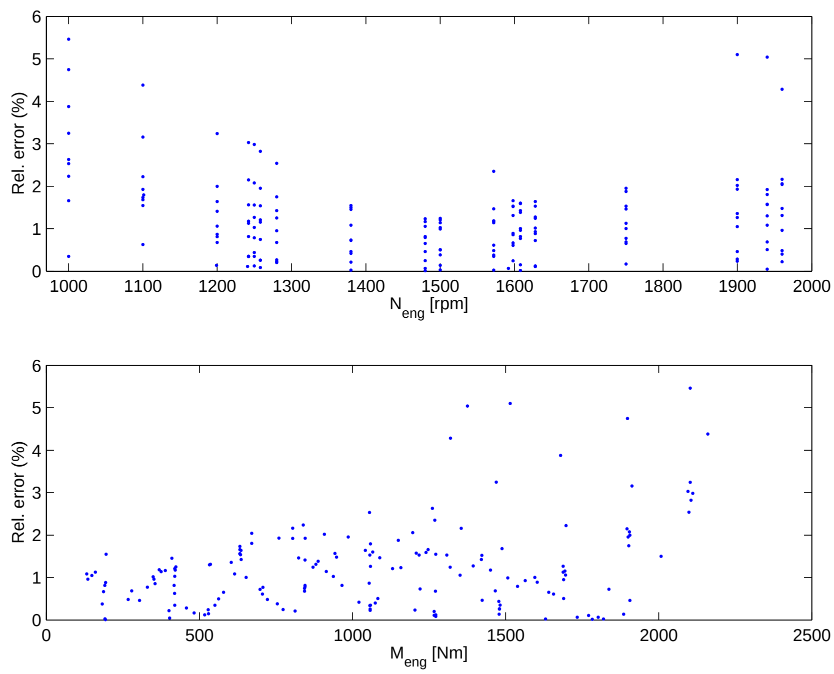

Figure 7: Relative error for simulated p_{im} sorted w.r.t. N_{eng} and M_{eng} respectively.

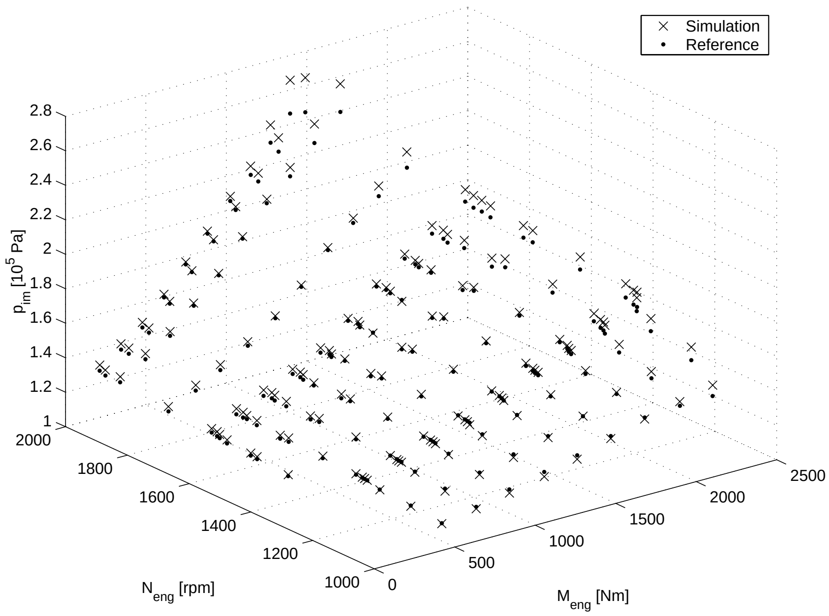

Figure 8: Measured and simulated p_{em}.

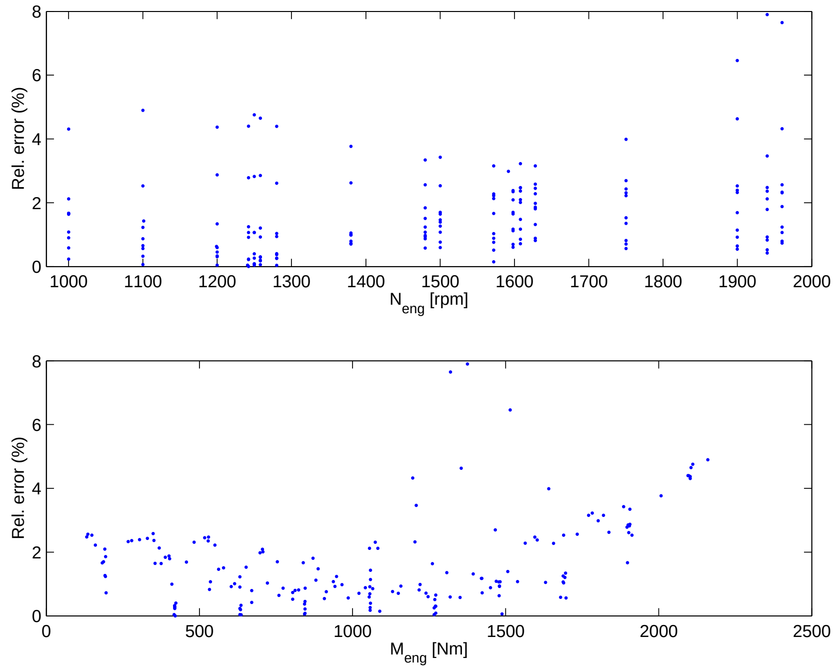

Figure 9: Relative error for simulated p_{em} sorted w.r.t. N_{eng} and M_{eng} respectively.

