# A CONTROL-ORIENTED MODEL FOR THE SIMULATION OF TURBOCHARGED DIESEL ENGINES

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Abstract. Theoretical models play an important role in the design of engine control and diagnostic systems, allowing to reduce development time and costs. The paper describes a control-oriented theoretical model for the simulation of an automotive Diesel engine built up using a “simulation library” developed in Simulink® environment. Library blocks, defined for engine components and subsystems, were assembled together in a Mean Value Engine Model (MVEM) of a typical automotive Diesel engine, with Common Rail injection system, waste-gated turbine and EGR. Simulation results were compared with experimental data (both in steady and transient conditions) showing a good agreement. The model satisfactorily estimated engine behaviour both in steady and transient operation with very short calculation times.

Keywords: Control oriented models, Diesel engines, Engine modelling, Internal Combustion Engines, Models, Software tools.

## 1. INTRODUCTION

The use of control systems in automotive applications has become wider and wider, as a result of several reasons (Heywood, 1988). The reduction of pollutant exhaust emissions, forced by legislative regulations, the improvement of engine performance (i.e., power output and fuel consumption), required by the customer, the complexity of present automotive applications, due to the introduction of several components and systems (e.g., electronic fuel injection, intake and exhaust manifolds, exhaust gas recirculation systems, turbochargers, aftertreatment devices, antilock and stability control systems, etc.), can be quoted as the most significant ones.

Many efforts are nowadays directed to the development of suitable control systems, being aware of the importance of the optimisation of engine and powertrain management in the improvement of vehicle performance. In this field theoretical simulation models can be powerful tools to reduce development time and costs: even if they do not replace experimental investigations, they allow a shorter way from the definition of design specifications to the road tests of the control system.

Engine models can help to carry out several important tasks (Weeks and Moskwa, 1995; Moskwa, *et al.*, 1997; Stobart *et al.*, 1998). The first one is testing control algorithms in order to theoretically investigate their effects on engine behaviour, hence to optimise management strategies. In this case the engine model may be used also to evaluate sensors and actuators models, or may be employed as a subsystem in a powertrain or vehicle dynamic model. Moreover, the growing number and complexity of design requirements is now forcing a shift from map based controls towards model-based algorithms, which have the advantage of supporting both control and diagnostics in engine management systems (Weeks and Moskwa, 1995; Moskwa, *et al.*, 1997; Truscott and Porter, 1997). A more recent application of engine models is the hardware-in-the-loop (HIL) simulation (Iserman *et al.* 1998; Woerman *et al.*, 1999), which allows to develop and test electronic control systems well before on-road testing.

Even if only in the last two cases real-time simulation is needed, it is apparent that a major requirement of control-oriented models is a manageable execution time, which can be obtained by means of proper simplified approaches. This means, of course, that these models have to catch the engine behaviour as regards control-related aspects, avoiding a detailed description of the others (Heywood, 1988). To this extent, they have to combine physical -and chemical- fundamentals (generally based on the conservation laws) with an empirical description of processes (usually through quasi-steady representations based on suitable characteristic maps), often with reference to averaged values of thermodynamic parameters ("Mean Value Engine Models", MVEM).

Within this scenario, a theoretical tool has been specifically developed for the simulation of automotive turbocharged internal combustion engines (ICE). Submodels were built up for engine components in a MATLAB/Simulink® environment, and a control-oriented model of a Diesel engine was assembled and tested in both steady and transient conditions.

## 2. THE SIMULATION MODEL

The theoretical tool for the simulation of present automotive I.C. Engines was developed taking account of several requirements, i.e., limited computation time, transferability and user-friendliness.

Starting from a previous steady-state Mean Value Engine Model (MVEM) developed in MATLAB® (Gambarotta and Capobianco, 1998) a specific Simulink® library has been built up in order to simulate typical automotive turbocharged Diesel engines (fig.1). To improve both portability and flexibility, the library is organised in a hierarchical structure which makes submodel blocks easier to be found, picked up and assembled by the user. They were defined within Simulink®3.0 environment to simulate components and subsystems of present automotive turbocharged engines, possibly fitted with exhaust gas recirculation system (EGR) and variable geometry turbine (VGT).

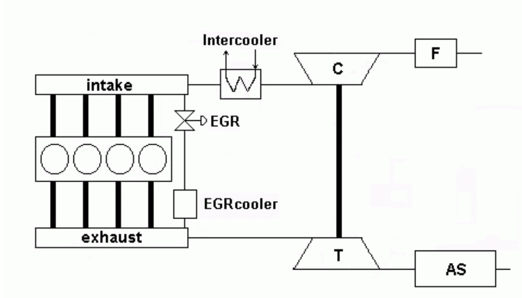


Fig.1. Scheme of a typical automotive Diesel engine.

