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Real-Time Simulation of Torque and Nitrogen Oxide Emissions in an 11.0 L Heavy-Duty Diesel Engine for Model-Based Combustion Control

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Abstract: A real-time combustion model was assessed and applied to simulate BMEP (Brake Mean Effective Pressure) and NOx (Nitrogen Oxide) emissions in an 11.0 L FPT Cursor 11 diesel engine for heavy-duty applications. The activity was carried out in the frame of the IMPERIUM H2020 EU Project. The developed model was used as a starting base to derive a model-based combustion controller, which is able to control indicated mean effective pressure and NOx emissions by acting on the injected fuel quantity and main injection timing. The combustion model was tested and assessed at steady-state conditions and in transient operation over several load ramps. The average root mean square error of the model is of the order of 110 ppm for the NOx simulation and of 0.3 bar for the BMEP simulation. Moreover, a statistical robustness analysis was performed on the basis of the expected input parameter deviations, and a calibration sensitivity analysis was carried out, which showed that the accuracy is almost unaffected when reducing the calibration dataset by about 80%. The model was also tested on a rapid prototyping device and it was verified that it features real-time capability, since the computational time is of the order of 300–400 μs. Finally, the basic functionality of the model-based combustion controller was tested offline at steady-state conditions.

Keywords: model-based; control; heavy-duty; diesel engine; NOx emissions; BMEP

1. Introduction

1.1. Background

The need to comply with the more and more stringent regulations in terms of CO2 and pollutant emissions is pushing the automotive industry to develop innovative technologies. Recent trends which may produce the highest benefits include engine downsizing [1], alternative fuels (such as Compressed Natural Gas (CNG) and biofuels) [2], innovative combustion concepts such as Homogeneous Charge Compression Ignition (HCCI) and Premixed Charge Compression Ignition (PCCI) [3], advanced high-pressure common rail systems [4,5], innovative combustion controls [6–8], innovative after-treatment system (ATS) technologies [9], kinetic and thermal energy recovery [10], vehicle road load reduction [11], powertrain electrification [12], and vehicle connectivity (V2X technologies) [13]. In particular, in ref. [1] the authors describe the potential benefits of the electrically-assisted turbocharger with VGT to enable heavy-duty diesel engine downsizing. The study reported in ref. [2] pointed out the benefits of the use of biodiesel on the Diesel Particulate Filter (DPF) particulate emissions. In ref. [3] an experimental study is reported, which shows the potential of PCCI
combustion in dramatically reducing engine-out nitrogen oxides and soot emissions in a 3.0 L diesel engine, with the possibility of reviewing the after treatment system. In ref. [4] an innovative common rail system with reduced accumulator is compared with a standard one, while in ref. [5] the same authors propose several guidelines for the reduction of leakage in injection systems. In refs. [6,7] the authors propose a controller of MFB50 and of BMEP/NO\textsubscript{x} emissions, respectively, for a 3.0 L diesel engine, while in ref. [8] a neural network-based approach is proposed for the control of torque and MFB50 for a 1.6 L diesel engine. The paper proposed in ref. [9] shows the main advancements in diesel and spark ignition engine emissions, including the impact of the after-treatment devices. In ref. [10], the authors investigate the available energy of the waste heat from a high boosted 6-cylinder heavy duty diesel engine which is equipped with high and low pressure loop Exhaust Gas recirculation (EGR) system, showing that the estimated improvement in fuel consumption by the Rankine combined cycle is about 2.7% in a single-stage turbocharged system and 2.9% in a 2-stage turbocharged system. In ref. [11] the authors compare reference and lightweight design solutions in terms of global warming potential. The study reported in ref. [12] presents a tool for the optimal design of hybrid vehicles equipped with diesel engines, and the consequent benefits in terms of costs and CO\textsubscript{2} emission reduction. Finally, in ref. [13] a cooperative adaptive cruise control system that combines precise positioning at the lane level and V2X communications was developed, with the aim of reducing CO\textsubscript{2} emissions in dense traffic conditions.

Among these technologies, model-based combustion controls seem to have a good potential in order to achieve both fuel consumption and pollutant emission reductions, compared to the traditional map-based approach. This has been made possible by the recent advances in the computational performance of engine control units (ECUs), which are now capable of executing more and more complex model-based algorithms in real-time. The advantages of model-based controllers against conventional map-based ones are due to several reasons. For example, a model-based approach allows an onboard optimization of the combustion process to be realized, by adjusting the main engine control variables in real-time, in order to achieve desired targets of performance or pollutant emission metrics. Moreover, model-based combustion controllers can be coupled with additional model-based control modules which could be available in the ECU (such as ATS management), in order to allow higher engine-out pollutant levels when the ATS efficiency is high (with benefits in terms of fuel consumption) and lower engine-out pollutant levels when the ATS efficiency is low. It should also be considered that model-based combustion controllers may be integrated with the emerging V2V and V2I technologies. For example, if the road characteristic and traffic conditions are known in advance (on the basis of an eHorizon system), the vehicle energy management supervisor may request the engine combustion controller to realize specific instantaneous targets of torque and/or engine-out pollutant emissions, in order to globally optimize the performance of the vehicle in the future time frame.

In addition to the previous aspects, it should be noted that model-based combustion controllers are advantageous, compared with traditional map-based controllers, also in terms of calibration effort, especially for that which concerns diesel engines. It should in fact be considered that modern diesel engines are characterized by a very high degree of complexity, and are managed by a large number of control parameters, which are defined by maps. These maps require a high calibration effort, with related consequences in terms of needed time and costs. Moreover, traditional engine maps need to be properly calibrated in order to take the variability of the environmental conditions (temperature, pressure, humidity etc.) into account, while a model-based approach is in general more robust with respect to these kinds of effects.

The previous considerations justify why the interest in the development of model-based combustion controls for ICEs (Internal Combustion Engines) and, more specifically, for diesel engines, has increased in recent years.

With reference to diesel technology, although the current social and political attitude towards this kind of technology is negative, it should be pointed out that recent innovations have allowed a dramatic reduction of NO\textsubscript{x} emissions (far less than European standards after 2020, see ref. [14]),
also exploiting artificial intelligence technology. Moreover, diesel technology is expected to remain the best solution for light-duty and heavy-duty long haul applications, in which it offers incontestable advantages in terms of both fuel consumption and fuel costs. These advantages may be amplified by means of powertrain electrification.

Given the previous background, a real-time combustion model was developed in this study, capable of predicting BMEP (Brake Mean Effective Pressure) and NO\textsubscript{x} emissions in an 11 L FPT Cursor 11 heavy-duty diesel engine. The model is intended to be used for the development of a model-based combustion controller to be implemented in the engine, within the IMPERIUM (IMplementation of Powertrain Control for Economic and Clean Real driving emission and fuel ConsUmption") EU H2020 collaborative research project.

This project started in 2016, and is currently ongoing. Its main objective is to achieve fuel consumption and urea reduction of up to 20% for long haulage heavy-duty trucks, while keeping the vehicle within the legal limits for pollutant emissions [15]. The consortium includes three major manufacturers of heavy duty trucks (DAF Trucks N.V, IVECO S.p.A. and Volvo Technology AB), eight major high end and large component and engineering service suppliers (FPT Motorenforschung AG, AVL List GmbH, Ricardo UK Ltd., Continental Automotive GmbH, FEV GmbH, Robert Bosch GmbH, and Honeywell SPOL. S.R.O), as well as six research and service centers specializing in simulations (Chalmers University, Politecnico di Torino, Politecnico di Milano, CTU Prague, RWTH Aachen University, Technical University Eindhoven).

The consortium will realize three heavy-duty demonstrators, in which the fuel consumption and urea reduction benefits will be achieved through several technologies, including the use of an energy management supervisor, dynamic e-horizon system, model-based engine control, waste heat recovery, micro-hybrid and smart auxiliaries.

1.2. State of the Art in Low-Throughput Combustion Modeling for Model-Based Combustion Control

In general, low computationally demanding simulation models are required for the development of model-based combustion controllers, due to the fact that the ECU must be capable of running these controllers in real time, often on a cycle-by-cycle basis.

Therefore, multidimensional or one-dimensional approaches are not suitable for this purpose.

The best candidates for the development of model-based combustion controllers are mean-value zero-dimensional models and artificial intelligent systems [7].

Mean-value zero-dimensional models are capable of simulating the combustion and/or pollutant formation processes on the basis of a physical approach, and require at the same time a low computational effort compared to 3D-CFD or 1D-CFD approaches. Some examples are provided in refs. [16–20]. In particular, in ref. [16] the potential of mean-value models, especially of the filling and emptying approach, is discussed for the simulation of transient conditions. In ref. [17], the authors propose a real-time mean value engine model, which is capable of simulating the in-cylinder pressure and NO\textsubscript{x} emissions. The model was tested at steady-state and transient conditions and showed a good accuracy over New European Driving Cycle (NEDC) and Worldwide harmonized Light-duty Test Procedure (WLTP) missions. In ref. [18] a simple, physics-based model for the NO\textsubscript{x} emissions of diesel engines was developed and validated using data from three different engines, showing an error during transients of 2.5% in average. In ref. [19], the previous model is embedded in a mean-value model for the air path. In ref. [20] a real time NO\textsubscript{x} calculation model is proposed, which takes into account fuel injection, ignition delay, premixed heat release rate and diffusion combustion heat release rate.

These models usually show very good performance at steady-state operation and an acceptable accuracy in mildly transient conditions. In general, they do not require a high calibration effort, due to their physics-based nature, and are quite robust outside the calibration range.