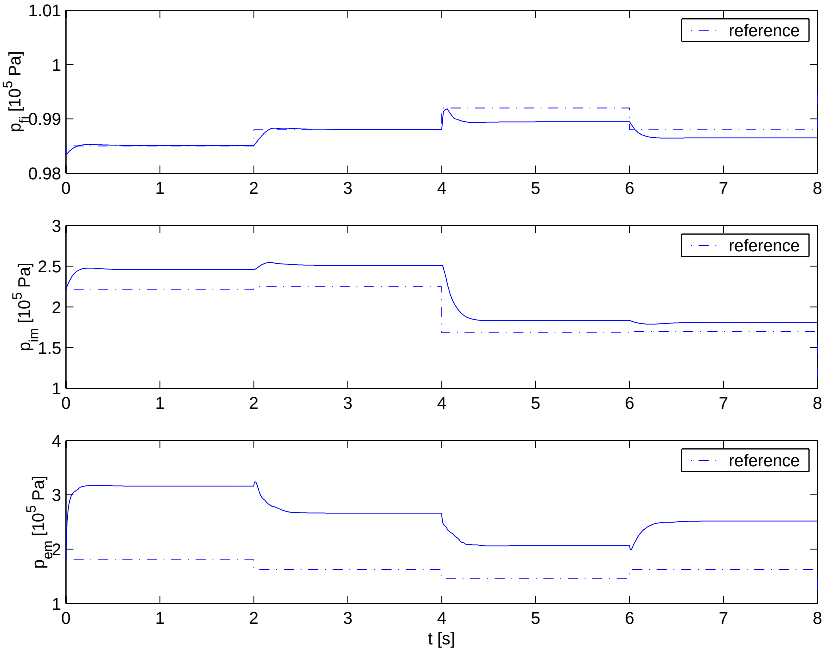

Figure 10: Step response for p_{fi}, p_{im} and p_{em}.

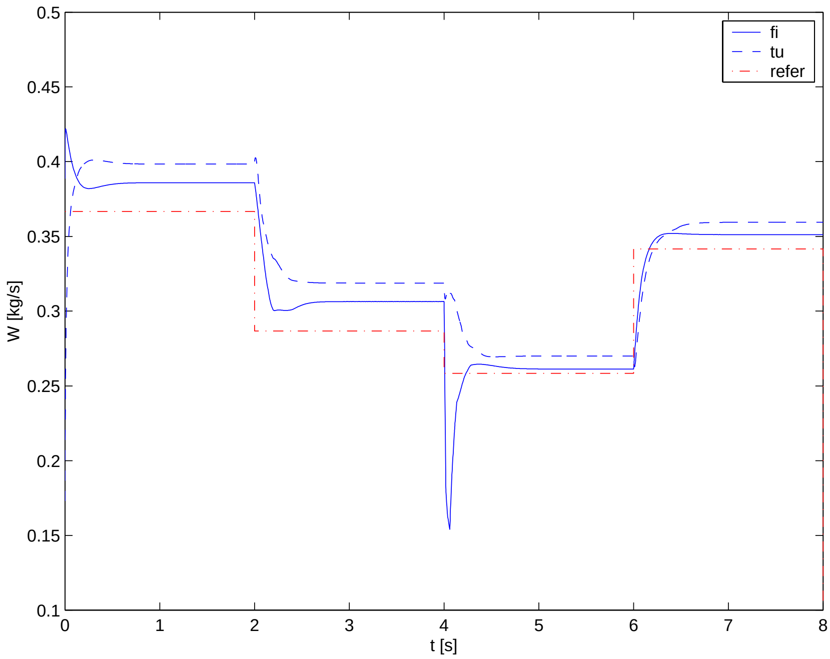

Figure 11: Step response for W_{fi}.

