

Fuel Consumption Optimization and Noise Reduction in a Spark-Ignition Turbocharged VVA Engine

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ABSTRACT

Modern VVA systems offer new potentialities in improving the fuel consumption for spark-ignition engines at low and medium load, meanwhile they grant a higher volumetric efficiency and performance at high load. Recently introduced systems enhance this concept through the possibility of concurrently modifying the intake valve opening, closing and lift leading to the development of almost “throttle-less” engines. However, at very low loads, the control of the air-flow motion and the turbulence intensity inside the cylinder may require to select a proper combination of the butterfly throttling and the intake valve control, to get the highest BSFC (Brake Specific Fuel Consumption) reduction. Moreover, a low throttling, while improving the fuel consumption, may also produce an increased gas-dynamic noise at the intake mouth.

In highly “downsized” engines, the intake valve control is also linked to the turbocharger operating point, which may be changed by acting on the waste-gate valve. Depending on the valve lift and the actual exhaust pressure, different internal-EGR levels can be achieved, requiring a variation in the spark advance as well. Of course, the introduction of new degrees of freedom to the engine control poses new problems in terms of engine calibration and tuning. To best exploit the potential offered by these systems, new methodologies and development tools must be utilized as well.

In this paper, an optimization procedure is presented, aiming to select the best combination of four control parameters (intake valve closure angle, butterfly valve opening, waste-gate opening and spark advance) of a twin-cylinder turbocharged engine in different operating conditions. A detailed 1D simulation model of the whole engine is firstly developed and validated against the experimental data in WOT conditions and in predefined low load operating points. The model is developed within GT-Power™ commercial code environment, but employs an in-house routine that is implemented to simulate the combustion process. This routine takes into account the effects on the heat release induced by the variations of the in-cylinder flow field, turbulence and internal-EGR. These, in fact, are substantially modified by the actual strategy specified for the engine control. An external module is implemented to evaluate the intake valve lift profile as a function of the closure angle and the engine speed. The 1D prediction of the gas-dynamic noise at the intake mouth is further validated through the comparison with the results obtained by the unsteady 3D CFD analysis of the flow field within the air-box device. The 1D computed pressure profile downstream the device is utilized as boundary condition for the 3D model.

The validated 1D model of the turbocharged engine is then coupled to a multipurpose commercial optimizer (ModeFRONTIER™). The optimization procedure identifies the best combinations of the previously selected control parameters with the goal of minimizing both the fuel consumption and the gas-dynamic noise, for defined low-load low-speed operating points. The procedure is validated against the experimental data obtained by a calibration of the engine at the test-bed. The values of the control parameters selected by the optimizer well agree with the experimentally identified ones. Alternative settings are also proposed with the aim to substantially reducing the radiated noise with limited or no penalty for the BSFC. The optimization results are also verified by a 3D CFD analysis.

Therefore, the procedure allows for the pre-calibration of a VVA turbocharged SI ICE on completely theoretical basis and proves to be very helpful in reducing the experimental costs and the engine time-to-market.

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INTRODUCTION

Modern internal combustion engines present more and more complex solutions with the aim of improving performance in terms of both delivered power/torque, fuel consumption, pollutants emissions and radiated noise. The turbocharging technique and completely flexible valve actuation systems are currently the most promising technologies for further improvement of spark ignition ICEs. Initially, valve actuation systems were only used to control the intake and/or exhaust valve phasing (VVT - Valve Variable Timing) [1,2,3] or employed to realize different cam profiles according to the engine speed [4,5]. Most recent solutions are able to independently define the valve opening, closing and the lift (VVA - Variable Valve Actuation) [6,7]. The flexibility offered by the modern VVA systems allows to adjust the engine load with very limited butterfly valve throttling [8,9], which drastically reduces the pumping work at low/medium load. Moreover, a reduction in NO_x emissions and an improved combustion stability at idle were experimentally verified [10,11,12,13,14]. The combustion stability could be further enhanced if a double lift mode in each engine cycle is available thanks to a higher in-cylinder turbulence.

Conversely, while in the past, turbocharging was used to increase the delivered power and torque, currently it is primarily employed to reduce fuel consumption and, consequently, CO₂ emissions [15], thanks to the so-called "downsized" engine architecture. It consists in achieving the required power/torque performance using smaller displacement engines coupled to a turbocharger group [16]. In this way, improvements of the fuel consumption are expected due to the lower mechanical and pumping losses. Further advantages are related to a higher torque at low engine speed and thus to a better vehicle drivability [17,18]. The previously described technologies are included in the engine that is analyzed in the present article. The engine is a small turbocharged twin-cylinders engine equipped with a VVA module, able to control the intake valve lift profile. The valve actuation is provided by an electro-hydraulic system that modulates the valve lift according to the operating conditions. In many cases, the valve lift greatly differs from the cam-derived profile. The system consists of a small capacity between the cam and the valve filled with lubricant oil, whose emptying is regulated by an electronically controlled electro-valve [19].

