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Pre-lift valve actuation strategy for the performance improvement of a DISI VVA turbocharged engine

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Abstract

Modern internal combustion engines (ICEs) are becoming more and more complex in order to achieve not only better power and torque performance, but also to respect the pollutant emissions and the fuel consumption (CO_2) limits.
The turbocharger, advanced valve actuation systems (VVA) and the EGR circuit allow the ICE's load control together with the traditional throttle valve and spark advance. Thus, an higher number of operating parameters are available for the calibration engineer to achieve the required performance target (minimum fuel consumption at part load, maximum power and torque at full load, etc.). On the other hand, the increased degrees of freedom may frustrate the potentialities of so complex systems because of the effort needed to identify the optimal engine control strategies. The development of proper numerical models may assist and direct the experimental activity in order to reduce the related times and costs.
Although VVA solutions could bring a reduction in the specific fuel consumption thanks to an important de-throttling of the intake system, unfortunately they can simultaneously lead to higher noise levels radiated by the intake mouth. In fact, in this case, the pressure waves travelling through the intake ducts are not properly damped by the throttle valve.
In this paper a numerical methodology is developed to define the engine calibration and the intake valve lift profile that simultaneously minimize the BSFC and the noise at part load. The engine object of the study is a turbocharged Spark-Ignition Direct Injection (SIDI) ICE equipped by a lost motion valve actuation system for the intake valves. In this study, the commercial 1D thermo fluid-dynamic code GT-Power^{TM} is provided with user routines for the description of the combustion process and the handing of variable valve lift profiles. The engine model is thus integrated with a commercial optimization code (modeFRONTIER^{TM}) to identify the optimized load control strategies to achieve the set objectives. The proposed methodology is also used for the definition of unconventional valve lift profiles. Particularly, the advantages related to the use of a small *pre-lift* before the main valve lift profile are estimated compared to a conventional EIVC strategy.

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Nomenclature

BSFC	Brake Specific Fuel Consumption
BMEP	Brake Mean Effective Pressure
EGR	Exhaust Gas Recirculation
ICE	Internal Combustion Engine
PID	Proportional Integrative Derivative
PMEP	Pumping Mean Effective Pressure
SPL	Sound Pressure Level
TDC	Top Dead Center
V	Instantaneous in-cylinder volume
V_{max}	Maximum in-cylinder volume
VVA	Variable Valve Actuation
VVT	Variable Valve Timing
WG	Waste-Gate Valve
α	Air to Fuel Ratio
β	Throttle Valve Opening
θ	Spark Advance
φ_1	Intake Valve Opening Angle
φ_2	Intake Valve Closure Angle

1. Introduction

The modern automotive internal combustion engines (ICEs) are characterized by more and more complex architectures in order to minimize the fuel consumption, without penalizing power and torque performance, and to comply with the European law concerning the pollutants and CO_2 emissions [1]. To this aim, regarding the SI-ICEs, a common strategy involves the reduction of the total displacement and the coupling to a turbocharger [2].

This approach, usually named *downsizing*, allows for the required performance target and a reduction of the fuel consumption with respect the naturally aspirated engine delivering the same power, as well. In fact, the reduced displacement and weight imply lower mechanical losses, both in absolute and relative terms. In addition, for an assigned load level, a smaller displacement determines a reduced throttling of the intake system, with a positive effect on the pumping work [1],[2]. The small size does not undermine the ICE power and torque performance thanks to the possibility to achieve a high boost pressure at full load. Those benefits can be further enhanced equipping the engine with a fully flexible valve actuation system and with a direct fuel injection.

The valve actuation systems currently available on the market allow, as a function of the operating point, for the variation of the phasing (VVT – Variable Valve Timing), [3],[4],[5] or, in addition, of the lift (VVA – Variable Valve Actuation) [6],[7]. The aim of both the solutions is the load adjustment with a reduced valve throttling; this implies a significant positive influence on the pumping work.

Moreover, in a SIDI ICE, since the fuel evaporation takes place in the cylinder, the temperature at the end of the compression stroke results lower; in this way, a reduced knock risk is achieved [8]. In addition, the direct injection allows for higher valves overlap. This involves a better cylinder scavenging without any fuel loss at the exhaust. In some cases, the direct injection is employed to realize a stratified charge in the cylinder with the aim of improving the combustion process and, as a consequence, reducing the fuel consumption by using lean fuel/air mixture [9].

