

Analysis of Knock Tendency in a Small VVA Turbocharged Engine Based on Integrated 1D-3D Simulations and Auto-Regressive Technique

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ABSTRACT

In the present paper, two different methodologies are adopted and critically integrated to analyze the knock behavior of a last generation small size spark ignition (SI) turbocharged VVA engine. Particularly, two full load operating points are selected, exhibiting relevant differences in terms of knock proximity. On one side, a knock investigation is carried out by means of an Auto-Regressive technique (AR model) to process experimental in-cylinder pressure signals. This mathematical procedure is used to estimate the statistical distribution of knocking cycles and provide a validation of the following 1D-3D knock investigations.

On the other side, an integrated numerical approach is set up, based on the synergic use of 1D and 3D simulation tools. The 1D engine model is developed within the commercial software GT-Power™. It is used to provide time-varying boundary conditions (BCs) for the 3D code, Star-CD™. Particularly, information between the two simulation tools are at first exchanged under motored conditions to tune an "in-house developed" turbulence sub-model included in the 1D software. 1D results are then validated against the experimental data under fired full load operations, by employing a further "in-house developed" combustion sub-model. BCs are finally passed back to the 3D code to carry out a detailed knock analysis for two full load points, namely 2100 and 4000 rpm. In particular, the knock intensity is predicted, for experimentally actuated and earlier spark advances, and the results are qualitatively compared to the AR model outcomes.

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INTRODUCTION

The development of current internal combustion engines (ICEs) has to comply with several simultaneous requirements. On one hand, in fact, customers are becoming increasingly sensitive to fuel consumption levels, mainly because of the uprising fuel costs; on the other hand, car manufacturers are requested not to reduce, or even to increase, power/torque performance, in order to guarantee the desired level of drivability and fun-to-drive. Furthermore, both tasks have to comply with increasingly stringent pollutants emission limitations. To fulfill the above listed - often conflicting - issues, ICE architecture is becoming more and more complex and new engine concepts are under continuous investigation and development [1]. Concerning spark ignition (SI) ICEs, advanced technical solutions, involving the combination of a

fully flexible actuation system of the intake valves (Variable Valve Actuation - VVA), a turbocharging system and gasoline direct injection are under continuous research and development [2,3]. Particularly, a VVA system allows the designers to combine a fine tuning of the pressure waves in the intake system at full load (maximum cylinder filling) and a flexible load control without throttling the intake system (thus reducing fuel consumption). On one hand, the use of either a turbocharging or a supercharging system allows for a further reduction of the brake specific fuel consumption at low- and medium-loads through the reduction of engine displacement (downsizing), while preserving (or even increasing) peak engine torque at high load / low engine speed and peak engine power at high load / high engine speed. On the other hand, the consequent higher pressure and temperature levels reached in the cylinders contribute to increase the engine knock tendency.

Knock arises when a portion of the fuel charge within the combustion chamber spontaneously ignites after the spark event, but before it has been reached by the flame front propagating from the spark plug [4, 5]. This phenomenon most likely occurs within the air/fuel charge portion that burns late during the combustion process (end-gas autoignition). A simple way to avoid knock is to limit engine compression ratio and spark advance, as well. This however significantly reduces the potential in gaining a lower fuel consumption and a higher power output. Increase in the heat removal by the cooling circuit to lower down the combustion chamber wall temperatures might be an option [6, 7], but this would result in a decreased engine thermal efficiency and in a relevant increase in the amount of unburnt hydrocarbons (UHC) at the exhaust.

As a consequence, an ideal engine control system should be able to operate as close as possible to knocking, without really moving inside it. The above consideration is remarkably true in a “downsized” turbocharged spark-ignition engine, like the one considered in the present work. To avoid knock, in fact, most turbocharged SI engines operate at full load with a rich air/fuel ratio and a very delayed spark. At low speeds, a negative value of the spark advance is sometimes required (the combustion begins after the TDC). This occurrence, although strongly penalizing the fuel consumption, is the only solution in these operating conditions to limit the peak temperature and pressure during combustion and avoid the knock. An additional problem is related to cycle-to-cycle variations, which are commonly observed during the combustion process of an SI engine [8, 9]. They mainly cause unpredictable and stochastic fluctuations in the rate of heat release, turning in fluctuations in the in-cylinder pressure cycle. As a consequence, faster-than-average cycles are most likely to knock. Therefore the optimal “knock-limited” spark advance must be accurately chosen taking into account cycle-by-cycle variations.

