The Application and Advantages of a Generic Component-Based SI/CI Engine Model with VVA Compatibility ⋆

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Abstract: Engine models developed for control purposes are often developed with a time schedule in mind under the pressure of deadlines without reusability in mind and end up being hard-coded with a single engine type usage in mind. This approach can lead to more work when a new engine is created and a model is to be developed, as it usually takes less time than changing or modifying the old one. To facilitate a more rapid development process, there is a desire to have control-oriented models that can be adapted to new types of hardware with ease while at the same time providing fundamental insights into the physics of the engine that limit the control performance. The main idea is to use a component-based structure where the components are designed to be reused for similar processes; when combined, it constitutes a generic engine model with parametrization and compatibility with VVT/VVA and SI/CI combustion. An open-source mean value engine model was created in MATLAB/Simulink to meet the objectives. The engine model describes components such as the air filter, intercooler, and exhaust system components as incompressible flow restrictions. Bypass, throttle, intake/exhaust valves, and wastegate are modeled as compressible flow restrictions. Adiabatic control volumes are placed between each component to keep track of masses, pressures, and temperatures. The remaining components are modeled separately, with unique functions for each model, while integrated into the component-based structure. To demonstrate the concept and the generality of the approach, two engines, a 6-cylinder 12.7-liter Scania diesel engine, and a 4-cylinder 2.0-liter Volvo petrol engine, are used as case studies. The generic simulation platform is parameterized and validated against experimental data for both engines.

Keywords: Engine modeling, Parametrization, Mean value model, Component-based structure

1. INTRODUCTION

Engine models are often developed with a specific control purpose, sometimes even hard-coded for a single engine type. This approach presents a challenge when new engines are developed, as creating a new model is usually quicker than modifying an existing one. Control-oriented models that can easily adapt to new hardware types are needed to expedite development. These models should also provide fundamental insights into the engine physics that limits control performance. An open-source mean value MATLAB/Simulink model is created to meet these objectives. This generic engine model is parametrized and compatible with VVT/VVA and SI/CI combustion.

The model is built on a component-based structure designed for reusability. It includes components for the air filter, intercooler, and exhaust system, modeled as incompressible restrictions. Bypass, throttle, valves, and wastegate are modeled as compressible flow restrictions. Adiabatic control volumes are placed between each component to monitor pressures and temperatures. The additional components are modeled separately with unique functions. To demonstrate the generality of this approach, we've used two engines as case studies: a 6-cylinder 12.7-liter Scania diesel engine and a 4-cylinder 2.0-liter Volvo petrol engine. The generic simulation platform was parameterized and validated against experimental data for both engines. This paper aims to create a generic engine model that captures the dynamics in both SI and CI engines with VVA. The goal is a model with a parameterizing structure that makes it easy to fit for different engines. The model will simulate the engine's behavior from the air inlet to the exhaust system.

This objective is then divided into three goals.

- Implement a model that can handle both SI and CI combustion with VVA.
- Utilize the ability to change model equations simulated easily.
- Structurally present equations, parameters, and validation.

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1.1 Reason for Research

The primary motivation for developing a more generic engine model is to facilitate the simulation of various engine configurations and dependencies, eliminating the need to create a new model each time. If a generic engine model can demonstrate a strong correlation with real engines across various engines and displacements. In that case, it can be inferred that the model applies to various engine configurations.

Such a model can be utilized in both engine simulations and controller development. It could also be employed in real-time applications, allowing for direct selection of control parameters in the Engine Control Unit (ECU).

To validate this model, it will be tested using two distinct engines: a 2.0-liter turbocharged petrol engine from Volvo and a 12.7-liter turbocharged diesel engine from Scania.

1.2 Related Research

In engine modeling, various models are utilized, including transfer function models, cycle mean value models (MVEM), zero- or one-dimensional models, and computational fluid dynamic models Theotokatos et al. (2018). MVEM compromises the simpler transfer function models and the more detailed zero- or one-dimensional models. These are particularly useful in the design of the engine control system, where rapid simulation times are crucial Theotokatos et al. (2018). This makes MVEM suitable for Hardware-in-the-Loop (HIL) simulations, as the model needs to capture engine behavior accurately Maroteaux and Saad (2015).

The MVEM approach is favored in modeling due to its low computational power requirements, stemming from using nonlinear ordinary differential equations Eriksson et al. (2002). This allows for faster-than-real-time simulations Llamas and Eriksson (2019), making it ideal for controller development and tuning. In 2001, a component-based MVEM was developed in Modelica. This included a detailed explanation of the model structures, the Modelica Language, and the models' validations. This methodology provides a deeper understanding of the potential of the component-based modeling approach Brugård et al. (2001). Component-based modeling involves utilizing derived models from literature and building your model one component at a time Eriksson (2003). In engine modeling, components can include the air filter, throttle, intake manifold, and cylinder. These models must then be validated to capture the system dynamics accurately.

Different parts of the engine model assume different types of flows, which can generally be categorized as compressible or incompressible. Most gas flows in engine pipes are incompressible and turbulent. However, a compressible flow should be assumed over most types of valves. This is particularly important for components like the throttle and the opening of an intake/exhaust valve Eriksson and Nielsen (2014).

Several open-source engine models are currently available. One such model is a modified MVEM for Spark Ignition (SI) engines with Exhaust Gas Recirculation (EGR) using the Simulink environment. The modification involved replacing the usual isothermal models for manifolds with a model for temperature dependency. The results found that neglecting the instances that change the input regime in the manifold air temperature is approximately constant Mostofi et al. (2006).

MVEM can also be used to validate models for control purposes, such as the validation of a control model for an electronic throttle Chaing et al. (2007).

LiU-Diesel is a MATLAB/Simulink model simulating a Diesel Engine with EGR and Variable Geometry Turbocharger (VGT) Wahlström and Eriksson (2011). Further development of the LiU-Diesel, called LiU-Diesel2, adds compatibility of Throttle and Turbocharger with Wastegate Ekberg et al. (2018).

1.3 Description of Modelling Concept

The MVEM is utilized to analyze and design control and diagnostic systems. This is due to the MVEM's ability to describe variations that occur faster than an engine cycle. MVEM models are sometimes called control-oriented models or filling and emptying models.

The process of averaging over one or several cycles in MVEM modeling necessitates a thorough investigation of physical sensor and actuator dynamics. These components are crucial as they actuate and measure the actual engine Eriksson and Nielsen (2014).