Analyses discussed in the present paper concern a prototypal 4 cylinders turbocharged DISI ICE equipped by a fully flexible actuation system of the intake valves. Its flexibility is due to an electro-hydraulic system belonging to the lost motion family [10], [11]. The use of a VVA system to adjust the engine load implies, as previously mentioned, a reduction in the pumping work, but, at the same time, a deterioration of the engine performance in terms of gas-dynamic noise radiated at the intake orifice [13], [14]. In fact, the lower intake system throttling involves a less effective damping of the pressure waves moving through the intake system. The main characteristics of the analyzed engine are listed in Table 1:

Table 1 – Engine main characteristics

Model	Turbocharged, 4 cyl., 16 valves, VVA
Displacement	1368 cm^3
Stroke/Bore	84 mm / 72 mm
Connecting Rod Length	128.95 mm
Compression Ratio	10

The operating parameters available for load control are: the air-to-fuel ratio (α), the throttle valve opening (β), the spark advance (θ), the intake valve opening and closing angle (φ_1 and φ_2) and the waste-gate opening (WG).

The first three parameters are commonly used to calibrate the ICE, while the others are related to the VVA system and to the turbocharger. Each operating point can be obtained with infinite combinations of the above listed quantities; as a consequence, it is mandatory to identify the optimal calibration that allows to achieve the required performance targets (minimum specific fuel consumption at part load, maximum power and torque at full load, etc.). In this work, a 1D model of the analyzed engine is implemented in the GT-Power™ environment and is utilized to predict BSFC and gas-dynamic noise [14], [15], [16] The model is integrated within a commercial multi-purpose optimization software (modeFRONTIER™) [12], [17], to identify the engine calibrations able to optimize BSFC and noise emissions at part load. An operating point of interest in the homologation European Urban Driving Cycle is selected, 2000 rpm and 2 bar of BMEP (in the following named 2000@2).

The same automatic procedure is used to estimate the effects of unconventional valve lift profiles that enhance the internal EGR, allowing for a further fuel consumption reduction. In particular, a small *pre-lift* of the intake valve occurring during the exhaust stroke is analyzed at different *pre-lift* heights and angular durations.

2. Model description

As previously mentioned, the engine model is developed within the GT-Power 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 described by sub-models introduced in GT-Power through user routines. Specifically, the combustion process is modeled by the “fractal combustion model” [18], 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 0D sub-model capable to sense the valve opening and closing timing [19], [20]. The combustion process modeling is properly linked to the in-cylinder turbulence evolution in order to detect the effects of different control settings on the engine performance.

The analyzed VVA system allows for different strategies such as the LIVO (Late Intake Valve Opening), the EIVC (Early Intake Valve Closing), a combination of the previous ones, and a multi-lift strategy, as well. To simplify the engine calibration issue, only the adjustment of the valve closing angle (φ_2) is taken into account, corresponding to a EIVC strategy. Fig. 1a shows a set of valve lift profiles related to different φ_2 angles.

The valve lift profiles depicted in Fig. 1a are derived by using a 1D model of the valve actuation system. In particular, the different lift profiles are calculated for various engine speeds and φ_2 values, as well. These data constitute a valve lift “database” used by a user routine in GT-Power to evaluate, by interpolation, the actual lift profile, depending on the engine operating conditions.

The shown valve lift profiles refer to the actual operation of the engine, since they are derived by the actual geometry of the cam, by the inertial data (valve mass, etc.), by the spring preload and by the thermo-physical properties of the lubricant oil. In the design of a new engine, just modifying the cam shape, it is possible to realize a lift profile that includes a small, almost constant, lift during the exhaust stroke (*pre-lift*). This opportunity is also included in the valve lift user routine that allows to assign, together with the closure angle φ_2, also the *pre-lift* duration, $\Delta\theta$, and height, h, (Fig. 1b).