During the engine calibration phase, a typical problem in the definition of the borderline spark timing consists in the definition of a proper metrics for the estimation of a knock index [10]. As known, standard methods are based on the analysis of the in-cylinder pressure signal. The signal is usually band-pass filtered and processed, in order to obtain an index that is then compared to a predefined threshold level. The latter however depends on engine features, transducer position [11,12] and operating conditions [13], as well. Its determination comes out from a number of consideration based on extensive engine calibration at the test-bench, manufacturer know-how, durability tests, etc.

In order to deepen the understanding of knock onset and theoretically support the definition of proper threshold levels, in this paper, a knock analysis of a small turbocharged twin-cylinder VVA engine is carried out following two different approaches. The first methodology consists of the use of statistical analysis by applying an Auto-Regressive (AR) technique to in-cylinder pressure measurements. The approach, extensively described in [14], allows to estimate the

distribution of a selected knock index (noise variance) on a wide set of consecutively acquired pressure cycles, and to evaluate the knock safety margin for the considered spark timing. More precisely, knocking cycles are identified by a parametric model coupled to a predictive algorithm in order to select the appropriate coefficients able to describe the system output. The second methodology consists of a hierarchical 1D/3D modelling approach, where the first step is the schematization of the whole engine in the 1D code GT-Power™. Preliminary simulations are carried out under motored operations to provide reliable boundary conditions (BCs) for the subsequent 3D-CFD analyses. The 3D results, in terms of temporal evolution and spatially averaged in-cylinder flow fields, are used to calibrate an “in-house developed” turbulence sub-model, included as a “user routine” in the employed 1D commercial code [15]. Then, the engine performance are predicted by the calibrated 1D simulation for full load (fired) operations, using a further “in-house developed” sub-model describing the combustion process (fractal combustion model). The 1D engine model is validated against the overall engine performance parameters (power, torque, BSFC, etc.) and ensemble averaged in-cylinder pressure cycles [16]. Finally, a detailed 3D-CFD investigation of the in-cylinder phenomena, employing the 1D code-derived BCs, is conducted, both for the experimentally actuated spark advance and for more advanced values.

THE ENGINE

The engine is a small-size turbocharged VVA twin-cylinder engine characterized by an innovative electro-hydraulic intake valve actuation system [17] which allows to control the air intake without substantially throttling the intake system.

Table 1. Engine main features

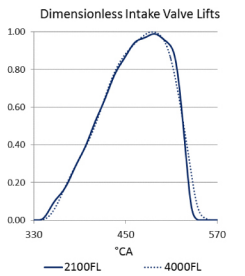
Model	2 cylinders, 8 valves	
Displacement	875 cm ³	
Stroke/Bore	86 mm / 80.5 mm	
Connecting rod length	136.85 mm	
Compression Ratio	9.9	
Max Brake Power	64.6 kW @ 5500 rpm	
Max Brake Torque	146.1 Nm @ 2500 rpm	

Table 1 reports some basic engine features, as well as the dimensionless intake lift profiles for the investigated engine conditions, corresponding to the full-load operation at 2100 rpm and 4000 rpm, respectively.

EXPERIMENTAL ACTIVITY

The engine summarized in Table 1 is installed at the test-bench, and full load operation is analyzed in the speed range (1500-5500 rpm). The intake valve closure is determined in order to guarantee the maximum cylinder filling. The overall

performance are, however, limited by some constraints related to the boost pressure level and to the turbocharger speed for low and high speeds, respectively. Moreover, a very rich air/fuel ratio is specified to respect the limits on the maximum allowable temperature of the turbine blades. The spark advance is set at each speed in order to prevent audible detonating combustions. Acquired data include air flow, fuel flow, torque, emissions, etc.. Instantaneous in-cylinder pressure measurements are carried out in both cylinders, too. In particular, 200 consecutive pressure cycles are acquired for each operating condition. A sampling resolution of 0.5 crank angles is adopted at each speed. A higher sampling frequency (0.1°) is instead specified during combustion angular window (−10°, 75°).