A quasi-steady approach was followed, with reference to cycle-averaged values of thermodynamic parameters, to reproduce the behaviour of engine components (compressor, turbine, intercooler, EGR system, etc.), while a filling-and-emptying technique was used for volumes (e.g., intake and exhaust manifolds). Since heat transfer has a significant effect on gas temperature at the outlet of exhaust manifold (which has a strong influence on the available enthalpy at the turbine inlet and on the behaviour of aftertreatment systems), a specific submodel has been defined to evaluate heat losses through the walls of exhaust manifold: conductive, convective and radiative heat transfer are considered through empirical relationships (Heywood, 1998) and flow unsteadiness is considered by means of a suitable "convective augmentation factor" (CAF, as described in Gambarotta, 1998).

Combustion processes were described following a single-zone approach, based on the prediction of a proper Fuel Burning Rate (FBR) (Heywood, 1988). Blocks can be easily linked together through mass and energy conservation equations to describe the whole engine system.

To enhance library flexibility, allowing an easy application not only to different set-ups, but also to different engines, interactive procedures have been developed to build components maps starting from available data.

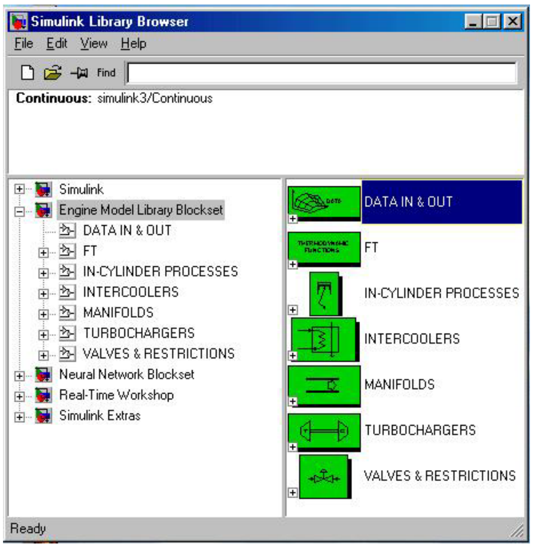


Fig.2. The root level of the Engine Model Library.

The "root" level of the library is shown in fig.2 and can be opened from Simulink® main window. Different sublevels can be reached, i.e., data input and output, thermodynamic functions, in-cylinder processes, intercoolers, manifolds, turbochargers, valves and restrictions. In each sublevel several simulation blocks are available, as described in (Gambarotta, 2001). Suitable functions are used and implemented in specific blocks to define the thermodynamic properties of the working fluid (considered as a mixture of perfect gases).

*2.1 In-cylinder processes.*

The simulation of in-cylinder processes was limited to the combustion, while the effects of valve geometry and timing will be considered in further steps of the work. The combustion in Diesel engines is a very complex process, and in many applications simplified models were proposed and used in the considered literature (Heywood, 1988; Watson and Janota, 1982; Ramos, 1989). With particular reference to control-oriented applications, requirements in terms of calculation time usually lead to very simple approaches (examples may be found in Moskwa and Weeks, 1995; Truscott and Porter, 1997; Watson and Marzouk,1977).
Within the presented work, combustion process is simulated following a single-zone method with the introduction of an apparent fuel burning rate (FBR). On the basis of the method proposed by Watson (Watson and Janota, 1982 ), combustion is modelled as an homogeneous process with a uniformly distributed heat release (related to the apparent FBR) given in the following form:

$$\frac{dm_f / d\theta}{(m_f)_{tot}} = \beta \cdot f_p(\theta) + (1-\beta) \cdot f_d(\theta) \quad (1)$$

where $m_f$ is the fuel mass, $(m_f)_{tot}$ is the total injected fuel mass, $f_p$ and $f_d$ represent the premixed and diffusive burning functions respectively, $\theta$ is the crank angle (nondimensionalized with reference to the total combustion duration), and the coefficient $\beta$ is a "mode of burning factor", which expresses the cumulative fuel burnt during premixed combustion as a fraction of the total fuel injected. Burning functions may be defined through the following expressions (Watson and Janota, 1982):

$$f_p(\theta) = C_{p1} \cdot C_{p2} \cdot \theta^{C_{p1}-1} \cdot (1-\theta^{C_{p1}})^{C_{p2}-1} \quad (2)$$

$$f_d(\theta) = C_{d1} \cdot C_{d2} \cdot \theta^{C_{d2}-1} \cdot exp(-C_{d1} \cdot \theta^{C_{d2}}) \quad (3)$$

To evaluate the apparent FBR through equations (1)÷(3) in any given engine operating condition, the values of five parameters are needed (i.e., $C_{p1}$, $C_{p2}$, $C_{d1}$, $C_{d2}$, and $\beta$) (Watson and Janota, 1982): to this purpose, different methods were used.
A specific MATLAB® procedure has been developed to evaluate the five coefficients: starting from in-cylinder pressure measurements, apparent FBR is calculated from experimental data and then the best-fitting values of $C_{p1}$, $C_{p2}$, $C_{d1}$, $C_{d2}$, and $\beta$ are calculated.
Simple correlations were firstly used to extrapolate the FBR coefficients: linear functions (obtained through a least-square method) were used to estimate of $C_{p1}$, $C_{p2}$, $C_{d1}$, $C_{d2}$, and $\beta$ for any engine operating condition (defined giving the mass of trapped gas, EGR fraction, mass of injected fuel and intake manifold thermodynamic conditions). However, to achieve a better approximation, different non-linear extrapolating methods have been considered. In co-operation with the Department of Biophysical and Electronic Engineering (DIBE) of the University of Genoa, the application of Learning Machines (LM) to the estimation of FBR coefficients was studied. The first results obtained by means of Neural Networks (NNW) and Support Vector Machines (SVM) seem to be promising: these methods are now being used and tested in future steps of the research.
After the apparent FBR, the in-cylinder simulation block (fig.3) estimates the thermodynamic conditions by solving the energy equation at defined crank-angle steps.