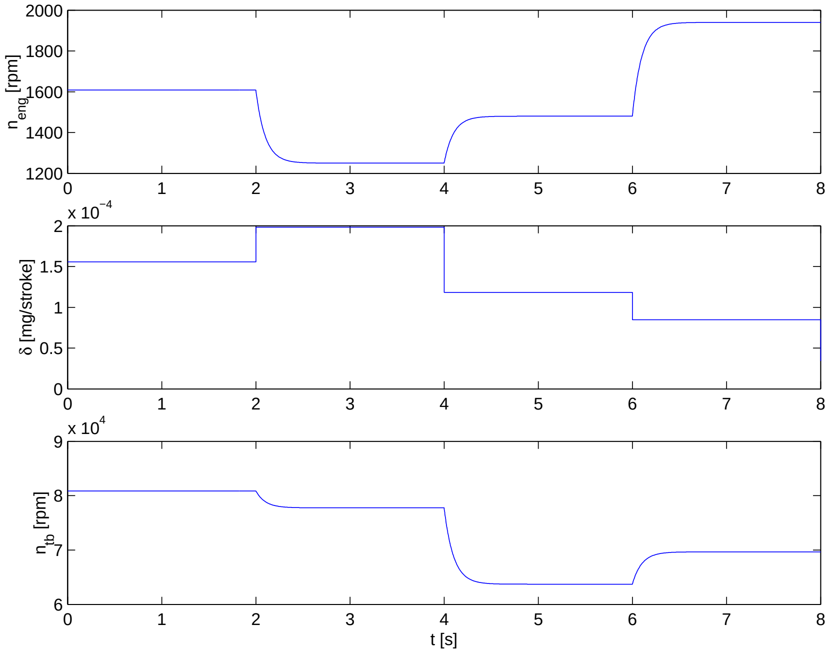

Figure 12: Reference values for the step response.

6 Model Errors and Further Improvements

The most notable model errors and model defects are listed below. Listed are also suggestions for how these defects can be removed.

EGR: No accurate model of the EGR system has been implemented.

In Section 3.7, an EGR model is described. This model has to be tested and verified during static and dynamic measurements.

Heat losses: Heat losses have not been implemented. The heat losses with large impact on the system are the losses in inlet- and exhaust manifolds. The heating of gases in inlet manifold and cooling of gases in exhaust manifold will impact the dynamic behavior of the system.

In (Müller, 1998) a model for the exhaust manifold heat losses is described. This model can be implemented in the engine model. To validate the model, dynamic measurements are needed. During a step response the time constants for the heat losses are visible.

Turbo: The turbo speed is assumed to be an input in the second model.

It seems difficult to implement a good turbo speed model. Note that (Pettersson, 2000) failed in the implementation of dynamic turbo. A successfull implementation is described in (Müller, 1998).

When performing tests in test cells the turbo speed can be seen as input. If the turbo speed can't be measured, a solution is to map the turbo speed and use this as input to the model. The turbo dynamics can partly be reproduced with low pass filtering, i.e., time constant found w.r.t. turboshaft inertia.

Low mass flow: For low mass flows the simulations fails.

This is a numerical problem that has to be isolated.

6.1 Further Measurements

For a complete validation of the model, the measurements in Table 1 have to be performed.

Table 1: Required measurements.

Type	EGR	Description	Identified parameters
Static	Closed	Mapping of engine.	k_{fi}, $\{k, s\}_{co}$, t_{tu}, $\{k, \eta\}_{ic}$, η_{vol}, k_{ep}
Static	Nominal	Mapping of engine.	$\{\eta, A\}_{egr}$
Dynamic	Closed	Step responses.	Q_{im}, Q_{em} (Validation: V_x etc.)
Dynamic	Nominal	Transient cycle.	Complete validation

6.2 Sensors

Besides the "standard" sensors the following variables have to be measured.

- n_{tb} – Turbo speed.
- α_{egr} – The true angle of the egr valve. The angle has to be measured to be able to validate a correct relationship between actuator signal and valve angle.

From the ECU, δ, α, u_{egr} and sensor data have to be collected.

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A EGR Valve Angle Model

Note: Some parts of this text have been removed due to corporate secretes.

The amount of EGR gases in inlet manifold are predicted with the EGR model. In the model it is assumed that EGR valve angle, α_{egr}, can be predicted from EGR control signal, u_{egr}. In the fault free case this is correct, since pneumatic actuators are designed to be linear, i.e., $\alpha_{egr} = f(l_{egr})$ and $l_{egr} \propto u_{egr}$. However, in some cases hysterics can cause large deviation from linearity.

In this section the results from a test of the EGR linearity are presented. Note that this is only a first experiment and its conclusions are not fully proved.

A.1 Experiment and Experimental Setup

Figure 13 shows a schematic overview of the experimental setup. The EGR valve is moved by the EGR arm. The EGR arm is connected to the EGR actuator. An inductive length sensor is attached to the EGR arm. For relatively small angles there is a linear relationship between actuator and sensor. The sensor is attached to minimize the angle.

The sensor was not correctly adjusted before experiment. The result of this is that for large EGR valve angles the sensor give constant values. The limit is about 49°, closed valve is x^{**}.

A.2 Model Construction

The model objective is to predict α_{egr} from u_{egr}. Figure 14 shows the trigonometric problem.

The sensor is linear and l is found from sensor signal y_{egr},

$$l = l_{min} + \frac{1}{0.32}(y_{egr} - min(y_{egr})).$$

Minimum $l_{min} = x$ and maximum $l_{max} = x$. The angle β is with law of cosines,

$$\beta(l) \quad = \quad \arccos\frac{a^2 + b^2 - l^2}{2ab},$$

where $a = x$ and $b = x$. Since minimum $\alpha_{egr} = 0$ and $\beta(l_{min}) = x$,

$$\alpha_{egr}(l) \quad = \quad \beta(l) - x.$$

Maximum $\alpha_{egr} = x$. The model for the EGR actuator is

$$l_{egr}(t) = C_1 u_{egr}(t + d) + C_2,$$

where $C_{1,2}$ and d are constants. The time delay is introduced to model time lag in the actuator. The time lag d is identified with least square minimization.

