Available online at www.sciencedirect.com

ScienceDirect

Energy Procedia 45 (2014) 839 – 848

Energy Procedia

68th Conference of the Italian Thermal Machines Engineering Association, ATI2013

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

Paolo Casoli[a], Agostino Gambarotta[a,*], Nicola Pompini[a]
Ugo Caiazzo[b], Enrico Lanfranco[b], Annachiara Palmisano[b]

[a]*University of Parma, Department of Industrial Engineering, Parco Area delle Scienze 181/A, 43124 PARMA, Italy*
[b]*Fiat Research Center-CRF S.C.p.A., Strada Torino 50, 10043 ORBASSANO (TO), Italy*

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.

* Corresponding author. Tel.: +39-0521-905864; fax: +39-0521-905705.
E-mail address: agostino.gambarotta@unipr.it

doi:10.1016/j.egypro.2014.01.089

Keywords: Internal Combustion Engines, Real-Time modeling, HiL/SiL systems

Nomenclature

bmep	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:

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

- energy conservation equation:

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

- chemical species conservation equation (i-species):

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

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:

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

$$\dot{m}_{gen,H_2O} = \frac{{}^{n_H}\!/_{2} \cdot \mu_{H_2O}}{\mu_{fuel}} \cdot FBR \tag{5}$$

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

$$\dot{m}_{gen,fuel} = -FBR \tag{7}$$

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:

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

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:

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

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

$$f(\theta) = \beta \cdot f_p(\theta) + (1-\beta) \cdot f_d(\theta) \tag{10}$$

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]:

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

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

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:

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

and

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

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 $h=f(\theta)$).

2.3. Sub-model from the existing MVM Library

Components of the intake and exhaust systems other than valves and cylinders are simulated using the blocks of the original library, described in details in [4,25,26,27]. Volume components (e.g., manifolds) are described mathematically with a F&E approach, and non-volume components (e.g., valves, restrictions, pressure losses, compressors, turbines, etc.) are simulated through QSF methods.

3. Development and validation of the Engine model

The described mathematical tools can be used to build up "crank-angle" 0-D models of naturally aspirated or turbocharged engines, both SI and Diesel, taking account of causality. In addition, a specific procedure allows their identification from OEM's data and/or from experimental data measured in steady operating conditions [27]. A first application referred to an eight-cylinder naturally aspirated spark ignition engine was developed and validated as reported in [28].

A four-cylinder Diesel engine has been considered in the present work, with low-pressure EGR circuit with a three-ways valve (made up of an EGR and air throttle valves mechanically linked together to control both EGR flow and intake air pressure drop) and variable geometry turbine turbocharger, with reference to the layout reported in Figure 1. The sub-model of the assembly {intake valve}-{cylinder}-{exhaust valve} has been replicated and coupled with existing sub-models following the causality scheme of Figure 2. The structure of the model (alternating volume and non-volume blocks) avoided numerical problems and algebraic loops.

Input parameters are engine rotational speed, fuel mass flow rate, driving signals for VGT and EGR , ambient temperature and pressure. Outputs can be every single one of parameters estimated by the engine model, e.g., torque, *bmep*, effective power output, state parameters in the intake and exhaust manifolds (i.e., p, T, composition), etc.

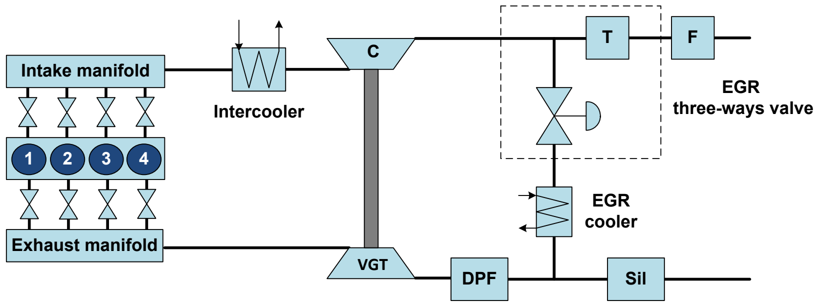

Fig. 1. Schematic of the modeled Diesel engine.