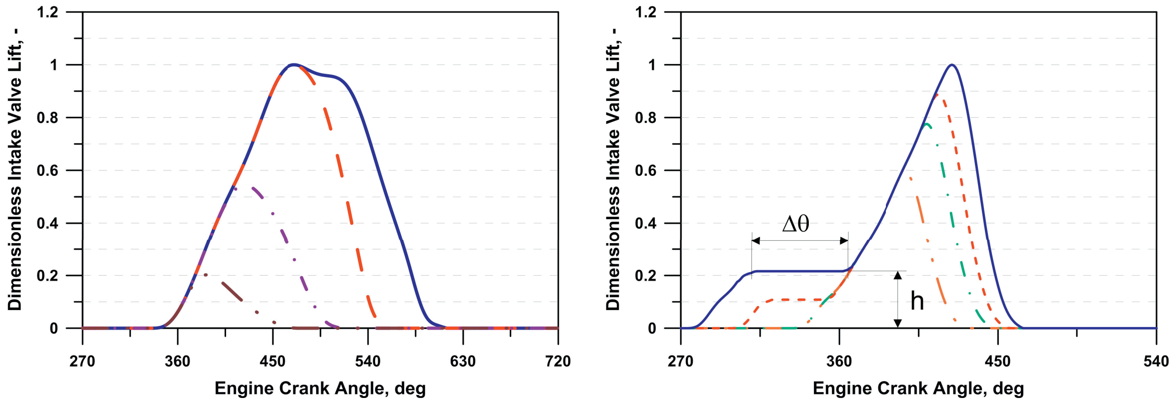


Fig. 1. (a) Typical valve lift profiles of an electro-hydraulic VVA system; (b) Valve lift profiles with different *pre-lift* parameters

The 1D schematization of the tested engine includes a "virtual microphone object" located at 1 cm from the intake orifice. 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 [21]. This allows to simultaneously compute BSFC and Sound Pressure Level (SPL) of the gas-dynamic noise in each operating condition.

3. Model tuning

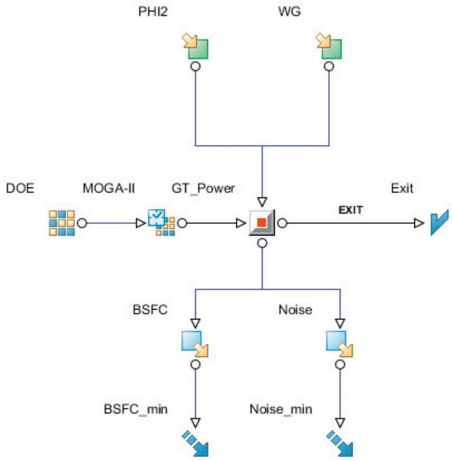
Since the design of the analyzed engine is not completely concluded, experimental data are not yet available. Nevertheless, the combustion and turbulence sub-model tunings are borrowed by an engine of the same family. All the details about the similar engine and the sub-model tuning are included in [12], [19], [20].

4. Part load optimization at 2000 RPM-2 bar of BMEP

The 1D model is the core of the optimization process aiming to define the engine calibration strategies that minimize the BSFC or the SPL at 2000@2. It is included in an automatic procedure, implemented within the commercial optimizer modeFRONTIER™ (Fig. 2a). At the beginning of each iteration, the selected decision variables, i.e. the WG opening and φ_2 angle, are assigned, and a single analysis is performed. A PID controller included in the 1D model allows to identify the throttle valve opening required to get the target BMEP (2 bar).

The simulation is carried out by fixing the spark advance, so that the crank angle related to 50% burned mass reaches a literature advised value, i.e. 8 degrees ATDC. In addition, the modeled fuel injector adjusts the fuel amount according to the air flow rate, in order to realize a stoichiometric air-to-fuel ratio. At the end of each simulation, the optimizer uses the computed BSFC and SPL levels to set the decision variables values for the next iteration according to the genetic algorithm logic (MOGA-II).

Previous works have already demonstrated that this kind of optimization procedure is able to identify with great accuracy the experimentally applied strategy realizing the lowest BSFC [12]. As usual in a multi-objective optimization problem, multiple optimal solutions arise, corresponding to the trade-off between BSFC and SPL as shown in Fig. 2b†. Because of the confidentiality required by the engine manufacturer, the BSFC-SPL values are normalized by subtracting the corresponding values of the solution that realizes the minimum BSFC. The latter is labeled "0" in the following, and is assumed as a reference solution. The same normalization is applied to all the considered quantities excepting the waste-gate opening; the latter is in fact represented as a fraction of its maximum opening.