Note that the valve is closed for values below $x - x\% u_{egr}$. It is fully open for values above $x - x\% u_{egr}$. The effects from this is not in the model. The lower plot of Fig. 15 shows this effect for low values.

**Corporate secret.

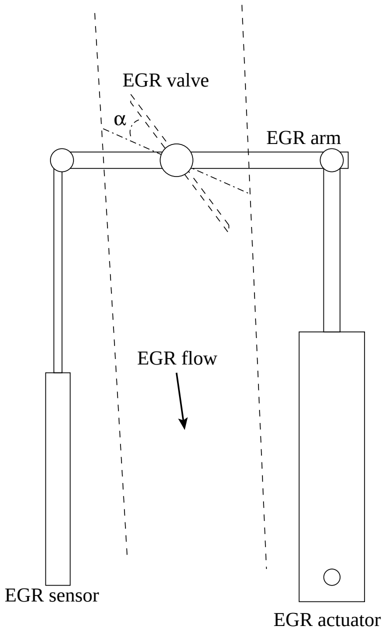

Figure 13: Schematic overview of experimental setup.

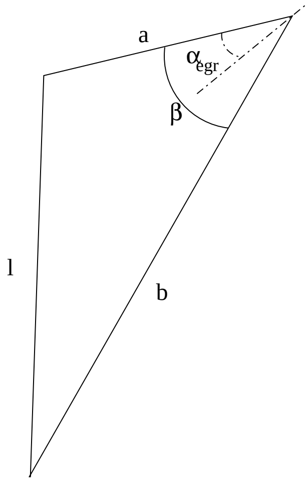

Figure 14: Trigonometric problem.

A.3 Identification and Validation

A transient cycle is used for identification and validation. Figure 15 shows EGR control signal and measured EGR valve angle. Solid line is EGR control signal and dashed line is measured EGR valve angle. As noted above the sensor is limited to 49°. Figure 16 shows four different time windows during a transient cycle. Solid lines are measured length. Dashed lines are model prediction. The time windows are marked in Fig. 15 with "*".

The data from time window two is used to identify the model described in Section A.2. The time windows are chosen so that the EGR valve is open during the entire time window. All time windows are used to validate the model. The model is identified with least square minimization. The time lag is xs.

The model prediction for time window four has a bias fault. The model prediction is good for time window two and three.

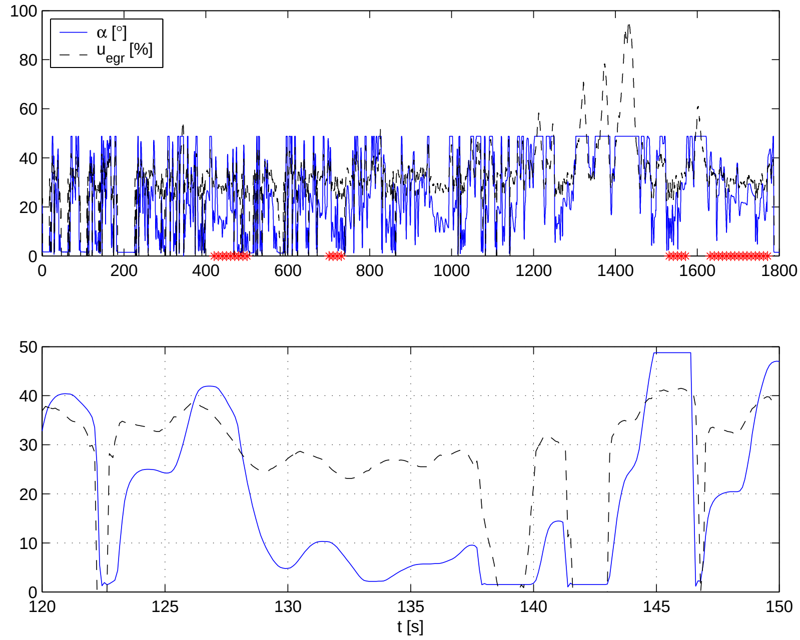

Figure 15: Solid line is EGR control signal and dashed line is measured EGR valve angle. The time windows used to identify the model are marked with “*”.