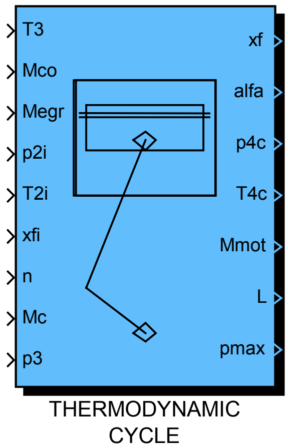


Fig.3. In-cylinder processes simulation block.

Heat losses through cylinder walls are calculated according to Woschni's correlation (Heywood,1988). Scavenge flow is neglected and the volumetric efficiency, evaluated through an empirical relationship (Gambarotta and Capobianco, 1998)), yields the mass of trapped gas. Gas composition is determined at each step considering both recirculated and residual exhaust gases, and assuming that apparently injected fuel (as determined from the FBR) burns instantaneously and completely to form $CO_2$ and $H_2O$.

Block input data are pressure and temperature in the intake manifold, air and EGR mass flows, injected fuel quantity and engine rotational speed. Pressure and temperature in the exhaust manifold are required to estimate residual gas fraction. Following the calculation of thermodynamic process on the basis of FBR, block output allows to export several parameters, e.g., pressure, temperature and gas molar fractions at the end of expansion, overall air/fuel ratio, indicated work and peak pressure.

The thermodynamic conditions of exhaust gases entering the manifold have significant effects on heat transfer in the exhaust system and on aftertreatment devices. However, the estimation of gas temperatures and pressures in the exhaust manifold is very difficult, since expansion and blow-down are very complex processes: in the present work it has been modelled following the procedure suggested in (Zinner, 1978), which has been coded in a dedicated library block.

In order make calculation time shorter, a simpler and faster model based on the application of LM has been tested to describe the in-cylinder processes. Starting from experimental data, the SVM developed at DIBE was trained to evaluate exhaust conditions and bmep directly from engine operating parameters: the calculation procedure has been implemented in a simulation block (fig.4). The first results worked out to be promising, showing very short computational time: this model will be enhanced and widely applied in the future developments of the research programme.

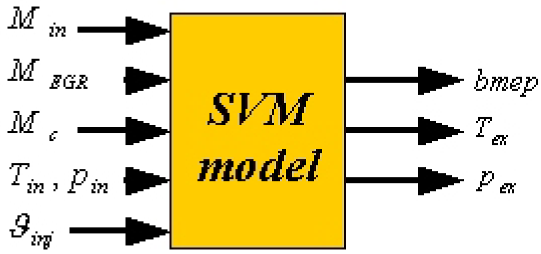


Fig.4. The simplified combustion block.

*2.2 Turbochargers*

The exhaust turbocharger is simulated by means of quasi-steady techniques for compressor and turbine, taking account of its dynamics. Blocks were built up to simulate components on the basis of their steady behaviour: since both compressor and turbine were considered as they were not state-determined systems, block input and output variables are state parameters and mass and energy flows, respectively (fig.5).

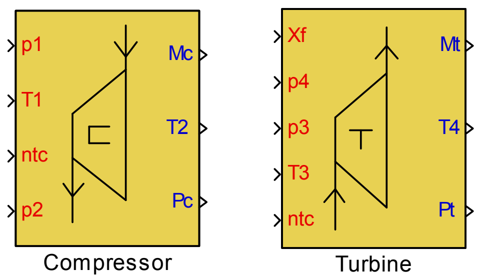


Fig.5. Compressor and turbine blocks.

Compressor performance are described through two maps which relate pressure ratio $\beta$ with reduced mass flow rate $M_{c,r}$ and reduced power $P_{c,r}$ respectively at constant values of reduced rotational speed $n_{tc,r}$ , in the following form:

$$M_{c,r}=f_{c1}(\beta,\ n_{tc,r}) \quad \text{and} \quad P_{c,r}=f_{c2}(M_{c,r},\ n_{tc,r}) \quad .(4)$$

Similar maps are used for the turbine, in terms of mass flow parameter $M_{t,nd}$ and isentropic efficiency $\eta_t$ respectively, vs. expansion ratio $\varepsilon_t$ and rotational speed parameter $n_{t,nd}$ , and to blade speed ratio $u/c_s$ (Watson and Janota, 1982). Therefore, maps were defined as follows:

$$M_{t,nd}=f_{t1}(\varepsilon_t, n_{tc,nd}) \text{ and } \eta_t=f_{t2}(u/c_s). \quad (5)$$

It should be noted that suitable devices have to be used to control turbine mass flow rate (i.e., variable geometry (VG) devices or waste-gate (WG) valve). To take account of the related control parameter $A$ (e.g., waste-gate opening degree, or VG position (Capobianco and Gambarotta, 1992)), multiple maps can be defined at $A$=const. and between them a linear or quadratic interpolation can be used. A different approach was introduced for waste-gated turbines, modelling them as two nozzles in parallel (even if care is required when dealing with small automotive turbochargers, where significant interactions may occur between turbine and waste-gate flows (Capobianco *et al.*, 1990)).