The MVEM will employ the 'filling and emptying' method for the control volumes, including the intake and exhaust manifold. This method divides volumes into sections, as the manifolds are represented as finite volumes Heywood (1988).

1.4 Model Limitations

Items that are outside the scope of this paper are:

- Fuel spray models
- Thermal stress and solid mechanics that affect geometries
- Validation of EGR model
- Warm-up process of the engine
- Formation of emissions
- Implementation of control system
- Variable compression by varying stroke length
- Start-stop technology and functionality

2. ENGINE COMPONENTS

The engine consists of multiple components. The components can be seen in Fig. 1, and the flow is positive when flowing to the right. The engine components include the air filter, compressor, intercooler, throttle, intake and exhaust, EGR, WG, turbine, after-treatment, and the rest of the exhaust system.

2.1 Parameter Estimation

There are three different types of measurement data

- Stationary (also called the engine map)
- Dynamic

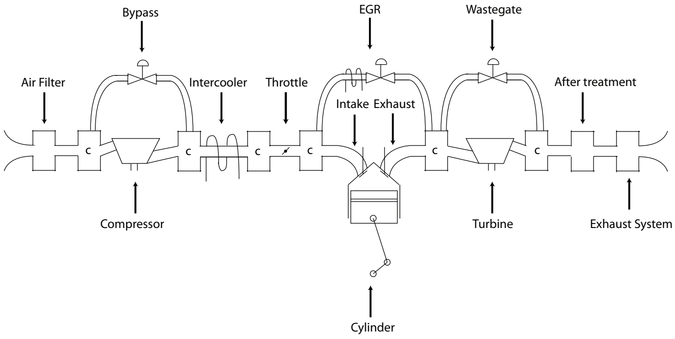

Fig. 1. The engine components include air filter, compressor, intercooler, throttle, intake and exhaust, EGR, WG, Turbine, after treatment, and the rest of the exhaust system. In this paper, the after-treatment and exhaust system is modeled as the "Exhaust system."

- Crankshaft based.

From that data, all the parameters can be found mainly using the least-squares method. The least-squares method can be divided into two categories: linear and non-linear. For a more detailed description of the least-squares method, see Björck (1996).

2.2 Incompressible Flow Model

There are different kinds of flow restrictions among the engine components. The flow can be assumed incompressible for the air filter, intercooler, and exhaust system (including the catalyst and muffler). This means that the throttle and cylinder flows will be modeled differently.

The model selected for the incompressible flow restrictions is found in Eriksson and Nielsen (2014)

$$\dot{m} = \begin{cases} C_{tu}\sqrt{\dfrac{p_{us}}{RT_{us}}}\sqrt{p_{us}-p_{ds}}, & \text{if } p_{us}-p_{ds} \geqslant \Delta p_{lin} \\ C_{tu}\sqrt{\dfrac{p_{us}}{RT_{us}}}\dfrac{p_{us}-p_{ds}}{\Delta p_{lin}}, & \text{otherwise} \end{cases} \tag{1}$$

$$T_{flow} = T_{us} \tag{2}$$

The parameters for this model are C_{tu} and Δp_{lin}.

If p_us = p_ds the Lipschitz condition is not fulfilled as the derivative goes towards infinity, which causes problems for the ODE solver. In this paper, ODE15s were used. p_li is the linear pressure region, and it is assumed p_li $\geq$ 1000 pa.

2.3 Control Volume Model

Control volumes can be divided into two different models. One is the isothermal model, where it can be assumed that the temperature is constant for the whole control volume. The other one is the adiabatic model, which means the heat transfer is often set to zero. All control volumes in the implemented model are adiabatic also to see temperature variations. The model implemented has pressure or mass and temperature as states, p, m, and T .

$$\frac{dT}{dt} = \frac{RT}{pVc_v}\left[\dot{m}_{in}c_v(T_{in}-T) + R(T_{in}\dot{m}_{in} - T\dot{m}_{out}) - \dot{Q}\right] \tag{3}$$

$$\frac{dp}{dt} = \frac{RT}{V}(\dot{m}_{in} - \dot{m}_{out}) + \frac{p}{T}\frac{dT}{dt} \tag{4}$$

$$\frac{dm}{dt} = \sum_i \dot{m}_n \tag{5}$$

The parameter for this model is V, the volume of each control volume. $\dot{m}_{in}$ is the sum of all positive flows into the volume, and $\dot{m}_{out}$ is the sum of all negative flows into the volume. T_{in} is the temperature of the gas flowing into the volume and is modeled as a mean value for all the flow flowing into the volume, assuming the same c_v for all flowing fluids in the volume. This means

$$T_{in} = \begin{cases} \dfrac{\dot{m}_1 \cdot T_1}{\dot{m}_{in}} + \dfrac{\dot{m}_2 \cdot T_2}{\dot{m}_{in}} + ... + \dfrac{\dot{m}_n \cdot T_n}{\dot{m}_{in}} & \dot{m}_{in} > 0 \\ 0 & \text{otherwise} \end{cases} \tag{6}$$

where n is the flow with the corresponding temperature for inflows into the control volume.

when the mass state is used, pressure is determined using the ideal gas law,

$$p = \frac{mRT}{V}. \tag{7}$$

If the control volume is placed in the intake or exhaust manifold, the gas composition, x, is modeled as the fractional content of the specimens in the total gas mixture,

$$\frac{dx}{dt} = \frac{1}{m}\sum_j (x_j - x)\dot{m}_j \tag{8}$$

where the index j indicates the contents of the specimen in the respective flow.

For the isothermal model, the incompressible flow restriction is

$$\frac{dp}{dt} = \frac{RT}{V}(\dot{m}_{in} - \dot{m}_{out}) \tag{9}$$

$$T = T_{in} = T_{out} \tag{10}$$

where the only parameter is V, the volume of each control volume.

2.4 Compressible Flow Model

The flow can be assumed to be compressible over the throttle, bypass, wastegate, and valves. Validation for the throttle can be found in Lind Jonsson (2021). The flow model is based on a compressible flow approach Eriksson and Nielsen (2014).

$$\Pi = \max\left(\frac{p_{ds}}{p_{us}}, \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}}\right), \tag{11}$$

$$\dot{m} = \frac{p_{us}}{\sqrt{RT_{us}}}A_{eff}\Psi_{li}(\Pi), \tag{12}$$

$$\Psi_0 = \sqrt{\frac{2\gamma}{\gamma-1}\left(\Pi^{\frac{2}{\gamma}} - \Pi^{\frac{\gamma+1}{\gamma}}\right)}, \tag{13}$$

$$\Psi_{li} = \begin{cases} \Psi_0 & \text{if } \Pi \leq \Pi_{li} \\ \Psi_0\dfrac{1-\Pi}{1-\Pi_{li}} & \text{otherwise,} \end{cases} \tag{14}$$

where the parameters are Π_{li} that is defining the linear region.

