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Development and validation of a “crank-angle” model of an automotive turbocharged Engine for HiL Applications

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Abstract

Management and diagnostic functions are playing a key role in the improvement of engines performance and in the reduction of fuel consumption and pollutant emissions especially in automotive applications. As widely documented in the open literature, design, validation, and testing of control systems take actually advantage of theoretical models to a great extent, due to their capabilities to reduce development time and costs. However, the increasing complexity of present engines and related management systems give rise to challenging issues in the development and applications of mathematical models.

The paper describes the improvements introduced in the original Library set up by the authors in Simulink[®] for “control-oriented” simulation of Internal Combustion Engines (ICE) and powertrains. The tool has been initially developed to build up Mean Value Models (MVMs) of automotive engines for “real-time” simulations, and in that version has been used in several HiL applications. Due to the enhancing requirements in engine control functions, the Library has been recently improved to allow for “crank-angle” simulation of the engine. To this extent models of intake and exhaust valves and of in-cylinder processes have been built up (where combustion process is described following a classic single-zone approach based on a proper Heat Release Rate, HRR). An original algorithm has been developed to run the model at a computational speed comparable with real time even with a resolution of 1 degree CA for in-cylinder calculation.

Modeling tools have been applied to the simulation of a four-cylinder turbocharged Diesel engine with Exhaust Gas Recirculation. Through a specific calibration procedure, the model was fitted on a typical layout of an automotive Diesel engine and then validated comparing simulation results with experimental data measured by the OEM on a test bench. With a very low computational time, the model showed interesting capabilities in the simulation of the behavior of automotive engines with “crank-angle” resolution and therefore has been used in an original HiL application developed by the authors.

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Nomenclature

b_{mep}	brake mean effective pressure
e	specific internal energy
f	burning function
h	specific enthalpy, heat transfer coefficient, valve lift
k	ratio of specific heat coefficients ($= c_p/c_v$)
m	mass
\dot{m}	mass flow rate
n	rotational speed (in [rpm])
p	pressure
t	time
\tilde{x}	mass fraction
μ	molecular mass
A	flow or surface area
C	coefficient
C_d	discharge coefficient
\dot{Q}	heat flow
R	gas constant
T	temperature, period
V	volume
θ	crank angle
τ	ignition delay

Subscripts

c	combustion
cyl	cylinder
ex	exhaust
f	fuel
in	intake, inlet
out	outlet
tot	total
w	cylinder walls

1. Introduction

The constant need of a continuous improvement of the performance of automotive engines and powertrains (i.e., specific power/torque and at the same time fuel consumption and emissions) is continuously pushing for both an enhancement of existing subsystems (e.g., fuel injection, valve actuation systems, etc.) and the introduction of innovative solutions (e.g., dual-stage turbocharging, low- and high-pressure Exhaust Gas Recirculation systems, aftertreatment components, etc.) and smart sensors (e.g., piezoelectric transducers for in-cylinder pressure measurement). As a matter of fact, the complexity of actual powertrains and the large number of involved variables require unceasing improvements in the computational capabilities of ECUs and in the comprehension of processes that affect the behaviour of the controlled systems.

The application of innovative technologies is really effective only if a proper design of the control system and of related management strategies is developed side by side. Physical modelling and simulation represent a solid

support to outline system behaviour, which is strongly non-linear. Mathematical models are powerful tools to estimate the effects of control parameters and strategies on engine and powertrain performance and to reduce development time and costs, shortening the way from design specifications to on-road testing.

In this field, in-cylinder pressure is an important variable for combustion analysis and diagnostics. The actual availability of cost-effective transducers paved the way to use this parameter as a feedback signal for combustion control. As a consequence, mathematical models able to simulate in-cylinder processes still keeping calculation time very short, and even lower than real-time, are required. Proper simplifications have to be introduced in order to catch the real behaviour of the system from the behaviour of its components, joining physical –and chemical– principles (generally based on conservation laws) with an empirical description of more complex phenomena (i.e., taking account of relevant aspects and avoiding a detailed description of the others).