Artificial intelligence systems include different methodologies, such as, support vector machine (SVM), fuzzy logics, and artificial neural networks (ANNs). These methods belong to the black-box category, and are often used for model-based control purposes since they require a low computational
effort. They are capable of capturing the non-linear behaviors of complex systems, without the need of modeling the physics of the process to be simulated. The main drawback of these types of models is that they require, in general, a high number of experimental tests for robust training, and tend to lose accuracy when extrapolating outside the calibration range. Moreover, they can be subject to overfitting. Some examples are reported in refs. [21–25]. In particular, in ref. [21] an ANN-based model was developed and applied to predict the variation of energy efficiency, exergy efficiency and exhaust temperature based on biodiesel type and engine load. In ref. [22] an ANN-based model was implemented and used to predict combustion product composition and gas thermodynamic properties. In ref. [23], a recurrent non-linear autoregressive with exogenous input (NARX) neural network is proposed, for reconstructing cylinder pressure in multi-cylinder IC engines using measured crank kinematics. The study proposed in ref. [24] illustrates NO\textsubscript{x} and smoke emission models based on an artificial neural network approach with Multi-Layer Perceptron (MLP) architectures. The results show that both models are capable of capturing the highly nonlinear engine-out emissions high accuracy and strong generalization performance over NEDC and WLTP missions. Finally, in ref. [25] the authors propose hierarchical models, namely “neuro-fuzzy model trees”, to predict transient particulate mass and NO\textsubscript{x}.

In this paper, a low-throughput mean-value zero dimensional model was considered, due to its physical consistency and low requirements in terms of calibration effort.

1.3. Contribution of the Present Study

The main contributions of this study are summarized hereafter.

1. A real-time predictive combustion model, that was previously developed by the authors for small and medium displacement engines [6–8,17], was assessed and applied to simulate BMEP and NO\textsubscript{x} emissions in a 11.0 L FPT Cursor 11 heavy-duty diesel engine. Several improvements and adaptations were implemented in the previous model version, in order to optimize its performance for a large displacement engine.

2. The model was extensively validated over about 2200 experimental tests, which include engine maps with and without pilot and EGR rate, sweep tests of injection pressure and injection timing, as well as sweep tests of EGR and VGT rack position. An extended validation of this type has not been addressed in the past, with reference to previous applications.

3. The model was also validated in transient operation, considering load ramps at different engine speeds.

4. A model robustness analysis to input parameters was carried out, in order to analyze the effect of the uncertainty of the main model input parameter on the accuracy of the main model outcomes.

5. A calibration sensitivity analysis was carried out, in order to investigate the impact of the number and type of experimental tests used for model calibration on the accuracy.

6. An analysis of the computational time that is required by the model, when it is run on an ETAS ES910 rapid prototyping device, was carried out. This is a preliminary step in order to check the real-time capability of the proposed approach and its suitability for the development of a model-based controller.

7. The model was inverted in order to derive a model based combustion controller, which receives targets of IMEP (Indicated Mean Effective Pressure) and engine-out NO\textsubscript{x} emissions, and sets the injected fuel quantity and main injection timing in order to achieve the desired targets. This controller was developed starting from a previous version [7]. However, several features were added in order to improve its performance and robustness in view of the implementation on a real engine. A preliminary assessment of the controller was carried out at steady-state conditions.
2. Experimental Setup and Engine Conditions

The experimental tests were measured for a FPT C11 11.0 L diesel engine. The main engine specifications are reported in Table 1.

<table>
<thead>
<tr>
<th>Engine Specifications</th>
<th>FPT Cursor 11 diesel engine</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine type</td>
<td>FPT Cursor 11 diesel engine</td>
</tr>
<tr>
<td>Number of cylinders</td>
<td>6</td>
</tr>
<tr>
<td>Displacement</td>
<td>11.1 dm³</td>
</tr>
<tr>
<td>Bore × stroke</td>
<td>128 mm × 144 mm</td>
</tr>
<tr>
<td>Conrod length</td>
<td>228 mm</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>20</td>
</tr>
<tr>
<td>Valves per cylinder</td>
<td>4</td>
</tr>
<tr>
<td>Turbocharger</td>
<td>VGT type</td>
</tr>
<tr>
<td>Fuel injection system</td>
<td>High pressure Common Rail</td>
</tr>
</tbody>
</table>

The engine scheme, including the main measurement sensors, is shown in Figure 1. A short-route EGR system equipped with a cooler is installed on the engine. The EGR valve is installed upstream from the EGR cooler. A flap is present in the exhaust pipe downstream of the turbine, in order to control the temperature of the exhaust gas flowing to the after-treatment system and to increase the EGR rate if necessary. The ATS system includes a Diesel Oxidation Catalyst (DOC), a DPF, an Selective Catalytic Reduction (SCR) and a Clean-Up Catalyst (CUC).

The test engine was instrumented with standard K-type thermocouples and piezo-resistive pressure transducers in order to acquire temperature and pressure at several locations, such as upstream from the compressor (pC1, Tc1), upstream and downstream from the intercooler (pIC1, TIC1, pIC2, TIC2), upstream and downstream from the EGR cooler (pEGRc,1, TEGRC,1, pEGRc,2, TEGRC,2), in the intake manifold, in the exhaust manifold, and downstream from the exhaust flap (pEF2, TEF2), upstream and downstream from the SCR (pSCR,1, TSCR,1, pSCR,2, TSCR,2), and in the urea injection system.
intake manifold ($p_{\text{IMF}, T_{\text{IMF}}}$), in the exhaust manifold ($p_{\text{EMF,cyl1-3}, T_{\text{EMF,cyl1-3}}, p_{\text{EMF,cyl4-6}}, T_{\text{EMF,cyl4-6}}}$),
downstream from the turbine ($p_{T2}, T_{T2}$), downstream from the exhaust flap ($p_{EF2}, T_{EF2}$), upstream
and downstream from the ATS system ($p_{ATS,1}, T_{ATS,1}, p_{ATS,2}, T_{ATS,2}$). The temperature was also
measured upstream and downstream from DOC ($T_{\text{DOC,1}, T_{\text{DOC,2}}}$) and SCR ($T_{\text{SCR,1}, T_{\text{SCR,2}}}$).
Moreover, for the ATS management, the ECU received information concerning the pressure differential across
the DPF ($\Delta p_{\text{DPF}}$), the NOx concentration upstream and downstream from ATS ($\text{NOx}_{\text{ATS,1}}, \text{NOx}_{\text{ATS,2}}$),
and the NH$_3$ concentration downstream from the SCR (NH$_3$). The NOx concentrations upstream and
downstream from the ATS system were obtained from two NOx sensors, characterized by an error of the order of
5–10%.

A Kistler 6045A high-frequency piezo-electric transducer ($p_{\text{cyl1}}$) was placed inside cylinder 1,
in order to acquire the in-cylinder pressure time-history on a crank basis. The in-cylinder pressure
traces were pegged on the basis of the intake absolute pressure that was measured by means of a
high-frequency Keller M510 piezo-resistive transducer, located in front of cylinder 1 ($p_{\text{I1,hf}}$). On the
exhaust side, a high-frequency cooled Kistler 4075A10 piezo-resistive transducer was also installed
($p_{e1,hf}$) to measure the absolute exhaust pressure.

The tests were carried out at a dynamic test rig at the FPT Motorenforschung in Arbon. Figure 2
shows a scheme of the test bench.

The test bench is equipped with a Schenck dynamometer (maximum power/torque/speed: 570 kW/3200 Nm/4000 rpm) and a PLU 401/121 fuel meter (measurement range: 0.35–150 L/h).

The raw engine-out gaseous emissions were measured by means of a Pierburg AMA4000 gas analyzer.

The developed combustion model was tested on an ETAS ES910 rapid prototyping device, in order
to check its real-time capability. The device specifications are shown in Table 2.

Finally, Table 3 reports a summary of the accuracy of the main test bench sensors that were used
to derive the experimental variables used for the model assessment.
Table 2. Main specifications of the ETAS ES910 device.

<table>
<thead>
<tr>
<th>ETAS ES910 Device</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Main processor</td>
<td>Freescale PowerQUICC™ III MPC8548 with 800 MHz clock Double precision floating point unit</td>
</tr>
<tr>
<td>Memory</td>
<td>512 MByte DDR2-RAM (400 MHz clock) 64 MByte Flash 128 kByte NVRAM</td>
</tr>
</tbody>
</table>

Table 3. Accuracy of the main test bench sensors.