The model has been identified on the basis of steady-state experimental data from the OEM, which were used to define look-up tables and coefficients of interpolating functions through least-square methods (i.e., flow coefficients of intake/exhaust valves, pressure loss coefficients of air filter and exhaust system, etc.). Compressor and turbine models were identified on the basis of their characteristics from the Manufacturer. Parameters of Woschni's equation were defined as suggested in the literature [9]. Coefficients of eqs.(11) and (12) were estimated from experimental data and then evaluated by means of interpolation functions over engine speed and load. Examples of calculated vs. measured in-cylinder pressure diagrams are reported in Figure 3.

The algorithm developed for the integration of eqs.(1)÷(3) allows the use of a variable time step for in-cylinder processes to keep an angular step of approx. 1deg CA independently of engine speed n. The algorithm implemented in the in-cylinder block allowed running the comprehensive model on a 2GHz Quad Core PC with 8 GB RAM at a satisfying computational speed, near to real time target.

The model was validated with reference to 48 steady state operating conditions (Figure 4) comparing experimental data from OEM (different from those used for the identification phase) with results given by the model.

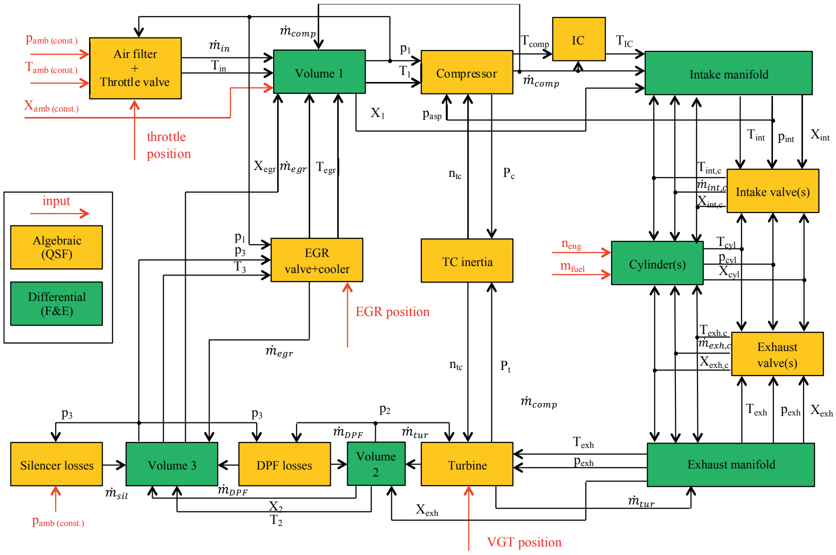

Fig. 2. Causality scheme for the Diesel engine model.

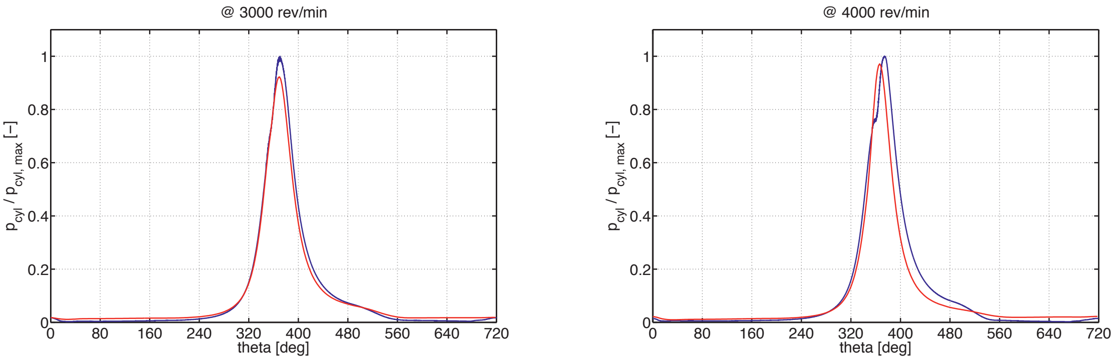

Fig. 3. In-cylinder experimental (blue) and simulated (red) pressure.

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.

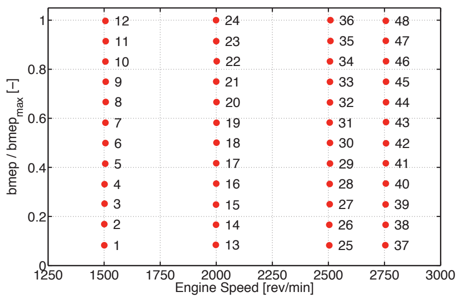

Fig.4. Steady state operating conditions considered for model validation.