Cycle-by-cycle WOT pressure measurements at 2100 and 4000 rpm are shown in Figure 1 (a) and (b), superimposed with ensemble averaged ones. A relevant cyclic dispersion can be noticed, with differences on the peak pressure of about 20-30 bar, and an IMEP Coefficient of Variation (CoV - defined as the ratio between the standard deviation of the IMEP and the related average level) around 5% in both cases. In Figure 1 (c), two individual cycles are selected and compared to highlight the presence (see the zoomed areas around combustion peak pressure) of high frequency pressure fluctuations, indicating the local occurrence of soft-knock phenomena. Although the magnitude of the above fluctuations is higher at 4000 rpm, it cannot be concluded that a more intense knock is actually taking place. At high engine speeds, in fact, a background knock-unrelated pressure fluctuation is present, mainly due to the rapidly changing in-cylinder volume (see the zoomed area just before the TDC). The above disturbances may amplify the knock-induced fluctuations, causing increased knock metrics at high speed. Actually, the most critical operating conditions, which may cause severe engine damages, verify at low-speed, since end-gases are exposed for a longer time to high temperature. A delayed spark timing is in fact specified at low speed (see Figure 1 (a) and (b)). Due to the above considerations, in order to distinguish knock and knock-free operations, a speed-increased threshold level is usually selected, whichever is the employed knock metrics [18].

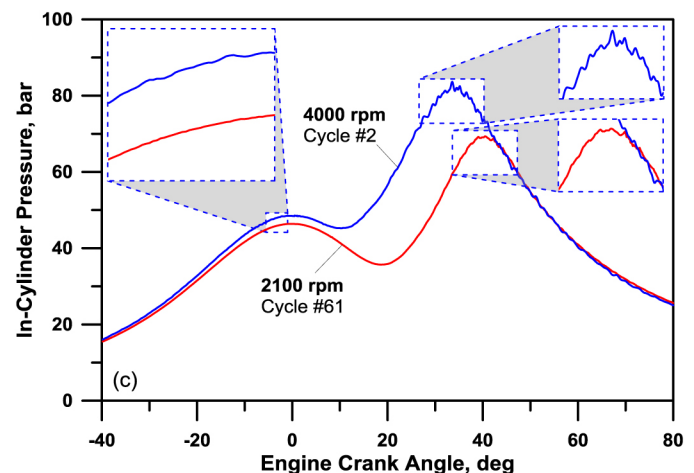
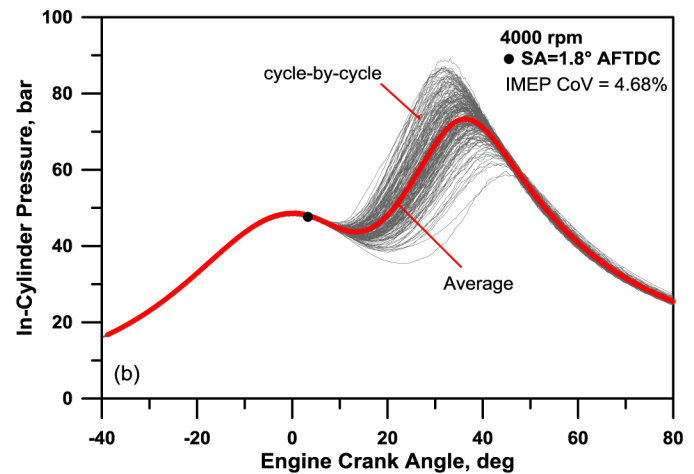
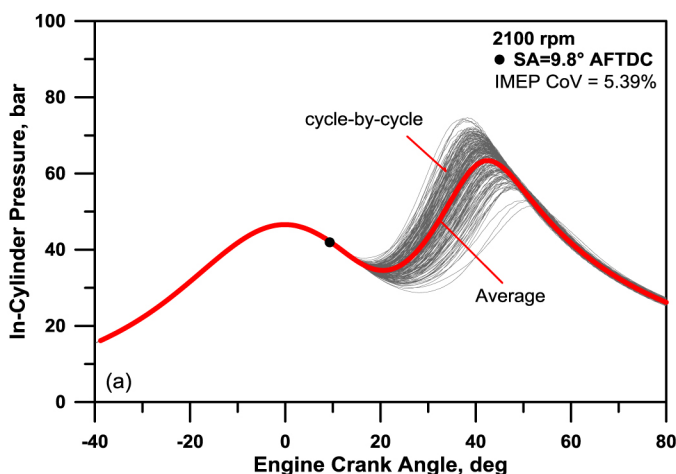