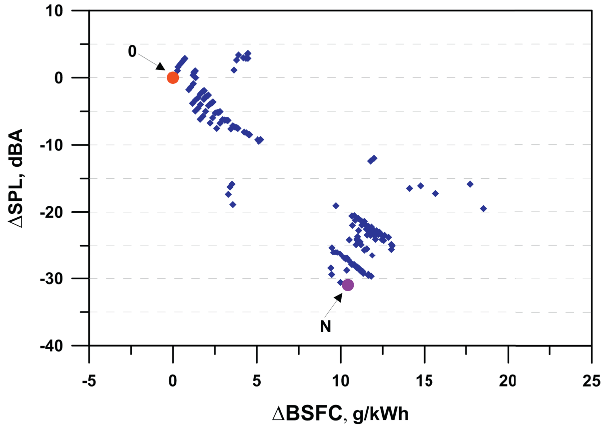
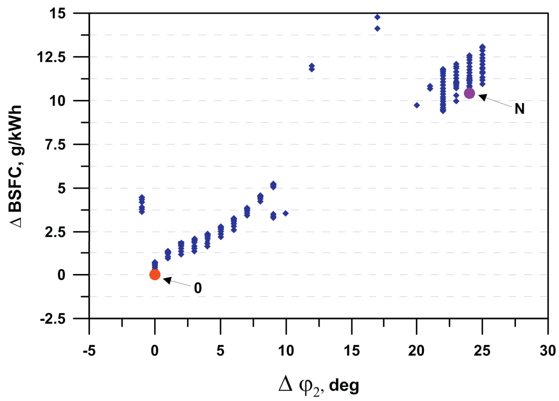


Fig. 2. (a) Workflow of the optimization process; (b) Trade-off between SPL and BSFC normalized values

Among the optimal solutions, the calibration realizing the minimum gas-dynamic noise (labeled "N") is selected, too. Observing the Fig. 2b, it can be noted that a relevant reduction in the noise emission can be attained (-31.0 dBA), even if a penalty in the BSFC has to be accepted (+10.4 g/kWh).

Fig. 3 shows that the solution "N" is characterized by values of the angle φ_2 much higher than the reference calibration "0". This strategy determines an higher PMEP (Fig. 3b), due to an higher throttling of the intake system (the intake plenum pressure is reduced by 0.36 bar, see Table 2 in the next section). As a consequence, an higher BSFC is obtained (Fig. 3a), as usually occurs in a traditional throttle-valve controlled engine. This is clearly put into evidence in Fig. 4, where the negative pressure loop is highlighted for solutions "0" and "N".



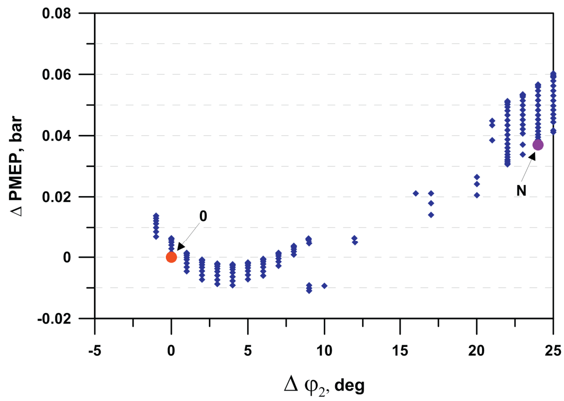


Fig. 3. (a) ΔBSFC vs. $\Delta\varphi_2$ variation; (b) ΔPMEP vs. $\Delta\varphi_2$

† The plotted quantities are the differences between actual and "0" solution values; the same rule is adopted for the plots concerning the other operating and overall performance parameters, with the exception of ΔPMEP, that is calculated as the opposite.