<table>
<thead>
<tr>
<th>Measurement Device</th>
<th>Accuracy</th>
</tr>
</thead>
<tbody>
<tr>
<td>Encoder</td>
<td>Resolution: 0.1° CA</td>
</tr>
<tr>
<td>Fuel meter</td>
<td>Error &lt; ±0.1% of read value</td>
</tr>
<tr>
<td>Torque meter</td>
<td>Error &lt; ±0.95 Nm</td>
</tr>
<tr>
<td>In-cylinder pressure sensor</td>
<td>Linearity error: &lt; ±0.4% of Full Span (250 bar)</td>
</tr>
<tr>
<td>NO\textsubscript{x} CLD device in the gas analyzer</td>
<td>Error &lt; ±5% of read value when NO/NO\textsubscript{x} = 90%</td>
</tr>
<tr>
<td>O\textsubscript{2} POD device in the gas analyzer</td>
<td>Error &lt; ±10% of read value when NO\textsubscript{2}/NO\textsubscript{x} = 90%</td>
</tr>
</tbody>
</table>

Experimental Activity

The experimental tests which were considered in this paper for model assessment and validation were carried out at steady-state and transient conditions. More in detail, the following tests were considered for the model calibration and assessment:

1. a full engine map, carried out with pilot injection/EGR and without pilot injection/EGR (Figure 3, red circles).
2. sweep tests of SOI\textsubscript{main} (main injection timing) and p\textsubscript{f} (injection pressure) carried out at selected key-points (Figure 3, black circles). In these tests, for each engine operating point, SOI\textsubscript{main} and p\textsubscript{f} were varied of ±6 deg and of ±20% around the nominal values, respectively. The pilot quantity and the dwell-time between the pilot and main pulses were maintained constant. These tests were carried out in torque control mode. These sweep tests were carried out with pilot injection/EGR and without pilot injection/EGR.
3. EGR- and VGT-sweep tests at fixed key-points (Figure 3, blue circles).

The total number of steady-state tests is of the order of 2200.
With reference to the transient tests, which were used for model validation, the details are reported in the “Results and Discussion” section.

With reference to the fuel used in the present investigation, the engine was fed with conventional diesel fuel (according to EN 590 regulations), whose main properties are listed in Table 4.

### Table 4. Main properties of the diesel EN590 fuel.

<table>
<thead>
<tr>
<th>Property</th>
<th>Units</th>
<th>Diesel EN 590 Class 0</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cetane number</td>
<td>-</td>
<td>54</td>
</tr>
<tr>
<td>Flash Point</td>
<td>°C</td>
<td>68</td>
</tr>
<tr>
<td>Density at 15 °C</td>
<td>kg/m³</td>
<td>836</td>
</tr>
<tr>
<td>Density at 40 °C</td>
<td>kg/m³</td>
<td>818</td>
</tr>
<tr>
<td>Viscosity at 20 °C</td>
<td>mm²/s</td>
<td>4.49</td>
</tr>
<tr>
<td>Viscosity at 80 °C</td>
<td>mm²/s</td>
<td>1.49</td>
</tr>
<tr>
<td>Lower heating value</td>
<td>MJ/kg</td>
<td>43.0</td>
</tr>
</tbody>
</table>

3. Model Description

The scheme of the real-time combustion model, as well as of the needed input variables, is reported in detail in Figure 4.

![Figure 4. Scheme of the real-time combustion model.](image)

In particular, in Figure 4, N indicates the engine speed, $p_{IMF}$ and $T_{IMF}$ the intake manifold pressure and temperature, respectively, $p_{EMF}$ the exhaust manifold pressure, $m_{air}$ and $m_{EGR}$ the mass flow rate of fresh air and of EGR, respectively, $DT_{pil,j}$ the dwell-time of pilot injections, $q_{pil,j}$ the injected fuel quantity of the pilot injections, $T_{amb}$ the ambient pressure, $Rh_{amb}$ the ambient relative humidity, and $q_{f,inj}$ the total injected fuel quantity.

Basically, the heat release is evaluated first, on the basis of the accumulated fuel mass (AFM) approach. This allows the calculation of the chemical energy release ($Q_{ch}$) and of the related metrics (e.g., MFB50) to be performed.

Then, by using a heat transfer model, the net energy release ($Q_{net}$) is estimated, which in turn is used as input for the in-cylinder pressure model. The latter model is capable of estimating the
in-cylinder pressure trace and the related metrics, such as IMEPg (gross Indicated Mean Effective Pressure) and PFP (Peak Firing Pressure).

Finally, on the basis of friction (FMEP) and pumping (PMEP) models, the net IMEP (IMEPn) and BMEP (Brake Mean Effective Pressure) are estimated.

With reference to the NOx estimation, a slightly revised version of the approach described in ref. [26] was adopted. This model is based on the estimation of the deviations of NOx emissions, with respect to the nominal engine-calibration map values, as a function of the deviations of the intake oxygen concentration and MFB50.

The detailed description of each submodel is reported in detail in previous papers (see [17,26]).

However, a summary of the equations is reported in Tables 5 and 6. It should be noted that the model tuning parameters have been underlined in the equations. In particular, Table 5 reports the equations of the chemical and net energy release models, while Table 6 reports the equation of the pressure and of the NOx models.

Table 5. Main equations of the chemical and net energy release models. j indicates the generic injection pulse, K and τ are the combustion rate and ignition delay coefficients. Qfuel,j and Qch,pil,j indicate the fuel evaporation heat and the heat exchanged by the charge with the walls over the combustion cycle. m_t,inj is the total injected fuel mass per cycle/cyl.

<table>
<thead>
<tr>
<th>Q_ch Model</th>
<th>Q_net Model</th>
</tr>
</thead>
</table>
| \[ \frac{dQ_{ch,main}(t)}{dt} = q_{ch,main}(t) \] | \[ Q_{net,ht} \approx Q_{ch,net} - \sum_{j=1}^{n} Q_{ch,j} \]
| \[ Q_{l,inj}(t) = \int_{0}^{t} m_{t,inj}(t)H_{2}dt \] \( t \leq EOC_{ij} \) | \[ Q_{fuel,ht} \approx Q_{c,ht} \] | \[ Q_{fuel,pil,j} \approx Q_{c,ht} \] |
| \[ Q_{l,inj}(t) = \int_{EOC_{ij}}^{t} m_{t,inj}(t)H_{2}dt \] \( t > EOC_{ij} \) | \[ \] |

Table 6. Main equations of the pressure and of the NOx models. pIMF indicates the intake manifold pressure, pIVC the in-cylinder pressure at IVC, n and n’ the compression and expansion polytropic exponents. The combustion phase simulation derives from a single-zone heat release model [27].

<table>
<thead>
<tr>
<th>Pressure Model</th>
<th>NOx Model</th>
</tr>
</thead>
<tbody>
<tr>
<td>Starting condition ( p_{IVC} ) : PIVC = pIMF + ΔpIMF</td>
<td>NOx = NOx0( N, \text{IMEP}_n ) + ΔNOx (ΔMFB50, ΔO2, ( \text{IMEP}_n ), NOx0N)</td>
</tr>
<tr>
<td>Compression phase (IVC to SOC) : pIVC = ( p_{SOC} )</td>
<td>( g_{NOx} = g_0 \times \left[ -\text{abs}(\Delta \text{MFB50})^{\alpha_5} \times \text{sign}(\Delta \text{MFB50}) + \Delta \text{O}<em>2 \right] \times \left( \frac{\text{IMEP}<em>n}{\text{IMEP}</em>{n0}} \right)^{\beta</em>{NOx}} \times NOx_0^N )</td>
</tr>
<tr>
<td>Combustion phase (SOC to EOC): ( p' = \frac{\Delta \text{Q}<em>g}{\text{IMEP}</em>{n0} \gamma (\gamma - 1)} )</td>
<td>ΔMFB50 = MFB50 - MFB50_{n0}(N, \text{IMEP}_n) ; \ ΔO2 = \text{O}_2 - \text{O}_2(N, \text{IMEP}_n)</td>
</tr>
<tr>
<td>Expansion phase (EOC to EVO): : pV'' = ( p_{V''} )</td>
<td>( )</td>
</tr>
<tr>
<td>IMEP_p = ( \frac{\text{IMEP}_{n0}}{\gamma - 1} )</td>
<td>( )</td>
</tr>
<tr>
<td>IMEP_{n0} = IMEP_p + PMEP</td>
<td>( )</td>
</tr>
<tr>
<td>BMEP = IMEP_n - PMEP</td>
<td>( )</td>
</tr>
</tbody>
</table>

With reference to the NOx model, it can be seen from Table 6 that the engine-out NOx emissions as the sum of two contributions: the nominal NOx value ‘NOxN’, which represents the emissions when the engine operates at nominal conditions, and an NOx deviation term (‘ΔNOx’) which is null when MFB50 or intake O2 concentration deviate with respect to the nominal engine map values MFB50N and O2N.

The \( \left( \frac{\text{PMFP}}{q_{t,inj}} \right)^{ak} \times NOx_0^{\frac{a}{N}} \) term is a scaling factor which accounts for the fact that the range of NOx deviations changes as a function of the engine operating points, even for constant deviations of MFB50 and intake O2 concentration. The \( \frac{\text{PMFP}}{q_{t,inj}} \) term takes into account the effect of the air-to-fuel ratio.
(AF). AF was not used directly since it is affected by a higher degree of uncertainty than that of the $p_{\text{IMF}}/q_{\text{inj}}$ term.

The nominal values of NO$_x$ emissions, MFB50 and O$_2$ have been tabulated as a function of engine speed $N$ and IMEP$_n$, and these tables are based on the measurements performed at steady-state conditions. Finally, it should be recalled that the NO$_x$ model equations shown in Table 6 are valid for a given set of ambient temperature and humidity. If the model is applied to predict NO$_x$ emissions when the ambient conditions are varied, the NO$_x$ are corrected according to the recommended practice proposed in ref. [28]. The procedure is illustrated in ref. [26].

Estimation of the Intake O$_2$ Concentration

The intake O$_2$ concentration is a very important variable for the model, since it affects the combustion process and the NO$_x$ formation process to a great extent.