The effective area is

$$A_{eff} = A_0 + A_1\alpha_{th} + A_2\alpha_{th}^2 \quad \text{for throttle,} \tag{15}$$

$$= C_D \frac{D_v^2\pi}{4} f([0..1]) \quad \text{for bypass/wastegate,} \tag{16}$$

$$= C_{D,in/ex}\frac{D_v^2\pi}{4} \quad \text{for intake/exhaust valves.} \tag{17}$$

where C_D for the intake and exhaust valves are

$$C_{D,in} = f_{\text{lookup}}(L_{in}(\theta)) \tag{18}$$

$$C_{D,ex} = f_{\text{lookup}}(L_{ex}(\theta), Pr) \qquad Pr = \frac{p_{cyl}}{p_{em}}. \tag{19}$$

2.5 Turbo

The compressor and turbine model is presented and validated in Lind Jonsson (2021) for the Volvo engine. The ellipse model tuned and validated in LiU CPgui Llamas and Eriksson (2018) was used for the Scania engine.

2.6 Turbo Dynamics Model

The turboshaft friction is modeled according to

$$Tq_{tc,fric} = c_{tc}\omega_{tc} \tag{20}$$

where the friction is assumed to be low since the shaft is oil lubricated, meaning $c_{tc} = 1 \cdot 10^{-6}$ Nm/(rad/s). Newton's second law of rotation is then used to model the turbo shaft speed

$$\frac{d\omega_{tc}}{dt} = \frac{1}{J_{tc}}\left(Tq_t - Tq_c - Tq_{tc,fric}\right) \tag{21}$$

The parameters for this model are J_{tc}, which is the inertia of the shaft.

2.7 Intercooler Model

The Intercooler is modeled as an incompressible flow.What differentiates the intercooler from the incompressible flow model is the model for the temperature Eriksson and Nielsen (2014).

$$T_o = T_i - \epsilon(\dot{m}_{cool}, \dot{m}_{ic}, ...)(T_i - T_{cool}) \tag{22}$$

$$\epsilon = a_0 + a_1\left(\frac{T_i + T_{cool}}{2}\right) + a_2\dot{m}_{air} \tag{23}$$

where T_o is the temperature of the gas flowing out of the intercooler, T_i is the temperature of the gas flowing into the intercooler, T_{cool} is the temperature of the air surrounding the intercooler, which in this model is set to T_{amb}, $\dot{m}_{air}$ is the mass flow of the gas flowing into the intercooler and $\dot{m}_{cool}$ is the mass flow of the gas cooling the intercooler. The parameters for this model are a_0, a_1 and a_2.

2.8 Cylinder Model

To fully capture the dynamics of the oxygen concentration, cylinder- temperature, and pressure using VVA, each cylinder has to be simulated separately with full-cycle cylinder states Johansson (2019).

Cylinder Flow Four different flows—intake, exhaust, CRB, and blowby—are modeled as the compressible flow restriction.

Combustion Modeling Combustion is affected by the amount of available oxygen. Therefore, the oxygen concentration $X_{O,cyl}$ is a state. This also gives the $X_{burned} = 1 - X_{O,cyl} - X_{fuel}$, which is the amount of oxygen that has reacted in the combustion. A perfect stoichiometric combustion is assumed, meaning particle formation has been neglected. Combustion is assumed to be a reaction between air and hydrocarbons.

As the most common element in the air is nitrogen, all elements except oxygen are clumped together, creating a common assumption Eriksson and Nielsen (2014)

$$\mathrm{C}_{air} = \mathrm{O_2} + 3.773\mathrm{N_2} \tag{24}$$

Exhaust composition depends on what fuel is assumed to be burning during combustion. The fuel used is cetane for CI combustion and isooctane for SI combustion. The burned composition is.

$$\mathrm{C}_{burned} = \mathrm{aCO_2} + \mathrm{b/2H_2O} + 3.773\,(\mathrm{a} + \mathrm{b/4N_2}) \tag{25}$$

The chemical reaction in the combustion is

$$\begin{aligned} &n_{fuel}C_aH_b + n_{air}\mathrm{C}_{air} + \mathrm{n_{burned}}\mathrm{C}_{burned} \longrightarrow \\ &(n_{fuel} + n_{burned})\mathrm{C}_{burned} + \\ &+ (n_{air} - n_{fuel}(a + b/4))\,\mathrm{C}_{air} \end{aligned} \tag{26}$$

where the fuel composition is C_aH_b.

The number of moles in the reaction is calculated at IVC.

$$n_{tot,air} = 1 + 3.773 \quad n_{tot,burned} = a + \frac{b}{2} + 3.773 \tag{27}$$

$$n_{air} = \frac{m_{IVC}X_{O,cyl}}{M_{air}n_{tot,air}} \quad n_{fuel} = \frac{m_{fuel}}{M_{fuel}} \tag{28}$$

$$n_{burned} = \frac{(1 - X_{O,cyl})\,m_{IVC}}{M_{burned}n_{tot,burned}} \tag{29}$$

where m_{IVC} is the mass in the cylinder at IVC. The mass in the cylinder is calculated by

$$m_{cyl} = \frac{p_{cyl}V_d}{R_{cyl}T_{cyl}} \tag{30}$$

The $X_{O,cyl}$ state is modelled similarly as the fractions x in the manifolds, with the stoichiometric combustion assumption

$$\frac{dx}{dt} = \frac{1}{m_{cyl}}\sum_i (x_i - x)\,\dot{m}_i - S_{AF_s}C\frac{dx_b}{dt} \tag{31}$$

where the Vibe function in equation (33) gives $\frac{dx_b}{dt} = \frac{dx_b\theta}{d\theta}\omega_e$, S_{AF_s} scales the composition with regards to the fraction burned, assuming (AF_s-1) used air per 1 used fuel and C is the scaling of the Vibe function to make the Vibe function scale properly to equation (26).

$$x_b(\theta) = \begin{cases} 0, & \theta < \theta_{SOC} \\ 1 - e^{-a\left(\frac{\theta - \theta_{SOC}}{\Delta\theta}\right)^{m+1}}, & \theta \geq \theta_{SOC} \end{cases} \tag{32}$$

x_b determines the fraction burned, from 0 to 1. $\Delta\theta$ and a are related to combustion duration, m affects the shape and θ_{SOC} denotes the start of combustion.