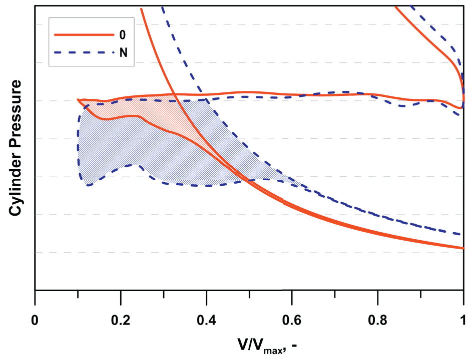


Fig. 4. Comparison of solutions “0” and “N”. In-cylinder pumping pressure cycle

Concerning the effects of valve strategy on burning rate, Fig. 5a shows that a limited difference in combustion duration is obtained between solutions “0” and “N”. Actually, due to the longer time available for turbulence decay, the EIVC strategy of solution “0” determines a minor turbulence level at TDC [19]. Solution “N”, instead, as a consequence of the lower pressure in the intake plenum, is characterized by a higher residual fraction at IVC. The above effects practically compensate and a similar burning rate is definitely realized. Fig. 5b confirms indeed that a delayed φ_2 angle is the main responsible for a simultaneous reduction of the SPL at the expense of an increased BSFC (Fig. 3a).

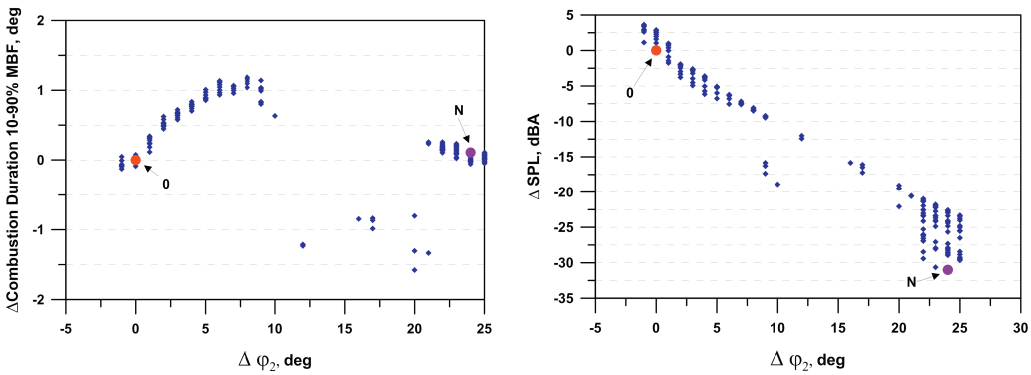


Fig. 5. (a) Δ Combustion Duration vs. $\Delta\varphi_2$; (b) ΔSPL vs. $\Delta\varphi_2$.

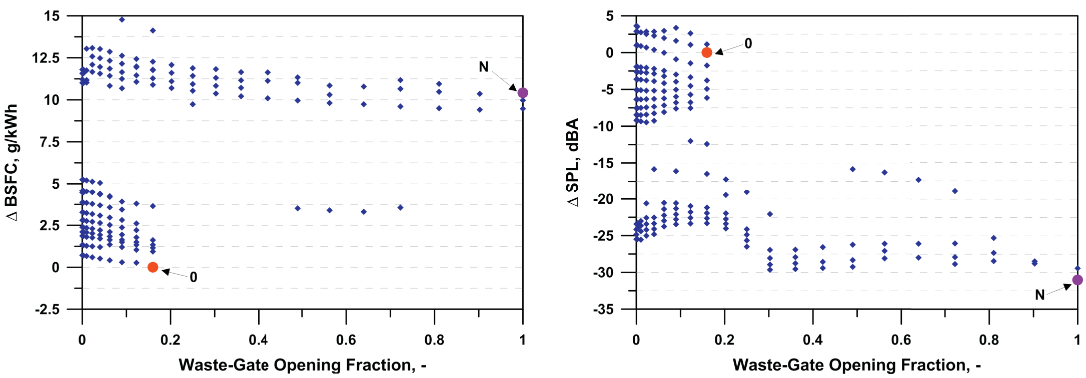


Fig. 6. (a) ΔBSFC vs. WG opening fraction; (b) ΔSPL vs. WG opening fraction

Regarding the effects of the waste-gate opening, Fig. 6 displays that it plays a minor effect on BSFC and SPL, as well. This is mainly due to the reduced boost pressure required for the specified 2000@2 condition. For an assigned opening fraction, large variations of both BSFC and SPL may occur, depending on the local value of the φ_2 angle, representing the most important control parameter. The above results hence allow to get a wide picture of the overall engine behavior and, as a consequence, to select proper compromise solutions among the two conflicting objectives.