Figure 1. (cont.) Average and instantaneous pressure cycles at 2100 rpm (a) and 4000 rpm (b); high frequency pressure oscillations (c) [14]

THE ADOPTED METHODOLOGIES

Knock Detection Based on AR Technique

Since 3D knock analyses reported in the following paragraphs only refer to the ensemble averaged cycle computation, it is important to collect preliminary experimental information on the effect of cyclic dispersion on knock occurrence on individual cycles.

To this aim, an innovative knock detection technique is applied to the sets of experimentally acquired pressure cycles. The procedure is based on an Auto-Regressive (AR) model, which was proven to be preferable [14] to standard knock detection techniques, based on a band-pass filtering of the acquired pressure cycles [19]. The AR model [20] is able to predict the in-cylinder pressure at time t , p_t , basing on its own previous values p_{t-i} plus an error function e_t , that is ascribed to the knock occurrence:

$$p_t = a_1 p_{t-1} + \dots + a_n p_{t-n} + e_t$$

Figure 1. Average and instantaneous pressure cycles at 2100 rpm (a) and 4000 rpm (b); high frequency pressure oscillations (c) [14]

The $[a_1, \dots, a_n]$ coefficients are the AR parameters, that relate the pressure at time t to pressure values at previous times (hence auto-regressive). The goal of the AR model is hence to select the a_i coefficients in order to predict the pressure signal p_t . The first important step of the AR model set up is the specification of a suitable order (number of coefficients) for the model [20, 21]. Generally, the AR order (n) is unknown and it is usually selected to minimize the error function (e_t). However, a suitable model should be made up of the smallest number of variables. Unnecessarily high orders, in fact, introduce spurious artifacts in the resulting response. For a prefixed order, a variable set of a_i coefficients is automatically selected to minimize the error between the actual and predicted outputs. Their estimation is based on least-squares minimization criteria.

Once the AR model has been set up, the variance of the error in the pressure prediction (noise variance, σ^2) is supposed to be proportional to the intensity of knock disturbances. Namely, in a knock-free operation, the noise function is expected to exhibit a close-to-zero variance. Knocking process indeed introduces relevant changes in the pressure signal, turning in a sudden increase in the noise variance. By monitoring the noise variance level it is definitely possible to detect the knock occurrence.

1D Analysis and 1D/3D Integration

In this section, a brief summary of the integrated 1D/3D methodology is reported. A detailed description is included in [15,16].

The geometrical data provided by the engine manufacturer are used to build the 1D model of the whole engine and to carry out preliminary 1D simulations under motored operations. In particular, four operating points are selected, characterized by different rotational speeds and valve lift profiles typical of both full and part load operations (Table 2).

Table 2. Tested engine operating points

Case #	Engine speed, rpm	Intake valve closure angle, ° AFTDC
1	2500	560
2	5100	548
3	2000	462
4	3000	448

Then, multi-cycle motored 3D analyses, applying 1D computed time-varying pressure and temperature traces as boundary conditions, are performed. Detailed information about the in-cylinder mean and turbulent flow fields are extracted from the 3D analyses over a whole engine cycle. The latter are utilized to calibrate a 1D turbulence model coupled to the phenomenological combustion model later resumed. In particular, the mass averaged integral length scale, specific turbulent kinetic energy and mean flow kinetic energy are calculated all over the 3D domain.