For heat release calculations, the Vibe function is derived

$$\frac{dx_b\theta}{d\theta} = \frac{a(m+1)}{\Delta\theta}\left(\frac{\theta-\theta_{SOC}}{\Delta\theta}\right)^m e^{-a\left(\frac{\theta-\theta_{SOC}}{\Delta\theta}\right)^{m+1}} \quad (33)$$

The molar fraction x_{air} is the amount of free air that has not yet reacted in the combustion. The amount of free air is directly correlated to the amount of oxygen according to the air assumption in equation (24). The scaling factor C is modeled as the amount of air at IVC and air left after combustion.

$$n_{comb} = n_{air} - n_{fuel}\,(a + b/4) \quad (34)$$

$$\tilde{x}_{air} = \frac{n_{comb}n_{tot,air}}{n_{comb}n_{tot,air} + n_{comb}n_{tot,Burned}} \quad (35)$$

$$x_{air} = \frac{\tilde{x}_{air}M_{air}}{(1-\tilde{x}_{air})\,M_{Bunrned} + \tilde{x}_{air}M_{air}} \quad (36)$$

This gives the scaling of the Vibe function

$$C = x_{air,IVC} - x_{air,aC} \quad (37)$$

Since the gas constant and heat capacity change with oxygen concentration and temperature, NASA polynomials are used to decide the heat capacities $c_{p,Burned}$ and $c_{p,air}$ according to equation (38).

The specific heat ratio depends on the temperature. NASA polynomial is a database for how different chemical species' specific heat ratio changes with temperature. The NASA polynomial can be read more in McBride (2002).

$$\frac{\tilde{c}_p(T)}{\tilde{R}} = a_1 + a_2T + a_3T^2 + a_4T^3 + a_5T^4 \quad (38)$$

The gas mass-specific constant and specific heat is calculated as

$$R = (1 - X_{o,cyl})R_{Burned} + X_{o,cyl}R_{air} \quad (39)$$

$$c_p = (1 - X_{o,cyl})c_{p,Burned} + X_{o,cyl}c_{p,air} \quad (40)$$

The ideal gas assumption is used to get the specific heat constant c_v

$$c_v = c_p - R \quad (41)$$

The parameters for the IVC event use intake temperature from the engine map, but the rest of the measurements are based on crankshaft measurements. The parameters for the Vibe function use pressures and temperatures from the crankshaft-based measurements.

Energy equations The energy flows in the cylinder are heat release, heat transfer, the work carried out, and internal energy. The heat release is calculated

$$\frac{dQ_{HR}(\theta)}{d\theta} = m_f Q_{LHV}\eta_{co}\frac{dx_b(\theta)}{d\theta} \quad (42)$$

The heat transfer is calculated using the Woschini method. However, the model for the heat transfer coefficient is taken from Eriksson and Nielsen (2014)

$$h = C_0 B^{-0.2}p^{0.8}w^{0.8}T^{-0.53} \quad (43)$$

where $C_0 = 1.30 \cdot 10^{-2}$, B is the cylinder bore, p is the pressure in the cylinder, T is the temperature in the cylinder and w is the characteristic velocity equation given by

$$w = C_1\bar{S}_p + C_2\frac{VT_{IVC}}{V_{IVC}p_{IVC}}\,(p - p_m) \quad (44)$$

where $\bar{S}_p = \frac{2aN_e}{60}$ is the mean piston speed, p_m is the motored pressure, and the constants C_1 and C_2 are dependent on what stroke the engine has, and can be seen in Table 1.

Table 1. Constants used in Woschini's model for the heat transfer coefficient.

	Gas Exchange	Compression Compression	Combustion and Expansion
C_1	6.18	2.28	2.28
C_2	0	0	0.00324

The motored pressure is modeled with a polytope, where κ is the polytropic exponent.

$$p_m(\theta) = \begin{cases} p(\theta), & \text{if } \theta \le \theta_{SOC} \\ p\,(\theta_{SOC})\left(\frac{V\,(\theta_{SOC})}{V(\theta)}\right)^\kappa & \text{if } \theta > \theta_{SOC} \end{cases} \quad (45)$$

where the polytropic exponent κ is set to a constant value, optimized from a motored cycle. Another way to get the motored pressure is to use the pressure from a measurement in an engine test cell.

The power is calculated using Newton's second law of motion.

$$\dot{W} = p_{cyl}\frac{dV}{dt} \quad (46)$$

and the internal energy is modeled

$$u = c_vT_{cyl} \quad (47)$$

State equations The temperature and pressure states are T_{cyl} and p_{cyl}.

$$h = c_pT_{us} \quad (48)$$

$$\dot{T}_{cyl} = \frac{R_{cyl}T_{cyl}}{p_{cyl}Vc_{v,cyl}}\left(\dot{m}_{intake}\,(h_{intake} - u) - \sum_i \dot{m}_i\,(h_i - u) + \frac{dQ_{HR}(\theta)}{dt} - \dot{Q}_{HT} - W\right) \quad (49)$$

$$\dot{p}_{cyl} = \frac{R_{cyl}T_{cyl}}{V}\left(\dot{m}_{intake} - \sum_i \dot{m}_i\right) + \frac{p_{cyl}}{T_{cyl}}\dot{T}_{cyl} - \frac{W}{V} \quad (50)$$

where i is the flow for exhaust, CRB, and blowby.

Generated Engine Torque A simple instantaneous torque model that neglects friction is used in this paper Eriksson and Nielsen (2014)

$$M_{e,i}(\theta) = \sum_{j=1}^{n_{cyl}}\left(p_{cyl,j}\left(\theta - \theta_j^{\text{offset}}\right) - p_{amb}\right) A\,L\left(\theta - \theta_j^{\text{offset}}\right) \quad (51)$$

where θ_j^{offset} is the crank angle offset for cylinder j, A is the area and $L(\theta)$ is the crank lever.

$$A\,L(\theta) = \frac{dV(\theta)}{d\theta} = \frac{dV(\theta)}{dt}\frac{1}{\omega_e} \quad (52)$$

For example, the average torque is of interest so that it does not exceed any loads for the driveline. The average torque is then calculated over four strokes Eriksson and Nielsen (2014).

where M_f denotes the friction, neglected in this paper.

