2014-01-1078
Published 04/01/2014
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doi:10.4271/2014-01-1078
saeeng.saejournals.org

A New Model for Matching Advanced Boosting Systems to Automotive Diesel Engines

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ABSTRACT

Boosting technologies have been key enablers for automotive engines development through downsizing and downspeeding. In this situation, numerous multistage boosting systems have appeared in the last decade. The complexity arising from multistage architectures requires an efficient matching methodology to obtain the best overall powertrain performance.

The paper presents a model aimed to choose the best 2-stage boosting system architecture able to meet required criteria on boosting pressure, EGR ratios for both short and long route loops while respecting the engine thermo-mechanical limits such as in-cylinder pressure, compressor outlet temperature and exhaust manifold temperature. The model includes filling-and-emptying 0D elements together with mean value. The engine model is set in a way that, for given requirements and boosting system layout, calculates in seconds if the requirements will be achieved and the position of variable geometry, waste-gate, EGR and by-pass valves. The model is thus inversed thanks to a new representation of turbine maps that converts the classical iterative matching procedure in straight forward. The model can be also used in a predictive manner to calculate the engine transient response.

The model has been calibrated to 3 different turbocharged diesel engines. The model gives good results provided that wave effects are not important. This is the case of compact exhaust manifolds, typically used in turbocharged diesel engines, below 3500 rpm. Tuned intake air lines can be taken into account through a tuning parameter affecting boosting pressure.

An example is given in the paper for the matching procedure in a 2-stage, double route EGR, including steady and transient results.

CITATION: Galindo, J., Lujan, J., Climent, H., Guardiola, C. et al., "A New Model for Matching Advanced Boosting Systems to Automotive Diesel Engines," *SAE Int. J. Engines* 7(1):2014, doi:10.4271/2014-01-1078.

INTRODUCTION

The complexity arising from multistage architectures requires an efficient matching methodology to obtain the best overall powertrain performance. With a large number of components interacting between each other's, optimization processes are more and more exigent and only a correct combination of experimental techniques and simulation tools can lead to good results under suitable development time. To complete these tools, the second part of the methodology is based on a whole engine model able to correctly match the boosting system in steady-state conditions and to reproduce its performance under transient operations.

Different types of models are present in engine development and can be classified according to their complexity and computational cost. Each type has its pros and cons and their selection must be adapted to the specific application with a right balance between physical description, predictive capabilities, computational speed and accuracy [1][2]. There is lot of information in the literature about these different modeling approaches. So only a brief description of their main characteristics will be provided here and interesting reviews of thermodynamics models applied to the Diesel engine can be found in [3][4][5].

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2D/3D Computational Fluid Dynamics (CFD), are the most predictive simulations because they solve the full unsteady Navier-Stokes equations set. They provide very detailed information of internal processes but require important computational resources. That is why they are generally restricted to local studies or in-cylinder simulations where fuel spray, turbulent combustion and chemical kinetics need a very small characteristic time scale to be predicted [6].

1D approach is a reduction of CFD models assuming only one dimension in the Navier-Stokes equations. Well established 1D codes such as GT-Power, WAVE, AVL Boost or the open source available OpenWAM [7] solve the mass, momentum and energy conservation equations in piping systems with finite differential or finite volume schemes [8][9], and employ particular boundary conditions in the different connections. They give an accurate description of complex gas dynamic phenomena in the intake and exhaust lines [10] but their computational time is relatively slow, in the order of 100-1000 times greater than real time. Their highly resolved gas exchange and cycle simulation are normally employed in engine development processes where inertia or wave effects could be important such as air delivery ratio optimization, variable valve timing calibration, manifold design, etc.

0D Filling & Emptying Models (FEM) represent the engine components as a series of interconnected control volumes (cylinder, manifold, intercooler, etc.) and connecting elements (valve, restrictions, etc.). They solve the mass and energy conservation equations with time steps in the order of one crank angle to obtain instantaneous state of the gas within the different volumes. They exhibit for a moderate computational cost a relatively good prediction of the engine performance in both steady and transient conditions. Nevertheless, by ignoring the momentum conservation equation and without spatial resolution, they bring a further reduction to the Navier-Stokes equations which, even if engine aspiration is physically captured, make them unable to predict inertia and wave effects in the manifolds. Depending of the operating conditions and engine geometry, errors can be significant. Winterbone et al. [11] and Watson et al. [12][13] were the first in the late 1970's to introduce FEM in engine development to simulate turbocharger response. Since that, these models have also been largely used in control design and fault diagnosis [14][15].

