

Relating Knocking Combustions Effects to Measurable Data

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ABSTRACT

Knocking combustions heavily influence the efficiency of Spark Ignition engines, limiting the compression ratio and sometimes preventing the use of Maximum Brake Torque (MBT) Spark Advance (SA).

A detailed analysis of knocking events can help in improving the engine performance and diagnostic strategies. An effective way is to use advanced 3D Computational Fluid Dynamics (CFD) simulation for the analysis and prediction of the combustion process. The standard 3D CFD approach based on RANS (Reynolds Averaged Navier Stokes) equations allows the analysis of the average engine cycle. However, the knocking phenomenon is heavily affected by the Cycle to Cycle Variation (CCV): the effects of CCV on knocking combustions are then taken into account, maintaining a RANS CFD approach, while representing a complex running condition, where knock intensity changes from cycle to cycle. The focus of the numerical methodology is the statistical evaluation of the local air-to-fuel and turbulence distribution at the spark plugs and their correlation with the variability of the initial stages of combustion.

CFD simulations have been used to reproduce knock effect on the in-cylinder pressure trace. For this purpose, the CFD model has been validated, proving its ability to predict the combustion evolution with respect to SA variations, from non-knocking up to heavy knocking conditions.

The CFD model allowed relating measurable data (i.e., the simulated cylinder pressure signal) to other factors, representative of the phenomena actually taking place during knocking combustions: for each cell used in the CFD simulation, information such as pressure, heat release, etc. are available and can be traced over the angular domain. Furthermore, the analysis refers to hundredths of engine cycles, leading to a comprehensive correlation between standard cylinder pressure-based knock indexes and other indexes (only available in a simulation environment), more representative of the actual knock intensity.

CITATION: Corti, E., Forte, C., Bianchi, G., and Moro, D., "Relating Knocking Combustions Effects to Measurable Data," *SAE Int. J. Engines* 8(5):2015, doi:10.4271/2015-24-2429.

INTRODUCTION

The continuous demand of high performance and low emissions vehicles leads the manufactures to push the operating range of the engine within the knocking region. The capability of accurately simulating the engine behavior in such a critical condition is essential, not only in the design stage, but also for the definition of knock diagnosis strategies. The aim of the paper is to assess how commonly used (cylinder pressure-based) knock indexes are related to the actual knock intensity.

The knock phenomenon is not entirely deterministic: knock tendency is obviously related to compression ratio, spark advance, inlet temperature, etc., but a stochastic component is also present, leading to both knocking and non-knocking engine cycles for the same running condition [1].

Both the deterministic and stochastic elements must be taken into account by the simulation setup, in order to be representative of measurements based on the cylinder pressure signal: knock intensity

evaluations are often based on the statistical analysis of many engine cycles, thus a multi-cycle simulation is mandatory to fulfill the objective of the present analysis.

The modeling of Cycle-to-Cycle Variation plays a crucial role in the simulation of combustion on a multi-cycle basis. Often CCV modeling consists in a forced perturbation of the combustion evolution in the chamber: many methodologies today available in the literature are based on this mechanism, but the key issue is the way these perturbations are implemented. The imposition of a change in the combustion evolution has to be based on actual engine behavior, in order for the model to be predictive. Different approaches are possible for the introduction of a combustion perturbation generating CCV: from simple models based on the Monte Carlo approach [2], to two-zone thermodynamic models combined with flame propagation sub-models [3], from 1D/0D simulation methodologies fed by experimental and numerical (3D) data for the perturbation [4], to analysis based on the Large Eddy Simulations (LES) [5, 6, 7, 8]. Other interesting contributions can be found in [2, 10, 11, 12].

In a previous work [13], the authors investigated the root causes of CCV increase associated to leaner combustion strategies, by means of a joint numerical and experimental approach: it was shown that the combustion sensitivity to the initial perturbation of the mixture air index (equivalence ratio) at the spark location and to the level of in-cylinder air index homogeneity increases due to the lower laminar combustion speed of the leaner mixtures. The relationship between the variability of IMEP and the fluctuation of local mixture air index (λ) was clearly showed in the work of Ikeda et al. [14], where the chemiluminescence technique was used to analyze the mixture homogeneity and the composition at the spark plug for a high performance engine.