Examples of “control-oriented” engine models proposed in the literature for the simulation of intake and exhaust systems flows are reported in [1,2,3,4], while a comprehensive scenario is outlined in [5,6,7]. Typically Filling-and-Emptying (F&E) and Quasi-Steady Flow (QSF) approaches are used to build up 0-D, lumped parameter and cycle averaged Mean Value Models (MVM).

When dealing with the simulation of “cycle-to-cycle” phenomena (typically intake and exhaust flows and combustion process) F&E and QSF methods can be still used, but more complex models usually result, whose equations have to be integrated on a crank angle base. Calculation time is usually a challenge in this case, since the estimation of in-cylinder pressure and temperature histories requires more detailed approaches than those used for Mean Value Models [5,8]. The complexity of chemical and physical processes which take place in the cylinder would be properly described through “multi-dimensional” methods [9,10], but for “fast” applications simplified “zero-dimensional” single-zone models proposed in the past [11,12] seem still the best option. In-cylinder processes can be simulated on a crank-angle base following a F&E approach joined with a QSF technique for gas flows through valves by applying the energy and mass conservation equations to a gas mixture which is considered homogeneous within the cylinder (“single-zone” models). Combustion process can be simulated through the definition of a proper fuel burning function [9,11,12]: to this extent an apparent Heat Release Rate (HRR) can be defined for example by coupling two Wiebe equations identified from experimental data [11]. Similar models have been proposed for control-oriented applications [3,5,6,13,14,15] and used in several “crank-angle” commercial tools (e.g., en-DYNA[®] THEMOS[®] by Tesis Dynaware [16], Automotive Simulation Models-AMS from dSpace[®] [17], AMESim[®] Engine library [18], IFP HiL systems [19], real-time models based on Cruise[®] from AVL [20], GT-Power[®] from GTSOFT [21], WAVE[®] from Ricardo [22], etc.).

Within this scenario, a “crank-angle” model of a four-cylinders turbocharged Diesel engine based on a single-zone scheme has been set up by the authors through an improvement of the original library [23–26] developed for Mean Value engine Modelling. In-cylinder and gas exchange processes were described using a QSF approach for intake and exhaust valves and a F&E method for the cylinder. Causality has been carefully considered in order to allow for an easy use of the cylinder and valves models with the existing sub-models. After calibration and validation, the proposed model has been validated by comparing calculated results with experimental data measured by the OEM on the test bench.

2. Development of a Library for RT modeling of ICE: improvement to “crank-angle” simulation

An original simulation library was set up by the authors in the last decade for the assembling of “real-time” Mean Value Models (MVMs) of automotive engines and powertrains [4,23,24,25,26]. Based on F&E and QSF methods, and developed within Matlab[®]/Simulink[®] environment to improve portability and flexibility, the library has been organised in a hierarchical structure so that sub-model blocks can be found, picked up and assembled following the desired system layout. Dedicated procedures have been defined for the identification of each block [27].

Intake and exhaust system components are modelled as volume components (i.e., capacitances) through a F&E approach (e.g., manifolds), or as non-volume components (i.e., resistances) with a QSF methodology (e.g., valves, compressors, turbines, etc.). Following similar approaches and a coherent causality [4], new sub-models have been developed for cylinder and intake/exhaust valves and then assembled to set up a model of a four-cylinder turbocharged Diesel engine with EGR system. In the following, most important sub-models are shortly outlined.