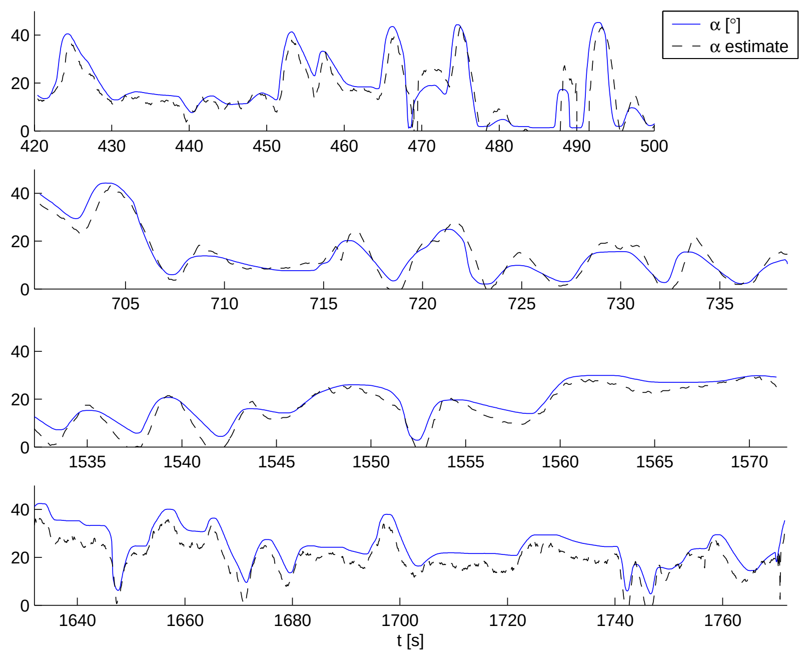

Figure 16: EGR valve angle. Measured and simulated.

B Notation

In the following table notations used in the model is described. See Table 2 for description of abbreviations.

Symbol	Value	Description	Unit
		PHYSICAL PROPERTIES	
component $x \in \{air, exh\}$			
c_{pX}	Con	Spec. heat capacity, constant pressure	$J/(kgK)$
c_{vX}	Con	Spec. heat capacity, constant volume	$J/(kgK)$
R_x	$c_{pX} - c_{vX}$	Gas constant	$J/(kgK)$
γ_x	c_{pX}/c_{vX}	-	-
		FLOW	
F	$[W, \nu, \chi]^T$	Flow between control volumes	-
W	Var	Mass flow	kg/s
ν	Var	Energy flow	J/s
χ	Var	Amount of exhaust gas	[0,1]
		CONTROL VOLUME	
m_{air}	S.Var	Mass of air	kg
m_{exh}	S.Var	Mass of exhaust gas	kg
χ	Var	Amount of exhaust gas	-
U	S.Var	Internal energy	J
$\dot{m}_{air}$	Var	Change of air mass	kg/s
$\dot{m}_{exh}$	Var	Change of exhaust gas mass	kg/s
$\dot{U}$	Var	Change of energy	J/s
p	Var	Pressure	Pa
T	Var	Temperature	K
V	Con	Volume	m^3
Q	Var	Heat losses	W
		FILTER	
k_{fi}	Con	Restrictor constant	$Pas^2/(m^3kg)$
		COMPRESSOR	
f_w	Map	Compressor flow	kg/s
f_e	Map	Compressor efficiency	-
M	Var	Compressor moment	Nm
F_{in}	Var	Flow in to compressor	-
F_{out}	Var	Flow out of compressor	-
		INTERCOOLER	
k_{ic}	Con	Restrictor constant	$Pas^2/(m^3kg)$
T_{icSurr}	T_{amb}	Cooler temperature	K
η_{ic}	Con	Cooler efficiency	-
		COMBUSTION CHAMBER	
δ	Act	Amount of injected fuel	$kg/stroke$
α	Act	Ignition angle	rad
ρ_{im}	Var	Density in inlet manifold	kg/m^3

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Symbol	Value	Description	Unit
V_d	Con	Displacement volume (1 cylinder)	m^3
V_{cyl}	Con	Cylinder volume (1 cylinder)	m^3
r_c	x	Compression ratio	$-$
η_{volEm}	Var	Theoretic volumetric efficiency	$-$
η_{vol}	Map	Volumetric efficiency	$-$
f_{temp}	Map	Temperature increase	K
n_{cyl}	x	Number of cylinders	$-$
N_{eng}	Var	Engine speed	rpm
λ_{eng}	Var	Air/fuel equivalence ratio	$-$
$(A/F)_s$	Con	Stoichiometric air to fuel ratio.	$-$
		EGR	
$T_{egrSurr}$	Con	Cooler temperature	K
η_{egr}	Con	Cooler efficiency	$-$
A_{egr}	Map	Effective area of EGR valve opening	m^2
u_{egr}	Act	EGR valve control-signal	-
α_{egr}	Var	EGR valve angle	rad
l_{egr}	Var	Length of egr actuator	m
		TURBINE	
f_w	Map	Turbine flow	kg/s
f_e	Map	Turbine efficiency	$-$
M	Var	Turbine moment	Nm
J_{tb}	Con	Turbo inertia	s^2Nm
u_{tb}	Act	Turbo variable geometry signal	-
		EXHAUST PIPE	
k_{ep}	Con	Restrictor constant	$Pas^2/(m^3kg)$

Table 2: Abbreviations used in this report.

Abbreviation	Explanation
Act	Actuator
Con	Constant
CV	Control Volume
ECU	Electronic Control Unit
EGR	Exhaust Gas Recirculation
HDD	Heavy Duty Diesel
MVEM	Mean Value Engine Model
OBD	On Board Diagnostics
RPM	Revolutions Per Minute
S.Var	State variable
Var	Variable
VGT	Variable Geometry Turbo-charger
VNT	Variable Nozzle Turbine

C Simulink blocks

The main parts of the simulink models are included in this appendix. Figure 17 shows the top level of the engine model. Figure 18 shows the first layer, it includes the inlet , see Figure 19, the combustion chamber, see Figure 20, and the outlet, see Figure 21. The models are collected in a simulink library shown in Figure 22.