When the model is assessed using the test bench data, an accurate measurement of O$_2$ is available, by means of the test bench gas analyzer. However, if the model is used for onboard combustion control, this measurement is not available, and an alternative method is required to estimate it. Two approaches are possible for this purpose:

- The intake O$_2$ concentration is measured by means of an intake O$_2$ commercial sensor.
- The intake O$_2$ concentration is estimated on the basis of a sub-model.

In this paper, the impact of the method used to estimate the intake O$_2$ concentration on the model accuracy is investigated. The following sub-model was used to estimate O$_2$ [29]:

$$O_2 = x_1 \frac{X_{r,EGR}}{RAF} + x_2$$

(1)

where $x_1$, $x_2$ are fitting parameters, RAF is the relative air-to-fuel ratio, $X_{r,EGR}$ is the EGR rate.

The EGR rate $X_{r,EGR}$ is defined as follows:

$$X_{r,EGR} = \frac{m_{\text{EGR}}}{m_{\text{EGR}} + m_{\text{air}}}$$

(2)

where $m_{\text{EGR}}$ is the trapped EGR mass and $m_{\text{air}}$ is the trapped air mass.

The relative air-to-fuel ratio is defined as follows:

$$RAF = \frac{AF}{AF_{st}} = \frac{m_{\text{air}}}{m_{\text{inj}}AF_{st}}$$

(3)

where $m_{\text{air}}$ is the trapped air mass, $m_{\text{inj}}$ is the injected fuel mass, AF is the air-to-fuel ratio and AF$_{st}$ is the stoichiometric air-to-fuel ratio (taken as 14.4 in this work).

In this study, RAF was estimated from an UEGO sensor, while the air and EGR mass were estimated from a sub-model embedded in the ECU. Details of these submodels cannot be provided for confidentiality reasons. However, the general approach is as follows:

- The total trapped mass is evaluated by estimating the volumetric efficiency of the engine
- The EGR mass is estimated using an approach based on the pressure differential across the EGR valve and the EGR valve opening position (see [29])
- The air mass is estimated as the difference between total trapped mass and EGR mass.

The average root means square error in the estimation of the EGR mass flow rate, at steady-state operation, is of the order of 24 kg/h, and the average root mean square error of EGR rate is of the order of 3% (absolute error). The EGR mass flow rate error was estimated by comparing the EGR mass flow rate estimated by the ECU with the EGR mass flow rate derived from the experimental measurements. In particular, the latter was calculated by means of Equation (2), where the experimental value of
X_{\text{EGR}} was obtained on the basis of the intake and exhaust CO$_2$ concentrations, while the air mass flow rate was measured directly on the basis of the test bench sensors.

4. Model Calibration

Basically, the aim of the calibration phase is to identify physically-consistent correlations for the model calibration parameters, which have been underlined in Tables 5 and 6 and are listed in Table 7.

<table>
<thead>
<tr>
<th>Submodel</th>
<th>Calibration Parameter</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat release model</td>
<td>$K_{\text{pil,j}}$, $K_{\text{1,main}}$, $K_{\text{2,main}}$, $\tau_{\text{pil,j}}$, $\tau_{\text{main}}$</td>
</tr>
<tr>
<td>Net energy release model</td>
<td>$Q_{\text{f,evap}}$, $Q_{\text{ht,glob}}$</td>
</tr>
<tr>
<td>Pressure model</td>
<td>$\Delta P_{\text{IMF}}$, $n$, $n'$</td>
</tr>
<tr>
<td>PMEP model</td>
<td>PMEP, FMEP</td>
</tr>
<tr>
<td>NOx model</td>
<td>$a_1$, $a_2$, $a_3$, $a_4$, $a_5$</td>
</tr>
</tbody>
</table>

The model calibration was carried out according to the following steps.

- **Heat release model**: first, the experimental net and chemical heat release curves are derived on the basis of the measured in-cylinder pressure traces. Afterwards, the calibration parameters of the heat release model are tuned, test by test, in order to achieve the best possible matching between the predicted and experimental $Q_{\text{ch}}$ profiles. A least square fitting algorithm is used to this aim. At the end of this phase, a dataset of optimal values of $K$ and $\tau$ for all calibration tests is available. This dataset is used to derive physically-consistent correlations, as a function of significant engine variables. The selection of the input variables is done as a result of a trade-off between prediction accuracy (which takes benefit of a large number of input variables) and model robustness (which suffers from the adoption of a large number of variables).

- **Net energy release model**: the experimental values of $Q_{\text{f,evap}}$ and $Q_{\text{ht,glob}}$ are derived, for each calibration point, on the basis of the measured in-cylinder pressure and of the related net energy release. The dataset of optimal values of $Q_{\text{f,evap}}$ and $Q_{\text{ht,glob}}$ is then used to derive the physically-consistent correlations.

- **Pressure model**: the experimental values of $\Delta P_{\text{IMF}}$, $n$ and $n'$ are derived, for each calibration point, on the basis of the analysis of the measured in-cylinder pressure trace. The dataset of optimal values is then used to derive the physically-consistent correlations.

- **PMEP and FMEP models**: the experimental values of PMEP and FMEP are derived, for each calibration point, on the basis of the measured in-cylinder pressure trace. In particular, PMEP is obtained from the analysis of the pressure curve during the gas exchange process, while FMEP is obtained as the difference between the pressure-derived IMEP and the measured BMEP. The dataset of optimal values is then used to derive the physically-consistent correlations.

- **NOx model**: the experimental values of the deviations $\delta$NO$_x$, $\delta$MFB50, and $\delta$O$_2$ are estimated, for each calibration point, on the basis of the test bench measurements. The coefficients of the NO$_x$ model reported in Table 6 are then tuned on the basis of the least square method.

It should be noted that the indicated parameters for model calibration were extracted from one of the engine cylinders (i.e., cyl 1).

Therefore, any cyl-to-cyl variability of the main combustion or emission metrics is indirectly taken into account by the tuning of the model parameters. For example, consider the estimation of the FMEP parameter (see Table 6). During the model calibration phase, FMEP is estimated as the difference between the IMEP value of cyl 1 (which is derived from the in-cylinder pressure) and the BMEP value which is measured at the test bench. If cylinder 1 is characterized by a systematic higher performance than the other cylinders in terms of IMEP, this will result in an overestimation of FMEP with respect to the real value. As a consequence, the pressure model will provide a systematic overestimation of IMEP.
with respect to the average value of the cylinders (being calibrated using the data of cyl. 1), and the
friction model will provide a systematic overestimation of FMEP (being calibrated on experimental
values which are overestimated). However, when the full model is applied to estimate BMEP, there will
be a compensation of the effects, since the overestimation of IMEP provided by the model will be
compensated by the overestimation of FMEP, and a correct calculation of BMEP will be obtained.

First, the model calibration was carried out by using the engine map tests and the SOI\textsubscript{main}/P\textsubscript{f}
sweep tests (with and without EGR) shown in Figure 3. The VGT/EGR sweep-tests were instead used
for the model assessment. With reference to the heat release model, some simplifications were done on
the basis of the analysis of the heat release traces of the engine considered in this study. In particular, it
was verified that the contribution of the pilot shot is very small, and the dwell-time between the pilot
and main shots is generally short. Therefore, the start of combustion of the pilot shot was set equal to
the start of combustion of the main pulse in order to simplify the calculations, and a single correlation
was identified for the main ignition delay:

\[
\tau [\text{deg}] = 0.031 P_f^{-0.634} P_{SOI\textsubscript{main}}^{-0.682} N^{1.500} \tag{4}
\]

The following correlations were identified for the combustion rate parameters of the heat
release model:

\[
K_{1,\text{main}} \left( \frac{\text{deg}}{N} \right) = 16.789 p_f^{0.305} \rho_{SOI}^{0.074} \phi_{inj}^{0.163} P_{IMF}^{0.979} \tag{5}
\]

\[
K_{2,\text{main}} \left( \frac{\text{deg}}{N} \right) = 1.059 \times 10^{-3} p_f^{0.394} \rho_{SOI}^{0.980} O_2^{2.298} N^{-0.401} q_{\text{main}}^{0.483} \tag{6}
\]

\[
K_{\text{pil}} \left( \frac{\text{deg}}{N} \right) = 0.01 \tag{7}
\]

With reference to the net energy release model, the following correlations were found:

\[
Q_{\text{f, evap}} [\text{kJ}] = 8.65 E - 10 T_{IMF}^{0.715} N^{1.472} q_{\text{inj}}^{0.287} f_{inj}^{0.020} \tag{8}
\]

\[
Q_{\text{ht, glob}} [\text{kJ}] = 2.863 \rho_{SOI}^{0.724} N^{-0.295} q_{\text{inj}}^{1.372} O_2^{-1.0196} \tag{9}
\]

With reference to the pressure and BMEP models, the identified correlations are:

\[
\Delta p_{IMF} [\text{bar}] = A_0 p_{IMF}^{a_1} N^{a_2} \tag{10}
\]

\[
n = 3.084 T_{IMF}^{-0.167} N^{0.020} P_{IMF}^{0.017} q_{\text{inj}}^{-0.0017} (IVC \leq CA < 320^\circ) \tag{11}
\]

\[
n = 69.189 \times 10^{-132} N^{-0.011} P_{IMF}^{0.014} \phi_{\text{SOI first}}^{0.561} (320^\circ \leq CA > SOI) \tag{12}
\]

\[
n' = 8.778 \times N^{-0.030} q_{\text{inj}}^{-0.046} P_{IMF}^{0.067} \phi_{\text{SOI main}}^{-0.253} \tag{13}
\]

\[
PMEP [\text{bar}] = A_0 P_{EIMF}^{a_1} N^{a_2} - B_0 P_{IMF}^{b_1} N^{b_2} \tag{14}
\]

\[
FMEP [\text{bar}] = a_0 N + a_1 N^2 + a_2 \cdot PF \cdot F + a_3 \tag{15}
\]

\[
O_2 = -21.94 \frac{X_r}{RAF} + 20.57 \tag{16}
\]

