

A Modeling Study of Cyclic Dispersion Impact on Fuel Economy for a Small Size Turbocharged SI Engine

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ABSTRACT

In this paper, the results of an extensive experimental analysis regarding a twin-cylinder spark-ignition turbocharged engine are employed to build up an advanced 1D model, which includes the effects of cycle-by-cycle variations (CCVs) on the combustion process. Objective of the activity is to numerically estimate the CCV impact primarily on fuel consumption and knock behavior.

To this aim, the engine is experimentally characterized in terms of average performance parameters and CCVs at high and low load operation. In particular, both a spark advance and an air-to-fuel ratio (α) sweep are actuated. Acquired pressure signals are processed to estimate the rate of heat release and the main combustion events. Moreover, the Coefficient of Variation of IMEP (CoV_{IMEP}) and of in-cylinder peak pressure (CoV_{pmax}) are evaluated to quantify the cyclic dispersion and identify its dependency on peak pressure position.

In parallel, the experimentally tested engine is fully schematized in a 1D framework. The 1D model, developed in the GT-Power™ environment, makes use of user defined sub-models for the description of combustion, turbulence and knock phenomena. 1D analyses are carried out for various engine speeds, load levels, α ratios, and spark timings, without changing any tuning constant. In a first stage, the model is validated in terms of overall engine performance parameters, and ensemble-averaged pressure traces inside the cylinder, and within the intake and exhaust ducts, as well. A more detailed comparison is also performed with reference to the average rate of heat release in different operating conditions.

In a subsequent step, the effects of CCVs are introduced in the model in terms of Gaussian distributed modifications of the burning rate, predicted with reference to the ensemble-average operation. Consistently with the experimental data, applied burning rate are modified to simulate a train of pressure cycles statistically equivalent to the measured ones. The influence of the CCVs on the instantaneous peak position, air flow rate, and Indicated Specific Fuel Consumption (ISFC) is consequently investigated on a cycle-by-cycle basis, and compared to the average operation.

Numerical analyses show that CCVs cause a reduced ISFC penalty, which can be considered significant only in case of delayed combustions and increased CoVs. Knock limited spark advance is also identified with and without CCV, highlighting some additional fuel economy penalties.

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INTRODUCTION

The continuously growing demand for better performance, fuel economy, and reduced emissions of internal combustion engines is still motivating engine manufacturers towards the development of new technologies, engine architectures and advanced components.

During last years, however, a greater attention is focused on fuel consumption, to comply with the European legislative demands concerning the CO₂ emissions [1].

With reference to Spark-Ignition (SI) engines, a well assessed engine architecture is represented by the downsizing technique, consisting in the reduction of total displacement, and the employment

of a turbocharger to preserve the power output. This solution, possibly coupled to the adoption of Variable Valve Actuation (VVA) devices [4], guarantees a reduced Brake Specific Fuel Consumption at part load [5,6]. At the same time, however, fuel economy of downsized turbocharged SI engines is highly penalized at full load, since pre-ignitions or heavy knocking combustions may arise [7,8]. Pre-ignition mainly occurs at low speed and is usually avoided reducing the boost level. Knock phenomena, indeed, can be controlled by moving the combustion phasing very late in the expansion stroke. This, while penalizing the thermodynamic efficiency [9], causes an increase in the exhaust temperature and, in particular, at the turbine inlet. This additional concern is usually compensated by an enrichment of the air-fuel mixture, with further fuel economy penalties [10].

Research on this topic is moving toward the adoption of Low-Pressure cooled Exhaust Gas Recirculation (LP-EGR) circuits [11, 12, 13] or, alternatively, employing a Water Injection (WI) [14, 15, 16, 17]. Both these solutions allow to reduce over-fueling and obtain a more favorable combustion phasing. At low load, on the contrary, the road-map to obtain further fuel savings could be given moving the air-to-fuel ratio in the lean or extremely lean zone, eventually providing enhanced combustion technologies, such as the homogeneous charge compression ignition [18].

As known, however, these technologies (EGR, WI, lean mixtures, etc.) add further complications to the engine management system and introduce additional constraints, also related to the control of cyclic variability. A proper combination of the air-to-fuel ratio, EGR or WI level and Spark Advance (SA) must be realized [13] in fact to also keep under control cyclic dispersion phenomena, and ensure an optimal vehicle drivability.