2.1. Cylinder model

Processes which take place in the cylinder have been modelled through a F&E method based on mass and energy conservation equations defined for an open thermodynamic system and applied through a single-zone approach [9], i.e.:

- mass conservation equation:

$$\frac{dm_{cyl}}{dt} = \dot{m}_{in} - \dot{m}_{ex} \quad (1)$$

- energy conservation equation:

$$\frac{dT_{cyl}}{dt} = \frac{1}{m_{cyl} \cdot c_v} \left[\dot{m}_{in} \cdot (h_{in} - e_{cyl}) - \dot{m}_{ex} \cdot (h_{ex} - e_{cyl}) - \dot{Q}_w - \dot{Q}_c - p \cdot \frac{dV_{cyl}}{dt} \right] \quad (2)$$

- chemical species conservation equation (i-species):

$$\frac{d\tilde{x}_{cyl,i}}{dt} = \frac{\dot{m}_{in} \cdot (\tilde{x}_{in,i} - \tilde{x}_{cyl,i})}{m} + \dot{m}_{gen,i} \quad (3)$$

assuming the working fluid as a mixture of ideal gases. T_{cyl} , V_{cyl} , m_{cyl} and \tilde{x}_{cyl} are referred to thermodynamic conditions within the cylinder and can be evaluated as functions of time t (or crank angle θ) by integration of eqs.(1), (2) and (3). The term $\dot{m}_{gen,i}$ represents generation of i^{th} -species during combustion. Assuming that no other species except CO_2 and H_2O are produced, following correlation between i -species mass generation rate and Fuel Burn Rate FBR can be introduced:

$$\dot{m}_{gen,CO_2} = \frac{n_C \cdot \mu_{CO_2}}{\mu_{fuel}} \cdot FBR \quad (4)$$

$$\dot{m}_{gen,H_2O} = \frac{n_H / 2 \cdot \mu_{H_2O}}{\mu_{fuel}} \cdot FBR \quad (5)$$

$$\dot{m}_{gen,O_2} = - \frac{\left(n_C + \frac{n_H}{4} \right) \cdot \mu_{O_2}}{\mu_{fuel}} \cdot FBR \quad (6)$$

$$\dot{m}_{gen,fuel} = -FBR \quad (7)$$

with n_C and n_H fuel Carbon and Hydrogen ratio, and μ_i the molecular mass of the i^{th} -species.

Heat flux through walls is evaluated using the classical heat exchange equation:

$$\dot{Q}_w = \frac{1}{4\pi} \int_0^{4\pi} [h \cdot A_w \cdot (T_{cyl} - T_w)] d\theta \quad (8)$$

where A_w is the combustion chamber surface area, T_w is the wall temperature and h is the heat exchange coefficient evaluated following the Woschni's equation [9].

Combustion term in eq.(2) has been estimated through a Heat Release Rate (HRR) and therefore defined as:

$$\dot{Q}_c = \frac{d\theta}{dt} \cdot \frac{(Q_c)_{tot}}{\Delta\theta_c} \cdot f(\theta) = \frac{d\theta}{dt} \cdot \frac{(m_f)_{tot} \cdot LHV}{\Delta\theta_c} \cdot f(\theta) \quad (9)$$

where $f(\theta)$ can be specified following the method proposed by Watson [9,11]:

$$f(\theta) = \beta \cdot f_p(\theta) + (1 - \beta) \cdot f_d(\theta) \quad (10)$$

assuming the coefficient β as a “phase proportionality factor” [9,11]. Premixed and diffusive burning functions f_p and f_d can be identified in the following form [9,11]:

$$f_p(\theta) = C_{p1} \cdot C_{p2} \cdot \theta^{C_{p1}-1} \cdot (1 - \theta^{C_{p1}})^{C_{p2}-1} \quad (11)$$

$$f_d(\theta) = C_{d1} \cdot C_{d2} \cdot \theta^{C_{d2}-1} \cdot \exp(-C_{d1} \cdot \theta^{C_{d2}}) \quad (12)$$

Coefficients β , C_{p1} , C_{p2} , C_{d1} , C_{d2} and the ignition delay τ (evaluated through Hardenberg and Hase correlation [9]) can be estimated from experimental data as functions of engine speed and load (i.e., through experimental look-up tables).