It should be noted that the units of K and \( \tau \) parameters are expressed as a function of the crank
angle. This is due to the fact that the heat release model equations reported in Table 5 are implemented
on a crank angle domain. It can be seen that two distinct correlations have been worked out for the
polytropic coefficient \( n \). In fact, for the present engine, it was noted that the pressure trace simulation
and the accuracy in the estimation of IMEP can be largely improved when adopting two distinct
definitions of the polytropic coefficient, the first one being used for the first part (IVC to 320\(^\circ\)) and the
second one for the last part (320\(^\circ\) to pilot SOI) of the compression phase. In particular, it was verified
that using one constant value of the polytropic coefficient over the entire compression phase brought a systematic overestimation of IMEP, especially at high load conditions. The main reason is due to the fact that a large displacement engine, such as the C11 engine under study, typically features an injection pattern which includes a small pilot injection with relatively delayed SOI, with a short dwell time with respect to the main injection. As a consequence, the compression phase interval prior to the pilot SOI is quite long, and the thermal capacity of the charge varies over this interval, with a consequent variation in the value of the polytropic coefficient. It was verified that the introduction of two distinct values of the polytropic coefficients during the compression phase brought a reduction of the RMSE of IMEP with respect to the original approach. This can be seen in Table 8, which reports the RMSE of IMEP derived from the simulated pressure traces during the point-by-point calibration phase, using the two approaches. It can be seen that adopting two distinct values of n during the compression phase leads to significant improvement in the reconstruction of the in-cylinder pressure and IMEP estimation.

Table 8. RMSE of IMEP derived from the simulated pressure traces during the point-by-point calibration phase, when adopting a constant or two distinct definitions of the polytropic coefficient during the compression phase.

<table>
<thead>
<tr>
<th>Number of Definitions for the ‘n’ Coefficient</th>
<th>Engine Map Tests w/o EGR</th>
<th>SOImain/Pf Sweep Tests w/o EGR</th>
<th>Engine Map Tests with EGR</th>
<th>SOImain/Pf Sweep Tests with EGR</th>
<th>EGR/VGT Sweep Tests</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.509 bar</td>
<td>0.517 bar</td>
<td>0.326 bar</td>
<td>0.317 bar</td>
<td>0.212 bar</td>
</tr>
<tr>
<td>2</td>
<td>0.193 bar</td>
<td>0.199 bar</td>
<td>0.109 bar</td>
<td>0.107 bar</td>
<td>0.073 bar</td>
</tr>
</tbody>
</table>

The numerical values of the coefficients of Equations (10), (14) and (15) are not reported for confidentiality reasons.

Finally, for that which concerns the NO\textsubscript{x} model, the following formulation was identified for the NO\textsubscript{x} deviation:

\begin{equation}
\delta \text{NO}_x [\text{ppm}] = 0.024 \left[ -\text{abs}(\delta \text{MFB50})^{1.10} \cdot \text{sign}(\delta \text{MFB50}) + 3.85 \cdot \delta \text{O}_2 \right] \left( \frac{\text{P_{IMF}}}{\text{q_{inj}}} \right)^{0.431} \cdot \text{NO}_x^{1.39} (\delta \text{MFB50} \geq 0) \tag{17}
\end{equation}

\begin{equation}
\delta \text{NO}_x [\text{ppm}] = 0.025 \left[ -\text{abs}(\delta \text{MFB50})^{1.270} \cdot \text{sign}(\delta \text{MFB50}) + 3.34 \cdot \delta \text{O}_2 \right] \left( \frac{\text{P_{IMF}}}{\text{q_{inj}}} \right)^{0.499} \cdot \text{NO}_x^{1.495} (\delta \text{MFB50} < 0) \tag{18}
\end{equation}

5. Model-Based Combustion Controller

Within the IMPERIUM project, a model-based vehicle controller has been realized, which has the aim of optimizing the energy flows and the vehicle component operation, in order to achieve fuel consumption and urea consumption reduction of up to 20%. This is realized by means of an energy management supervisor, which is able to exploit real-time information concerning the traffic and the road characteristics, on the basis of a dynamic eHorizon system, so as to perform an on-board optimization of the vehicle operation. The energy manager supervisor is coupled with other model-based modules, including the ATS manager and the combustion controller, which was developed in this study. A scheme of the developed combustion controller is provided in Figure 5.
 Basically, the combustion controller receives targets of engine-out NO\textsubscript{x} (NO\textsubscript{xtgt}) and net IMEP (IMEP\textsubscript{tgt}) which are requested in real-time by the energy manager supervisor and the ATS management module, and predicts the values of the injected fuel quantity q\textsubscript{f,inj} and of the injection timing of the main pulse, i.e., SOI\textsubscript{main}, that allow the desired targets to be achieved. This is done through an iterative procedure, which is performed within the engine cycle and which stops when the difference between the predicted and the required values of the target variables fall below the predefined thresholds ε\textsubscript{IMEP} and ε\textsubscript{NOx}. During the iterative procedure, SOI\textsubscript{main} is corrected on the basis of the NO\textsubscript{x} error, while q\textsubscript{f,inj} is corrected on the basis of the IMEP error. The injection pressure p\textsubscript{f} is also corrected, with respect to the baseline value, using a correction map which is a function of the engine speed and load, in order to compensate for the soot increase that would be obtained when SOI\textsubscript{main} is delayed to achieve low targets of NO\textsubscript{x} emissions.

The proposed controller structure derives from that proposed in ref. [7], which had been developed for a 3.0 L diesel engine but not yet assessed for the implementation on a real engine. However, several improvements were made in comparison with the previous version in order to speed-up the convergence and improve its robustness in view of the implementation on the engine. The first difference is that the previous controller was based on a higher computationally demanding NO\textsubscript{x} model, which required a 3-zone thermodynamic approach, while the new controller is based on the new NO\textsubscript{x} model [26], which is much simpler and requires a shorter computational time.

The correction algorithm of q\textsubscript{f,inj} is the same as that described in ref. [7], i.e.,:

\[
q_{\text{f,inj},i+1}^j = q_{\text{f,inj},i}^j + K_{\text{IMEP},i}^j \cdot S_{\text{IMEP},i}^j \cdot \text{Err}_{\text{IMEP},i}^j
\]  

(19)

where ‘j’ indicates ‘the generic engine cycle, ‘i’ indicates the iteration, Err\textsubscript{IMEP,i} is the BMEP error at cycle ‘j’ and iteration ‘i’ (i.e., the difference between the target value and the predicted value), K\textsubscript{IMEP,i} is a modulation factor, which was introduced in order to optimize the response of the controller and to guarantee stable operations, S\textsubscript{IMEP,i} is an fuel-to-IMEP sensitivity factor which is derived from the average engine fuel conversion efficiency. It was shown in ref. [7] that the modulation factor K\textsubscript{IMEP,i} varies in the [0.2, 1] range, and its value is halved if the sign of the IMEP error changes between two consecutive iterations, or is doubled otherwise.

With reference to the SOI\textsubscript{main} correction, a new algorithm was proposed, which exploits the low-throughput capability of the NO\textsubscript{x} model used for the present application and its dependency
on MFB50 (see Table 6). Basically, for a given NO$_x$ target at cycle ‘j’ (i.e., NO$_{x\text{j_tgt}}$) an MFB50 sweep is built, which includes MFB50 variations of 0.1°, 1°, 3°, and 6° with respect to the nominal value of the current engine point. The NO$_x$ model is then applied for all of the sweep points, keeping all the other parameters constant, and the related NO$_x$ emissions are estimated for each MFB50 variation. Finally, using the results of the simulated sweeps, the value of MFB50 variation which leads to the desired target of NO$_x$ (i.e., ∆MFB50) is found by means of interpolation, and is used to correct SOI$_{\text{main}}$:

\[ \text{SOI}_{\text{main,}\text{i+1}} = \text{SOI}_{\text{main,i}} + \Delta\text{MFB50*} \quad (20) \]

It was verified that the adoption of this correction scheme allows the target of NO$_x$ to be achieved in a low number of iterations.

The use of the new NO$_x$ model also brings an improvement of robustness and accuracy when using real time ECU signals as input. In fact, the previous 3-zone-based NO$_x$ model, despite a similar performance when calibrated and validated using accurate input signals, showed a deterioration of accuracy when ECU-estimated input variables were used, due to its great sensitivity to the EGR and air mass.

Finally, several additional improvements were made in order to improve robustness and increase the resistance to input variables estimation failures, in view of the implementation on a real engine, such as:

- Introduction of upper/lower saturation for all the input variables
- Introduction of boundary safety limits for SOI$_{\text{main}}$ and $q_{\text{f,inj}}$. In particular, the SOI$_{\text{main}}$ boundary depends on engine speed and load, the $q_{\text{f,inj}}$ boundary is constant. These boundaries are introduced in order not to exceed the engine limits in terms of peak firing pressure and exhaust gas temperatures.
- Introduction of low thresholds of engine speed and injected fuel quantity, beyond which the controller is deactivated and the engine runs in standard ECU-based mode. This was done in order to improve robustness during the engine start phase and during the transitions from positive brake torque to cut-off conditions.
- Introduction of an output buffer for the control variables SOI$_{\text{main}}$ and $q_{\text{f,inj}}$, which allows a smoothing of the trends before being sent to the engine ECU.