2.9 Complete Model

The complete Simulink model can be seen in Fig. 2 and is run by first running the init.m file setting up the necessary model input and loading the parameter struct.

3. RESULTS

3.1 Validation Points

Stationary measurements are done after the engine has stabilized, meaning all dynamic behavior has had time to subside. Typically, measurements are taken over a brief period, usually a few seconds, and then an average value for that specific operating point is calculated and recorded. The stationary measurements are systematically conducted in a series, moving from one operating point to the next, ensuring comprehensive data collection. An example of a complete engine map can be seen in Fig. 3. The middle point is presented in this paper, and the other operating points are given in Lind Jonsson (2021).

During the validation, pressure is used instead of mass, as it is easier to relate to pressures than masses. Some components were not present on the Volvo and Scania engines and were not simulated. This meant some inputs were kept the same for each iteration. The inputs for the Scania engine were kept the same: throttle angle, wastegate, bypass, and EGR actuation, and they were set to zero. The inputs for the Volvo engine that were kept the same were bypass, EGR, and CRB actuation, which were set to zero. Setting inputs to zero has the same effect as if they were non-existent.

The middle operating point is presented here to validate the model, while all are covered in Lind Jonsson (2021). In Tables 3 to 4 below the parameters are pressure, temperature, turbo shaft speed, engine torque, mass flows and oxygen concentration. The parameters for pressure are after the compressor, p_c, in the intake manifold, p_im, and after the exhaust manifold, p_em. The temperature parameters are after the compressor, T_c, in the intake manifold, T_im, after the exhaust manifold, T_em, and after the turbine, T_t. The parameter for the turbo shaft speed is w_tc, and the parameter for the generated engine torque is Tq_e. The parameter for oxygen concentration is after the exhaust manifold, Xo_em, but a reference for this measurement is only available for the Scania engine. The units used to validate the model are *bar* for pressure, $^\circ C$ for temperature, *kRPM* for turbo speed, *Nm* for generated torque, *g/s* for mass flows and % for oxygen concentration.

Results are presented when the model has reached a steady value. The simulation time was set to 10 seconds, ensuring that all dynamics were gone at the end of the simulation. Measured data was investigated on a cycle-to-cycle basis. The simulation reached a steady state after about 50 cycles. The results presented below are taken at the 9.5-second mark for the cycle data and a mean of the last 50 simulated cycles for the Volvo engine and the last 25 simulated cycles for the Scania engine for mean and max error. This is due to the difference in engine speed between the engines. The cycle data is over 720 degrees, a complete cycle for the four-stroke engine.

3.2 Middle Operating Point

The non-constant parameters needed to run this operational point for the different engines are presented in Table 2. In Table 3 and 4, mean value data is presented for different parameters, and in Figs. 6 and 7 cycle data is presented for the Scania and Volvo engine respectively.

Table 2. Operating points run for Scania and Volvo engines of the middle operating point.

Parameter	Scania	Volvo
Engine Speed [RPM]	1300	2250
Throttle Angle [Deg]	-	8.42
Fuel Flow [mg/stroke]	126.49	17.97
SOI [Deg]	-8.04	-8.00
Crank Angle Intake Offset [Deg]	15.06	-48.00
Crank Angle Exhaust Offset [Deg]	-15.00	30.00

Table 3. Mean error and max error for different parameters for the Scania engine on the mean operating point.

Parameter	Modelled value	Measured value	Units	Mean error [%]
p_c	1.76	2.02	bar	-13.00
p_im	1.72	1.98	bar	-13.02
p_em	1.68	1.80	bar	-7.14
T_c	105.43	104.32	C	1.06
T_im	27.83	26.56	C	4.77
T_em	353.55	371.50	C	-4.83
T_t	310.33	330.88	C	-6.21
w_tc	65.28	70.50	kRPM	-7.40
Tq_e	857.17	1249.92	Nm	-31.42
Lambda	1.85	2.24	-	-17.20
W_af	247.16	266.72	g/s	-7.33
W_c	247.16	266.72	g/s	-7.33
W_ic	247.13	266.72	g/s	-7.34
W_cyl_in	247.12	266.72	g/s	-7.35
W_cyl_out	262.29	348.93	g/s	-24.83
W_t	262.76	348.93	g/s	-24.69
W_es	262.84	348.93	g/s	-24.67
Xo_em	42.29	12.00	%	30.29

Table 4. Mean error and max error for different parameters for the Volvo engine on the mean operating point.

Parameter	Modelled value	Measured value	Units	Mean error [%]
p_c	1.03	1.03	bar	0.00
p_im	0.77	0.76	bar	0.95
p_em	1.06	1.03	bar	2.52
T_c	37.92	37.79	C	0.35
T_im	26.55	32.45	C	-18.20
T_em	883.79	665.21	C	32.86
T_t	881.55	637.69	C	38.24
w_tc	24.19	32.84	kRPM	-26.34
Tq_e	99.78	79.86	Nm	24.95
Lambda	1.04	0.99	-	4.60
W_af	15.23	18.75	g/s	-18.77
W_c	15.23	18.75	g/s	-18.77
W_ic	15.23	18.75	g/s	-18.77
W_th	15.23	18.75	g/s	-18.77
W_cyl_in	15.23	18.75	g/s	-18.79
W_cyl_out	17.80	19.09	g/s	-6.78
W_t	17.80	19.09	g/s	-6.77
W_es	17.70	19.09	g/s	-6.77