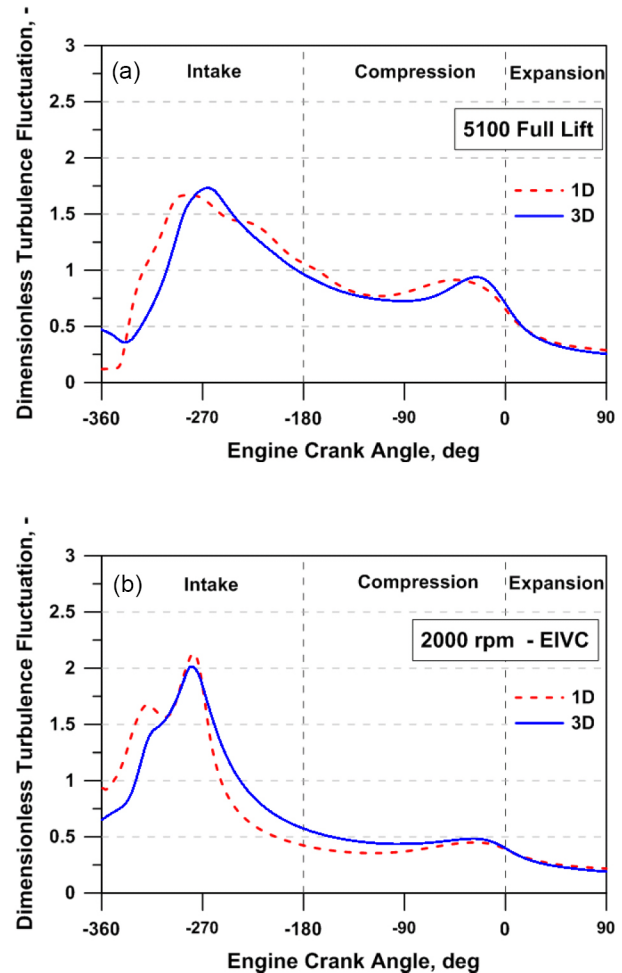


Figure 2. Dimensionless turbulence fluctuation 1D/3D comparison at (a) 5100 full lift and (b) 2000 rpm EIVC [15]

As an example, in Figure 2 the comparison between the 3D and 1D turbulence intensity results, normalized with the mean piston speed, is shown for the cases 2 and 3 of Table 2. It can be appreciated how the 1D turbulence sub-model outcomes adequately resemble the 3D-derived in-cylinder turbulence levels during the combustion-critical cycle portions, i.e. intake, compression and expansion phases. In particular, the agreement is very satisfactory around the compression TDC. It is worthwhile to remark that a unique set of tuning constants feasible for all the considered points is identified.

Once the turbulence sub-model is tuned, the 1D engine simulation is carried out under fired operations. Specifically, the combustion process is modeled in the 1D analyses by a “fractal combustion model” [16, 22], 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 combustion process description is properly linked to the previously tuned in-cylinder turbulence model. The combustion model tuning constants are selected in order to fit the in-cylinder pressure cycles at full load operation. A very good agreement between the numerical and experimental results is

obtained [16]. The model consistency, in terms of fuel consumption predictiveness, has been proved at part load, too [16]. Once validated, the 1D model can provide reliable boundary conditions also under fired operations, to be adopted in the subsequent 3D analyses.

In the next step, the 3D combustion model is validated against the measured in-cylinder pressure cycles for the experimentally actuated spark advance. Detailed 3D knock analyses are then carried out for earlier spark advances. Although knock is strongly related to instantaneous and cycle-dependent events, 3D-CFD simulations are performed within the RANS framework, and are therefore focused on the characterization of “ensemble averaged” fields and behaviors. Since experimental data with increased SA are not available, 3D results are qualitatively compared, in terms of probability of knock inception, to the Noise Variance levels provided by the AR model.