Even if in the presented application a single injection was considered, in-cylinder model has been developed to handle multiple injections as results from actuation signals to the injectors. To this purpose a customised procedure defined by the authors to shape the HRR through a superimposition of multiple Wiebe functions, outlined in [29], will be implemented in the model in the next future.

An original algorithm has been developed to solve conservation equations in the cylinder with a suitable time step, while keeping a larger overall time step. The numerical integration of eqs.(1), (2) and (3) is carried out iteratively on a crank-angle basis keeping a definition of approx. 1deg CA, which means that the in-cylinder time step is estimated from the engine speed n and kept constant within the overall time step. This procedure has been implemented in the in-cylinder block model to catch fast dynamics of related processes without compromising real-time capabilities of the comprehensive engine model.

2.2. Intake and Exhaust Valves

Valves are considered as components where no accumulation of energy and/or mass [4,5] is allowed and therefore they are modelled through QSF techniques. Mass flow rates through intake and exhaust valves are estimated by means of the well-known flow equation in the following form:

$$\dot{m} = \frac{C_d \cdot A \cdot p_{in}}{\sqrt{R \cdot T_{in}}} \sqrt{\frac{2k}{k-1} \cdot \left[\left(\frac{p_{out}}{p_{in}} \right)^{\frac{2}{k}} - \left(\frac{p_{out}}{p_{in}} \right)^{\frac{k+1}{k}} \right]} \quad \text{when} \quad \left(\frac{p_{out}}{p_{in}} \right) \geq \left(\frac{2}{k+1} \right)^{\frac{k}{k-1}} \quad (13)$$

and

$$\dot{m} = \frac{C_d \cdot A \cdot p_{in}}{\sqrt{R \cdot T_{in}}} \sqrt{k \cdot \left(\frac{2}{k+1} \right)^{\frac{k+1}{k-1}}} \quad \text{otherwise} \quad (14)$$

Exit temperature T_{out} can be determined assuming a polytropic expansion.

Flow coefficient C_d in eqs.(13) and (14) can be determined from experimental or tabulated data and valve flow area A is a function of valve lift h (described as a function of crank angle θ through look-up tables as $h=f(\theta)$).

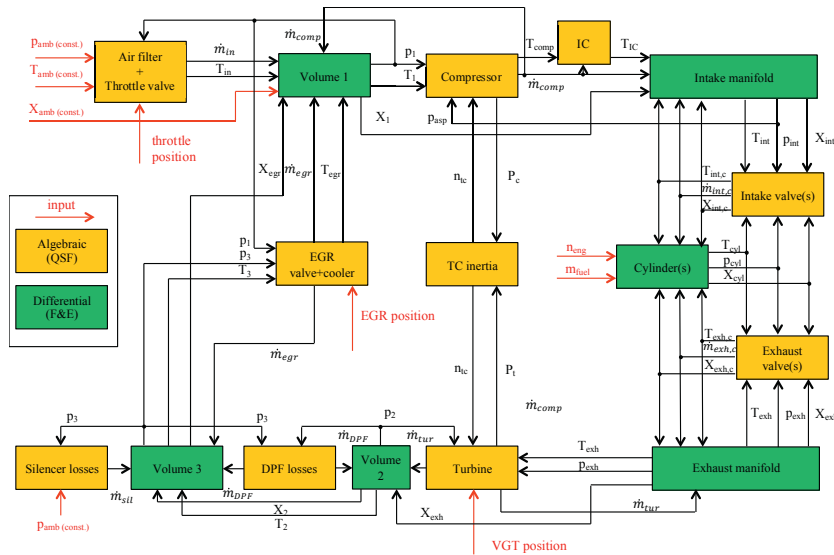


Fig. 2. Causality scheme for the Diesel engine model.

