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Simulation Environment for Analysis and Controller Design of Diesel Engines

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Abstract: Novel combustion concepts and multi-injection cylinder-wise control methods are needed in large marine diesel engines for increased performance and to reduce the green house gas emissions. Even though diesel technology in cars might be reducing there is no replacement of dual fuel diesel technology in large marine engines to be seen in the near future. The paper discusses a rapid grey-box modeling technique, which can be used to predict cylinder pressure and heat release in engine cylinders. The model can be used to design effective cylinder-wise control algorithms which increase the engine performance and save fuel under constraint of emissions.

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Keywords: Diesel engine; grey-box modeling; identification; combustion; combustion control; emissions; cycle-to cycle control; heat release; optimization

1. INTRODUCTION

The challenges in combustion technology have increased considerably, as the regulations of emissions are becoming more and more stringent (IMO, 2009). Renewable energy sources are being utilized and the related technologies developed as fast as possible in current society. For example, it is anticipated that the number of cars driven by diesel engines is getting smaller and smaller, even to the point that diesel engines would soon cease to be used in cars. However, for large diesel engines, like those used in maritime solutions, there is no substitution to be foreseen in the near future. Combustion dual-fuel diesel engines are the only practical solution to power large marine engines for a long period of time (Čampara et al., 2018).

In order to develop combustion technology for the future there are major challenges for technology. It appears to be extremely difficult to increase the efficiency and reduce fuel usage and emission levels by doing mechanical reconstructions to the engine itself. The technology here is so developed that new breakthroughs are not likely to happen. However, there are other approaches, which involve new possibilities for effective research, the other side being that these approaches are extremely challenging. They involve new combustion concepts like RCCI (Reactivity Controlled Combustion Ignition), HCCI (Homogenous Charge Compression Ignition), PPC (Partially Premixed Combustion) etc., which make cylinder-wise control with advanced heat release analysis necessary. The techniques being developed are actually based on optimal control of each cylinder separately, using well-timed pilot injections and generally multi-injections with the optimization target to minimize fuel consumption and maximize engine efficiency under the

constraint of limiting NO_x and CO_x emissions below given limits (Merker et al., 2006).

The purpose of the paper is to introduce a fast modeling approach to construct a simulation model to be used in rapid prototyping of diesel engine operation. The model uses both first-principles and data based -modeling, which utilize test runs of a diesel engine in order to develop a model to predict the behaviour of key variables in diesel engine under operation. The mean-value model is used to give initial values to the numerical ARX model (Auto Regressive Model with Exogenous Input), which predicts the dynamic behaviour of in-cylinder pressures, and hence heat release in cylinders. The model is then used to show, how it can be used to control the injections such that the combustion variables θ_{50} (crank angle for 50 % fuel burnt) and $IMEP$ (Indicated Mean Effective Pressure) are held within the desired levels.

The paper is organised as follows. In Section 2 the simulation schematic is first presented in order to show, how the model to be developed can be used for advanced cycle-by-cycle control of the engine cylinders. The modeling of the intake manifold, which is used as part of the input-output data modeling is presented next, followed by the ARX model, which simulates the cylinder pressure as a function of charged air pressure, injection pressure and start of ignition value. The in-cylinder pressure is given in each crank angle over the combustion cycle. In Section 3 experiments and the data analysis of real engine data are explained, to be used in the tuning of the model. In Section 4 simulation results are shown and the related heat release / in-cylinder pressure rates are shown and compared to real measurements obtained from the test engine. Conclusion and discussion are given in Sections 5 and 6.

2. METHODOLOGY

2.1 Simulation schematic

The simulation environment consists of two PI controllers and four main model blocks which are shown in Fig. 1.

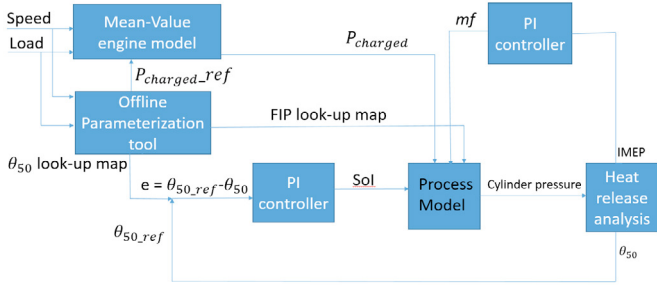


Fig. 1. Design of the simulation environment.