It should be noted that compressor and turbine maps have to be usually derived from experimental data (Capobianco and Gambarotta,1992) or from turbocharger supplier's data: however, this procedure gives rise to several problems. The first one is due to the limited domain of available data: this requires not only a proper interpolation inside the known region, but also a meaningful extrapolation outside of it, which is needed in the simulation of transients.

Therefore, a specific procedure has been developed in MATLAB® for map generation and handling. In order to enhance user-friendliness and flexibility, a graphical user interface has been built up for the purpose through several MATLAB® windows. In order to limit simulation time, map pre-processing is developed off-line (i.e., outside compressor and turbine blocks) and is based on sequential steps.

First, the user can interactively extend the maps beyond the range of available data by adding proper "boundary" operating conditions. Then, splines are used to represent constant speed lines, overcoming the constraints imposed by the use of a single polynomial: following a piecewise polynomial (pp) approximation, splines coefficients are evaluated through a least-square algorithm. Curves are represented on characteristic planes, allowing the user to interactively modify and optimise their shape.

At the second step further constant-speed lines are generated inside the normal operating range: a quadratic interpolation procedure is used starting from the closer available curve.

The third step regards the definition of characteristic curves at lower and higher rotational speeds, where no data are usually available. A theoretical approach has been followed, based upon relationships between pressure and enthalpy changes and rotor speed, and introducing a few simplifications. For example, the following equation can be written for the compressor:

$$\beta \cong \left[1 + C \cdot n_{tc,r}^2\right]^{\frac{k}{k-1}} \quad (6)$$

which for similar operating points, yields:

$$\frac{\beta_{i+1}}{\beta_i} = \left[\frac{1 + C \cdot n_{tcr,i+1}^2}{1 + C \cdot n_{tcr,i}^2}\right]^{\frac{k}{k-1}} \quad (7)$$

Moreover, among operating conditions belonging to the same characteristics number, the reduced mass flow rate $M_{c,r}$ can be considered proportional to the reduced rotational speed $n_{tc,r}$ and reduced power $P_{c,r}$ proportional to the third power of reduced rotational speed $n_{tc,r}$, i.e.,

$$M_{c,r} \propto n_{tc,r} \quad \text{and} \quad P_{c,r} \propto (n_{tc,r})^3 \quad (8)$$

After equations (6)÷(9), constant speed lines can be defined for out-of-data range: this is particularly important in the simulation of engine transients, since the turbocharger rotational speed may easily fall to very low values (which are not included in the range of available data).

Finally, the extended characteristic maps (fig.6) are transferred as data matrices to simulation submodels through MATLAB® workspace, and then embedded in the compressor block (fig.5) by means of Simulink® look-up tables.

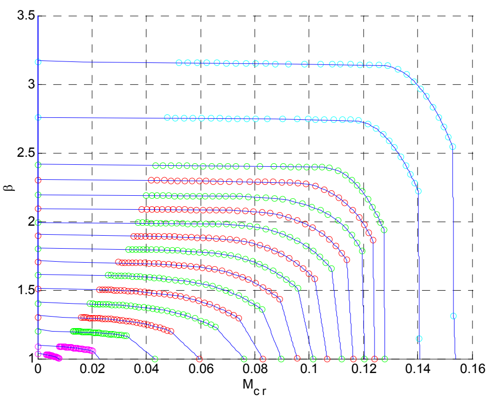


Fig.6. Extended compressor mass flow constant speed lines (Garrett GT15).

The same procedure is used for the turbine, with the definition of constant speed curves in the form of eq.(5). Starting from available data, the first step regards the introduction of zero mass flow rate conditions for different rotational speed: the corresponding value of expansion ratio can be theoretically estimated by computing the effect of centrifugal forces acting on the fluid in the turbine rotor. The energy conservation equation yields:

$$\varepsilon_{\dot{m}=0} = \left[1 + C \cdot n_{tc,nd}^2\right]^{\frac{k}{1-k}} \tag{9}$$

At the second step the user can interactively extend the maps outside of the range of available data by adding the choked flow conditions, i.e. defining the upper limit of the expansion ratio (above which the flow is choked) and extrapolating along constant speed lines.