6. Results and Discussion

In this section the real-time combustion model is assessed and validated at both steady-state (Section 6.1) and transient (Section 6.2) conditions. Section 6.3 reports instead a model robustness analysis, in order to verify the effect of the input parameter uncertainty on the accuracy of the model outcomes.

Section 6.4 is focused on a calibration sensitivity analysis, in order to check the effect of the number of type of calibration data on the model accuracy. This analysis demonstrates that the proposed model is robust even when a low number of points is used for calibration, thus it is physically consistent.

Section 6.5 reports the results of the computational time that is required by the model when it runs on an ETAS ES910 rapid prototyping device. This check is mandatory in order to verify the possibility to adopt the proposed model for the development of a model-based controller.

Finally, Section 6.6 reports a preliminary assessment of the functionality of the model-based controller.

6.1. Model Assessment at Steady-State Conditions

First, the model was assessed for the steady-state tests reported in Figure 3. It should be noted that the model performance is affected by the quality of the input parameters to a great extent. In order to verify this effect, three cases were considered:

- All the input variables are the result of the measurement of the test bench sensors (best case)
• The input variables are derived from the engine sensors or from the sub-models embedded in the ECU, except for the intake O$_2$ concentration.

• All the input variables are derived from the engine sensors or from the sub-models embedded in the ECU.

For the three considered cases, Figure 6 reports the predicted vs. experimental levels of the indicated quantities, i.e., MFB50 (Figure 6a–c), normalized PFP (Figure 6d–f), and IMEP$_n$ (Figure 6g–i), while Figure 7 reports the predicted vs. experimental values of BMEP (Figure 7a–c) and engine-out NO$_x$ emissions (Figure 7d–f). The experimental NO$_x$ levels were obtained from the engine test cell gas analyzer. The prediction accuracy of each model was quantified by the squared correlation coefficient ($R^2$) and by the Root Mean Squared Error (RMSE), which are reported in each figure. With reference to Figure 7d–f, the lines corresponding to ±20% with respect to the bisector have also been reported, in order to better analyze the error of the NO$_x$ model. The PFP values were normalized for confidentiality reasons.

**Figure 6.** Predicted vs. experimental values of MFB50 (a–c), normalized PFP (d–f) and IMEP$_n$ (g–i) for the steady-state tests reported in Figure 3. (a,d,g): input variables from test bench sensors; (b,e,h): input variables from ECU sensors/models except for intake O$_2$; (c,f,i): input variables from ECU sensors/models.
Figure 7. Predicted vs. experimental values of BMEP (a–c) and NO\textsubscript{x} emissions (d–f) for the steady-state tests reported in Figure 3. (a,d): input variables from test bench sensors; (b,e): input variables from ECU sensors/models except for intake O\textsubscript{2}; (c,f): input variables from ECU sensors/models.

The tests indicated in red refer to the engine map tests and SOI\textsubscript{main}/p\textsubscript{T} sweep tests without EGR, the tests indicated in black refer to the engine map tests and SOI\textsubscript{main}/p\textsubscript{T} sweep tests with EGR and finally tests indicated in blue refer to the VGT/EGR sweep tests.

It can be seen in the figures that, in general, the accuracy of the model is good. It was verified that the error distributions are centered around 0 for all parameters and feature Gaussian-like shapes (not shown here for the sake of brevity). Moreover, the accuracy in the estimation of MFB50, PFP, IMEP\textsubscript{n}, and BMEP parameters is slightly affected when ECU-derived input parameters are used instead of bench-derived parameters.

With reference to NO\textsubscript{x} estimation, the RMSE of the model is of the order of 80 ppm when the input variables are estimated from test bench sensors, and it is not affected if they are obtained from ECU sensors except for intake O\textsubscript{2} (see Figure 7c). However, if the ECU model is used to estimate the intake O\textsubscript{2} concentration (Figure 7f), the RMSE of the NO\textsubscript{x} model increases to 111 ppm.

Table 9 shows the values of RMSE of the NO\textsubscript{x} model for the three types of tests, in the two cases in which the input quantities are taken from the test bench sensors and from the ECU sensors or models.

Table 9. RMSE of the NO\textsubscript{x} model for the three types of tests, in the two cases in which the input quantities are taken from the test bench sensors and from the ECU sensors or models.

<table>
<thead>
<tr>
<th>Type of Test</th>
<th>Input Variables from Bench Sensors</th>
<th>Input Variables from ECU Sensors/Models</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine map and sweep tests of SOI\textsubscript{main}/p\textsubscript{T} without EGR</td>
<td>101 ppm</td>
<td>95 ppm</td>
</tr>
<tr>
<td>Engine map and sweep tests of SOI\textsubscript{main}/p\textsubscript{T} with EGR</td>
<td>75 ppm</td>
<td>117 ppm</td>
</tr>
<tr>
<td>EGR/VGT sweep tests</td>
<td>44 ppm</td>
<td>127 ppm</td>
</tr>
</tbody>
</table>
It can be seen in Table 9 that the accuracy is almost unaffected for the tests without EGR, while the RMSE increases significantly for the tests featuring EGR when using the ECU-derived sensors or models, mainly as a consequence of the inaccuracy of the EGR model embedded in the ECU.

The previous results suggest that the adoption of an intake O\textsubscript{2} sensor is recommended for model-based combustion control applications, in order to keep the accuracy of the NO\textsubscript{x} model at a high level.

6.2. Model Validation over Transient Conditions

After the assessment at steady-state operation, the model was then validated in transient operation. The results reported in this section are related to two different transient tests. The first test type is constituted by a series of tip-in and tip-out load maneuvers at N = 1500 rpm. The second test type is constituted by a series of up/down load ramps at N = 1900 rpm. Each type of test was carried out with and without EGR. Figures 8 and 9 report the results of the tip-in and tip-out maneuvers without (Figure 8) and with (Figure 9) EGR, while Figures 10 and 11 report the results of the load ramps without (Figure 10) and with (Figure 11) EGR. In each figure, the time histories of the predicted (blue) and experimental (red) values of MFB\textsubscript{50} (Figures 8a, 9a, 10a and 11a), normalized PFP (Figures 8b, 9b, 10b and 11b), BMEP (Figures 8c, 9c, 10c and 11c) and NO\textsubscript{x} emissions (Figures 8d, 9d, 10d and 11d) are reported. The experimental NO\textsubscript{x} levels were obtained from the engine NO\textsubscript{x} sensor. The values of R\textsuperscript{2} and RMSE are also reported in each chart. With reference to the NO\textsubscript{x} emissions, two modeled curves are reported: the light-blue trace refers to the raw model output (“Model”), while the blue trace was obtained by filtering the raw model output using a cut-off frequency of 0.66 Hz (“Model filt”). This filtering was adopted in order to better compare the model outcomes with the experimental trace obtained from the engine NO\textsubscript{x} sensor, since the latter is characterized by a certain degree of smoothing and delay due to the response time of the sensor and of the mixing of the gases in the exhaust manifold. The PFP values were normalized for confidentiality reasons.

With reference to the experimental indicated quantities (MFB\textsubscript{50} and PFP), they were processed and obtained directly from the test bench acquisition system.

With reference to the tip-in and tip-out load maneuvers (Figures 8 and 9), it can be seen from the charts that the average errors of MFB\textsubscript{50}, PFP, and BMEP are of the order of 1.6–1.7 degree, 6 bar, and 0.46–0.48 bar, respectively, and these errors are slightly influenced by the use of EGR. On the contrary, the prediction accuracy of NO\textsubscript{x} emissions is strongly influenced by the use of EGR. In fact, it can be seen that when EGR is not adopted (Figure 8d), a very good matching is obtained between the predicted and experimental trace, especially if the filtered model trend is taken, with a value of RMSE of the order of 85 ppm. On the contrary, when EGR is adopted, the model accuracy deteriorates (the RMSE value is of the order of 305 ppm for the filtered model trace). It can be seen in Figure 9d that the model over predicts NO\textsubscript{x} emissions at the end of the tip-in maneuver, while the experimental trace shows a drop of NO\textsubscript{x} emissions in the same intervals. The main reason is due to the fact that the EGR model embedded in the ECU, although being accurate at steady-state operation, is not able to capture highly transient phenomena which occur in the intake manifold: as a consequence, the intake O\textsubscript{2} concentration, given to the NO\textsubscript{x} model as input, is much higher that the real one, and the model over predicts the NO\textsubscript{x} levels. An improvement of the EGR model embedded in the ECU, or the installation of an intake O\textsubscript{2} sensor may lead to significant improvements in the estimation of the NO\textsubscript{x} emissions when EGR is adopted.
Figure 8. Predicted and experimental trends of MFB50 (a), normalized PFP (b), BMEP (c), and engine-out NOx emissions (d) for the tip-in/tip-out load maneuvers at 1500 rpm, without EGR. The RMSE values are also reported.
With reference to the tip-in and tip-out load maneuvers (Figures 8–9), it can be seen from the charts that the average errors of MFB50, PFP, and BMEP are of the order of 1.6–1.7 degree, 6 bar, and 0.46–0.48 bar, respectively, and these errors are slightly influenced by the use of EGR. On the contrary, the prediction accuracy of NOx emissions is strongly influenced by the use of EGR. In fact, it can be seen that when EGR is not adopted (Figure 8d), a very good matching is obtained between the predicted and experimental trace, especially if the filtered model trend is taken, with a value of RMSE of the order of 85 ppm. On the contrary, when EGR is adopted, the model accuracy deteriorates (the RMSE value is of the order of 305 ppm for the filtered model trace). It can be seen in Figure 9d that the model over predicts NOx emissions at the end of the tip-in maneuver, while the experimental trace shows a drop of NOx emissions in the same intervals. The main reason is due to the fact that the EGR model embedded in the ECU, although being accurate at steady-state operation, is not able to capture highly transient phenomena which occur in the intake manifold: as a consequence, the intake O2 concentration, given to the NOx model as input, is much higher that the real one, and the model over predicts the NOx levels. An improvement of the EGR model embedded in the ECU, or the installation of an intake O2 sensor may lead to significant improvements in the estimation of the NOx emissions when EGR is adopted.