A typical drawback of a VVA system is, however, the worsening of gas-dynamic noise emissions at low loads [20]. The reduced throttling of the intake ducts, while allowing a

reduction of the pumping work, also results in a less effective attenuation of the pressure wave in the intake system [21,22]. Additionally, at very low loads, the decreased duration of the intake phase may determine a reduced turbulence level at the spark time and a slower burning speed. For the above reasons, precise engine control parameter settings must be specified to obtain the best compromise among different objectives.

Table 1. Engine Main Features

Model	2 cylinders, 8 valves, VVA
Displacement	875 cm ³
Stroke/Bore	86 mm / 80.5 mm
Connecting rod length	136.85 mm
Compression Ratio	9.9
Max Brake Power	64.6 kW @ 5500 rpm
Max Brake Torque	146.1 Nm @ 2500 rpm

The engine analyzed here, whose main features are listed in the Table 1, allows for the opportunity to operate in a specified speed-load point using different control strategies. In particular, it is possible to independently define the following parameters:

- α air to fuel ratio (A/F)
- β throttle opening angle
- θ spark ignition timing
- ϕ_1 intake valve opening angle
- ϕ_2 intake valve closing angle
- WG turbine waste-gate valve opening

α , β , and θ are traditionally used to calibrate a naturally-aspirated SI engine, while ϕ_1 - ϕ_2 and WG are related to the control of the VVA and the turbocharger, respectively. To identify the set of parameters that allows the engine to achieve the required performance targets (minimum fuel consumption and pollutant emissions at part load, maximum power and torque at full load, combustion stability at idle, etc.), DoE (*Design of Experiment*) methodologies are successfully used limiting the number of experimental tests [23].

A further reduction in the experimental activity can be obtained using proper numerical models that, once tuned, are able to achieve a pre-calibration of the engine following a fully-numerical approach. Activity at test-bench can, hence, be limited to verify and refine the desired performance targets.

In this paper an example of the methodology described above is defined in detail. First, a 1D engine model is developed and validated at full and part load. The model shows the capability to predict the main overall performance parameters and the radiated noise with good accuracy. 3D CFD analyses are also carried out with the aim of verifying 1D gas-dynamic noise predictions at full and part load, as well. Subsequently, the engine model is included in a general purpose optimizer (ModeFRONTIER) [24,25] that is used to identify the calibration parameters that contemporary minimize the fuel consumption and the gas-dynamic noise at the intake mouth [26,27]. Finally, 3D analyses are once again utilized to confirm the optimization results, in terms of radiated noise reduction.

MODEL DESCRIPTION

The engine model is developed in the GT-Power commercial software, based on a 1D description of the flow inside the intake and exhaust pipes. The combustion process and the turbulence phenomenon are described using sub-models introduced in GT-Power through “user routines”. Specifically, the combustion is modeled using the “fractal combustion model” [28,29,30], which is a phenomenological model sensing both the combustion system geometry (head and piston shape, spark plug position, etc.) and the operating parameters, such as the spark advance and the air to fuel ratio.

The turbulence phenomenon is described by a sub-model capable to sense the valve opening and closure timing [31,32]; this model allows the variations in the burning speed related to different specified control settings to be taken into account.

The strategies involved by the valve actuation system equipping the tested engine include LIVO (Late Intake Valve Opening), EIVC (Early Intake Valve Closure), a combination of the previous ones and a multi-lift strategy. As previously observed, the same load level can be achieved by employing different valve strategies. To simplify the engine calibration issue, among the various available strategies, only adjusting the valve closure angle (ϕ_2) is selected. The Fig. 1 shows the comparison between some valve lift profiles related to the different opening angles of the VVA system electro-valve. It is clear that, while advancing the VVA electro-valve opening, the intake phase duration is shortened and, for a lower ϕ_2 , a relevant valve lift reduction is obtained, as well.

The valve profiles seen in Figure 1 are derived using a model of the valve actuation system developed in the AmesimTM environment. This model provides the valve dynamics as a function of the cam profile, the inertial data (valve mass, etc.), the spring preload, the engine speed, the electro-valve opening and closure angles and the thermo-physical properties of the lubricant oil (especially, the oil viscosity). It is clear that the absence of a mechanical link between the cam and the valve may determine a valve lift profile that is different from the traditional symmetric bell-

like shape; moreover, the lift profiles are strongly affected by all the previously listed parameters.