Mean Value Engine Models (MVEM) have the same origin than FEM. They integrate transfer components and different volumes where the mass and energy conservation equations are resolved. But their resolutions are time-based and predict only mean values variations [16][17][18]. With time step in the order of one revolution, sub-cycle processes such as combustion or flow through valves are simplified by empirical correlations or map based models. MVEM are therefore not predictive and rely mainly on data obtained from measurements or high fidelity models. Their calibrations employ identification techniques and require a large amount of comprehensive engine data over a wide operating range, especially when nonlinearity and interaction between engine variables are significant. The main advantage of MVEM reside in their real time capabilities which make them particularly favorable for control and observer design. Hendricks et al. [19][20][21] were the first to develop MVEM for control supervision. Then numerous authors have integrated other empirical or semi-empirical methods (analytical derivations, radial basis functions [22][23], neural networks [24][25], etc.) to improve simulation accuracy for their specific applications.

The conventional model level to perform matching calculations has been for years the 1D approach. This approach presents advantages in term of physical description but has become rather time-demanding with the arrival of multistage boosting systems and HP/LP EGR circuits. To face the challenge of fast engine development processes, a new approach has therefore to be defined with a more appropriate balance between model complexity and CPU performance.

Matching calculations are generally performed at an early stage of the engine development process when only little geometrical information is defined and no experimental data are available for the base engine. In these conditions, lookup tables or MVEM are not appropriate to simulate combustion processes and a predictive model is required with, as a minimum, a crank angle resolution in cylinders. 1D multizone models give a good description of in-cylinder heterogeneities involved in the local combustion and pollutant production terms [26]. However, as pollutant prediction is not essential for architecture matching, 0D phenomenological models are preferred for their sufficient physical representative capability and reasonable CPU expenses [27]. To calculate the engine scavenging behavior and the exhaust temperature, 1D wave action model are particularly accurate. Nevertheless, they require for correct wave effects predictions the design information of the intake and exhaust systems (runner diameters, pipe lengths, etc.) and a precise description of their acoustic behavior. This information which can be measured in specific impulse test rigs [28] or extracted from 3D CFD models [29], is not available at an early engine development stage. So, 1D model capabilities cannot be fully exploited and the lower computational cost of 0D FEM's become more attractive to reproduce the most important thermodynamics processes. For the different air path components such as turbochargers, intercooler, air filter, etc., large data bases with results of characterizations performed in specific test benches are generally provided. These data are in form of characteristics maps or lookup tables over wide operating ranges. MVM have therefore a sufficient complexity to model these different engine components.

From these considerations, it results that the adequate trade-off between maximum physical description and minimum computational cost to perform efficient matching calculations is a combination of 0D phenomenological combustion model, with a 0D FEM for the multi-cylinders and manifolds, and a MVM for the other air path elements. In that way, advantages of

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reasonable computational cost from the 0D-FEM and MVM approaches are combined with the physical depth of detailed cycle resolved cylinder models.

In addition, matching objectives are generally defined in terms of engine performance with boost pressure targets. The variables of interest in matching calculations are the ones describing turbochargers behavior, position of turbine actuators, etc. Nonetheless, equations resolutions performed by typical matching codes such as wave action models use the actuators variables as an input data to predict engine performance (actuators oriented model). In that way, control strategies for each engine actuators need to be developed and a large number of iterations are required to reach given performance objectives controlling actuator positions. This approach is low efficient and time consuming for steady-state calculations. That's why a new resolution algorithm has to be defined to run the simulations in a straightforward manner, from engine performance objectives to charging system configuration (boost pressure oriented model). In transient operations, the gas path handle different relevant physical effects such as mass accumulation transport of species, inertias, etc. and control strategies have an important impact on the boosting system performance. In this case, a boost pressure oriented algorithm cannot be employed and the conventional actuators oriented scheme remains the most appropriate. To reduce as much as possible CPU expenses during matching calculations, the simulation model has therefore to integrate both approaches, boost pressure oriented model for steady state conditions and actuators oriented model for transient operations.