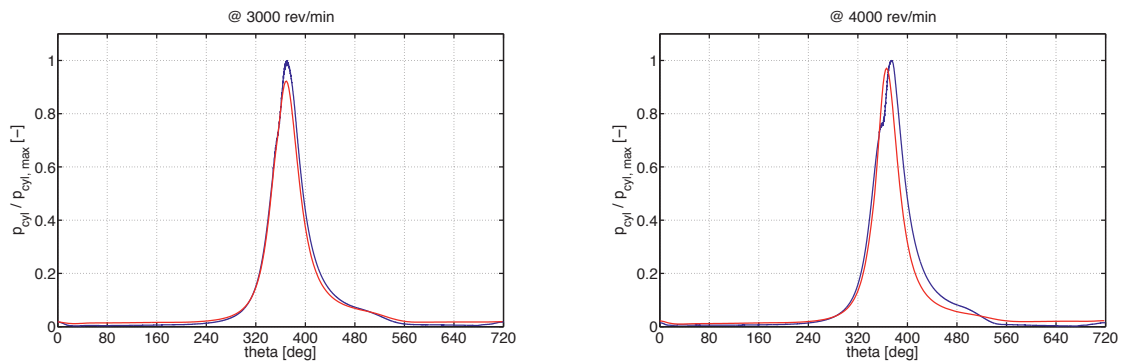


Fig. 3. In-cylinder experimental (blue) and simulated (red) pressure.

Figure 5 reports a comparison between simulated and experimental values of *bmep*. Figure 6 shows model capabilities in the estimation of state parameters (pressure and temperature) in the manifolds, and in Figure 7 the comparison between simulated and experimental mass flow rates is reported.

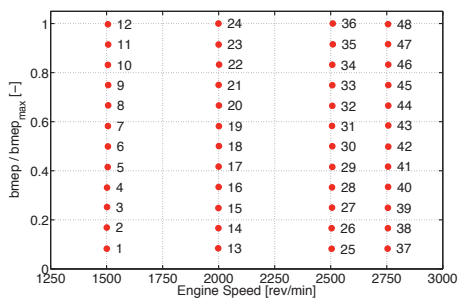
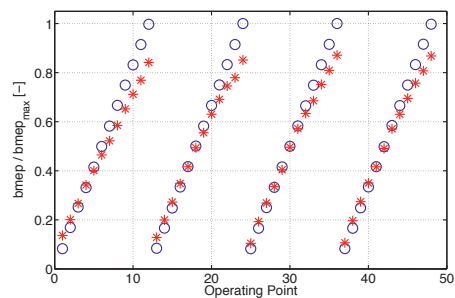


Fig. 4. Steady state operating conditions considered for model validation.

Fig. 5. Non-dimensional *bmeep* (blue-experimental, red-simulated).

