

Strategies for Improving Fuel Consumption at Part-Load in a Downsized Turbocharged SI Engine: a Comparative Study

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ABSTRACT

It is commonly recognized that the paths for improving fuel consumption (BSFC) in a spark-ignition engine at part-load require more advanced valve actuation strategies, which largely affect the pumping work. Since several years, many different solutions have been proposed, characterized by different levels of complexity, effectiveness, and cost. Valve systems currently available on the market allow for variable phasing (VVT - Variable Valve Timing), and/or lift (VVA - Variable Valve Actuation). Usually VVT devices are applied on intake and exhaust camshafts, in the "phased" or "unphased" configuration, as well. VVA devices are instead commonly mounted on the intake camshaft. More recent VVA systems also allow for a double intake valve lift during a single engine cycle (multi-lift), or may include a small intake pre-lift during the exhaust stroke. The latter solutions may determine further BSFC reductions. Alternatively, an external-EGR circuit can be considered, as well.

Each system introduces additional parameters to the standard engine control variables (throttle opening, spark advance, waste-gate valve opening in turbocharged engines). In order to quantitatively compare the potentialities of the above systems, proper numerical models can be employed.

In this paper, a comparative study is performed on a small-size turbocharged engine to estimate the maximum BSFC improvement which can be obtained with the employment of the above mentioned advanced techniques. Firstly, a 1D model of a commercial engine, equipped with a VVA device, is validated at full and part-load; then, different valve and EGR strategies are investigated starting from the base engine architecture. Analyses are carried out with the commercial 1D thermo fluid-dynamic code GT-Power™, provided with user routines for the modeling of the combustion process and the handling of different valve strategies. The 1D model is coupled to an external optimizer (modeFRONTIER™) which selects the various engine control parameters to minimize the fuel consumption at a prescribed part load and speed condition.

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INTRODUCTION

Recent spark-ignition internal combustion engines are becoming highly complex systems with the aim of improving their performance. Many development activities, in particular, have been focused on variable valve control systems (VVA - Variable Valve Actuation). The huge increase in patent applications concerning VVA systems in the past two decades confirms the great effort made in this area within the automotive industry [1]. Fully flexible valve actuation systems have good potential to meet future powertrain requirements for further improvement of fuel economy, emissions and delivered power for spark ignition engines. Originally, variable valve actuation systems were only able to control the intake and/or exhaust valve phasing (VVT - Valve Variable Timing) [2,3,4] or to switch from two different valve lift profiles according to the engine operating conditions (two-step VVA) [5,6]. Most recent

solutions are able to independently define the valve opening, closing and the lift (fully flexible VVA) [7,8]. These systems can continuously vary the valve lift and timing over the operating range and most of them use the Early Intake Valve Closing (EIVC) as the primary strategy to reduce pumping work and improve fuel economy through a very limited intake system throttling for engine load regulation [9,10].

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 maximize the BSFC reduction. Anyway, the use of different strategies can also allow for an improved combustion stability at idle [11,12]. Moreover, the combustion stability could be

further enhanced through unconventional valve lift profiles as a “Multi-Lift” strategy, able to promote a higher in-cylinder turbulence.

A further approach used to enhance the engine performance is the turbocharging. While in the past it was used to increase the delivered power and torque, currently it is primarily employed to reduce fuel consumption and, consequently, CO₂ emissions [13]. Following the so-called “downsizing” philosophy, the turbocharger allows to achieve the required power/torque performance using smaller displacement engines [14]. On the other hand, at part load operations, a fuel consumption reduction is expected thanks to lower mechanical and pumping losses. Further advantages are related to a higher torque at low engine speeds and thus to a better vehicle drivability [15].

The previously described technologies are included in the engine that is analyzed in the present article. It is a small turbocharged twin-cylinders engine equipped with a VVA module, able to control the intake valve lift profile. The valve actuation includes 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 [16].

The strategies involved by the valve actuation system equipping the tested engine include the EIVC, the LIVO (Late Intake Valve Opening), a combination of the previous ones, and a multi-lift strategy.

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 here analyzed, whose main features are listed in Table 1, can operate in a specified speed-load point by using different control strategies. In particular, it is possible to independently define the throttle opening angle, the turbine waste-gate valve opening, and the intake valve closure angle, ϕ_2 . The air to fuel ratio and the spark advance cannot be considered additional control parameters. In fact, the first, at part load, is usually set at a stoichiometric value to allow for a correct operation of the three-way catalytic converter, while, at full load, it is properly enriched to improve burning rate, prevent knock occurrence and reduce the turbine inlet temperature. The spark timing, at part load, is selected in order to realize an optimal combustion phasing, with a MFB50% (50% of mass burned fraction) 8-10 degrees after the TDC, while, at full load, it is properly delayed to avoid the occurrence of knocking combustion.