The most important block in this schematic is the process model in which the cylinder pressure is generated from four inputs: start of injection (SoI), fuel injection pressure (FIP) set point, charged air pressure ($P_{charged}$) and fuel mass flow (mf). The model is developed by using ARX models with training data collected from a real engine testbed. The four inputs of the process model are outputs from the other blocks in the schematics.

The $P_{charged}$ is a result from the mean-value nonlinear engine model (MVM) in which the airpath of the engine is modeled based on the engine's first principle equations. This model calculates the charged air pressure with respect to a reference point of $P_{charged}$ at given speed and load.

Reference set-points for the charged air pressure is given from a calibration block called Offline parameterization tool (Khac and Zenger, 2019). This tool models and optimizes the engine's fuel consumption under the emission regulations and then produces the optimal set-points to be used as references at each operating point (speed, load) of the following engine's parameters: $P_{charged}$, FIP and crank angle position where 50% of the heat is released (θ_{50}). The θ_{50} references are then used in the SoI-controller.

The last block is called heat release analysis, which calculates the apparent heat release rate (without heat loss consideration), cumulative heat release and indicated mean effective pressure (IMEP). The θ_{50} is then calculated from the cumulative heat release and used as a feedback signal for the SoI controller. On the other hand, the IMEP values are used as a feedback signal for the other PI controller which adjusts the fuel mass flow of the engine. The references of IMEP are read from the look-up table of the factory engine's controller.

2.2 Engine topology

A simplified topology of a diesel engine is presented in Fig. 2. A diesel engine can be divided into four major parts: intake manifold, exhaust manifold, engine cylinders and variable geometry turbocharger (VGT). In Fig. 2, intake manifold is shown in blue and exhaust manifold in red.

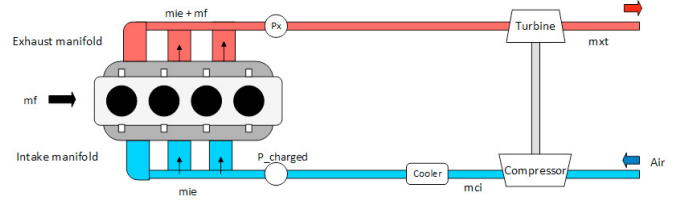


Fig. 2. Topology of a diesel engine

2.3 Mean-value model

As presented in the previous section one of the inputs of the system is charged air pressure (also often called intake manifold pressure). $P_{charged}$ is achieved through a developed mean-value model. Since the system represents the process inside the cylinder, only mean-value model of $P_{charged}$ is presented in this paper. Models of exhaust manifold pressure and turbocharger power can be found in (Samokhin, 2018), (Wahlström and Eriksson, 2009).

Mass flows

The conversation of air mass is equal to mass flows through the intake and exhaust manifold (Wahlström and Eriksson, 2009). The air mass flow through intake manifold is the difference between the air mass flows into the compressor, m_{ci} ($Kg \cdot s^{-1}$), and cylinders, m_{ie}

$$m_i = m_{ci} - m_{ie} \quad (1)$$

The pressure in the intake manifold can be presented with the first law of thermodynamics and ideal gas law (Wahlström and Eriksson, 2009). Intake manifold pressure is thus defined as

$$\frac{dP_{charged}}{dt} = \frac{R_i T_i}{V_i} m_i \quad (2)$$

where, R_i ($J \cdot mol^{-1} \cdot K^{-1}$), T_i (K) and V_i (m^3) are gas constant, temperature and volume of the intake manifold respectively. By substituting equation (1) into (2), the pressure of intake manifold becomes

$$\frac{dP_{charged}}{dt} = \frac{R_i T_i}{V_i} (m_{ci} - m_{ie}) \quad (3)$$

Air mass flow through the cylinders can be presented as a function of engine speed, ω_e ($rad \cdot s^{-1}$), volumetric efficiency of the engine, η_v , and $P_{charged}$

$$m_{ie} = \frac{\eta_v(\omega_e)\omega_e P_{charged} V_d}{2\pi\nu R_i T_i} \quad (4)$$

where, V_d (m^3) is engine displacement volume and ν is number of revolutions per engine cycle. η_v is defined as

$$\eta_v = a_{v1} + a_{v2}\omega_e + a_{v3}\omega_e^2$$

where a_{v1} , a_{v2} and a_{v3} are tuning parameters. (Guzzella and Amstutz, 1998).