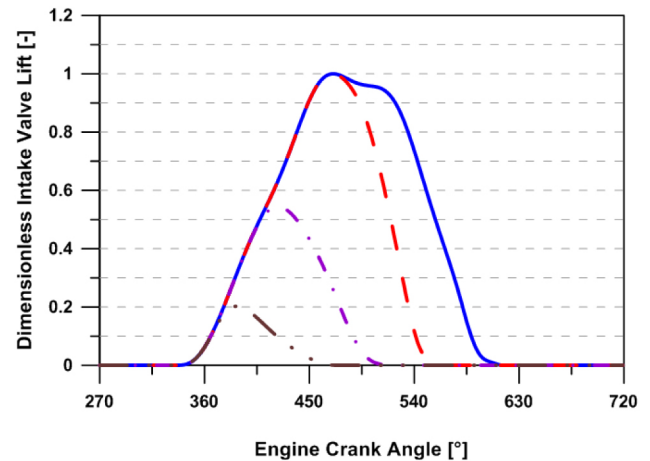


Figure 1. Electro-Hydraulic system typical valve lift

Employing the Amesim module and once the system geometry, the mechanical parameters and the lubricant oil properties are defined, the different valve lift profiles are calculated for various engine speeds (from 1000 rpm to 6000 rpm at a step 500 rpm) as well as the ϕ_2 values. These data constitute a valve lift “database” that is used by a “user routine” in GT-Power to evaluate by interpolation the actual lift profile, depending on the engine operating conditions.

The 1D schematization of the tested engine includes the presence of a “virtual microphone object” located at 1 cm from the intake mouth. This object computes the pressure field at the microphone location by treating the intake orifice as a simple pulsating monopole and assuming free field conditions [33].

1D MODEL VALIDATION

The experimental approach that was followed to obtain the full load (WOT) torque/power curves involved the tuning of the intake valve closure angle, ϕ_2 , which guarantees maximum cylinder filling. The overall performances are, however, limited by some constraints that are related to the boost pressure level and the turbocharger speed for low and high engine speeds, respectively. Moreover, a very rich A/F ratio is specified to respect the limits on the maximum allowable temperature of the turbine blades, while the spark advance is set at each speed in order to prevent detonating combustions.

The parameters mentioned above, namely the intake valve closure, the A/F ratio and the spark advance, are then provided as input to the 1D model. Because information about the experimental WG valve opening is not available, the latter has to be indirectly evaluated using a PID controller and a target experimental datum, like the boost level, the mass flow rate or the turbocharger speed. The usage of the experimental boost level may introduce some inaccuracies in the simulations, since at low-speed/full-load the compressor operates in close-to-surge map region, where iso-speed lines

are very reduced. Consequently, the map-derived mass flow rate cannot be precisely identified. On the other hand, the experimental mass flow rate or the turbocharger speed are usually affected by higher uncertainties. For this reason, an alternative strategy is proposed, aiming to identify the WG opening realizing the experimental pressure level at the firing TDC for the cylinder # 2. This level is in fact representative of the actual in-cylinder filling conditions since, at full-load, the combustion process, to avoid the knocking, always begins after the TDC. The proposed strategy provided the best agreement with experimental data.

The results shown in Figures 2,3,4 are obtained after the tuning of few combustion model parameters. They are kept constant for each operating condition analyzed. In particular, the figures show the numerical/experimental comparison in terms of power/torque, boost pressure, turbocharger speed, BSFC and gas-dynamic noise measured at the virtual microphone (labeled as SPL - Sound Pressure Level in Figure 4). Due to confidentiality reasons, each of these parameters is plotted normalized by the corresponding experimentally measured maximum value to obtain dimensionless parameters excepting the last one; in fact, the noise emissions are displayed in terms of difference with respect to the maximum measured value.