In this paper, a 1D model of the described engine is build up in the GT-Power™ environment. In a first stage, after a tuning at full load operation, the model is validated against the experimental data on the whole operating domain of the engine. Then, the base architecture is virtually modified to allow for the utilization of different intake valve strategies and external EGR configurations. The latter are compared at a prescribed low load, low speed operating point (3000 rpm and 3 bar of BMEP) with the aim of minimizing the fuel consumption. This is carried out by means of an automatic procedure [17,18,19] implemented in a general purpose optimizer (ModeFRONTIER™) [20,21].

ENGINE MODEL DESCRIPTION

As previously mentioned, the engine model is developed within a commercial software. It is based on a 1D description of the flow inside the intake and exhaust pipes. The combustion process and the turbulence phenomenon are simulated by sub-models introduced in the code through user routines. Specifically, the combustion process is modeled by the “fractal combustion model” [22,23,24,25,26], 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 simulated by a 0D sub-model that belongs to the K-k model family. It includes the balance equations of the macro-scale kinetic energy (K) and of the turbulent kinetic energy level (k) and describes the cascade mechanism from K to k. Based on a 0D approach, both the above cited quantities are assumed homogenous and isotropic within the cylinder. Proper source and sink terms are included in the model formulation to handle the energy production and destruction during the intake and exhaust phases. This allows the model to correctly sense the valves opening and closing timings. The k dissipation rate is related to the mass averaged integral length scale derived by preliminary 3D CFD simulations of the in-cylinder flow-field under motored operations [27].

The combustion process description is properly linked to the in-cylinder turbulence evolution in order to detect the effects of different valve control settings on the engine performance.

The engine model is provided by a further “user routine” to evaluate the actual intake valve lift profile, according to the engine operating conditions. The valve lift sub-model inquires a database including different lift profiles calculated, by a further incompressible 1D simulation of the VVA electro-hydraulic system, for various engine speeds and ϕ_2 values, as well.

Apart from the EIVC, the LIVO, and the multi-lift strategies, feasible by the actual device mounted on the considered engine, the valve user routine allows for the simulation of other

strategies such as a VVT, successfully employed in a lot of applications available on the market, or a more unconventional, "Pre-Lift" strategy [28].

ENGINE MODEL TUNING AT FULL LOAD

The experimentally actuated intake valve closure angle, ϕ_2 , air to fuel ratio, spark advance and throttle opening (fully opened) are imposed in the model. For the waste-gate opening identification, a different approach is followed with respect to the classical methodology based on a PID controller that targets an experimental datum (to this aim the manifold pressure, the air flow rate or the turbocharger speed can be equivalently employed). This is because, for the studied twin-cylinder engine, due to the largely pulsating flows in the intake system, the compressor operates in the unstable region of the map for a certain part of the engine cycle, especially for the lower engine speeds. This behavior cannot be correctly handled by the classical map-based approach [29] and, consequently, the standard strategies for the waste-gate opening identification may introduce some errors on volumetric efficiency prediction. On the other hand, a precise tuning of the combustion model, not affected by inaccuracies in the cylinder filling description, is mandatory to suitably carry out the following analyses at part load.

For this reason, the waste-gate is set in order to match the measured in-cylinder 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 knocking, always begins after the TDC.

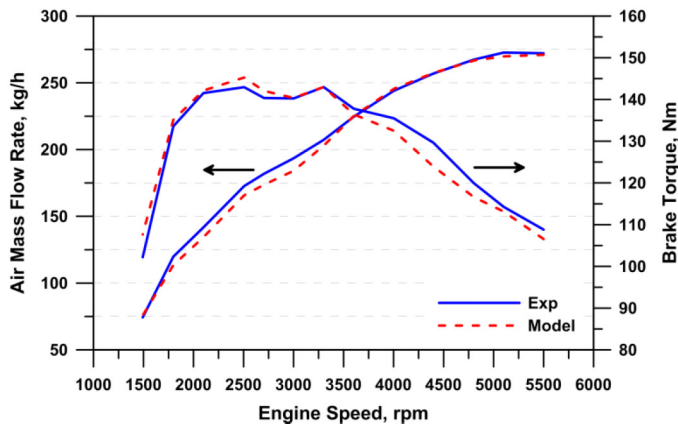


Figure 1. Brake power and torque numerical/experimental comparison at full load operation.

The tuning of the turbulence and combustion sub-models included in the present work follows a 0D/3D hierarchical approach, as described in detail in [26,27]. The first step is the computation of the in-cylinder flow field evolution by means of a 3D CFD analysis in motored conditions. The mass averaged values of the turbulence intensity, velocity magnitude and integral length scale are used to identify the tuning constants for the turbulence model. Then, the combustion model is tuned in order to fit the in-cylinder pressure cycles. A satisfactory

agreement between the numerical and experimental results is obtained both in terms of in-cylinder traces and overall performance (brake power/torque, mass flow rate, BSFC, turbocharger speed, etc.) [26,27].