In this paper, a 0D engine model especially developed for matching calculations will be described. First, the crank angle resolved model that regroups the combustion and the FEM will be introduced. Second, the intake and exhaust MVM will be presented with their different elements. An original representation of turbine performance is proposed for matching calculations. These models will be then interconnected to form a comprehensive engine model and both boost pressure and actuators oriented algorithms will be described. Finally, a complete validation process has been carried out to assess the new modeling approach. Comparisons between experimental and modeled data will be discussed in the last section.

MATCHING ORIENTED MODEL

The engine model proposed in this paper is a combination of FEM and MVM. Figure 1 shows a representation of the model components. The model core is a FEM of the engine cylinders, the intake and exhaust manifold and the HPEGR line with a cooler and a valve. The cylinders models include a diesel combustion model.

The basic engine model is connected to the air path model at intake and exhaust manifolds. Indeed, there are two different air path models depending on the type of calculations. For **matching** or boosting pressure oriented calculations the air path is represented by a MVM. In such calculations the input for the model is the boosting pressure (at the intake manifold inlet) and the output is the position of all the actuators allowing achieving this boosting pressure objective. For **transient** or control oriented calculations the air path model is also FEM in order to account for accumulation effects in the lines. In transient calculations the model inputs are the position of the actuators and the output would be the resulting boosting pressure and flow variables throughout the engine.

The whole model is coded in the Matlab environment. This allows using advanced algotithms to control the calculations for both matching and transient calculations.

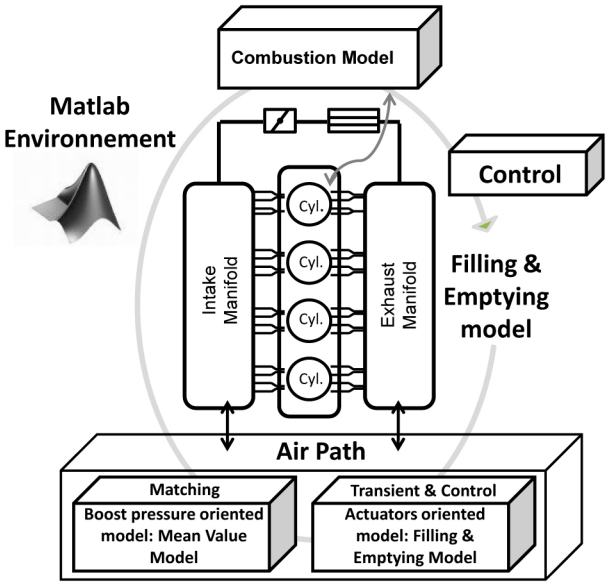


Figure 1. Elements of the proposed matching model

Crank Angle Resolved 0D Engine Model

As explained above the engine cylinders and manifolds are calculated by FEM able to take into account mass fluctuations during the gas exchange. In Figure 1, 4 cylinder engine is represented but the model is extendable to any number of cylinders. It is important to note that the HPEGR is included in the engine model since the flow is related with the instantaneous intake - exhaust pressure ratio.

Basically, the FEM used here is a collection of elements available in the code OpenWAM developed in the authors' lab. For the sake of simplicity these models will not described here, only each type of model will be pointed and some references to papers will be given. A detailed description of the models and even the source code can be retrieved at OpenWAM website [7].

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The cylinders and manifolds are calculated as an open control volumen. An explicit crank angle resolved solution of continuity, energy and state equations is performed. The cylinder model includes the effect of combustion chamber deformations due to pressure and inertia forces as explained in [30]. The thermal flows and wall temperatures are calculated by a nodal approach [31]. Convection coefficient is calculated following Woschni [32] with some corrections for the effect of swirl proposed by Payri et al. in [33][34]. For a proper trapped mass calculation the effect of scavenging [35] and blow-by [30] are also considered. The gas properties are estimated as a mass weighted average between air, fuel ($C_{10.8}H_{18.7}$) and stoichiometric burnt gas properties obtained from JANAF database [36] and revised by SANDIA laboratories [37].

The diesel combustion model used here is the one proposed by Arregle et al. [38][39]. The model constructed around a spray-mixing model is based on the gaseous steady jets theory and on the conceptual injection and combustion models proposed by Dec [40] and Hiroyasu [41]. This model evaluates the Apparent Combustion Time (ACT) the delay from the time when a given fraction of the fuel has been injected and the time when this fuel has burnt. It is possible to rely then the injection rate with the RoHR with only a calibration parameter to tune for a given engine. The model gives particularly good results for conventional mostly diffusive combustion, with low EGR rates and low swirl numbers. It also gives an estimate of NOx formation through the Zeldovich mechanism.