Cycle-by-cycle variations (CCVs) are in fact commonly observed during the combustion process of a SI engine [19,20]. They mainly cause fluctuations in the rate of heat release, turning in oscillations in the Indicated Mean Effective Pressure (IMEP), power output, fuel consumption and exhaust emissions [21].

Detailed experimental analyses have been carried out to understand CCVs origins and impact on engine performance. As an example, optical techniques have been applied to follow the evolution of the combustion process, with a special focus on the early stages of flame propagation [23,24]. It is widely accepted in fact that CCVs arise during the early flame development, and then deeply influence the subsequent combustion phase [19,22]. A cyclic variability of the air-to-fuel ratio may be determined by deviations in injection timing and duration. They are in fact continuously adjusted by the electronic control unit, working in a closed-loop system, depending on a number of signals affected by time delays and noise disturbances. Moreover, within the cylinder, the mixing of air, fuel and residuals may be not complete, resulting in a not fully homogeneous mixture at the spark time. The random nature of the spatial and temporal fluctuations of the turbulent flow field inside the cylinder also determines cycle-by-cycle variations of the fluid-dynamic conditions near the spark electrodes. Although the complex interaction of the above phenomena is not yet fully understood, it is widely recognized that cyclic variability especially increases in the cold starting, idling, low load, and diluted/

lean mixture operation, where the burning velocity is slow and flame development is difficult. Due to the above considerations, experimental and theoretical studies on cyclic variability have been mainly focused on lean mixture operation and low load conditions [25]. Cyclic variability however plays a fundamental role at high load, too, where knock most likely occurs [26, 27, 28]. The optimal Knock Limited Spark Advance (KLSA) is in fact usually selected taking into account cycle-by-cycle variations [29,30].

Current literature is plenty of papers trying to characterize cyclic dispersion under a numerical point of view, as well. Referring to most detailed 3D models, the employment of a LES approach may give further insights in understanding cycle-by-cycle variations basic origins and source mechanisms [31,32]. This methodology, however, is still limited by computational resources, and the possibility to estimate a statistically significant number of engine cycles is still unfeasible. In 1D models, on the other hand, cyclic dispersion cannot be directly predicted, but has to be externally introduced; its “intensity” has to also be tuned to properly reproduce the observed or LES computed fluctuations. As an example, in [33,34], random perturbations of the burning rate, assigned in terms of a Wiebe function, are introduced in the model. A more “physical” approach is followed in [35], based on the employment of phenomenological models for the description of the in-cylinder turbulence and combustion processes. In [36], CCVs are introduced through a random modification of the turbulence production during the intake phase. Similarly, in [37, 38, 39] CCVs are imposed through a perturbation of the additional parameters, such as: turbulence intensity, length scale, air-to-fuel ratio, residual fraction, etc. Although the above approach may help to physically estimate the effect of each parameter on combustion fluctuations, in any case, the need to select a proper distribution range for each perturbed parameter still remains, and an extensive tuning procedure of the CCV sub-model is required.

In this paper, a sort of “hybrid” procedure is developed. Following a recently proposed two-step analysis [40,41], the burning rate of the average cycle is firstly predicted by a phenomenological turbulent combustion model. As better explained in the following paragraphs, the average burning rate is then properly “stretched” to generate trains of pressure cycles characterized by the same Coefficient of Variation of maximum in-cylinder pressure ($CoV_{p_{max}}$) observed in the experiments. In this way, while preserving the predictive capability of the combustion model, just a single parameter, namely the burning rate stretch factor, has to be tuned in the CCV sub-model. The latter is automatically identified at run-time to reproduce an assigned $CoV_{p_{max}}$. This choice, in addition, is well-matched with the objective of the activity, consisting in a numerical estimation of the CCV impact primarily on fuel consumption and knock behavior.

To this aim, results of a wide experimental campaign are preliminary utilized to validate the 1D model in terms of overall engine performance parameters, and ensemble-averaged pressures. In a subsequent step, the CCV influence on Indicated Specific Fuel Consumption (ISFC) is assessed at high and part-load, and under knock limited operation, as well.

The paper is organized as follows: a paragraph illustrating the experimental setup, the in-cylinder measurements, and thermodynamic analyses is firstly presented; then the 1D model development and validation is described. A third paragraph explains in detail the cycle-by-cycle variations model, and results are presented under different operating conditions.

EXPERIMENTAL SETUP AND TESTS

The engine under investigation is a PFI downsized twin-cylinder SI VVA engine (Table 1). Main operating parameters, such as brake torque, fuel rate and air-to-fuel ratio are acquired. In addition, each cylinder is equipped with a piezo-quartz pressure transducer (accuracy of $\pm 0.1\%$) to detect in-cylinder pressure signal.