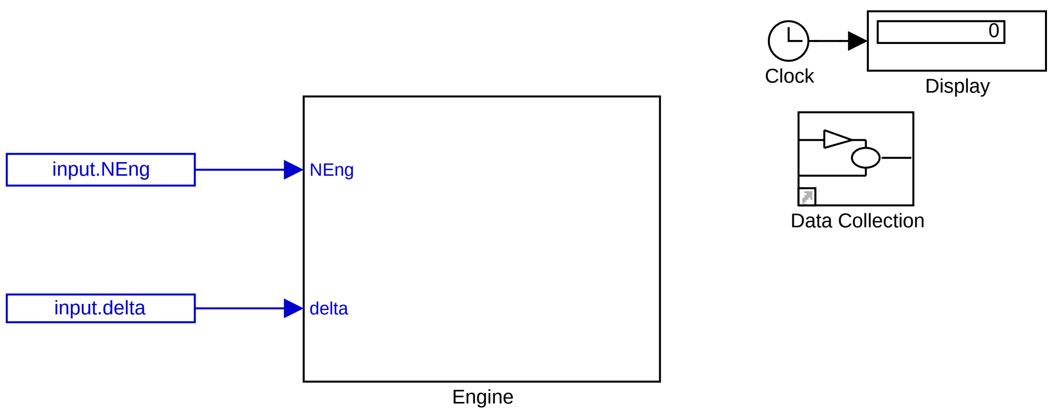

Figure 17: The top level of the simulink model.

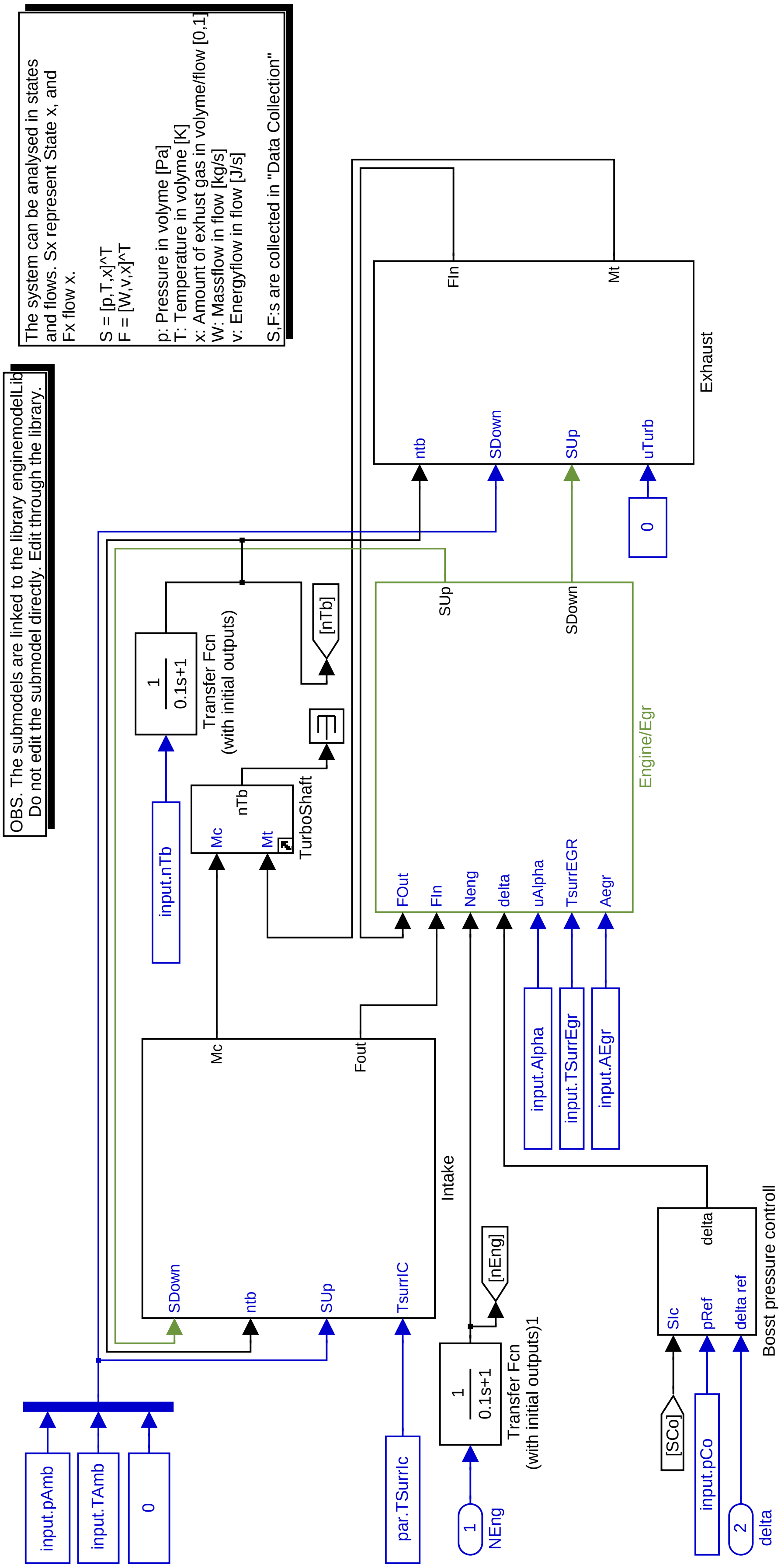

Figure 18: The first layer of the engine model. It includes the inlet, combustion, and finally outlet parts.