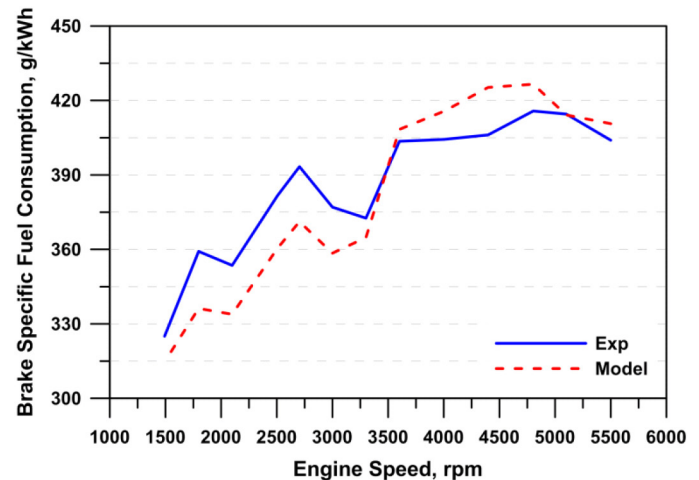


Figure 2. BSFC numerical/experimental comparison at full load operation

As an example, Figure 1 and Figure 2 show the numerical/experimental comparison in terms of air mass flow rate/brake torque and BSFC, respectively. The air flow and BSFC underestimation for the lower speeds can be justified by the above discussed problems in the description of the compressor behavior. In addition, the valve lift profiles employed in 1D analyses may partly differ from the ones really actuated by the VVA system. For the higher speeds, the air mass flow rate is correctly predicted, while the brake torque and BSFC denote some disagreement. They are mainly due to an overestimation of the pumping losses during the exhaust stroke, which may depend on some inaccuracies in the employed compressor and turbine efficiencies [30,31,32,33].

Despite the above discussed issues, the overall results of Figure 1 and Figure 2 show a satisfactory agreement, also considering that the tuning constants are kept unchanged for all the tested operating conditions.

ENGINE MODEL VALIDATION AT PART LOAD

After the above described tuning process at full load, the capability of the model in predicting the engine performance in terms of BSFC at part load is tested. Also in this case, the calculations are carried out by imposing the experimentally actuated spark advance, air to fuel ratio and intake valve strategy (EIVC). The latter is qualitatively depicted in Figure 3. As can be observed, the advanced closure is also coupled to a reduction of the maximum lift, with respect to the full-lift profile (lost-motion principle).

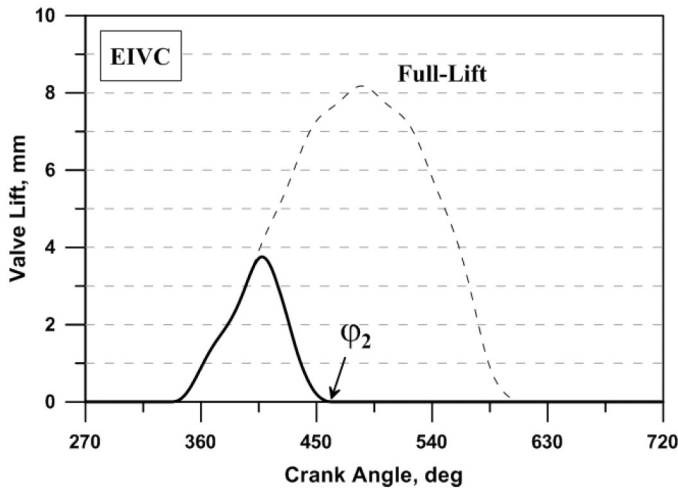


Figure 3. EIVC valve lift profile

The waste-gate opening is instead automatically selected by a PID controller aiming to reproduce the measured turbocharger rotational speed, while the throttle opening targets predefined loads. Figure 4, Figure 5, Figure 6, Figure 7, Figure 8, Figure 9 show the fuel consumption variation against the BMEP, for different speeds. It can be observed that the numerical results are in a very good agreement with the measurements, especially for the low-medium loads. Thanks to the EIVC strategy, the fuel consumption levels are substantially low and only increase at very low loads. This is mainly due to a more relevant incidence of the pumping losses, and to a detrainment of the combustion process efficiency, as well. The reduced intake phase duration (Figure 3) in fact produces a moderate turbulence level at the compression TDC, coupled to the well-known syringe effect. Both affect the combustion speed, mainly during its early stage. The latter phenomena are very well captured by the employed turbulence/combustion sub-models, as demonstrated by the highlighted results and by the next detailed combustion analysis.