By the way, it is also possible to use “recorded” RoHR coming from testing campaigns. Typically they are processed as mathematical functions such as a combination of 4 Wiebe laws [42][43].

The flow between cylinders and manifolds is estimated using the adiabatic nozzle equation and depends on upstream (in) pressure and temperature and on downstream (out) pressure.

$$\dot{m} = C_d A_{ref} p_{in} \sqrt{\frac{2\gamma}{R(\gamma-1)T_{in}}} \sqrt{\left(\frac{P_{out}}{P_{in}}\right)^{\frac{2}{\gamma}} - \left(\frac{P_{out}}{P_{in}}\right)^{\frac{\gamma+1}{\gamma}}} \quad (1)$$

For the poppet valves, the reference area Aref corresponds to the seat section while the discharge coefficient Cd is interpolated from the experimental data obtained in the flow test rig as a function of valve lifts and flow direction.

For the EGR valve, HP EGR rate XHPEGR appears generally as an objective in the matching calculations. The corresponding averaged mass flow passing through the HP EGR circuit depends of the breathing capability and is defined as:

$$\hat{\dot{m}}_{HPEGR} = \frac{X_{HPEGR}\left(\hat{\dot{m}}_{air} + \hat{\dot{m}}_{LPEGR}\right)}{1 - X_{HPEGR}} \quad (2)$$

where $\hat{\dot{m}}_a + \hat{\dot{m}}_{LPEGR}$ is the cycle-averaged mass flow rate entering in the intake manifold by the external section. The position of the HPEGR valve is thus readjusted during the simulations to obtain the desired mass flow applying to the effective section the following control:

$$C_d^{\alpha+720°} A_{ref} = \frac{X_{HPEGR}\left(\hat{\dot{m}}_{air} + \hat{\dot{m}}_{LPEGR}\right)}{1 - X_{HPEGR}} \frac{C_d^{\alpha} A_{ref}}{\hat{\dot{m}}_{HPEGR}} \quad (3)$$

Heat transfer is estimated by the effectiveness method in the HP EGR cooler and by the 1-node model with thermal resistance scheme [44] in the ports and manifolds. Different characteristics ducts are defined to reproduce the considered geometries and the same correlations presented for the exhaust line are employed for the exhaust manifold and all outer surfaces. For the other inner surfaces, heat transfer coefficients are calculated in the intake manifold and intake ports with the Nusselt-Reynolds correlation presented by Depcik and Assanis [45]:

$$Nu_{inner} = 0.0694\,\mathrm{Re}^{\frac{3}{4}} \quad (4)$$

while the correlations introduced by Caton [46] and modified by Reyes [47] are used in the exhaust ports. In the exhaust ports, the turbulence generated at the cylinders discharge has an important impact on the heat transfer and a distinction is made when the exhaust vales are open or closed:

$$Nu_{inner_{open}} = 0.5 \times 0.0075\,\mathrm{Re}$$

$$Nu_{inner_{closed}} = 0.022\,\hat{\mathrm{Re}}^{0.8} \quad (5)$$

When the valves are closed, in addition to different correlation coefficients, a cycle-averaged Reynolds number Re is considered instead of the instantaneous value.

Mean Value Based Gas-Path Description

Multistage boosting architectures regroup a large number of configurations that needs highly flexible simulation tools to efficiently describe different engine assemblies. The intake and exhaust line models consist thus of individual components that can be assembled arbitrarily to build a complete description of the gas path. The main architectures having some similarities, basic structures are also proposed here to give a better overview of possible assemblies.

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Intake Line

The base structure and the intake boosting components library can be observed in Figure 2. The air mass flow enters with atmospheric conditions in the air filter to exit in 1 where it undergoes a mixing process with the LP EGR mass flow to reach conditions in 1′. The evolutions in the LP and HP intake charging systems go from 1′ to 12 and 12′ to 2 respectively. Between both stages, an intercooler can perform a cooling process depending of the bypass valve control. After the HP stage, the mass flow passes through an aftercooler and a throttle valve to arrive in the intake manifold. The intake throttle is only used in combination with the HP EGR system when the pressure difference between the intake and exhaust manifold is not sufficient to obtain the desired HP EGR rate. For the LP and HP intake charging systems, the component library includes superchargers, centrifugal superchargers, eBoosters, centrifugal compressors in single or parallel configuration and connections for single boosting system.