Furthermore, the instantaneous pressure traces at compressor outlet, intake plenum and turbine inlet are measured through piezo-resistive low pressure indicating sensors. Instantaneous pressure signals are acquired for 1000 consecutive cycles and the ensemble-averages are computed by an automatic post-processing tool. Concerning in-cylinder indicated analysis, the cycle-by-cycle variations are statistically characterized in terms of $CoV_{p_{max}}$ and CoV_{IMEP} .

Table 1. Engine main characteristics

Twin-cylinder PFI Engine	
Displaced volume, cm ³	875.4
Layout	4 valve/cyl
Bore, mm	80.5
Stroke, mm	86
Connecting rod length, mm	136.85
Compression ratio	10:1
Maximum brake power, kW	63.7@5500 rpm
Maximum brake torque, Nm	146.7@2000 rpm

The engine is investigated under various operating conditions, at high and part load, and for various spark timings and air-to-fuel ratios. In particular, three high load operating points, at the speeds of 2160, 3500 and 4500 rpm, are selected, with a load level included in the range 13.5-17 bar IMEP, and a relative air-to-fuel ratio ($\lambda = \alpha/\alpha_{st}$) in the range 0.92-1.00. For each point, a spark sweep is performed, with the most retarded spark timing selected to ensure an acceptable combustion stability (CoV_{IMEP} below 6%), and with the most advanced spark limited in order to avoid the onset of knocking combustions. Part load operation is also inspected, at the speeds of 2500, 3000, 3500 and 4000 rpm, imposing a stoichiometric air/fuel mixture and a load level in the range 4.7-6.8 bar IMEP. Once again, a spark sweep is applied for each tested part load point. The overall test matrix, consisting of 71 operating conditions, is shown in Table 2.

Table 2. Test conditions

	Speed, rpm	SA, CAD ATDC	λ , -	IMEP, bar
High load	2160	+5.5 to -7.50	0.96 – 1.00	13.5 – 17.0
	3500	-8.0 to -16.0	0.92 – 1.00	14.5 – 16.3
	4500	-21.0 to -16.5	0.96 – 1.00	14.6 – 15.2
Part Load	2500	-42.0 to -26.0	1.00	4.7 – 5.0
	3000	-40.0 to -24.0	1.00	5.1 – 5.3
	3500	-36.5 to -20.5	1.00	5.7 – 5.9
	4000	-36.5 to -20.5	1.00	6.4 – 6.8