Mass flow through the compressor is a function the compressor isentropic efficiency η_c

$$m_{ci} = \frac{\eta_c P_c}{c_p T_a \left(\left(\frac{P_{charged}}{P_a} \right)^\mu - 1 \right)} \quad (5)$$

where, P_c is power of compressor, T_a is ambient temperature, $\mu = \frac{(\gamma-1)}{\gamma}$ and γ is the specific heat ratio.

2.4 ARX model

ARX stands for Autoregressive with Extra Input. It is an AR model with an exogenous input. The structure of the ARX-model is described as

$$\begin{aligned} y(t) + a_1y(t-1) + \dots + a_{na}y(t-na) \\ = b_1u(t-nk) + \dots + b_{nb}u(t-nb-nk+1) + e(t) \end{aligned} \quad (6)$$

Here, na and nb are the orders of the ARX-model and nk is the delay, $y(t)$ is the output, $u(t)$ is the input and $e(t)$ is white noise at time t (Ljung and Glad, 1994). Another way to write equation (6) is

$$A(q)y(t) = B(q)u(t-nk) + e(t) \quad (7)$$

where

$$\begin{aligned} A(q) &= 1 + a_1q^{-1} + a_2q^{-2} + \dots + a_{na}q^{-na} \\ B(q) &= b_1 + b_2q^{-1} + b_2q^{-2} + \dots + a_{nb}q^{-nb+1} \end{aligned}$$

and na is number of poles, nb is number of zeros and nk is number of samples occurring before the input affects the output. The ARX-model of a multi-input and single-output (MISO) system is defined as

$$\begin{aligned} A(q)y(t) &= B(q)_1u_1(t-nk_1) + B(q)_2u_2(t-nk_2) \\ &+ B(q)_{nu}u_{nu}(t-nk_{nu}) \end{aligned} \quad (8)$$

where nu is the number of inputs, na and nk are row vectors, in which the i th element and delay correspond to the i th element of the input vector $u(t)$ (Erdoğan and Güllal, 2009).

2.5 Process model

The inputs of the system as already presented in Section 2.1 are SoI , FIP , $P_{charged}$ and m_f . Output of the process model is the cylinder pressure (p_{cyl}). The cylinder pressure is modeled dynamically, where the current pressure is predicted by using the pressure of N previous cycles. This is achieved according to (9)

$$y(t) = \sum_{i=1}^N B_i u(t-i) - \sum_{i=1}^N a_i y(t-i) \quad (9)$$

which is the regression form of ARX model as described in (Ljung and Glad, 1994). In equation (9), the predicted output $y(t)$ is depended on previous inputs and outputs. The regression equation can be presented in matrix form as

$$y(t) = \phi(t)\theta^T \quad (10)$$

where $\phi(t)$ is a row vector of size $1 \times (n_b \cdot n_u + n_a)$ containing the inputs and outputs of the previous cycles. θ^T is a vector of size $(n_b \cdot n_u + n_a) \times 1$ containing unknown coefficients.

$$\phi(t) = \begin{bmatrix} u^T(t-nk) \\ u^T(t-nk-1) \\ \vdots \\ u^T(t-nk-nb+1) \\ -y(t-1) \\ \vdots \\ -y(t-na) \end{bmatrix}^T, \theta^T = \begin{bmatrix} b_{11} \\ b_{12} \\ \vdots \\ b_{1n_u} \\ b_{21} \\ \vdots \\ b_{2n_u} \\ b_{nb1} \\ \vdots \\ b_{nbn_u} \\ a_1 \\ a_2 \\ \vdots \\ a_{na} \end{bmatrix}$$

The coefficient vector is solved by least square regression method (Weisberg, 2005).

$$\theta = (\phi(t)^T \phi(t))^{-1} \phi(t)^T y \quad (11)$$

The regression model gives the values of p_{cyl} for each engine cycle.