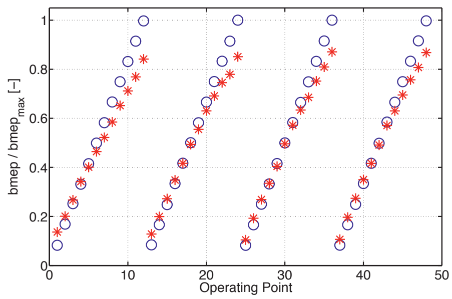

Fig. 5. Non-dimensional *bmep* (blue-experimental, red-simulated).

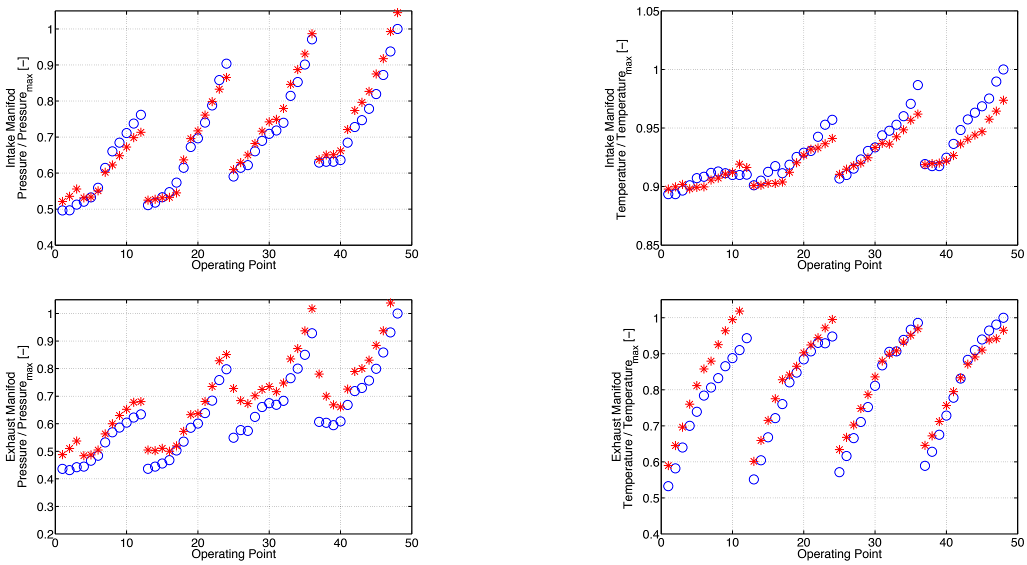

Fig. 6. Non dimensional intake & exhaust manifolds pressure and temperature (blue-experimental, red-simulated).

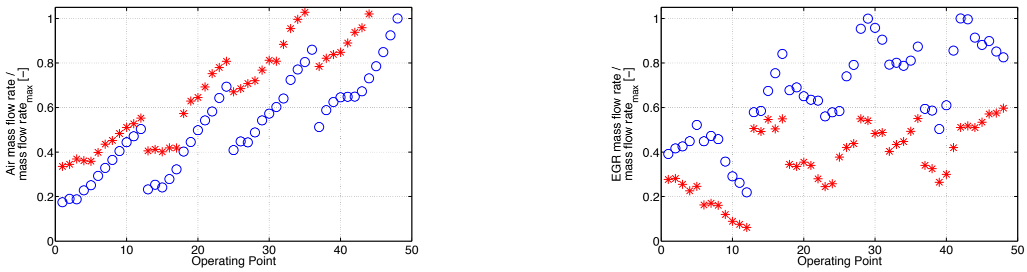

Fig. 7. Non dimensional air and EGR mass flow rate (blue-experimental, red-simulated).

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 ±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.

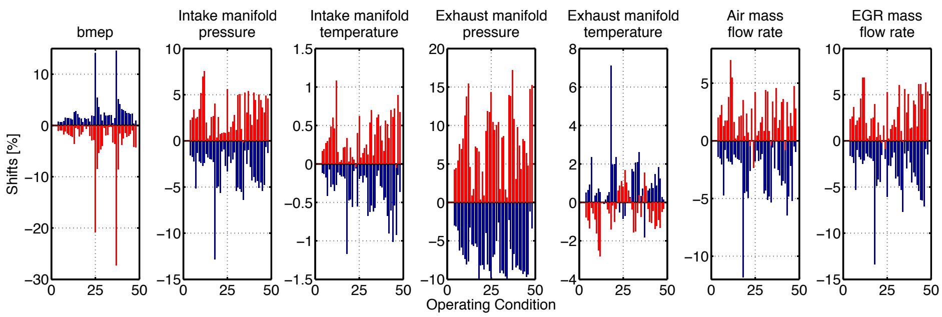

Fig. 8. Model output sensitivity vs.VGT position (blue: +10%, i.e., increasing turbine area; red: -10%, i.e., decreasing turbine area).