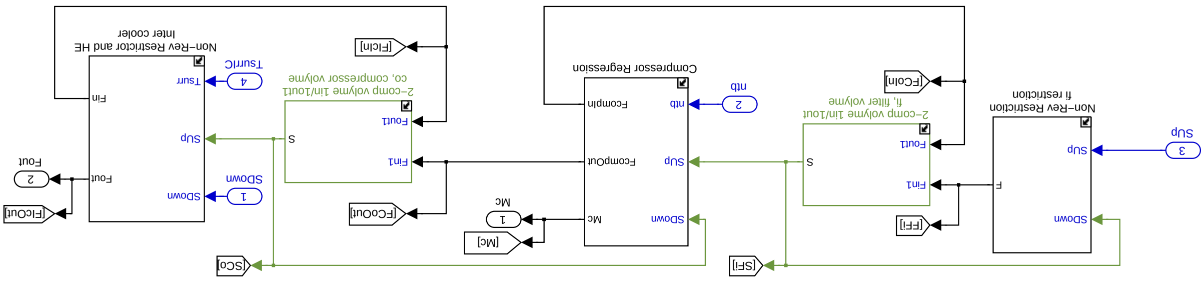

Figure 19: The inlet side of the engine-model.

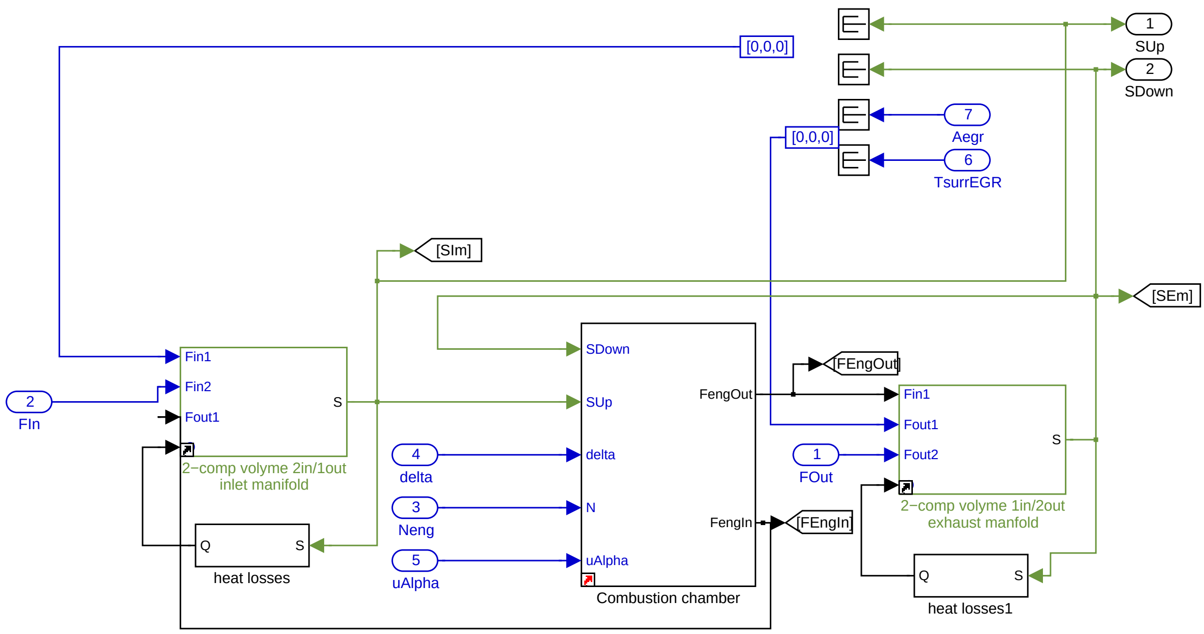

Figure 20: The central combustion part of the engine-model. Notice that this configuration of the engine does not include a EGR system.