2.6 Heat release analysis

The combustion reaction within the cylinder is modeled as a heat release rate (Heywood, 1988),

$$\frac{dQ_c}{d\theta} = \frac{\gamma}{\gamma-1} p_{cyl} \frac{dV_{cyl}}{d\theta} + \frac{1}{\gamma-1} V_{cyl} \frac{dp_{cyl}}{d\theta} \quad (12)$$

Here, V_{cyl} (m^3) is cylinder volume. The cylinder volume is modeled as a function of crank angle

$$V_{cyl} = V_c + \frac{V_d}{2} \left(R_v + 1 - \cos\left(\frac{\pi}{180}\theta\right) - \sqrt{R_v^2 - \sin^2\left(\frac{\pi}{180}\theta\right)} \right) \quad (13)$$

where V_d and V_c are displacement and clearance volumes, R_v is the ratio of the connecting-rod length to the crank radius and θ is crank angle. Pressure and volume rates can be obtained from (12)

$$\frac{dp_{cyl}}{d\theta} = \frac{\gamma}{V_{cyl}} \frac{dV_{cyl}}{d\theta} p_{cyl} + \frac{\gamma-1}{V_{cyl}} \frac{dQ_c}{d\theta} \quad (14)$$

Equation (12) is derived with the assumption that composed gas is uniform ideal gas with a constant mass (Turesson, 2018). In this work the crank angle is defined as an independent variable with different resolutions. The cumulated heat release during the combustion reaction as shown in Fig. 3. Measured p_{cyl} was utilized calculate the Q_c . The calculated θ_{50} for this operating point is approximately 19 crank angle degrees.

3. EXPERIMENT AND DATA ACQUISITION

3.1 Experiment procedure

In order to build the models for the Offline parameterization tool and the process model blocks, training data has been collected from a research engine using the Design of Experiment (DoE) method. The DoE method is a structural approach to design the experiments and to collect data, which sufficiently

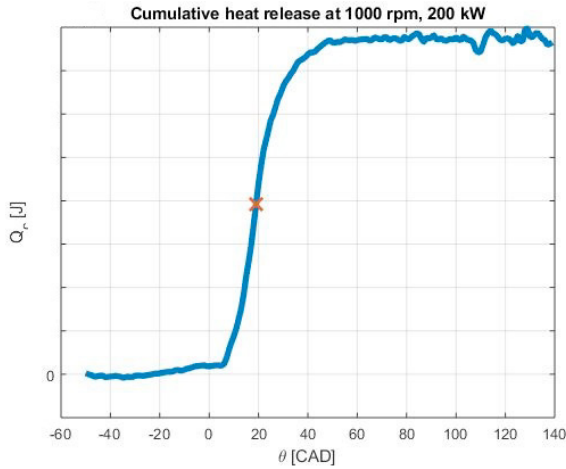


Fig. 3. Cumulative heat release during the combustion reaction with engine speed of 1000 RPM and load of 200 kW. The red mark represents the θ_{50}

capture output’s responses towards the input factors. The Box-Behken testing matrix design has been chosen for its simplicity and ability to avoid exceeding the operation range of the parameters. Figure 4 shows a geometric view of the design for a case with three inputs and one output. See Khac and Zenger (2019) for more details about the structure design and experiment procedure.

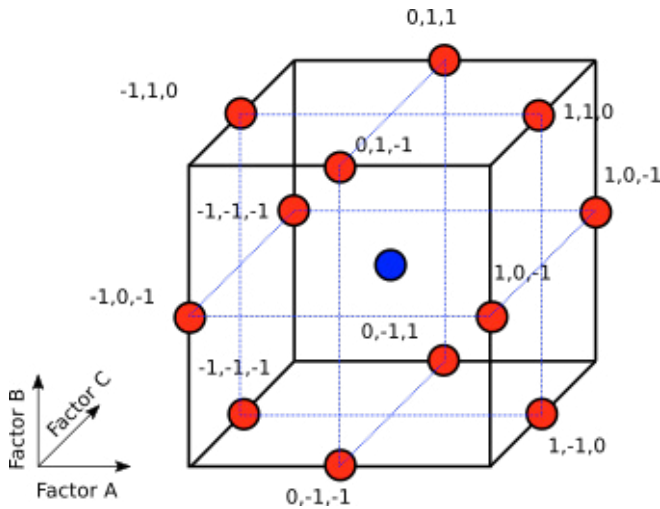


Fig. 4. Geometric view of a three-factor Box-Behken design.