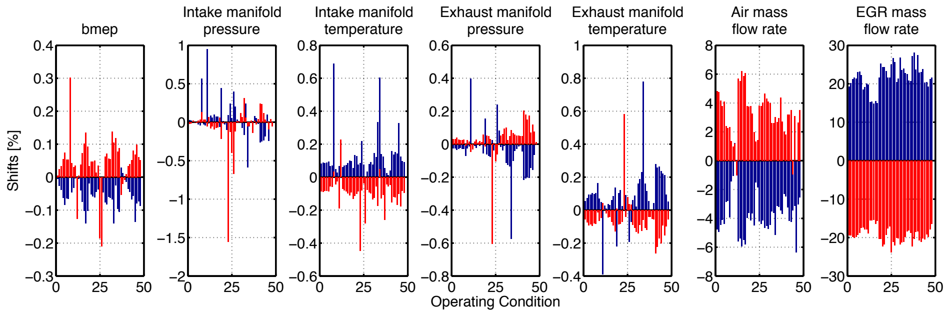

Fig. 9. Model output sensitivity vs. EGR position (blue: +10%, i.e, opening EGR valve; red: -10%, i.e., closing EGR valve).

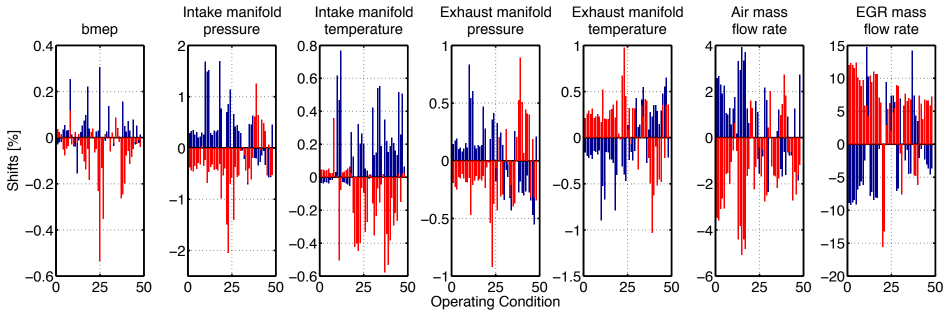

Fig.10. Model output sensitivity vs. throttle position (blue: +10%, i.e., opening throttle valve; red: -10%, i.e., closing throttle valve).

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 *bmep* 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.

References

[1] J.J.Moskwa, S.A.Munns, Z.J.Rubin, "The Development of Vehicular Powertrain System Modeling Methodologies: Philosophy and Implementation", SAE paper no.971089, 1997.

[2] L.Eriksson, "Modeling and Control of Turbocharged SI and DI Engines", Proc.E-COSM, Rencontres Scientifiques de l'IFP, Rueil-Malmaison, 2006.

[3] V.Talon, T.Tabet, S.Castric, P.Emery, "Engine Control Model-based Design with Achille library", Proc.E-COSM, Rencontres Scientifiques de l'IFP, Rueil-Malmaison, 2006.

[4] A.Gambarotta, G.Lucchetti, I.Vaja, "Real-time Modelling of Transient Operation of Turbocharged Diesel Engines", Proc. I.Mech.E., Part D: J. Automobile Engineering, Vol. 225, 2011, ISSN 0954-4070, DOI:10.1177/0954407011408943.

[5] L.Guzzella, C.H.Onder, "Introduction to Modeling and Control of Internal Combustion Engine Systems.", Springer-Verlag, Berlin, 2010.

[6] A.Dauron, "Model-based powertrain control: many uses, no abuse.", Proc.E-COSM, Rencontres Scientifiques de l'IFP, Rueil-Malmaison, 2006.