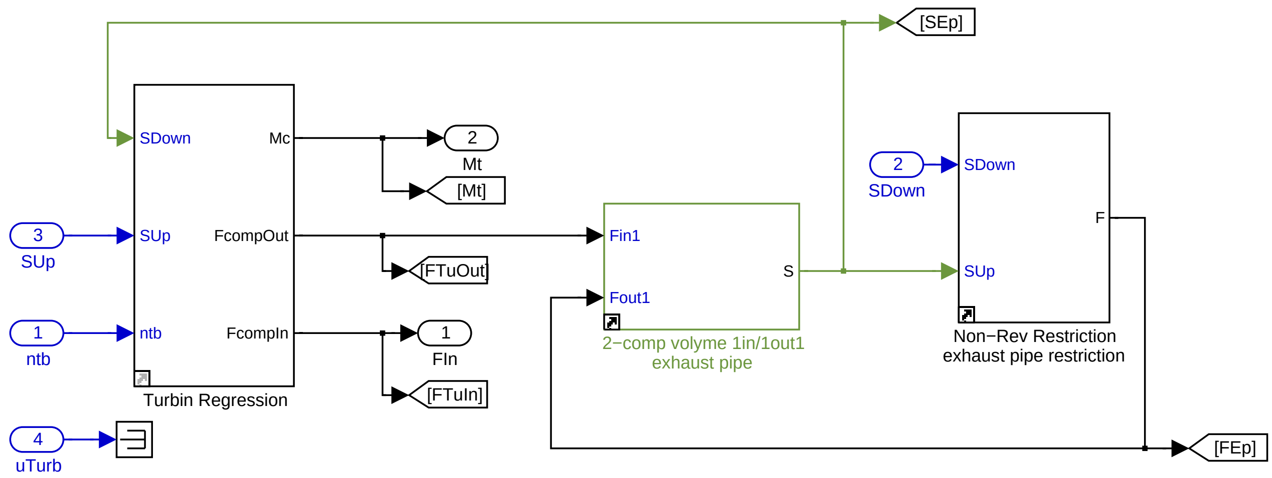

Figure 21: The outlet side of the engine-model.

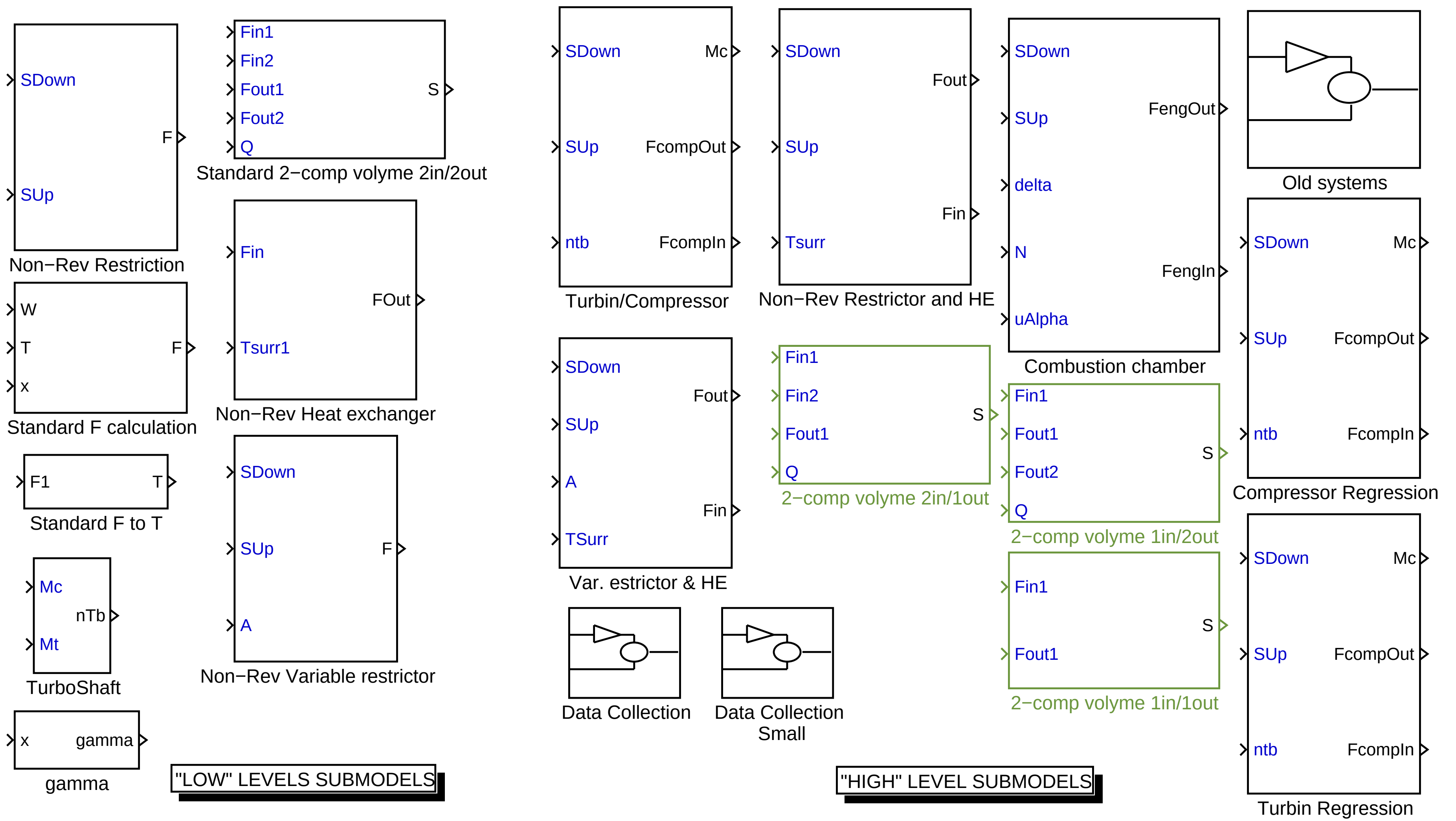

Figure 22: The library including the sub-models used to construct the engine-model.