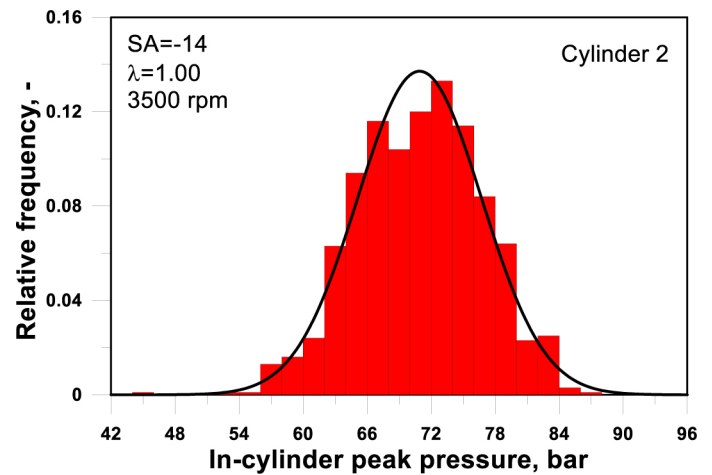


Figure 1. Typical distribution of experimental maximum in-cylinder pressure

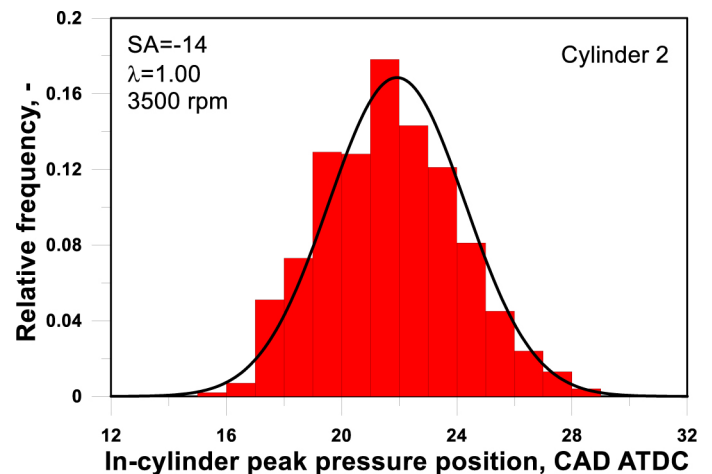


Figure 2. Typical distribution of experimental peak pressure position

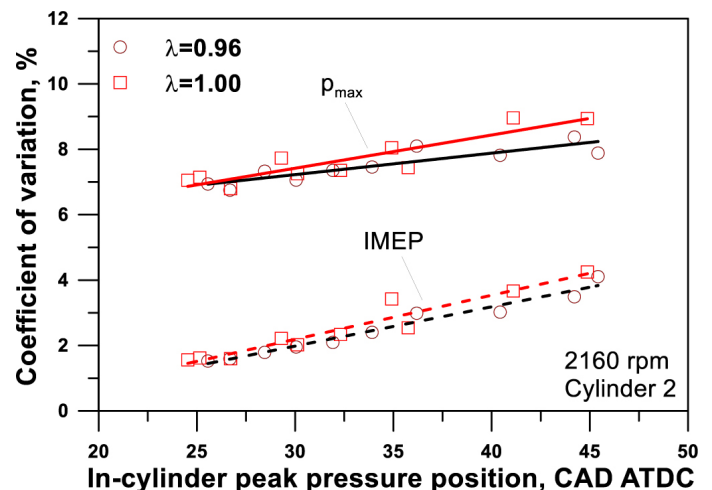
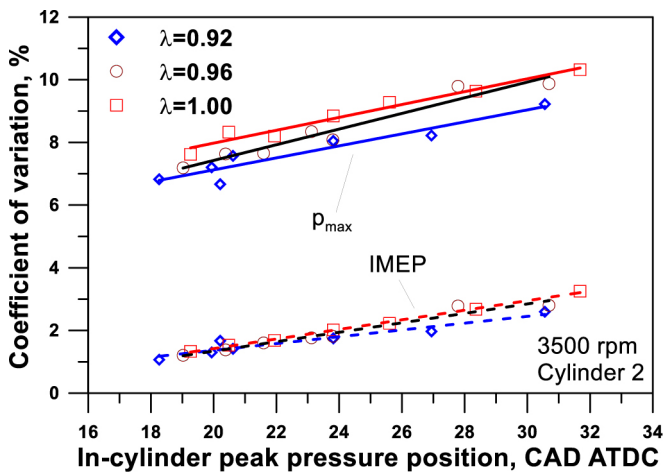
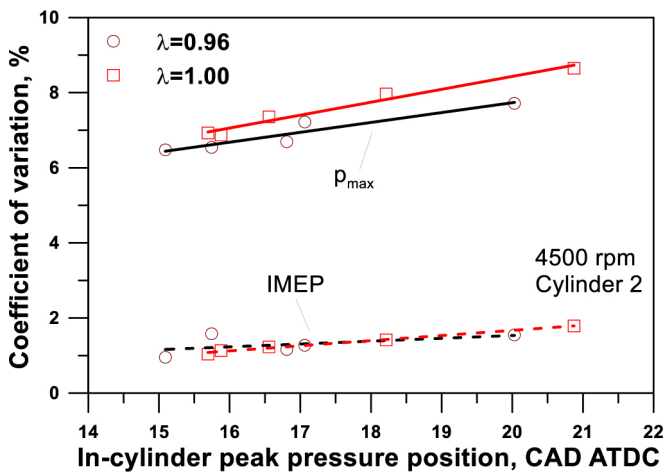
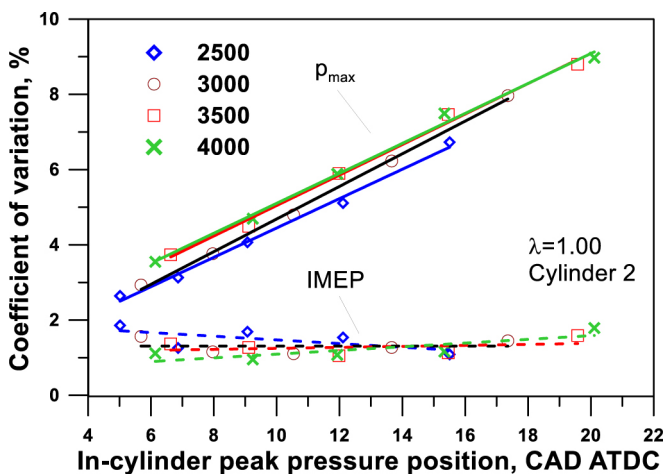