Finally, further constant-speed curves are generated from the Euler's equation (and considering an isentropic expansion) the following expression can be written:

$$\varepsilon = \left[1 + C \cdot n_{tc,nd}^2\right]^{\frac{k}{1-k}} \tag{10}$$

An application of the theory of similarity yields:

$$\frac{\varepsilon_{i+1}}{\varepsilon_i} = \left[\frac{1 + C \cdot n_{tc,nd,i+1}^2}{1 + C \cdot n_{tc,nd,i}^2}\right]^{\frac{k}{1-k}} \tag{11}$$

while outside of the choked region the mass flow rate can be considered proportional to the rotational speed, i.e.:

$$M_{t,nd} \propto n_{tc,nd} \tag{12}$$

Turbine isentropic efficiency is estimated as a function of the blade speed ratio (Watson and Janota, 1982) (see eq.(5)). It should be noted that usually turbocharger losses (i.e., mechanical losses in the bearings and friction losses on the back face of the rotor) are included in turbine efficiency (since it is very difficult in practice to split them (Watson and Janota, 1982)): therefore the actual enthalpy change (and the actual temperature change) across the turbine can not be determined on the basis of its efficiency. The problem has to be carefully considered in the evaluation of turbine outlet temperature, which has a remarkable influence on the efficiency of possible exhaust aftertreatment systems. Previous experience (Capobianco and Gambarotta, 1992) suggested the use of a theoretical method developed by the author to evaluate turbocharger friction losses: from the knowledge of turbocharger operating conditions, power losses are calculated by the turbine submodel to evaluate turbine outlet temperature.

Taking account of equations (9)÷(12), constant speed curves can be defined for a wide operating range following the same procedure described for the compressor. Maps, extended through the pre-processing utility (fig.7) are finally transferred as data matrices to simulation submodels through MATLAB® workspace, and then embedded in turbine block (fig.5) by means of look-up tables of Simulink® 3.

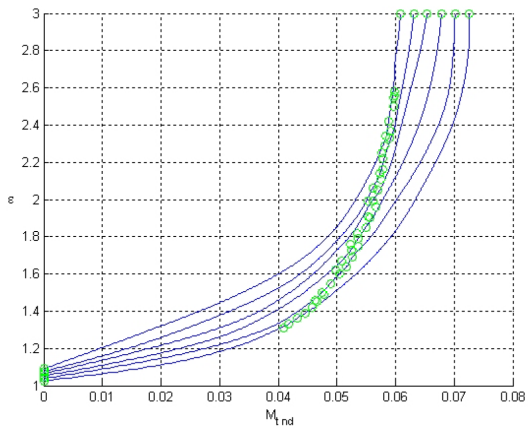


Fig.7. Extended turbine mass flow constant speed lines (Garrett GT15).

No correction factors are used to account for pressure pulsation effects on turbine performance. Previous investigations (Capobianco and Gambarotta, 1992; Capobianco *et al.*, 1990) showed that unsteady flow parameters (e.g., frequency and pulse amplitude) have a complex influence on turbine mass flow rate and power, and shifting from steady flow values may vary consistently with operating conditions. It was observed, however, that in the case of small automotive turbochargers these variations may result lower than 10 per cent (Capobianco and Gambarotta, 1992). As a consequence, the use of correction factors (or of a "corrected expansion ratio") seems to be possible only after a wide experimental investigation, which allows to define proper values with reference to actual turbine operating conditions.

Turbocharger dynamics is described by means of Newton's second law (Watson and Janota, 1982), in the form:

$$(P_t - P_c) = J_{tc} \cdot \omega_{tc} \cdot \frac{d\omega_{tc}}{dt} \tag{13}$$

where turbocharger power losses are included in the turbine power term $P_t$. By integrating eq.(13), turbocharger speed and acceleration can be determined, taking account of rotor inertia $J_{tc}$.

The described procedure can be easily applied to different turbochargers: starting with compressor and turbine available data, proper splines can be generated to describe constant speed curves, and their coefficients transferred to corresponding submodels.

## 3. APPLICATION TO A TURBOCHARGED AUTOMOTIVE DIESEL ENGINE

The simulation library has been used to build a Simulink® model of a typical automotive DI Diesel engine. For the test case, an Alfa Romeo 1.9 JTD, 1.910 litres displacement engine, with Common Rail, EGR and waste-gated turbocharger has been considered. A flow chart for this model is reported in fig. 8.