**Figure 9.** Predicted and experimental trends of MFB50 (a), normalized PFP (b), BMEP (c), and engine-out NOx emissions (d) for the tip-in/tip-out load maneuvers at 1500 rpm, with EGR. The RMSE values are also reported.
Figure 10. Predicted and experimental trends of MFB50 (a), normalized PFP (b), BMEP (c), and engine-out NOx emissions (d) for the load ramp tests at 1900 rpm, without EGR. The RMSE values are also reported.
With reference to the load ramps (Figures 10 and 11), a similar behavior of the model can be observed compared to that for the tip-in/tip-out maneuvers. In fact, it can be seen from the charts that the average errors of MFB50, PFP, and BMEP are of the order of 1.3–1.4 degree, 2.7–3 bar, and 0.45–0.48 bar, respectively, and these errors are slightly influenced by the use of EGR. The NO\textsubscript{x} model is characterized by a good accuracy when EGR is not adopted (Figure 10d, RMSE = 116 ppm for the filtered model trace) and by a lower accuracy when EGR is adopted (Figure 11d, RMSE = 173 ppm for the filtered model trace). Also in this case, it can be observed that the model over predicts NO\textsubscript{x} emissions at the end of the load ramps, as a consequence of an overestimation of the intake O\textsubscript{2} concentration during those phases.

With reference to the BMEP error, it is interesting to note that a significant contribution may be related to the uncertainty in the fuel quantity which is realized by the injectors with respect to the desired one. This can be verified in Figure 11c, which shows a comparison between the fuel quantity measured from the fuel meter (red line in the small box) and the ECU fuel setpoint (blue line in the small box). The ECU-derived fuel setpoint quantity was given to the model as input, and this can justify the BMEP underestimation in the considered time interval. The fuel quantity of the balance could not be used over the transients, due to a slow response in time.

On the basis of these results, an improvement of the accuracy of the injector maps is currently being performed using test bench results.
6.3. Model Robustness Analysis

A model robustness analysis has been carried out in this section, considering the steady-state tests reported in Figure 3. In particular, the procedure that was followed is reported hereafter.

- Combined variations numbering 2000 of the main model input variables were generated according to a Gaussian distribution, in which the standard deviation of each parameter was set according to the accuracy of the engine sensors or of the models used to estimate the parameters (see Table 10).
- The simulation of the steady-state points was performed considering all the 2000 combined variations of the main model input variables.
- The distribution of the RMSE values of the main model outcomes was evaluated.

<table>
<thead>
<tr>
<th>Input Model Parameter</th>
<th>Description</th>
<th>Standard Deviation σ</th>
</tr>
</thead>
<tbody>
<tr>
<td>PI_{IMF}</td>
<td>Intake manifold pressure</td>
<td>0.33% of read value</td>
</tr>
<tr>
<td>TI_{IMF}</td>
<td>Intake manifold temperature</td>
<td>0.33 °C</td>
</tr>
<tr>
<td>PE_{MF}</td>
<td>Exhaust manifold pressure</td>
<td>0.33% of read value</td>
</tr>
<tr>
<td>m_{air}</td>
<td>Trapped air mass</td>
<td>2.2E-1 g/stroke</td>
</tr>
<tr>
<td>m_{EGR}</td>
<td>Trapped EGR mass</td>
<td>1.1E-1 g/stroke</td>
</tr>
<tr>
<td>q_{inj}</td>
<td>Injected fuel quantity</td>
<td>6 mm³</td>
</tr>
<tr>
<td>P_{f}</td>
<td>Injection pressure</td>
<td>0.0033% of read value</td>
</tr>
<tr>
<td>SOI_{main}</td>
<td>Start of injection of the main pulse</td>
<td>0.067° CA</td>
</tr>
<tr>
<td>T_{amb}</td>
<td>Ambient temperature</td>
<td>0.33°</td>
</tr>
<tr>
<td>Rh_{amb}</td>
<td>Ambient relative humidity</td>
<td>0.67% of read value</td>
</tr>
</tbody>
</table>

The results of this analysis are reported in Figure 12. It can be seen that the most frequent values of RMSE are aligned with respect to the results shown in Figures 6 and 7, and that the maximum values of RMSE for the worst cases are of the order of 1.2 degree for MFB50, 6.5 bar for PFP, 1.4 bar for IMEP\textsubscript{n} and BMEP, and 250 ppm for the NO\textsubscript{x} emissions. It was verified that the main variable which affects the RMSE of IMEP\textsubscript{n} and BMEP is the injected fuel quantity, while the main variables which affect the RMSE of NO\textsubscript{x} emissions are the air and EGR mass. Therefore, an effort should be made in order to increase the accuracy of the injectors in realizing the desired fuel quantity and the accuracy of the submodels that estimate the air and EGR mass, in order to increase the robustness of the model. With reference to NO\textsubscript{x} simulation, it could also be beneficial to install an intake O\textsubscript{2} sensor in order to replace the sub-models of air and EGR mass.
Figure 12. Distributions of the RMSE values of (a) MFB50; (b) PFP; (c) IMEPn; (d) BMEP; and (e) NOx emissions, when performing 2000 model simulations with input parameter deviations according to the standard deviations shown in Table 5.
6.4. Model Calibration with Limited Number of Experimental Tests

A calibration sensitivity analysis is carried out in this section. In particular, the original calibration dataset was progressively reduced and the accuracy of the resulting model checked over the entire dataset.

The reduced calibration datasets are shown in Figure 13 and were identified as follows:

- Reduced calibration dataset 1: the engine map tests were reduced to 50% of the original dataset as shown in Figure 13a,b; the SOI\textsuperscript{main}-p\textsubscript{f} sweep tests were reduced by taking the maximum and minimum values of SOI\textsuperscript{main} and p\textsubscript{f} for each engine point, as well as additional tests as shown in Figure 13c (an example is reported in Figure 13c for the 900 × 2.4 engine point).
- Reduced calibration dataset 2: the engine map tests were reduced to 20% of the original dataset as shown in Figure 13a,b; the SOI\textsuperscript{main}-p\textsubscript{f} sweep tests were reduced by taking only the maximum and minimum values of SOI\textsuperscript{main} and p\textsubscript{f} for each engine point, as well as the central point (see Figure 13c).

![Figure 13](image-url)

**Figure 13.** Reduction of the number of calibration points with reference to the engine map tests (a,b) and SOI\textsuperscript{main}-p\textsubscript{f} sweep tests (c, an example is reported for the 900 × 2.4 engine point).

The model was tuned for each reduced dataset, and applied to the whole dataset, and the accuracy was estimated on the basis of the RMSE parameter. The result of this analysis is reported in Table 11 for the steady-state tests of Figure 3.
Table 11. Values of RMSE of the main model outcomes, for the different calibration datasets, when applying the model to the steady-state tests reported in Figure 3.

<table>
<thead>
<tr>
<th>Calibration Datasets</th>
<th>MFB50 RMSE</th>
<th>PFP RMSE</th>
<th>IMEP RMSE</th>
<th>BMEP RMSE</th>
<th>NOx RMSE</th>
</tr>
</thead>
<tbody>
<tr>
<td>Original</td>
<td>0.559</td>
<td>4.507</td>
<td>0.289</td>
<td>0.281</td>
<td>80.0</td>
</tr>
<tr>
<td>Reduced (50% of original)</td>
<td>0.557</td>
<td>4.495</td>
<td>0.286</td>
<td>0.279</td>
<td>79.4</td>
</tr>
<tr>
<td>Reduced (20% of original)</td>
<td>0.569</td>
<td>4.481</td>
<td>0.284</td>
<td>0.274</td>
<td>79.1</td>
</tr>
</tbody>
</table>

It can be observed in the tables that the model accuracy is virtually unaffected by the reduction of the number of calibration points. This indicates that the model is physically consistent since it does not require a large number of points in order to obtain an accurate calibration (20% of the original dataset proposed in Figure 3 is sufficient). Moreover, it can be noted that, in addition to the engine maps, detailed sweep tests of the main engine parameters (SOI_{main}, p_f, . . . ) are not required, since the use of the boundary values (i.e., the maximum and minimum values of the variation ranges) is sufficient.

6.5. Required Computational Time on ETAS ES910

The model was developed in the Matlab/Simulink environment and was then implemented on a rapid prototyping (RP) device (i.e., ETAS ES910), through ETAS Intecrio software.

The aim of this investigation was to test the real-time capability of the combustion model, for the subsequent development of the model-based controller.

The average computational time required by the model on the ETAS ES910 device is of the order 300–400 µs, when a crank angle integration step of 1 degree is used. Therefore, the model features real-time capability.