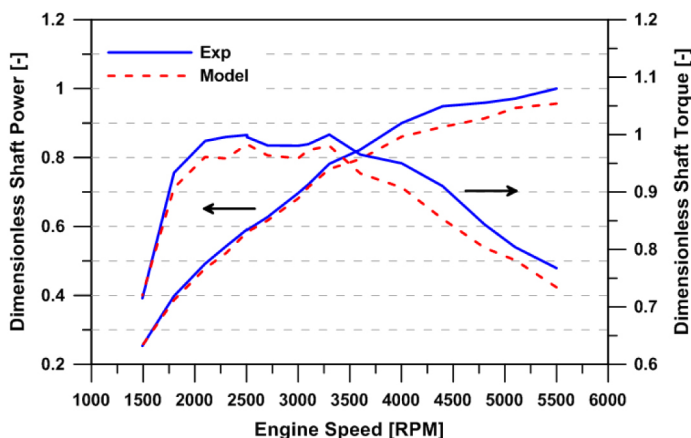


Figure 2. Dimensionless shaft power and torque

The agreement in terms of power and torque is satisfactory with the exception of the higher engine speeds; this result can be caused by an overestimation of the pumping work during the exhaust stroke. In additions, due to the two-cylinder arrangement, large mass flow rate and pressure ratio oscillations are expected. Engine-turbocharger matching is hence a critical issue for this engine, since a strongly unsteady operation is expected for the turbocharger. This may also affect BSFC predictions at higher speeds.

A similar satisfactory agreement is verified on the other performance parameters, with errors limited to 5-10%. The SPL trend is in particular correctly reproduced (Figure 4b), although slightly overestimated. This may be due to the standard “map-based” description of the compressor/turbine

characteristics. The maps' data in fact are determined from steady-state flow tests that do not include pressure waves [33]; their use within an unsteady calculation then implies that the pressure wave upstream the compressor cannot take into account the actual noise abatement through the device.

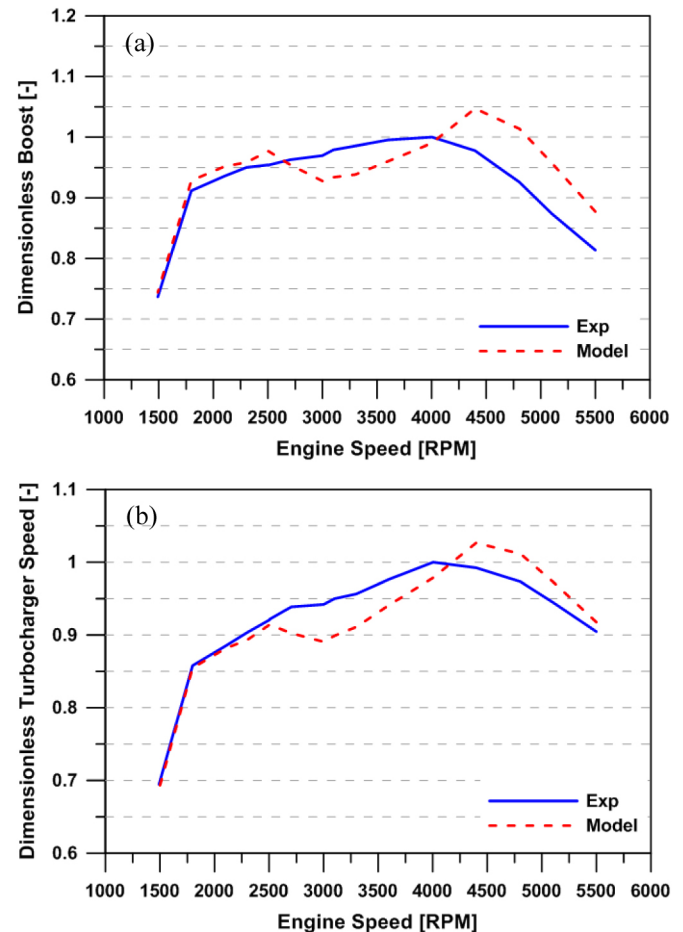


Figure 3. Dimensionless boost pressure and turbocharger speed

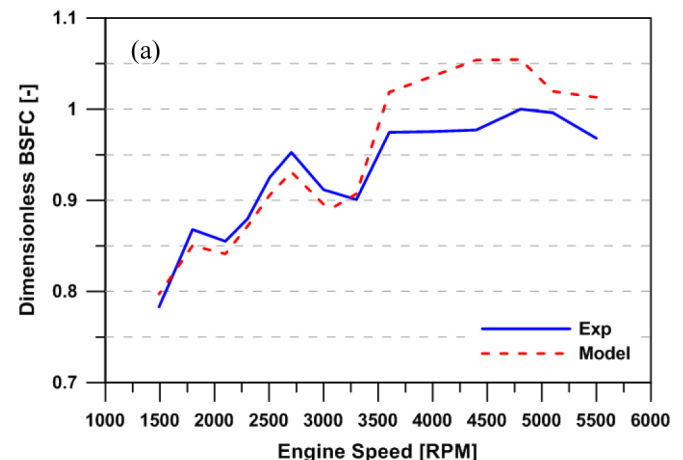


Figure 4. Dimensionless BSFC and ASPL