In Figure 3, a flowchart of the described integrated 1D/3D and AR methodologies is fully summarized.

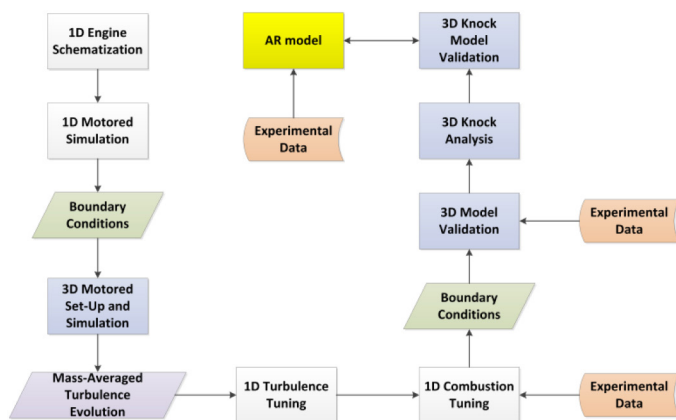


Figure 3. Flowchart of the integrated 1D/3D methodology

CFD-3D KNOCK ANALYSIS SETUP

The computational domain (see Figure 4) covers the combustion chamber together with the port portions within the engine head. Information about the 3D CFD model setup can be found in [15] and are not recalled here for the sake of brevity.

As reported in the previous section, time dependent pressure and temperature boundary conditions are derived from the 1D model of the whole engine. In addition to the previously presented CFD setup, a detailed methodology is now adopted for the analysis of abnormal combustion events and particularly knock. The methodology is based on a tabulated approach for the actual autoignition chemistry of real engine fuels proposed by the authors in [8, 9]. In the modeling framework, the combustion chamber is considered as a two-zone region, the former being the burning region, which is treated by the ECFM-3Z [23] combustion model, and the latter being the unburnt gas region, in which the progress of the autoignition

chemistry is tracked at a cell-wise level. Tracking is done through an off-line generated map of ignition delays based on pressure, temperature, mixture strength and EGR, computed using a constant-pressure reactor and based on the NICE PRF scheme for a RON95 gasoline [24].

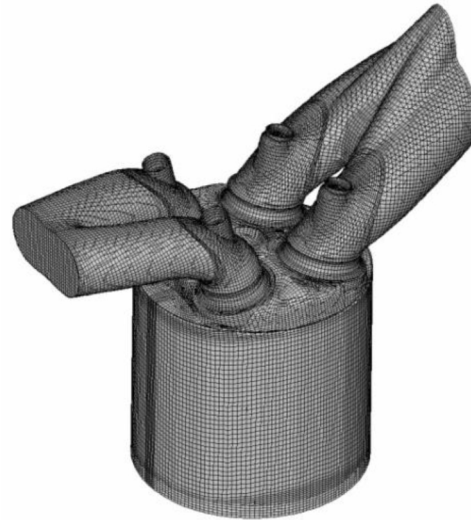


Figure 4. CFD-3D computational grid

A multi-linear interpolating routine is used on a cell-wise basis to accurately estimate the autoignition progress, which is tracked by a passive transported precursor scalar based on the correlation by Lafossas [25]:

$$\frac{dY_{IG}}{dt} = Y_{TF} \cdot \sqrt{1 + \frac{4(1-\tau)}{\tau^2} \cdot \frac{Y_{IG}}{Y_{TF}}} \quad (2)$$

Autoignition occurs as soon as the precursor concentration Y_{IG} locally equates the fuel tracer concentration Y_{TF} . It is then possible to define a Knock Tolerance KT scalar function, as the local difference between the fuel tracer and the intermediate species:

$$KT(\vec{x}, t) = Y_{TF}(\vec{x}, t) - Y_{IG}(\vec{x}, t) \quad (3)$$

The above defined scalar function is then post-processed over the whole combustion crank angle range using a user-implemented tool based on Matlab-Simulink, in order to better understand the knock tendency of the investigated engine operations. Particularly, local physical and chemical state is analyzed through the use of KT instantaneous PDF distributions and scatter plots [26].