The input parameters are $P_{charged}$, SoI and FIP. The recorded responses from the engine are the fuel consumption (fuel mass flow), NO_x emission and in-cylinder pressure.

3.2 Experiment Instrumentation

The experimental measurements were performed in Vaasa Energy Business Innovation Centre (VEBIC), Finland. The engine used is a 4 cylinders Wartsila line engine which is a 200mm bore diameter common rail diesel engine (4L20). It is connected to an ABB generator, a frequency converter and running against the local electrical grid. The test fuel is the commercial light fuel oil (LFO). The main specification of the test engine is given in Table 1.

Table 1. Wartsila 4L20 engine specifications.

Cylinder number	4
Cylinder bore	200 mm
Piston stroke	280 mm
Swept volume	0.0088 m ³
Rated speed	1000 rpm
Rated power	800 kW

Table 2. Analytic instruments.

Parameters	Devices
NO_x	Eco Physics CLD 822 M hr
CO, CO_2	Siemens Ultramat 6
Hydrocarbons	J.U.M.VE7
Cylinder pressure	Kistler KiBox
Injection timing	Current probe
Fuel consumption	HBM weight cell, Sartorius X3

This Wartsila 4L20 engine is equipped with 4 Kistler piezo-electric cylinder pressure sensors, 4 exhaust temperature sensors, one stage fixed geometry exhaust gas driven turbocharger, intake air pressure and temperature measurement system, and a fuel consumption measurement system. However, there is no exhaust after-treatment systems and wastegate installed. The analytic instruments are listed in table 2. The emission measurements data were obtained manually from the instruments listed in table 2 and it is processed according to the ISO 8178 standard with the simultaneously measured ambient condition.

According to the DoE test plan, the operating range of the engine is shown in table 3, where the selected 14 test points are marked by "√". Every point is run with the Box-Behken design method described earlier.

Table 3. Selection of operating points.

Speed (rpm)	1000	900	800	700	600
Load (kW)					
100 kW	√	√	√	√	
150 kW					√
200 kW	√	√	√	√	
300 kW			√		
400 kW	√	√			
600 kW	√				
800 kW	√				

4. SIMULATION RESULTS

4.1 Cylinder pressure estimation

The cylinder pressure has been modeled cycle by cycle, using data acquired at the operating points in table 3. Information from the two previous cycles is used to predict the pressure at the current cycle. The validation of the process model is conducted on 32 operating points which span from 600 revolution per minute (rpm) to 1000 rpm and from 150 kilo-Watts (kW) to 600 kW. Figure 5 shows a comparison between the measured cylinder pressure from the KiBox device and the estimated pressure achieved from the ARX process model at an example point. The model calculated the pressure trace with a nominal setting of the engine at 1000 rpm and 600 kW as the initial conditions.

As can be seen in Fig. 5, the modeled curve describes fairly well the compression phase and the expansion phase, which is a crucial factor for an accurate heat release calculation. The

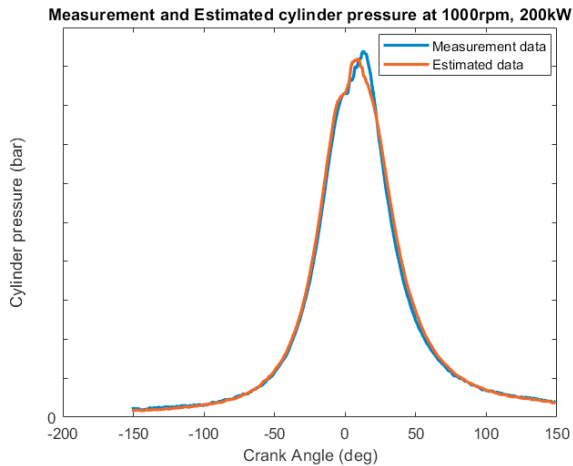


Fig. 5. Comparison of measured and estimated pressure.

pressure peaks are a bit different due to the uncertainties in the modeling and measuring procedure as it is quite sensitive to record the peak cylinder pressure correctly. Notice that the y-axis values have been removed due to confidentiality reasons.