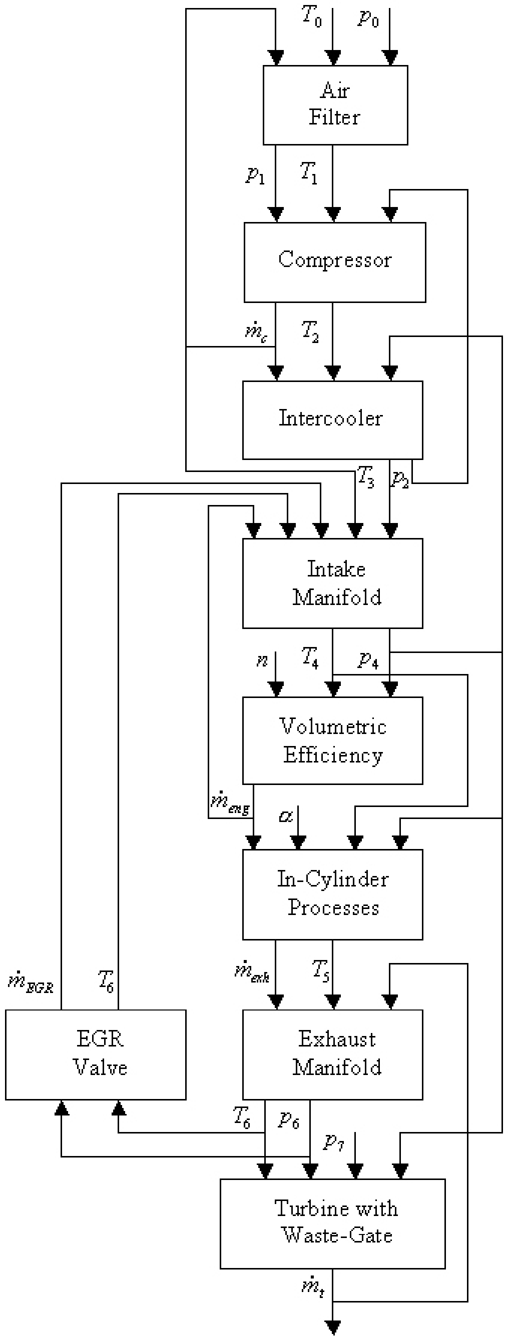


Fig.8. Flow chart of the engine model.

The steady state functions both for compressor and turbine were defined starting from manufacturer's data, and further processed with the previously described technique. Test results from the complete engine were employed to identify the quasi-steady flow parameters of the intercooler and the air filter. The volumetric efficiency is evaluated in the model by means of linear interpolations among data from engine dynamometer, assuming intake manifold density as a reference. A wide data base was considered with reference to in-cylinder pressure diagrams, that consists of constants in eq.(1)÷(3), in order to provide the input to the combustion block. Experimental data, taken both on the DIMSET test bench (Zamboni, 2001) and on the facilities of Fiat Research Center of Orbassano, were used for training the simulation procedures defined for the combustion process.

The model was firstly validated with reference to steady engine operation. To this extent 55 conditions were considered, covering the whole operating range (fig.9).

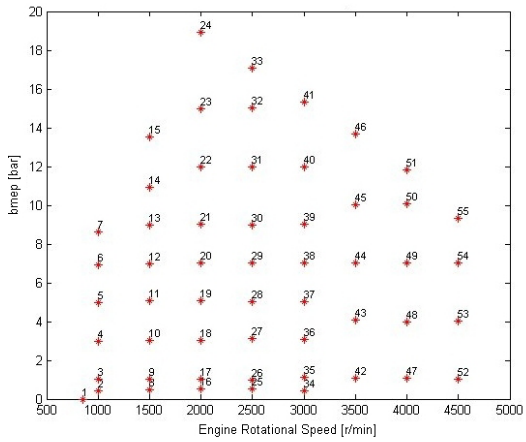


Fig.9. Engine steady operating conditions.

The model was run with constant values of the input parameters (referred to each operating condition): simulation was started and proceeded until the output reached a steady solution. Then the calculated results were compared with test data. Several examples are reported in figs.10,11,12,13, where exhaust and intake manifold pressure and temperature show a satisfactory agreement.

Although seldom mismatches could be noted, the model works out to be able to reproduce characteristic trends of the concerned parameters, i.e., to describe the effects of the independent variables on the behaviour of the engine.

Moreover, the introduction of simplified procedures, which on one hand leads to significant approximations, on the other allows for as short calculation times as they are requested by control and diagnostic applications.

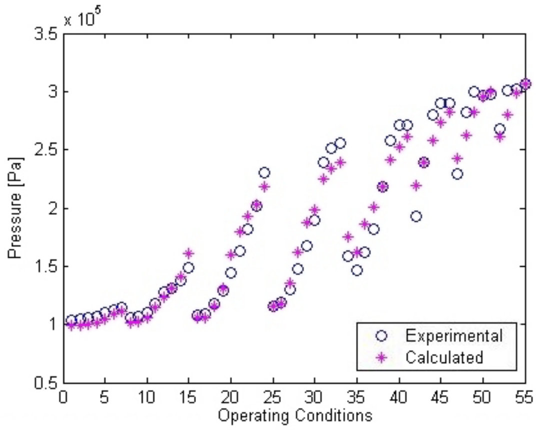


Fig.10. Exhaust manifold pressure.

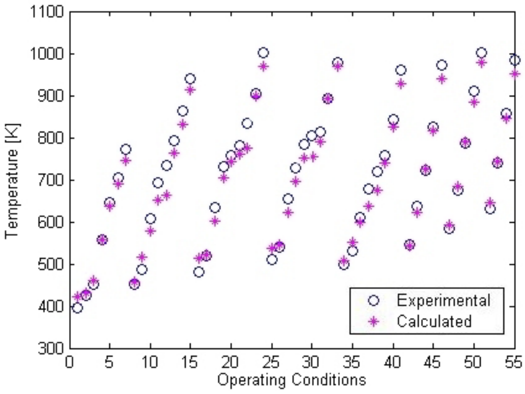


Fig.11. Exhaust gas temperature.