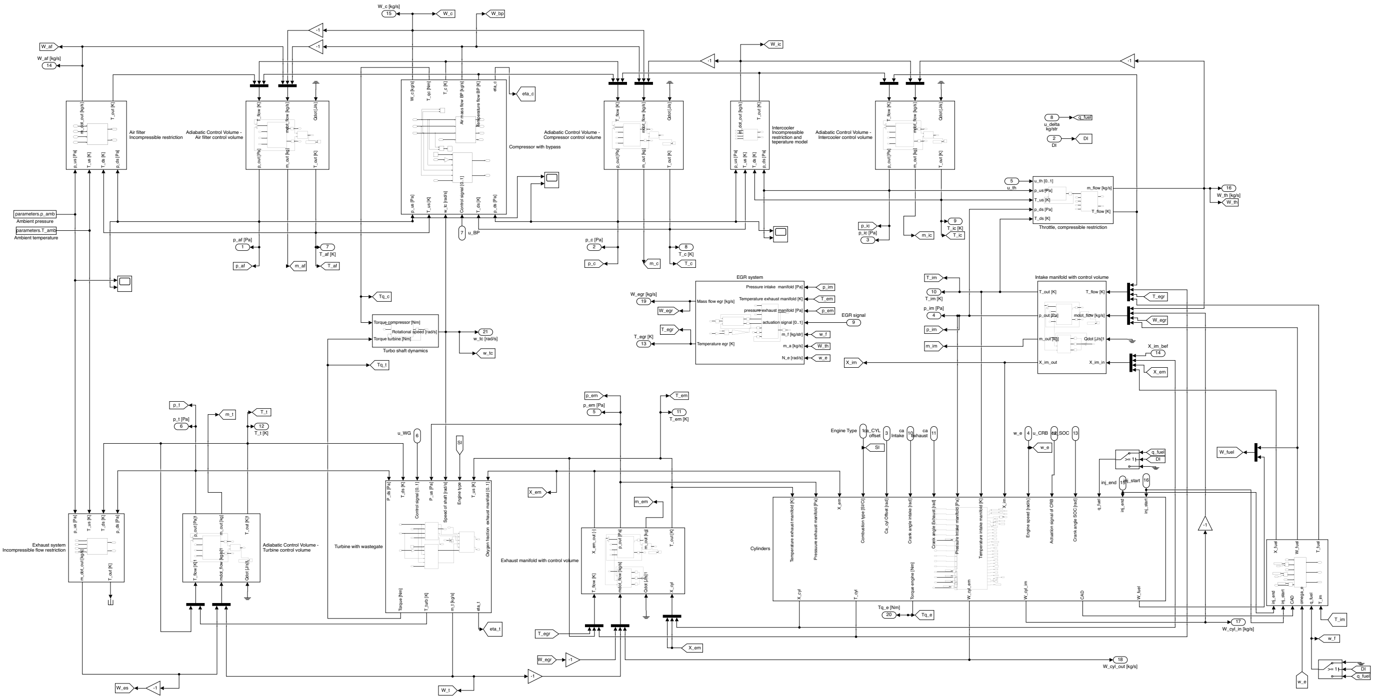

Fig. 2. The layout of the Simulink model

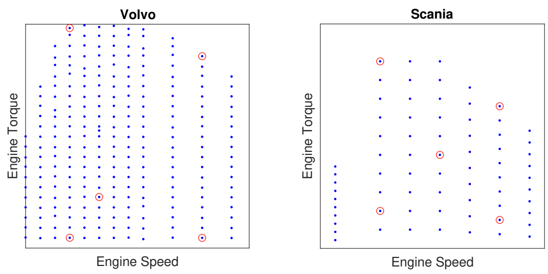

Fig. 3. A display of the different operating points of the simulated engines, with the chosen operating points marked with a red ring.

4. DISCUSSION

One can see the difference in combustion between SI and CI combustion by looking at the oxygen concentration in the cylinder in Figs. 6 to 7. Combustion in a CI is run lean to decrease the chances of creating particles, which also is a reason to use the EGR to reduce the available oxygen Eriksson and Nielsen (2014). In these simulations, the available oxygen is not combusted fully, leading to free air levels not going down to zero after combustion.All oxygen is combusted for SI, meaning the available oxygen is 0% at the combustion's end. The fill-up of available oxygen also follows the effective area. This proves the model can handle fresh air flow and residual gasses.

As seen in the Tables 3 to 4, the simulated pressures are closer for the Volvo model than the Scania model. This is due to the throttle and wastegate regulators regulating the pressures to the measured and desired value used in the Volvo model. Still, temperatures are closer for the Scania engine than the Volvo engine. The temperature deviation is because there is no direct correlation between pressure and temperature, so even if a regulator regulates

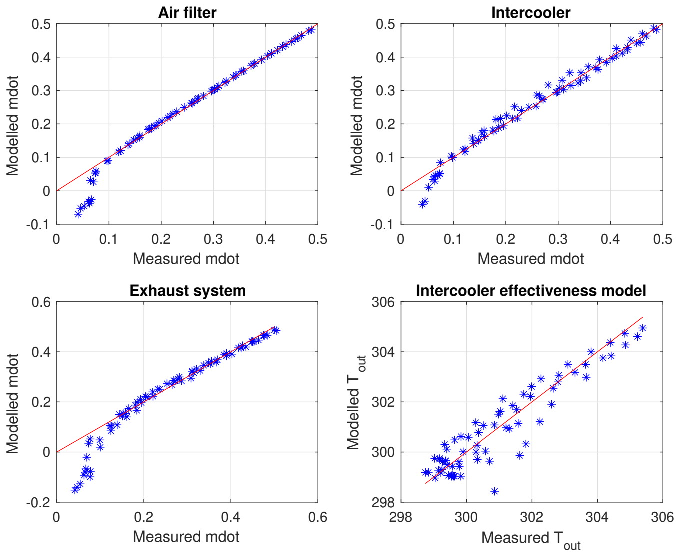

Fig. 4. The air filter, intercooler, and exhaust system flow validation, validating $C_{tu,af}$, $C_{tu,in}$ and $C_{tu,ex}$ for the Scania engine.

the pressure, it does not mean the temperature deviation will also improve.

Cylinder pressures are the only reference measurement available with cycle-to-cycle variations. In Figs. 6 and 7, they are used to validate combustion parameters. The vibe parameters are set for one operating point and not changed when the operating point is changed. As combustion varies depending on load, the vibe parameters should also change. This was, however, a simplification made.