Figure 6 and Fig. 7 present the modeling errors as standard deviation and coefficient of determination R^2 . These two indexes are calculated over the 32 validation points. It shows that the fitting is quite good and the model gives sufficiently accurate pressure trace with some errors at the zero crank angle degree and smaller error in the expansion and compression phases.

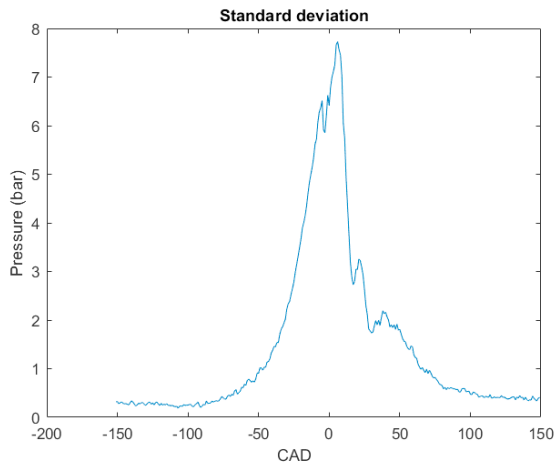


Fig. 6. Standard deviation.

4.2 Heat release analysis results

The heat release is calculated using (12) without considering the heat transfer through the cylinder wall. The results calculated from the “Heat release analysis” block are compared to the measured data from KiBox device. Figure 8 shows the comparison at 1000 rpm and 200 kW.

Figure 9 shows the comparison at 1000 rpm and 600 kW.

In both low load and high load cases, the calculation is quite accurate, the raising rate has been captured correctly. The curves appear to be horizontal after the combustion ends (around 40 to 60 crank angle degree) which indicates that there’s no more heat being released. Small differences between the measured

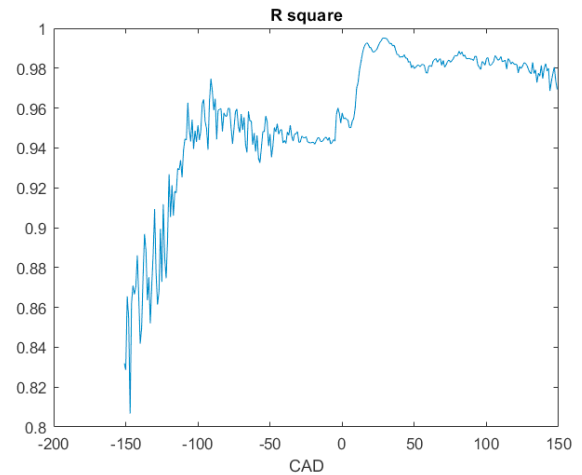


Fig. 7. Coefficient of determination R^2 .

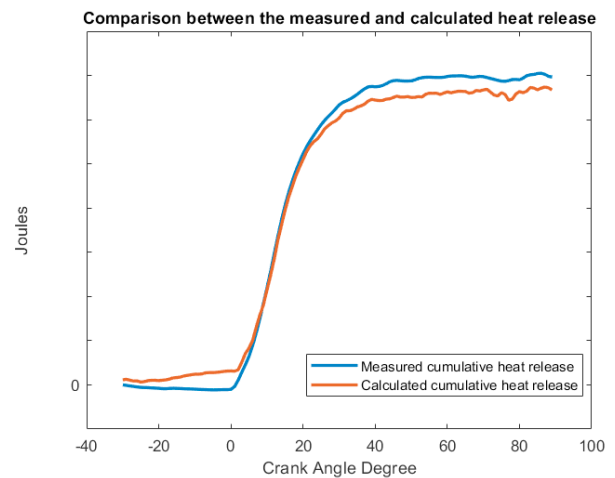


Fig. 8. Comparison of cumulative heat release at 1000 rpm and 200 kW.