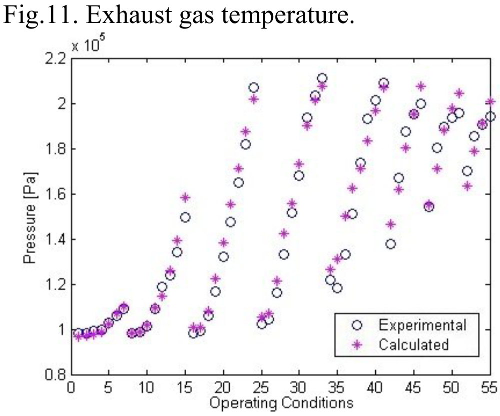


Fig.12. Intake manifold pressure.

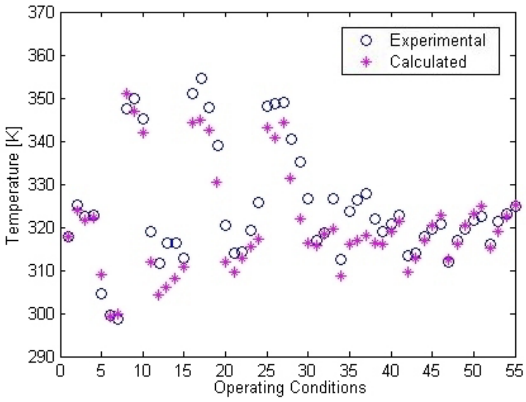


Fig.13. Intake manifold temperature.

The model was then tested in transient conditions to verify its capability to simulate engine dynamics. Two sets of experimental data were used, both from Fiat Research Centre facilities.

A sample of input functions is reported in fig.14 (engine speed, fuel mass flow and EGR opening ratio).

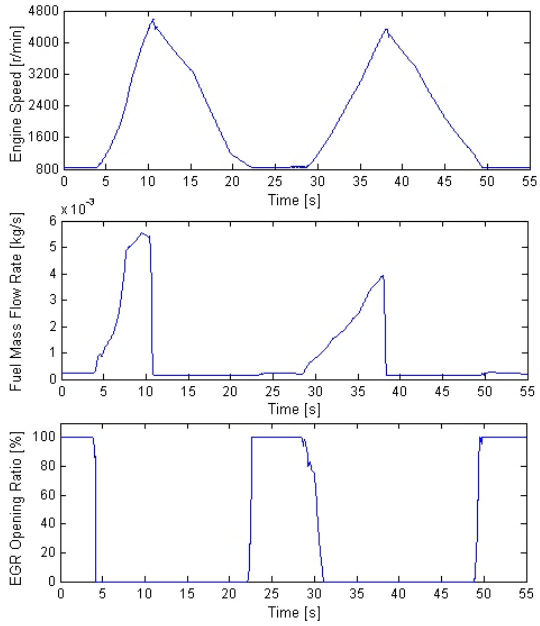


Fig.14. Input functions used for transient simulation.

Comparing the results of the simulation with experimental data allowed to highlight the good prediction capabilities of the model.

A few output data are reported in figs.15,16,17,18, showing air mass flow rate, turbine inlet pressure, boost pressure and turbocharger speed together with test results. A good agreement is apparent between the two families of data, proving that the model is able to predict engine behaviour also in transient conditions. Simulation time was always very short even on a 800MHz PC, resulting fairly close to real-time operation.

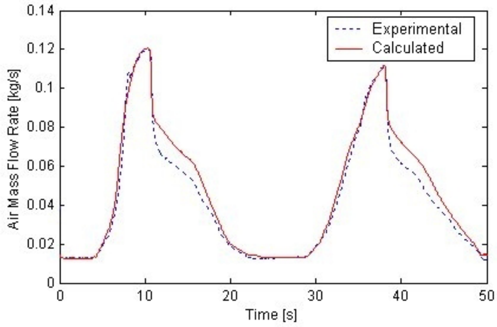


Fig.15. Intake air mass flow rate in transient conditions.

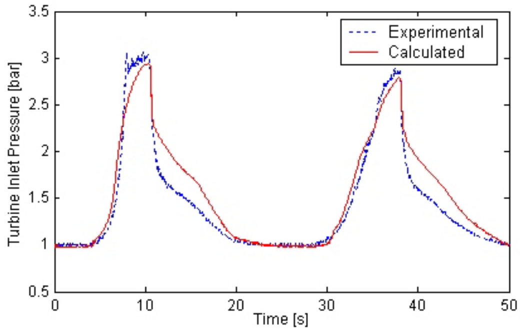

Fig.16. Turbine inlet pressure in transient conditions.