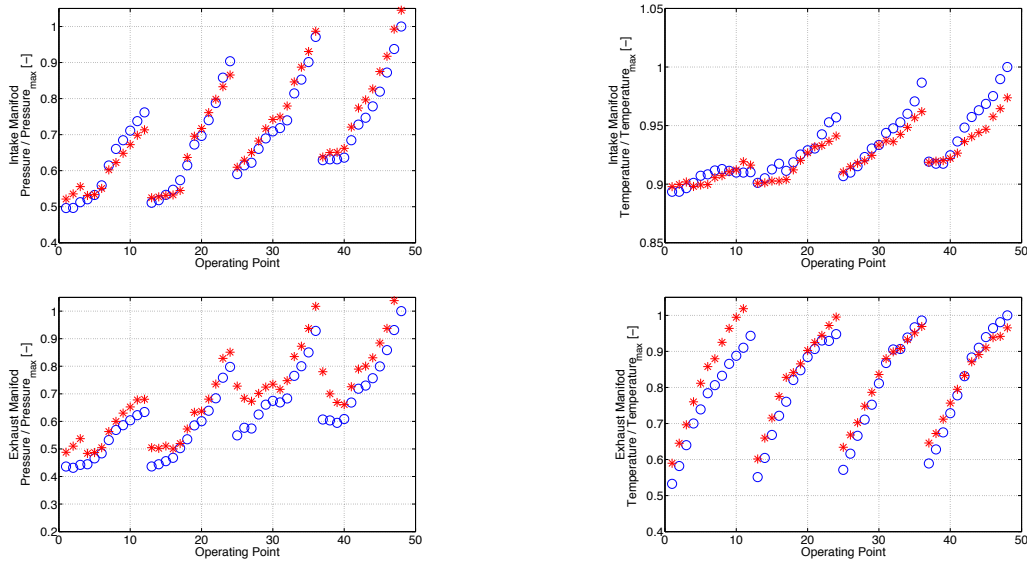


Fig. 6. Non dimensional intake & exhaust manifolds pressure and temperature (blue-experimental, red-simulated).

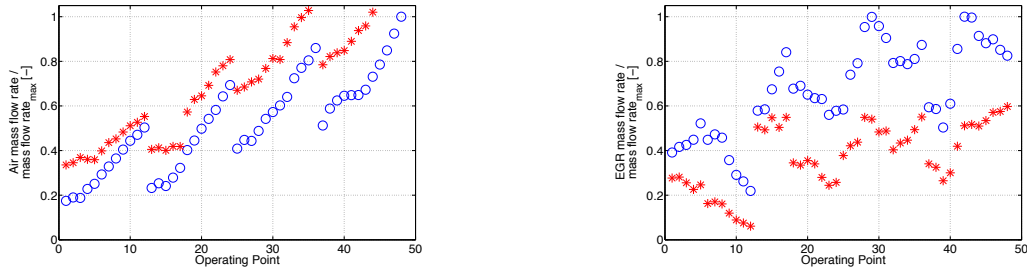


Fig. 7. Non dimensional air and EGR mass flow rate (blue-experimental, red-simulated).

The model is able to properly reproduce engine manifolds condition, with maximum errors lower than 9% (boost pressure), 3% (intake manifold temperature), 20% (exhaust manifold pressure), and 13% (exhaust manifold temperature). Even trends are correctly reproduced, however higher shifts between experimental and simulated values are apparent with reference to air and EGR mass flow rates.

An analysis of model sensitivity to input values has been developed by separately imposing $\pm 10\%$ shifts on VGT, EGR, and throttle position values (i.e., without taking account of EGR and throttle linkage). The corresponding effects on calculated outputs of the whole engine model are highlighted in Figures 8, 9 and 10.

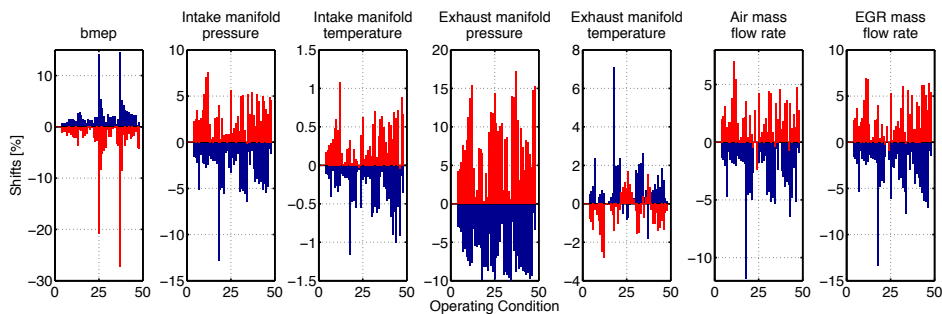


Fig. 8. Model output sensitivity vs. VGT position (blue: +10%, i.e., increasing turbine area; red: -10%, i.e., decreasing turbine area).