[7] A.Albrecht, O.Grondin, F.Le Berr, G.Le Solliec, "Towards a stronger simulation support for engine control design: a methodological point of view", Proc.E-COSM, Rencontres Scientifiques de l'IFP, Rueil-Malmaison, 2006.

[8] D.Anguita, P.Casoli, M.Canova, A.Gambarotta, F.Rivieccio, "A learning-machine based method for the simulation of combustion process in automotive I.C.Engines", Spring Conference of the I.C.Engines Division of the ASME, paper no.ICES2003-682, Salzburg, 2003.

[9] J.B.Heywood, "Internal Combustion Engines Fundamentals", McGraw-Hill, New York, 1988.

[10] J.I.Ramos, "Internal Combustion Engine Modelling", Hemisphere Publishing Corporation, 1989.

[11] N.Watson, M.S.Janota, "Turbocharging the internal combustion engine", John Wiley and Sons, 1982.

[12] J.H.Horlock, D.E.Winterbone, "The thermodynamics and gas dynamics of Internal Combustion Engines", vol.II, Oxford University Press, 1986.

[13] H.-Q.Liu, N.G.Chalhoub, N.Heinein, "Simulation of a Single Cylinder Diesel Engine under cold start conditions using Simulink", ASME Journal for Gas Turbines and Power, vol.123, p.117, January 2001.

[14] G.Q.Zhang, D.N.Assanis, "Manifold Gas Dynamics Modelling and its coupling with single-cylinder Engine Models using Simulink", ASME Journal for Gas Turbines and Power, vol.125, p.563, April 2003.

[15] C.D.Rakopoulos, D.C.Rakopoulos, E.G.Giakoumis, D.C.Kyritsis, "Validation and sensitivity analysis of a two zone Diesel engine model for combustion and emissions prediction", Journal of Energy Conversion and Management, vol.45, 2004, doi:10.1016/j.enconman.2003.09.012.

[16] Tesis website: http://www.tesis.de/dynaware

[17] dSpace website: http://www.dspace.de

[18] LMS website: http://www.lmsintl.com/internal-combustion-engine

[19] IFP Energies Nouvelles website: http://www.ifpenergiesnouvelles.com/developpement-industriel/moteurs/modelisation-et-simulation-moteurs-et-vehicules/simulation-moteurs-et-vehicules

[20] AVL List http://www.avl.com/

[21] Gamma Technologies website: http://www.gtisoft.com.

[22] Ricardo, "WAVE Documentation", http://www.ricardo.com/en-gb/What-we-do/Software/Products/WAVE/ and http://www.ricardo.com/en-gb/What-we-do/Software/Products/WAVE-RT/

[23] A.Gambarotta, "A control-oriented library for the simulation of automotive Diesel engines", 3rd International Conference on Control and Diagnostics in Automotive Applications, paper 01A3039, Sestri Levante, 2001.

[24] M.Canova, A.Gambarotta, "Automotive engine modelling for real time control using an object-oriented simulation library", 4th Int.Conf. on Control and Diagnostics in Automotive Applications, paper 03A2035, Sestri Levante, 2003.

[25] M.Canova, P.Fiorani, A.Gambarotta, M.Tonetti, "A real-time model of a small turbocharged Multijet Diesel engine: application and validation.", Proc.7th SAE-NA Int.Conf.on Engines for Automobiles, SAE paper no.2005-24-65, Capri, 2005.

[26] P.Fiorani, A.Gambarotta, E.Lanfranco, M.Tonetti, "A real-time model for the simulation of transient behaviour of automotive Diesel engines.", Proc.ATI/SAE Congress "The sustainable mobility challenge", SAE paper no.2006-01-3007, Perugia, 2006.

[27] A.Gambarotta, G.Lucchetti, M.Taburri, I.Vaja, "Mean Value Modeling of intake and exhaust systems of automotive engines: models identification and related errors", 10th Stuttgart International Symposium on Automotive and Engine Technologies, Stuttgart, 2010.

[28] A.Gambarotta, G.Lucchetti, "Control-oriented "crank-angle" based modeling of automotive engines", 10th International Conference on Engines for Automobiles, SAE paper no.ICE2011-24-0144, Capri, 9/2011.

[29] A.Gambarotta, "A method for Diesel combustion simulation in a "real-time" engine model for control applications", 12th Stuttgart International Symposium on Automotive and Engine Technologies, Stuttgart, 2012.