In the present work the authors use a RANS methodology for the evaluation of the combustion instabilities based on the mixture composition at the spark plug. The approach, presented in a previous paper [15], was introduced to compare single spark and twin spark combustions, focusing on the interaction of Cycle-to-Cycle Variation with knock intensity, over hundredths of engine cycles. The simulated cycles shows good agreement with cylinder pressure traces measured at the transducer position: IMEP, MFB50 and MAPO absolute values and distributions are comparable to experimental data. The same simulation approach is used in this paper to compare knocking combustions measurable effects (e.g., local pressure at the transducer) with other factors, representative of the phenomena actually taking place in the combustion chamber.

A selection of significant knock intensity parameters both in terms of cause (instantaneous fuel burning rate) and effects (local pressure and heat transfer on the piston surface) has been carried out in order to assess the cylinder pressure signal sensitivity to the phenomenon.

Many indexes are available in the literature for the evaluation of knock intensity [16]: the diagnosis is often carried out based on the so called 'knock amplitude', i.e., the maximum amplitude of the band-pass filtered cylinder pressure signal, considered over a knock-significant window of the engine cycle. The knock index resulting from this definition is often referred to as MAPO (Maximum Amplitude of Pressure Oscillations):

$$MAPO = \max(|p_{fi}|)_{i=1}^{i=N} \quad (1)$$

In order to assess the actual knock intensity several hundredth of engine cycles are necessary ([17]), and the cycle based MAPO index must be statistically analyzed. A possible approach ([18]) is to monitor a population of MAPO indexes, sorting their values in ascending order, and finally evaluating the intensity corresponding to a given percentile (e.g., 95%). The minimum number of engine cycles to include in the buffer can be decreased using probability density function (pdf) interpolation, as suggested by Naber et al. [19]. An alternative possibility suggested in [20] is to group the MAPO values based on intensity intervals, associating a weight factor to each interval, and then summing up the values to evaluate the knock severity of the operating condition.

Knock damages can be related to mechanical stress, causing erosion on the top land and first ring groove, or even on the second and third ring, depending on the intensity ([20]). Other mechanism however can be identified: Nates [21] investigated the contribution of thermal stresses to knock-induced erosion damage by modeling the temperature profile within the walls. Many authors studied the effect of the direct shock pressures on erosion, with minor attention on the thermal stress effect. Results demonstrate that knocking combustion causes thermal stresses between 2 to 3 times greater than that of non-knocking combustion. In [22] the authors show that MAPO is related non-linearly to the reduction in the piston ring gap, taken as a parameter for the damage caused by excessive heat flux. In [23] it is shown that the correlation coefficient between the erosion damage index and the Pressure value at the Knock Onset is good, leading to the consideration that MAPO indexes should be coupled to the corresponding peak pressure, in order to be representative of knock damages. Other works ([24, 25, 26, 27, 28]) show that during knocking cycles high Peak Heat Flux values can be measured: the correlation coefficient between peak heat values and knock intensity indexes is low for light knock, far higher for heavy knock, and strongly depends on the transducer position ([24]). In [26] Grandin et al. show that for knock intensities above 0.2MPa the heat flux is increased remarkably, and indexes such as Knock Onset angle and Knock Intensity (i.e., MAPO) are well related to the peak heat flux.

Local pressure on the piston surface, local heat flux transferred to the piston and instantaneous burning rate are available outputs of the CFD simulation that can be considered representative of actual knock intensity. The simulation of local pressure corresponding to the transducer position allows comparing the measurable knock-severity to the local one (piston surface).

EXPERIMENTAL SETUP

The engine under analysis is a Ducati-high performance unit, whose characteristics and running conditions are reported in Tab. 1. In order to increase the volumetric efficiency at high engine speed, a wide intake valve opening duration has been set, which makes the actual compression ratio lower than the geometric value. However, due to the ram and pressure waves tuning effects knocking combustions can happen even at high engine speed.

The engine was tested on the test bench equipped with cylinder pressure transducers. The pressure traces of 300 engine cycles have been recorded for each engine operating point. A combustion phasing sweep has been carried (changing the SA) out shifting MFB50 from 7 deg ATDC to 23 deg ATDC: the sweep allows assessing the model ability to represent the combustion sensitivity to SA in terms of performance (IMEP), combustion phase (MFB50) and knock intensity (MAPO). Only the highest knock intensity condition has been analyzed for the correlation of pressure data referring to the transducer position and the piston surface.