Another aspect differing from measured data is the model for the generated engine torque. This is due to the model not simulating engine friction. Most engine torques are

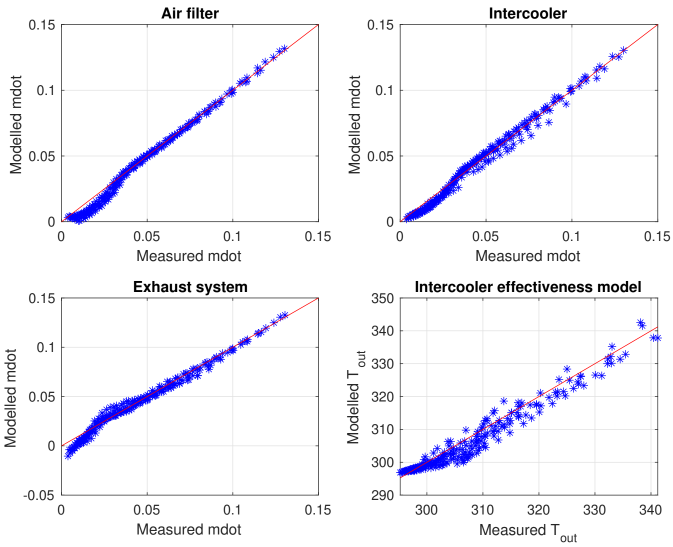

Fig. 5. The air filter, intercooler, and exhaust system flow validation, validating $C_{tu,af}$, $C_{tu,in}$ and $C_{tu,ex}$ as well as the intercooler effectiveness model, a_0, a_1 and a_2 for the Volvo engine. The red line marks the perfect model, meaning modeled equalling measured.

lower than measured because of how the model has been simulated. As the model uses variable step lengths, the cycle length in the measured data is often less than 720 degrees. However, if the cycle length were increased by one step, it would be longer than 720 degrees. The mass balance is fulfilled if the model is simulated with mass instead of pressure as a state. Currently, the model is tuned, focusing on the individual components. A better agreement between the model and data can be achieved if the total system model is fine-tuned with complete system behavior, as in Ekberg et al. (2018).

5. CONCLUSIONS

This paper aims to propose a generic engine model with VVA compatibility. Generic means the possibility of easily changing the equations used for each component and removing some components altogether and the possibility of interchange between CI and SI combustion. This was completed by making a separate MATLAB equation for each component, as each component is represented by one Simulink block. The generality is proven by the fact that the Volvo engine is SI and the Scania engine is CI, as well as by removing and exchanging components for the Scania engine, which used a different compressor and turbine function. That was fulfilled by changing the equations run in MATLAB and rerouting some signals in Simulink. As an added benefit of using this generic model, equations can easily be changed to improve the performance of the models used in this paper.

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REFERENCES

Björck, Å. (1996). *Numerical methods for least squares problems*. SIAM.

Brugård, J., Eriksson, L., and Nielsen, L. (2001). Mean value engine modeling of a turbo charged spark ignited engine - A principle study. Technical Report LiTH-R-2370, Department of Electrical Engineering, Linköpings Universitet, SE-581 83 Linköping, Sweden.

Chaing, W.P., Zhu, L., and Patankar, R. (2007). Mean value engine modeling and validation for a 4-stroke, single cylinder gasoline engine. *Trends in Applied Sciences Research*, 2(2), 124–131.

Ekberg, K., Leek, V., and Eriksson, L. (2018). Modeling and Validation of an Open-Source Mean Value Heavy-Duty Diesel Engine Model. *Simulation Notes Europe (SNE)*, 28(4), 198–203. doi:10.11128/sne.28.4.1044.

Eriksson, L. (2003). Mean value models for exhaust system temperatures. *SAE 2002 Transactions, Journal of Engines, 2002-01-0374*, 111(3).

Eriksson, L. and Nielsen, L. (2014). *Modeling and Control of Engines and Drivelines*. John Wiley Sons Inc.

Eriksson, L., Nielsen, L., Brugård, J., Bergström, J., Pettersson, F., and Andersson, P. (2002). Modeling of a turbocharged SI engine. *Annual Reviews in Control*, 26(1), 129–137.

Heywood, J. (1988). *Internal combustion engine fundamentals*. McGraw-Hill.

Johansson, L. (2019). *Full Cycle Cylinder State Estimation in DI Engines with VVA*. Master's thesis, Linköping University, Vehicular Systems.

Lind Jonsson, O. (2021). *A GENeric Internal Combustion Engine model : LiU-Genie*. Master's thesis, Linköping University, Vehicular Systems.

Llamas, X. and Eriksson, L. (2018). Liu cpgui: A toolbox for parameterizing compressor models. Technical Report 3102, Linköping University, Vehicular Systems.

Llamas, X. and Eriksson, L. (2019). Control-oriented modeling of two-stroke diesel engines with exhaust gas recirculation for marine applications. *Journal of Engineering for the Maritime Environment*, 223(2), 551–574.

Maroteaux, F. and Saad, C. (2015). Combined mean value engine model and crank angle resolved in-cylinder modeling with nox emissions model for real-time diesel engine simulations at high engine speed. *Energy*, 88, 515–527.

McBride, B.J. (2002). *NASA Glenn coefficients for calculating thermodynamic properties of individual species*. National Aeronautics and Space Administration, John H. Glenn Research Center ....

Mostofi, M., Shamekhi, A., and Gorji-Bandpy, M. (2006). Modified mean value si engine modeling (egr included). *Fuel*, 10, 5.

Theotokatos, G., Guan, C., Chen, H., and Lazakis, I. (2018). Development of an extended mean value engine model for predicting the marine two-stroke engine operation at varying settings. *Energy*, 143(15).

Wahlström, J. and Eriksson, L. (2011). Modeling diesel engines with a variable-geometry turbocharger and exhaust gas recirculation by optimization of model parameters for capturing non-linear system dynamics. *Journal of Automobile Engineering*, 225(7).