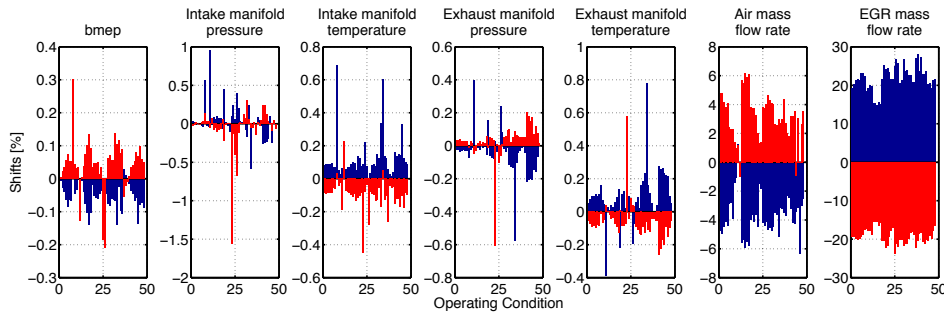


Fig. 9. Model output sensitivity vs. EGR position (blue: +10%, i.e., opening EGR valve; red: -10%, i.e., closing EGR valve).

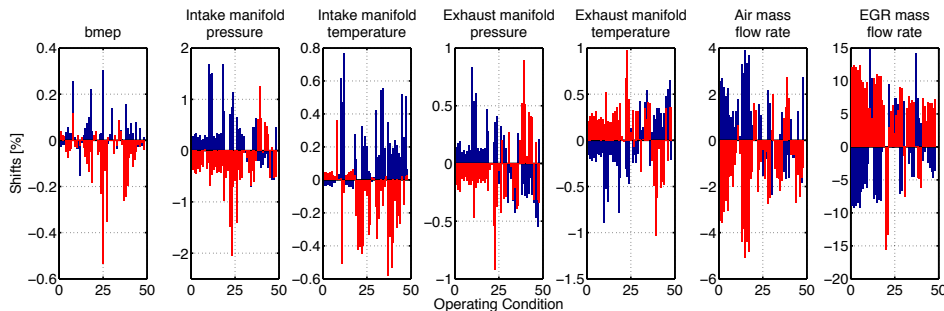


Fig.10. Model output sensitivity vs. throttle position (blue: +10%, i.e., opening throttle valve; red: -10%, i.e., closing throttle valve).

Reported results show that air path characteristics and regulation (i.e., throttle and EGR flow area and discharge coefficients) have a significant influence on mass flow rates, especially through EGR valve (up to 20%), while on the other hand they scarcely affect engine *bme_p* and manifolds pressure and temperature. VGT signal variations give rise to significant changes in both air and EGR mass flow rates and manifold parameters. These considerations may explain the higher discrepancies between simulated and experimental flow rates which can be due to an unsatisfying definition of the flow characteristics of EGR and throttle valve. The importance of a detailed calibration of sub-models of air path components, i.e., EGR valve, throttle valve, VGT characteristics, is therefore confirmed, as already pointed out by the authors in a previous investigation [27].

4. Conclusions

The model proposed in this work is an enhancement of MVMs of automotive engines previously developed by the authors. In order to comply with “crank-angle” simulations of the engine thermodynamic processes, blocks for intake and exhaust valves (with user-defined lift curves and variable actuation) and for in-cylinder processes were built up taking properly account of causality. Combustion process has been described following a classic single-zone approach based on the definition of a suitable Heat Release Rate (HRR).

An original algorithm has been developed for the in-cylinder processes simulation through the use of a variable time step in order to keep an angular step of approx. 1deg CA when varying engine speed n . The algorithm has been implemented in the in-cylinder block model to catch fast dynamics of related processes without compromising real-time capabilities of the comprehensive engine model.

The described mathematical tools have been used to build up a “crank-angle” 0-D model of a four-cylinder Diesel engine with low-pressure EGR and variable geometry turbine turbocharger which has been validated by comparing simulation results with experimental data measured by the OEM on a test bench. Even with very short computational times (comparable with real-time), the model was able to reproduce engine behaviour over a wide operating field with limited errors, with the exception of EGR mass flow rate. This result highlighted once more the

importance of a detailed characterisation of air path components in order to improve the calibration of corresponding sub-models.

With a very low computational time, and showing good capabilities in the simulation of the behavior of automotive engines with a “crank-angle” resolution, the model has been recently used in an original HiL application developed by the authors.

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