The indicating parameters have been extracted by filtering the pressure signal with a butterworth zero-delay low pass filter at 3kHz for IMEP and a butterworth band-pass filter between 3.5 kHz and 30kHz for the high frequency parameters.

Table 1. engine characteristics

Manufacturer	Ducati
Number of Cylinders	2
Valvetrain system	Desmodromic
IVO	4° BTDC→58° ABDC
EVC	58° BBDC→7° ATDC
Cylinder Displacement [cm³]	599
Bore [mm]	106
Stroke [mm]	67.9
Compression Ratio	12.5
Maximum Engine speed [rpm]	10000
Engine Test Conditions	
Operating point	WOT@8500rpm
Lambda	0.83
Spark configuration	Twin-spark
Combustion Phasing (MFB50)	Sweep from 7° to 23° ATDC

CFD RANS SIMULATION

The first step of the CFD simulation is an accurate reconstruction of the mixture composition in the chamber. A multi-step methodology [29] has been accomplished with the 3D CFD FIREv2011 (AVL) code, where the overall physics involved in the injection process is evaluated and validated. The engine is equipped with a multi-hole injector located in the intake duct, downstream the throttle. Ducati performed a full characterization of the injector spray (in terms of both droplet size distribution and flow patterns) and a semi empirical methodology has been used for the CFD injection setup. The spray is characterized in the same condition of work (i.e. injection duration, vessel pressure) and the droplet size distributions is registered over planes at different distance from the injector tip. The primary break-up is then by-passed with an ad hoc initialization of the probability distribution of the droplets size, while the secondary breakup consists in the WAVE model, tuned against experimental data available in literature. In order to better reconstruct the gas-dynamics in the intake duct, these steps have been followed:

1. Multi-cycle simulation without injection, until the fluid dynamics variables inside the cylinder reach a stationary condition. Tumble ratio, turbulent kinetic energy and volumetric efficiency need about three engine cycles to reach convergence. A fixed wall temperature is imposed on piston, head, liner and valves, based on the estimated results of 1D simulations [15]

2. Simulation of the injection for the number of engine cycles needed to reach the mixture air index target in the chamber. The engine under analysis takes three engine cycles to reach an ignitable mixture and six to achieve the equivalence ratio target value of 1.2.

Fig. 1 shows the equivalence ratio in the chamber at Inlet Valve Closing (IVC) and the location of the two spark plugs. The stars in the figure display the spark locations: one is at the center of the chamber, the other is located between the intake and the exhaust valves. The cylinder pressure transducer is placed on the opposite side of the combustion chamber, with respect to the second spark plug. As it can be seen in Fig 1, the fuel distribution is not homogeneous and the second plug is inside a leaner zone.

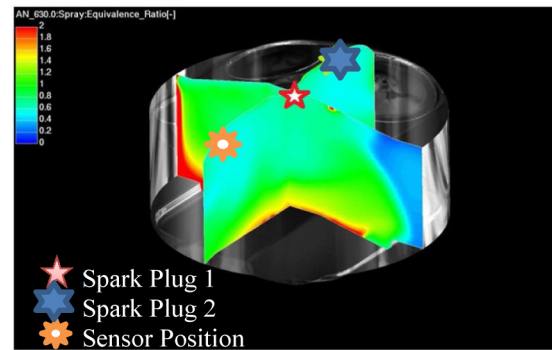


Figure 1. Distribution of the fuel in the chamber at IVC

The results of the injection simulations are then mapped on a modified version of the KIVA3D-code, developed at the university of Bologna. The main models implemented in this code are deeply described in [30] and listed in Tab. 2

Table 2. KIVA3D CFD MODELS

Turbulence	Standard k-ε
Law of the wall	Standard + Han & Reitz
Combustion	Extended Coherent Flamelet
Knock	Two step autoignition model based on [31]
Ignition	Lagrangian ignition model [32-33]

Simulation of the Combustion Process

The flamelet combustion model has been tuned in order to correctly represent the actual average behavior. Due to the cycle by cycle variability of SI engines all the pressure traces of the experimental data are widely scattered, representing very different combustion characteristics: as explained in [34], the cycles representing the average behavior are those showing combustion phase (MFB50) similar to the average. A comparison of the simulated versus experimental pressure traces for the cycles representative of the operating conditions is plotted in Fig. 2: the reconstruction of the pressure evolution in the chamber in terms of peak value and its angular position is good.