5. Part load optimization involving the *pre-lift* strategy at 2000@2

In this section, the possibilities offered by a *pre-lift* of the intake valves during the exhaust stroke are studied with the aim to further reduce the specific fuel consumption. A second optimization (Fig. 7a) is performed also including the additional *pre-lift* characteristic parameters, namely: the maximum *pre-lift* and its angular duration (see Fig. 1b). In Fig. 7b, the results of the two optimization procedures are compared in terms of Pareto frontiers collecting the whole set of optimal solutions. The Figure highlights that the introduction of a *pre-lift* provides a significant reduction in the BSFC, attaining its minimum value in the condition labeled as solution “B”; the latter also realizes a similar SPL level of solution “0”. The *pre-lift* is indeed completely disabled to get the minimum noise, and the same previously identified solution “N” is obtained.

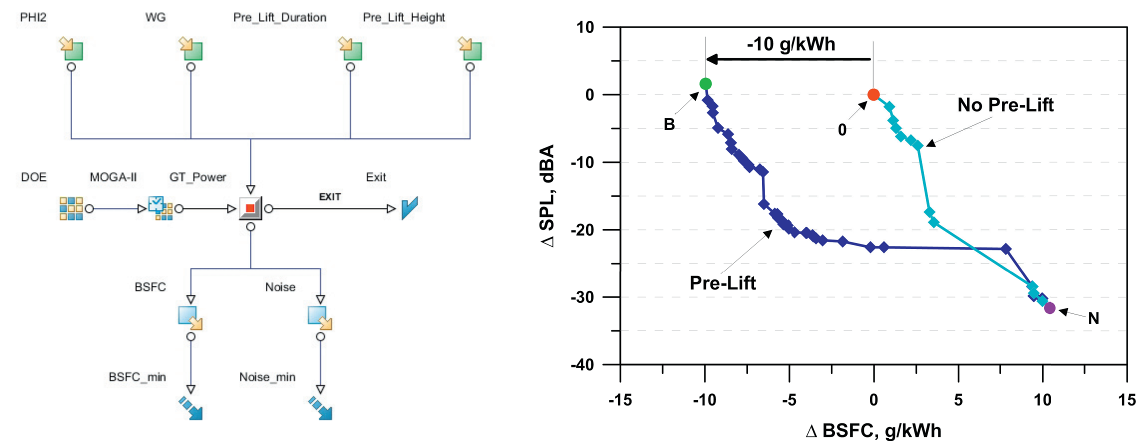


Fig. 7. (a) Workflow of the optimization process including the *pre-lift* design parameters; (b) Comparison between the Pareto optimal fronts for the strategies with and without *pre-lift*.

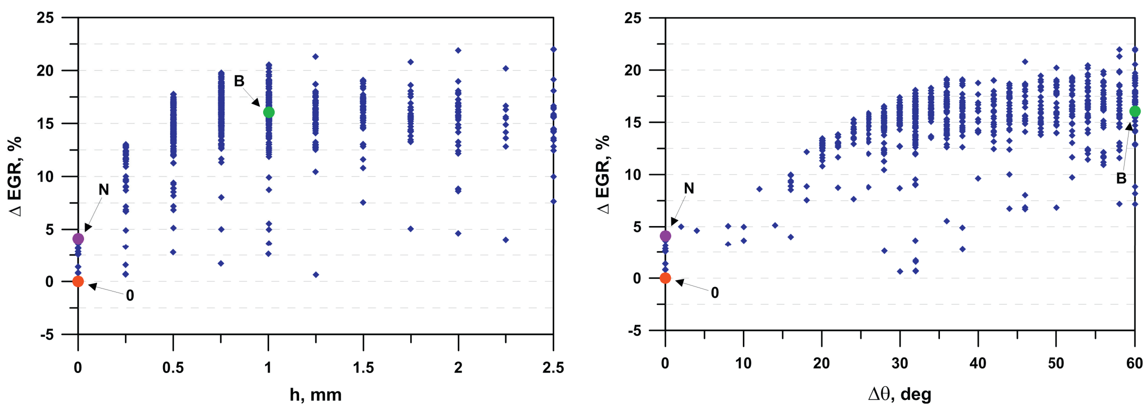


Fig. 8 (a) ΔEGR fraction vs. maximum *pre-lift* height; (b) ΔEGR fraction vs. *pre-lift* angular duration