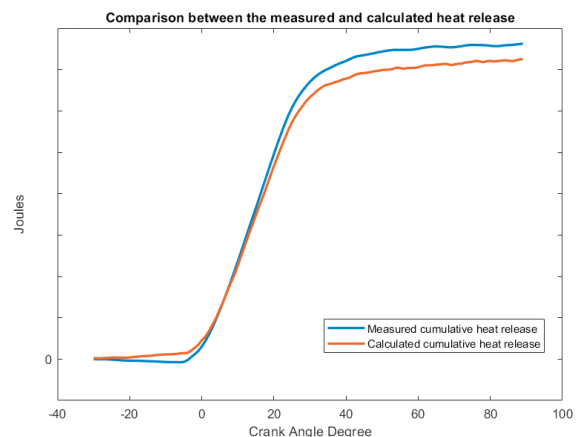


Fig. 9. Comparison of cumulative heat release at 1000 rpm and 600 kW.

and the calculated data could be caused by the deviation of the engine cylinder parameters and fuel characteristics.

5. CONCLUSIONS

This paper presents a fast modeling approach to construct a simulation environment to be used in rapid prototyping of diesel engines operation. The simulation model consists of the two main functional blocks, which can estimate the in-cylinder pressure and the heat release rate of the engine, alongside with the mean-value model and the offline calibration tool to give the initial values and reference set-points for the models. This environment is suitable to be used for different controller design purposes such as cycle-to-cycle cylinder-wise combustion control with multiple injection strategies. Several conclusions are made from this work:

1. A simulation environment are made for the controller design and testing purposes.
2. A grey-box model is built to dynamically estimate the behaviors of the cylinder pressure under the effects of other parameters.
3. The apparent heat release rate (regardless of the heat transfer) is derived from the cylinder pressure trace.
4. In the future works, a holistic combustion controller can be designed based on this simulation environment with some accuracy improvements such as considering the heat loss to the cylinder wall, the accuracy of the pressure measurement, etc.

6. DISCUSSION

There are several possible improvements that need to be considered in the future to make the simulation model more accurate and to capture better the engine response.

1. The in-cylinder pressure measurement is not hundred percent reliable as there are some uncertainties such as sensor errors, disturbance noise, etc. In order to compensate for the error in the measurement, some advanced method should be applied, such as pegging to correct the values of the pressure trace to the corresponding crank angle.
2. In Equation (12), the fuel specific heat ratio (γ) has an important role to the accuracy of the calculation. In this work, γ is set constant based on the estimation from the fuel supply. However, γ can be determined better by using more accurate estimation methods such as NASA polynomials, temperature dependent functions, etc.
3. As mentioned above, the heat transfer (or heat loss) is not considered in this work as it is challenging to model the heat loss precisely due to difficulties in measuring some parameters such as the cylinder wall temperature.
4. Another factor affecting the quality of the heat release calculation is the top dead center (TDC) offset of the cylinder. Due to installation or mechanical wear-out of the crank shaft, position of the top dead center might vary a few degrees. This offset affects the heat release calculation quite much. Figure 10 shows how different values of TDC offset affect the curves of the cumulative heat release.

ACKNOWLEDGEMENTS

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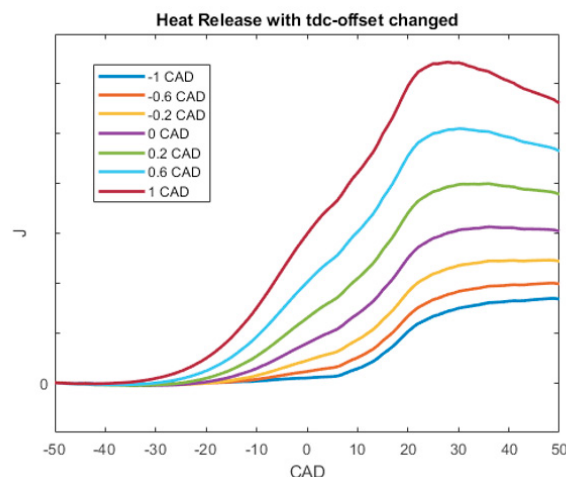


Fig. 10. Effects of TDC offset on heat release calculation.

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