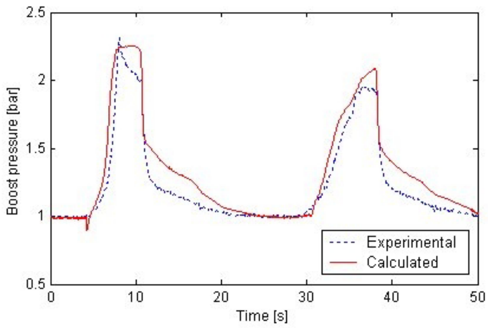

Fig.17. Boost pressure in transient conditions.

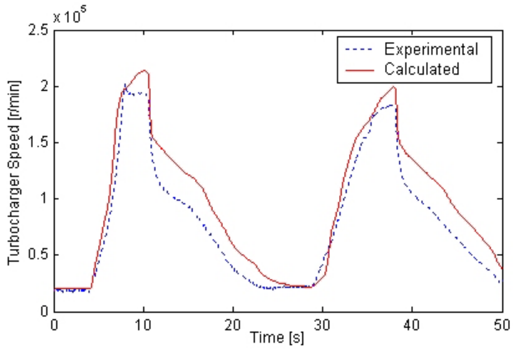

Fig.18. Turbocharger speed in transient conditions.

## 4. CONCLUSIONS

A library has been developed in Simulink® for the simulation of turbocharged automotive engines. Since it was intended for control and diagnostic applications, it has been constructed in a hierarchical structure. Several blocks were defined for the simulation of engine components and subsystems: in order to achieve a very short calculation time, quasi-steady and filling-and-emptying approaches were used, while in-cylinder processes were described either with a single-zone model or with a simplified method.
The library was employed to build up a simulation model of a typical automotive Diesel engine, thereby being validated in terms of accuracy, computational time and portability. The test was carried out both in steady and transient operation, comparing calculated results with data acquired on test benches. Good agreements were obtained with very short calculation times in every tested condition, and even if developed following simplified approaches, the model satisfactorily estimated engine behaviour both in steady and transient operation.
Flexibility and user-friendliness of the library were also verified, since it allows for easy applications to different engines and to different set-ups of intake and exhaust systems.
Further developments are now in progress in order to take account of the effects of multiple injections (as allowed by CR systems), of variable valve actuation, and of aftertreatment devices. However the features of the proposed library proved that it can be a flexible environment for several applications in the field of modern engine control and diagnostics, spanning from HIL testing to the development of model-based management strategies.

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## NOMENCLATURE

| | |
|---|---|
| $A$ | position of the turbine regulating system, parameter |
| $B$ | parameter |
| $C$ | parameter, compressor |
| $J$ | rotor inertia |
| $M$ | mass flow rate |
| $M_r$ | reduced mass flow $= M \cdot p_0 \cdot \sqrt{T} / p \cdot \sqrt{T_0}$ |
| $M_{nd}$ | mass flow rate parameter $= M \cdot \sqrt{T} / p$ |
| $P$ | power |
| $P_r$ | reduced power $= P \cdot p_0 \cdot \sqrt{T_0} / p \cdot \sqrt{T}$ |
| $P_{nd}$ | power parameter $= P / p \cdot \sqrt{T}$ |
| $T$ | temperature, turbine |
| $X$ | volumetric concentration |
| $a,b,c$ | premixed fuel fraction parameters |
| $c$ | absolute velocity |
| $f$ | function |
| $h$ | specific enthalpy |
| $k$ | isentropic coefficient |
| $n$ | rotational speed (in [rpm]) |
| $n_r$ | reduced rotational speed $= n \cdot \sqrt{T_0} / \sqrt{T}$ |
| $n_{nd}$ | speed factor $= n / \sqrt{T}$ |
| $p$ | pressure |
| $t$ | time |
| $u$ | blade speed |
| $\beta$ | compression ratio, premixed fuel fraction |
| $\varepsilon$ | expansion ratio, effectiveness |
| $\theta$ | non-dimensional crank angle |
| $\rho$ | density |
| $\omega$ | rotational speed (in [rad/s]) |

Subscripts

| | |
|---|---|
| *EGR* | exhaust gas recirculation |
| *c* | compressor |
| *d* | diffusive |
| *eng* | engine |
| *ex* | exhaust |
| *f* | fuel, friction |
| *in* | inlet, intake |
| *m* | mean value |
| *p* | premixed, constant pressure |
| *s* | isentropic |
| *t* | turbine |
| *tc* | turbocharger |
| *v* | constant volume |

Acronyms

| | |
|---|---|
| AS | aftertreatment system |
| CR | common rail injection system |
| CRF | Fiat Research Center |
| DI | direct injection |
| DIMSET | Department of Energetic Systems and Transportation |
| DIBE | Department of Biophysical and electronic engineering |
| EGR | exhaust gas recirculation system |
| F | filter |
| FBR | fuel burning rate |
| HIL | hardware-in-the-loop |
| ICE | internan combustion engine |
| LM | learning machine |
| MVEM | mean value engine model |
| NNW | neural network |
| SVM | support vector machine |
| bmep | brake mean effective pressure |