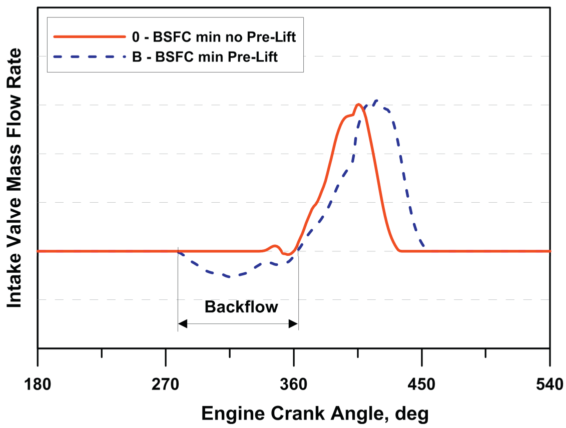
Table 2 - Performance and calibration parameters for the “0”, “B” and “N” solutions.

SOLUTION	“0”	“B”	“N”
DESCRIPTION	Min. BSFC no pre-lift	Min. BSFC pre-lift	Min. Noise
Pre-lift parameters $h/\Delta\theta$, mm/deg	0/0	1/60	0/0
BSFC, g/kWh	0.00	-9.98	+10.42
Noise, dBA	0.0	+1.6	-31.0
WG opening ratio, -	0.16	0.42	1.00
φ_2, deg	0	+16	+24
Comb. Duration, deg	0.0	+13.4	+0.1
EGR, %	0.0	+16.0	+4.1
Intake plenum pressure, bar	0.0	-0.033	-0.363

Fig. 8 shows that the presence of a *pre-lift* provides, as expected, a substantial increase of the internal-EGR. Comparing the instantaneous mass flow rates through the intake valve in cases “0” and “B” (Fig. 9a), it can be observed that a relevant exhaust gas backflow occurs during the *pre-lift* opening. The above gases return inside the cylinder during the first part of the subsequent intake stroke. For this reason, “B” solution involves a delayed valve closure to get the air-fuel mixture amount required for the prescribed load level (Fig. 9b). Simultaneously, an higher in-cylinder pressure at IVC and a greater value of the effective volumetric compression ratio[‡] are realized. Ultimately, the thermodynamic efficiency is improved and, despite a slightly increased negative pressure loop (shaded areas in Fig. 9b), a lower BSFC is actually attained.

The above behavior is however limited to precise values of control parameters h, $\Delta\theta$, which in turn determine an optimal φ_2 angle. It can be observed in fact that, over the φ_2 value corresponding to “B” solution, the fuel consumption once again increases (Fig. 10a). This occurrence can be explained with a less efficient combustion process: the higher φ_2, the larger is internal EGR percentage promoted by the *pre-lift*. Consequently, as shown in Fig. 10b, a slower and slower burning speed verifies.

The above discussed results highlight the strong coupling existing among many overlapping phenomena (pumping work, EGR percentage, effective compression ratio, etc.), each one affecting the fuel consumption in a different extent. The employment of an accurate phenomenological combustion model is hence mandatory in order to capture the effects of discussed issues on burning rate, in different operating conditions and control strategies.