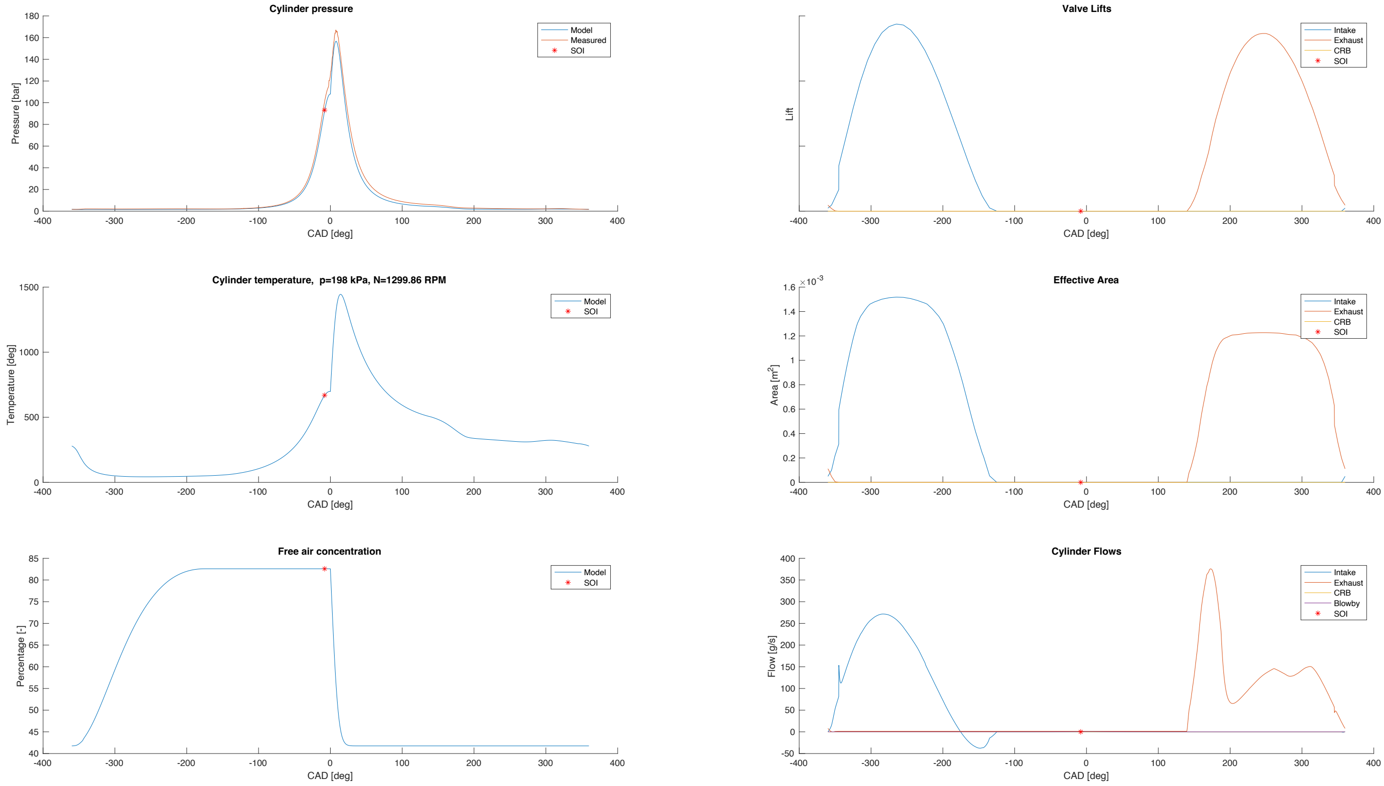

Fig. 6. Cylinder pressure, temperature, oxygen concentration, valve lift, effective area for the valves, and the cylinder flows over one complete cycle on the mean operating point for the Scania engine.

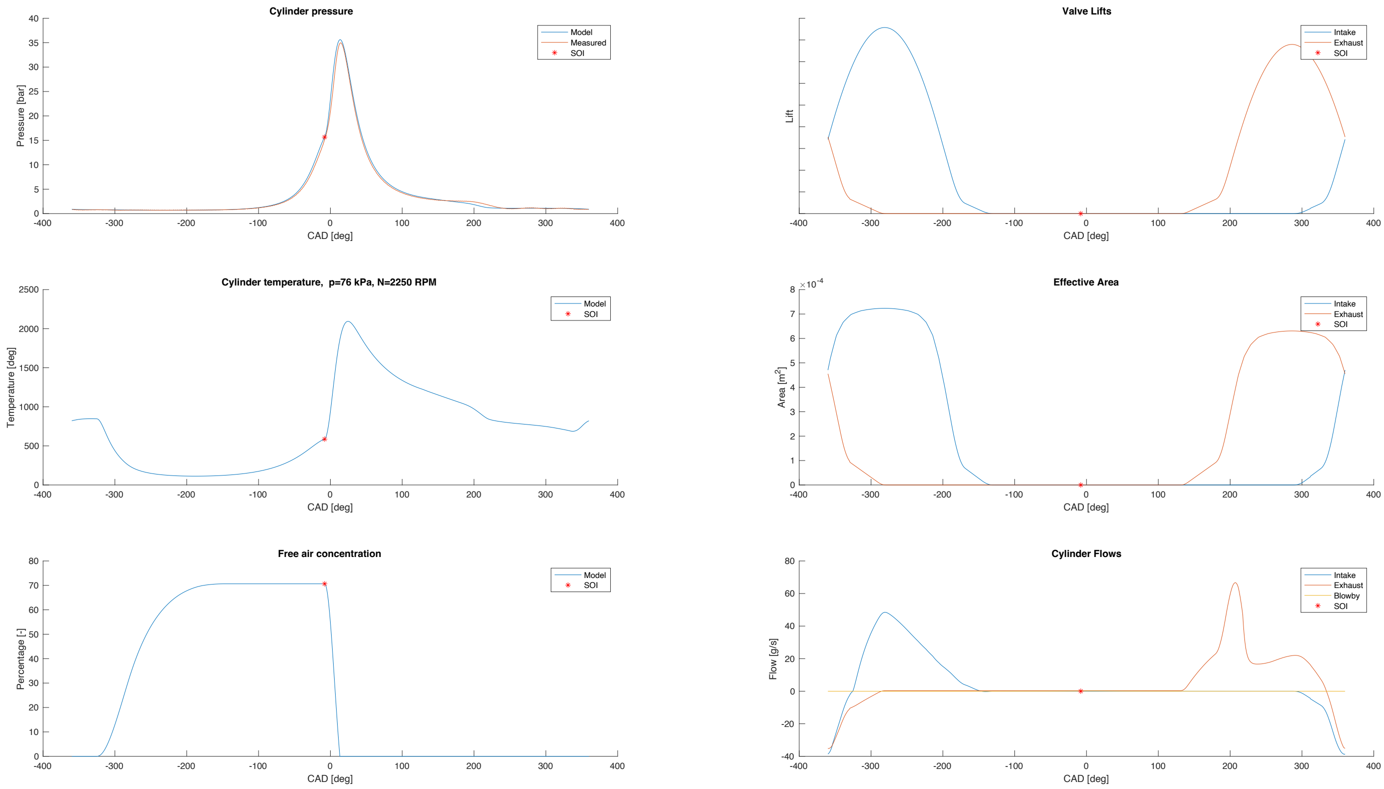

Fig. 7. Cylinder pressure, temperature, oxygen concentration, valve lift, effective area for the valves, and the cylinder flows over one complete cycle on the mean operating point for the Volvo engine.