6.6. Assessment of the Model-Based Combustion Controller at Steady-State Conditions

The model-based combustion controller was preliminary assessed at steady-state conditions. Basically, the experimental values of IMEP and NO_x emissions were set as targets of the controller, and the values of q_{f,inj} and SOI_{main} calculated by the controller were compared to the real actuated values. The results are shown in Figure 14.

![Figure 14](image_url)

**Figure 14.** Model-based controller: calculated vs. actuated values of SOI_{main} (a) and q_{f,inj} (b) for the steady-state conditions. The experimental values of IMEP and NO_x emissions were set as targets of the controller.
It can be seen that the RMSE of SOI\textsubscript{main} is of the order of 1 degree, while the RMSE of the injected fuel quantity is of the order of 3.4 mm\textsuperscript{3}/cyl/stroke.

An additional functionality test was carried out, in order to check its capability to respond to different targets of NO\textsubscript{x} emissions, with respect to the baseline experimental levels, while maintaining nominal values of IMEP targets. The results of this analysis are reported in Figure 15, which reports the values of SOI\textsubscript{main} estimated by the controller vs. the baseline actuated values, when setting NO\textsubscript{x} emissions targets equal to $-40\%$ (Figure 15a) and $+40\%$ (Figure 15b) with respect to the baseline experimental levels.

It can be seen that the controller responds adequately to different NO\textsubscript{x} target requests, by adjusting SOI\textsubscript{main} in advance or in delay with respect to the nominal values actuated by the ECU.

Finally, another useful piece of information is related to the number of iterations which is required to achieve convergence, since it is strictly correlated to the overall computational time required by the controller. This information is shown in Figure 16, which reports the percentage of points which requires, for convergence, a lower or equal number of iterations than those reported in the $x$-axis, for the three different NO\textsubscript{x} targets.

![Figure 15. SOI\textsubscript{main} values estimated by the controller vs. the baseline actuated values, when setting NO\textsubscript{x} emissions targets equal to $-40\%$ (a) and $+40\%$ (b) with respect to the baseline experimental levels.](image)

![Figure 16. Model-based controller: percentage of points which requires, for convergence, a lower or equal number of iterations than those reported in the $x$-axis. The maximum number of allowed iterations was set at 10.](image)
It can be seen that more than 90% of points requires a number of iterations which is lower or equal to 6. Therefore, when implemented on the real-engine, a maximum number of iterations equal to 6–8 can be set for the controller, which leads to an overall required computational time that is of the order of 2–3 ms.

7. Conclusions

A control-oriented real-time combustion model was assessed and applied to simulate BMEP (Brake Mean Effective Pressure) and NO\textsubscript{x} (Nitrogen Oxide) emissions in an 11.0 L FPT Cursor 11 diesel engine for heavy-duty applications. The activity was carried out in the frame of the IMPERIUM (IMplementation of Powertrain Control for Economic and Clean Real driving emission and fuel ConsUMption") H2020 EU Project. The model was used to derive a model-based controller, which identifies the values of injected quantity and main injection timing in order to achieve the desired targets of indicated mean effective pressure and NO\textsubscript{x} emissions.

First, the model was calibrated and assessed at steady-state conditions, considering engine map tests and sweep tests of injection timing and injection pressure, with and without EGR employment, as well as sweep-tests of VGT rack position and EGR. Within this analysis, several cases were considered, in which the input model parameters were obtained from accurate bench sensors or directly from the engine sensors. Subsequently, it was validated in transient operation over several load ramps at different engine speeds. In addition to BMEP and NO\textsubscript{x} emissions, additional combustion metrics were also included in the analysis, such as MFB50, peak firing pressure (PFP) and IMEP.

It was found that, at steady-state operation, the average error is of the order of 0.28–0.32 bar with reference to BMEP prediction, and of the order of 80–110 ppm with reference to NO\textsubscript{x} simulation, where the largest error occurs when ECU-derived input parameters are used. Similar accuracy levels were obtained in transient operation.

Subsequently, a statistical model robustness analysis was carried out, in order to verify the impact of the uncertainty in the input variables on the accuracy of the predicted quantities. It was found that the maximum values of RMSE are of the order of 1.4 bar for BMEP and 250 ppm for the NO\textsubscript{x} emissions, in the worst cases. The main variable which affects the BMEP estimation accuracy is the injected fuel quantity, while the main variables which affect the NO\textsubscript{x} emission estimation accuracy are the air and EGR mass. Therefore, an effort should be made in order to increase the accuracy of the injectors in realizing the desired fuel quantity and the accuracy of the sub-models that estimate the air and EGR mass, in order to increase the robustness of the proposed model. With reference to the NO\textsubscript{x} simulation, it could also be beneficial to install an intake O\textsubscript{2} sensor in the engine in order to improve the model accuracy especially in transient operation, and when EGR is adopted.

A calibration sensitivity analysis was also carried out, in order to verify how the model accuracy is affected by a reduction of the points used for calibration. It was found that the accuracy of the model is virtually unaffected by the reduction of calibration points, even when the original calibration dataset is reduced by 80%.

It was verified that the proposed model features real-time capability, since the required computational time on the ETAS ES910 rapid prototyping device is of the order of 300–400 µs. Finally, the basic functionality of the model-based combustion controller was tested at steady-state conditions.

Author Contributions: The authors equally contributed to the deployment of the paper. Conceptualization, R.F., O.M., E.S.; Methodology, R.F., O.M.; Software, O.M.; Formal Analysis, R.F., O.M.; Data Curation, R.F., O.M., A.M. (Alessandro Mancarella); Writing—Original Draft Preparation, R.F., O.M.; Writing—Review and Editing, R.F., O.M., G.H.; Supervision, A.M. (Antonio Mittica), E.S.; Resources; G.H.

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Conflicts of Interest: The authors declare no conflict of interest.
Abbreviations

AF air-to-fuel ratio
AF_{st} stoichiometric air-to-fuel ratio
ANN artificial neural network
ATS after-treatment system
BMEP Brake Mean Effective Pressure (bar)
CA crank angle (degree)
CFD Computer Fluid-Dynamics
CLD Chemiluminescence device
CNG Compressed Natural Gas
CUC Clean-Up Catalyst
DOC Diesel Oxidation Catalyst
DPF Diesel Particulate Filter
DT dwell-time
ECU Engine Control Unit
EGR Exhaust Gas Recirculation
EOI end of injection (degree)
EVO Exhaust Valve Opening
FMEP Friction Mean Effective Pressure (bar)
FPT Fiat Powertrain Technologies
HCCI Homogeneous Charge Compression Ignition
H_{L} lower heating value of the fuel
ICE Internal Combustion Engines
IMEP_{g} gross Indicated Mean Effective Pressure (bar)
IMEP_{n} net Indicated Mean Effective Pressure (bar)
IVC Intake Valve Closing
K combustion rate coefficient
m mass
m_{air} trapped air mass
m_{EGR} trapped EGR mass
m_{f,inj} total injected fuel mass (mg per cycle/cylinder)
m_{f,inj} fuel injection rate
MFB50 crank angle at which 50% of the fuel mass fraction has burned (degree)
n polytropic coefficient for the compression phase
n' polytropic coefficient for the expansion phase
N engine rotational speed (1/min)
NEDC New European Driving Cycle
O_{2} intake charge oxygen concentration (%)
p pressure (bar)
PCCI Premixed Charge Compression Ignition
P_{EMF} exhaust manifold pressure (bar abs)
P_{f} injection pressure (bar)
PFP peak firing pressure
P_{IMF} intake manifold pressure (bar abs)
PMEP Pumping Mean Effective Pressure (bar)
POD Paramagnetic Oxygen Detector
q injected fuel volume quantity (mm$^3$)
Q_{ch} chemical heat release
Q_{f,evap} fuel evaporation heat
q_{f,inj} total injected fuel volume quantity per cycle/cylinder
Q_{fuel} \quad \text{chemical energy associated with the injected fuel}
Q_{ht,glob} \quad \text{global heat transfer of the charge with the walls}
Q_{net} \quad \text{net heat release}
R^2 \quad \text{squared correlation coefficient}
RAF \quad \text{Relative Air-to-Fuel ratio}
Rh_{amb} \quad \text{ambient relative humidity}
RMSE \quad \text{root mean square error}
SOC \quad \text{start of combustion}
SCR \quad \text{Selective Catalytic Reduction}
SOI \quad \text{electric start of injection}
SOI_{main} \quad \text{electric start of injection of the main pulse}
SVM \quad \text{support vector machine}
t \quad \text{time}
T \quad \text{temperature (K)}
T_{amb} \quad \text{ambient temperature}
T_{EMF} \quad \text{exhaust manifold temperature}
T_{IMF} \quad \text{intake manifold temperature}
u^2 \quad \text{variance}
V \quad \text{volume}
V2I \quad \text{Vehicle-to-Infrastructure}
V2V \quad \text{Vehicle-to-Vehicle}
V2X \quad \text{vehicle-to-everything technology}
VGT \quad \text{Variable Geometry Turbine}
VPM \quad \text{Virtual Pressure Model}
WLTP \quad \text{Worldwide harmonized Light-duty Test Procedure}
X_{r,EGR} \quad \text{EGR rate}

Greek symbols
ρ \quad \text{density}
ρ_{SOI} \quad \text{in-chamber ambient density evaluated at the SOI instant}
σ \quad \text{standard deviation}
τ \quad \text{ignition delay coefficient}

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