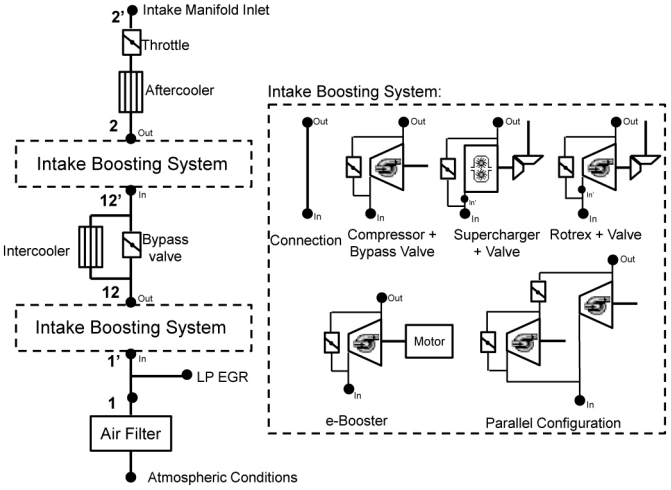


Figure 2. Based structure and components library of the intake line MVM.

In steady state operations, matching calculations are based on a boost pressure oriented resolution scheme. Engine performance objectives are defined in term of pressures in 2′ and 12′ corresponding to combustion and control requirements respectively. According to the boosting architectures, variations of P12′ are generally simulated to optimize exhaust backpressure and fuel consumption. Volume capacities with mass accumulation are not considered and mass flows are assumed to be constant along the whole air path. Heat transfers are discarded between the components although models like the ones used for the exhaust line can be integrated if their effects are judged significant. The submodels used to calculate the different elements are briefly described below.

Engine breathing capability is reproduced by the FEM which gives to the MVM the mass flow at the intake manifold inlet. This mass flow is imposed constant from 1′ to 2′. HP and LP EGR rates (X_{EGR}) are generally defined before matching calculations as combustion objectives to respect particular emissions regulations. LP EGR mass flow and, as a consequence air filter mass flow, are defined as:

$$\dot{m}_{LPEGR} = \dot{m}_{1'} X_{LPEGR} \left(1 + \frac{X_{HPEGR}}{1 - X_{HPEGR}}\right)$$

$$\dot{m}_{1} = \dot{m}_{1'} - \dot{m}_{LPEGR} \quad (6)$$

Air filters and coolers are characterized in a flow test rig where their pressure losses are measured as a function of corrected mass flow. Experimental data are accurately fitted with polynomial functions which are then directly integrated in the models. It is assumed the gas flow suffers an isothermal process in the air filter, whereas heat transfer in the coolers is calculated by the effectiveness ε method which evaluates the outlet temperature as:

$$T_{out} = T_{in} + \varepsilon\left(T_{ext} - T_{in}\right)$$

$$\varepsilon = f\left\{NTU, C_p \dot{m}\right\} \quad (7)$$

where T_{ext} stands for the external cooling fluid temperature and NTU the number of transfer units. If the coolers have not been characterized in a thermal flow rig, typical NTU values for heat exchangers can be found in [48].

Throttle and bypass valves are modeled as isentropic nozzles where the mass flow is expressed from the Saint Venant equation (Eq. 1)

From a control point of view the effective section in a valve is directly proportional to the actuators position. But here, with a boost pressure oriented resolution and knowing the mass flow, the throttle can be considered as a pressure drop to reach the HP EGR objectives and the effective section is calculated as a resultant. Bypass valves are generally used in on/off configurations. In this case, no pressure losses are considered across the valve.

The compressor model is based on the characteristic maps obtained in turbocharger test bench that give corrected shaft speed and efficiency as a function of corrected mass flow rate and pressure ratio. These maps are extrapolated in the low rpm zone with the isentropic enthalpy gain method [49] and interpolated in the model by polar basis functions. The compressor outlet temperature is derived from the isentropic efficiency.

The isentropic power used as turbocharger input data in the exhaust line MVM is expressed with the mass flow rate and total enthalpy changes. Taking into account a motor efficiency, this power corresponds for the eBooster to the electrical

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