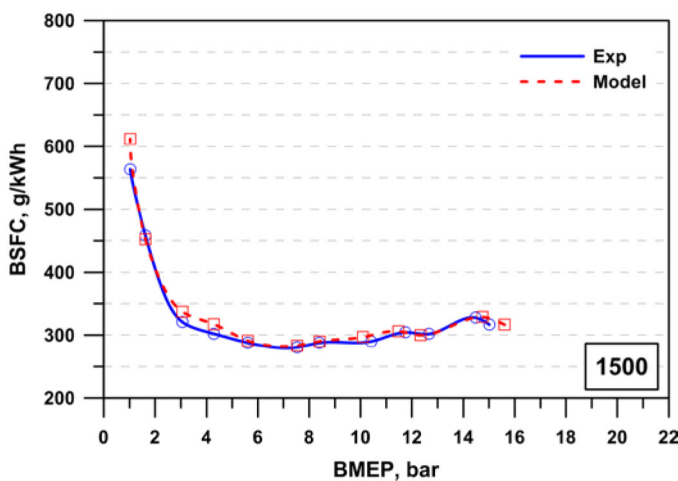


Figure 4. BSFC vs. BMEP at 1500 rpm

To further highlight the predictiveness of the used combustion model at part load operations, in Figure 10 the angular positions of 10%, 50% and 90% of the MFB are compared with the experimental data for an engine speed of 3000 rpm. It can be appreciated that the combustion center is correctly forecasted by the model for all the considered loads with a maximum error lower than 3 degrees. The combustion duration (10-90% MFB) is well captured for the medium and higher loads, while for the lower loads it is overestimated, mainly due to a longer combustion tail. However, as shown in Figure 6, this error has a negligible effect on the fuel consumption prediction.

The worsening of the fuel consumption for the higher loads is indeed related to the relevant mixture enrichment necessary to decrease the exhaust gas temperature at the turbine inlet. In addition, a further penalty is determined by the specification of a reduced (sometime negative) spark advance, required to avoid knocking phenomena. Also in this case, the model shows the capability to capture the described trends, except for the engine speeds of 4400 and 5500 rpm.

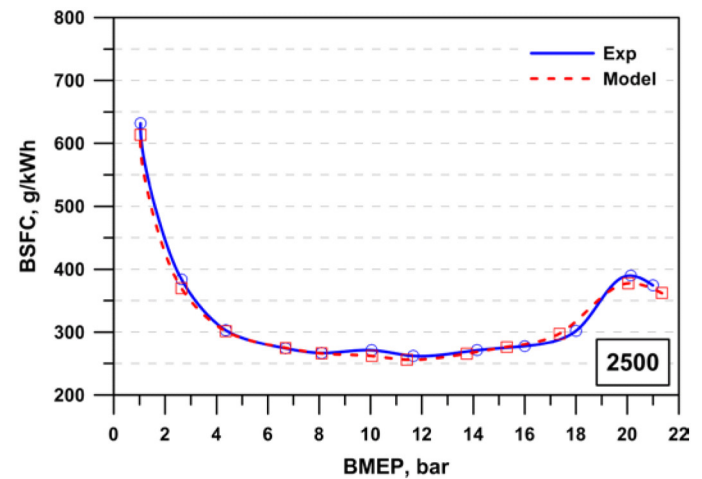


Figure 5. BSFC vs. BMEP at 2500 rpm

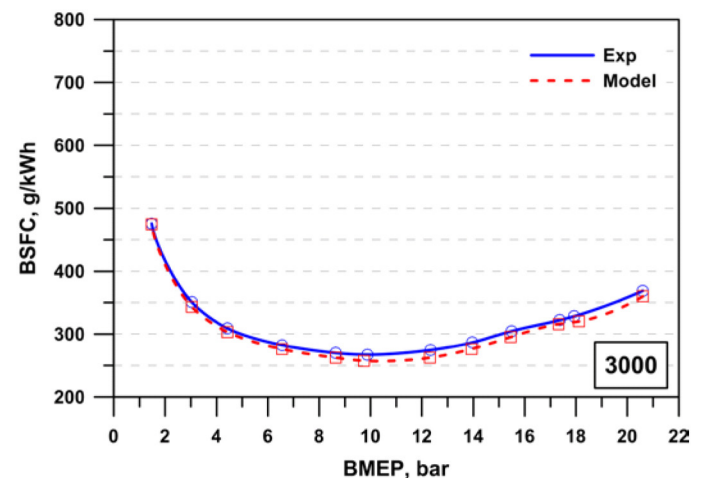


Figure 6. BSFC vs. BMEP at 3000 rpm