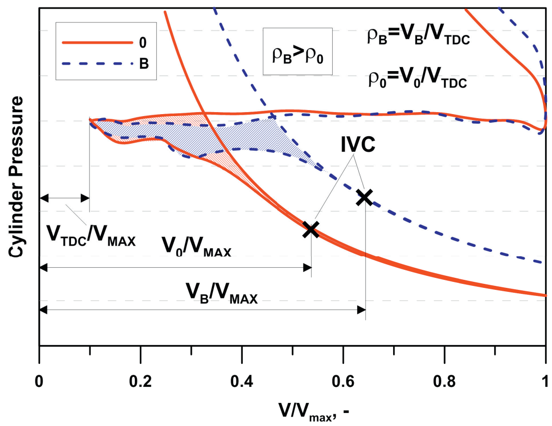


Fig. 9. Comparison of solutions “0” and “B”. (a) Instantaneous mass flow rate through the intake valve; (b) In-cylinder pumping pressure cycle

‡ The effective volumetric compression ratio is defined as the ratio between the cylinder volume at the IVC and at the TDC

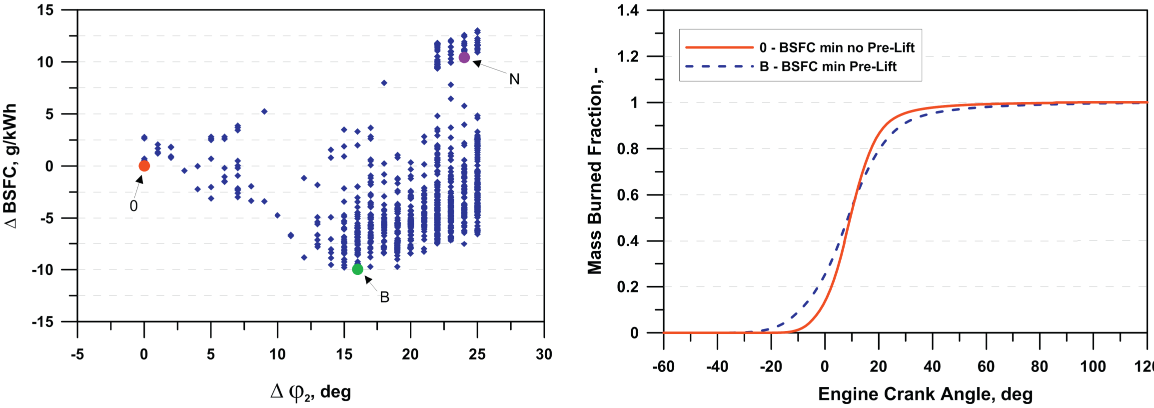


Fig. 10 (a) ΔPMEP vs. $\Delta\varphi_2$; (b) Mass burned fraction for the solutions “0” and “B”

6. Conclusions

The present paper describes a numerical methodology aiming to calibrate a turbocharged internal combustion engine equipped with a fully flexible VVA system applied to the intake camshaft. It is also able to investigate the effects of unconventional valve lift profiles on engine performance. The engine model is developed in GT-Power environment and is integrated with user routines for the turbulence/combustion and intake valve lift handling. The sub-model tunings are borrowed by a previous work regarding a similar engine.

The model is then included in the commercial optimization software ModeFRONTIER, with the aim of identifying the calibration strategies that minimize the fuel consumption and the gas-dynamic noise emissions in a part-load point of interest in the homologation European Driving Urban Cycle, i.e. 2000@2. Concerning the first optimization, the solution realizing the lowest BSFC (solution “0”) involves a very early intake valve closing that allows to substantially reduce the pumping work. The radiated noise in these conditions attains its maximum level. The optimizer simultaneously identifies the trade-off between BSFC and SPL, and gives information on the fuel consumption penalties to be paid for a prescribed radiated noise reduction.

In the selected part-load condition, instead, the engine and turbocharger matching does not seem to play a considerable role. The optimal turbocharger setting is always fixed to a fully opened waste-gate valve.

The effects of the introduction of a *pre-lift* are tested in a further optimization, where the *pre-lift* duration and height are included among the decision variables. Depending on these additional control parameters, a larger internal EGR is promoted, inducing a delayed closure of the intake valve. As a consequence, an higher effective volumetric compression ratio and a reduced BSFC is realized. Those effects are however limited by the combustion efficiency that reduces due to the EGR presence. An improved SPL-BSFC trade-off is definitely obtained.

The presented results demonstrate that the proposed methodology is able to highlight the strong coupling existing among many overlapping phenomena (pumping work, EGR percentage, effective compression ratio, etc.), each one affecting the fuel consumption in a different extent. The procedure can be hence successfully used for a fully numerical pre-calibration of an existing engine, but also to develop innovative strategies for the valve lift design.

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