Figure 3. $CoV_{p_{max}}$ and CoV_{IMEP} for cylinder 2 at 2160 rpm

Figure 4. $CoV_{p_{max}}$ and CoV_{IMEP} for cylinder 2 at 3500 rpmFigure 5. $CoV_{p_{max}}$ and CoV_{IMEP} for cylinder 2 at 4500 rpmFigure 6. $CoV_{p_{max}}$ and CoV_{IMEP} for cylinder 2 at part load

The statistical analysis of the indicated parameters, namely peak pressure and related crank angle position, points out that their distributions are well fitted by a Gaussian function. As an example, in Figure 1 and Figure 2, the above data are shown with reference to cylinder #2, for a high load operating point at 3500 rpm. A similar behavior is observed for the other investigated points and in the other cylinder. As pointed out in a number of works [19,22,29], the cyclic variability can be quantified in terms of CoV of IMEP and maximum in-cylinder pressure. In Figure 3, Figure 4, Figure 5, Figure 6, the

$CoV_{p_{max}}$ and CoV_{IMEP} are depicted for different λ values, as a function of the combustion phasing. The latter, usually evaluated by the crank angle of 50% mass burning rate (MFB_{50}) is here expressed in terms of peak pressure position.

According to cited references, the $CoV_{p_{max}}$ is always considerably higher than CoV_{IMEP} . Since the spark advance moves away from Maximum Brake Torque (MBT), as expected, both parameters increase delaying the combustion phasing, especially at high load (Figure 3, Figure 4, Figure 5). The latter would occur for increased (more negative) SA values with respect to the experimentally investigated ones, since, as said, the engine operation is limited by knock onset. At part-load, indeed, knock absence allows to select a close-to-MBT spark timing, realizing an optimal peak pressure position around 12-14 CAD ATDC. In these cases, a negligible variation of the CoV_{IMEP} is observed (Figure 6). Nevertheless, spark timing effect on CCVs is still present, as proved by the substantial increase in the $CoV_{p_{max}}$.

MODEL DESCRIPTION AND VALIDATION

The tested engine is carefully schematized in the GT-Power environment, as shown in Figure 7. The engine model takes into account a detailed description of the intake and exhaust piping, including the air-box, the turbocharger, the intercooler, the valves and the cylinders. The geometrical and fluid-dynamic schematization is based on the data provided by the engine manufacturer, while the turbocharger is modeled by a classic map based approach.

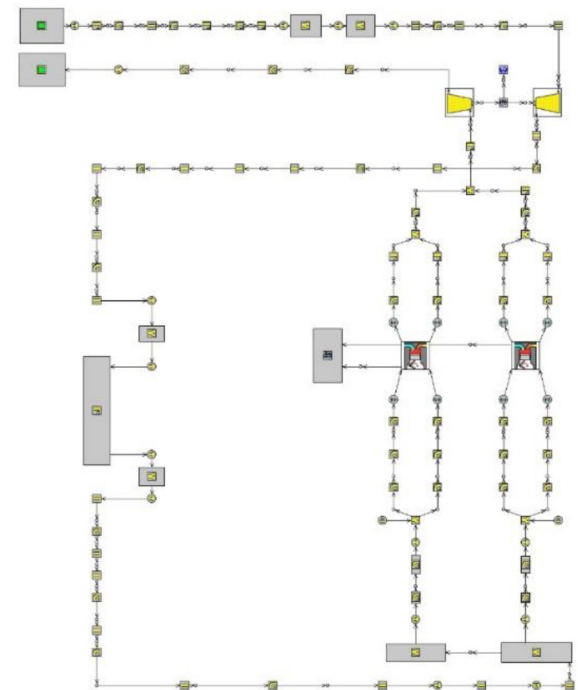


Figure 7. Layout of the 1D engine model in GT-Power environment

The experimentally actuated air-to-fuel ratio, spark advance and valve strategy are imposed as input data in the simulations. In the case of part load operation, the throttle is adjusted to match the experimentally derived pressure level at intake manifold, while it is fully opened for the high load cases. The waste-gate opening is adjusted by a PID controller to match the measured air flow rate.