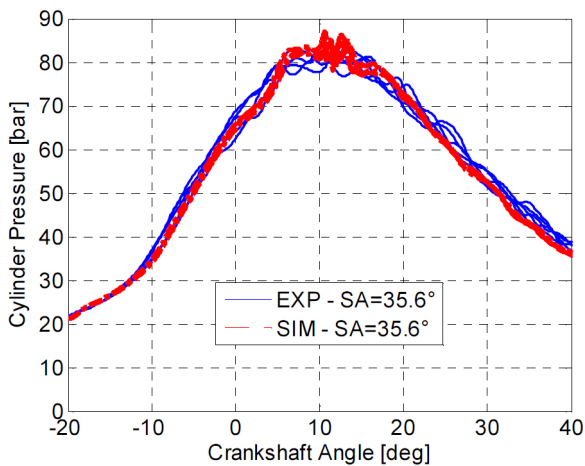


Figure 2. simulated and experimental pressure traces

The overall behavior on the whole spark advance sweep can be observed in Fig. 3: the simulation is able to represent the IMEP sensitivity to SA, which is crucial for a robust representation of the physical phenomena taking place during the combustion.

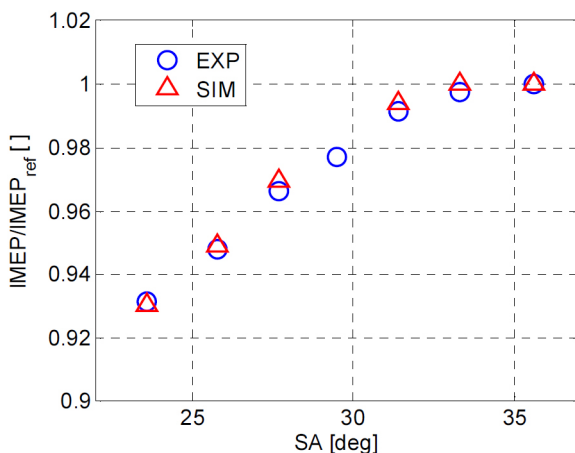


Figure 3. IMEP trend as a function of Spark Advance (degrees bTDC) at engine test speed of 8500 rpm.

RANS Simulation of CCV and Knocking Analysis

An effective way to simulate the cycle by cycle variation of a spark ignition engine is based on a forced perturbation of the combustion evolution, with particular emphasis on the early kernel development. The methodology of perturbation used in the presented approach is not based on a simple relationship between the perturbation strength and the amplitude of results, but it is founded on the spatial characterization of the mixture and turbulence at the spark plug. Tab. 3 describes the local condition of the mixture at the spark plug at 40 deg BTDC.

Table 3. Mixture condition inside a sphere with 12 mm radius at the spark plug, 40 deg BTDC

Parameter	1 st Spark Plug	2 nd Spark Plug
Average Local Lambda	0.98	1.10
Lambda Standard Deviation at the spark plug	0.08	0.14
u'/U_p	0.45	0.5

Lambda mean value and standard deviation are evaluated as mass-weighted values inside a sphere with 12mm radius, and the turbulent intensity is made dimensionless using the mean piston speed.

The perturbation of the combustion is thus imposed on two different physical phenomena:

- The laminar velocity is modified to consider lambda variations at the ignition. The variations are defined according to the spatial distribution of lambda in a sphere with 12 mm radius
- The turbulent combustion takes into account the variability of turbulence inside the chamber by modifying the source term of surface density of flamelet

The methodology is based on the hypothesis that the variability of mixture involves the combustion until a sphere of 12mm (for the configuration under analysis) of burned volume is reached. The perturbation is a linear interpolation between the stochastic variation for initial radius (flame kernel) and zero variation when the sphere surface is reached. The perturbation of lambda has direct consequences on laminar flame speed. The described methodology for the generation of the perturbation allows maintaining the total amount of energy released in the system, only influencing the combustion velocity. Keeping constant the total energy released while varying the flame development will cause MFB50 (and IMEP) to vary, leading to the typical scatter in the plot IMEP-MFB50 (Figure 4). As regards turbulent combustion, the disturbance acts on the turbulent distribution at the spark zone modifying the flamelet source term with a standard deviation of 5% (value of turbulent kinetics energy distribution resulting from the RANS simulation inside the initial control sphere). In both laminar and turbulent perturbation the stochastic variation is selected following a Monte Carlo methodology.

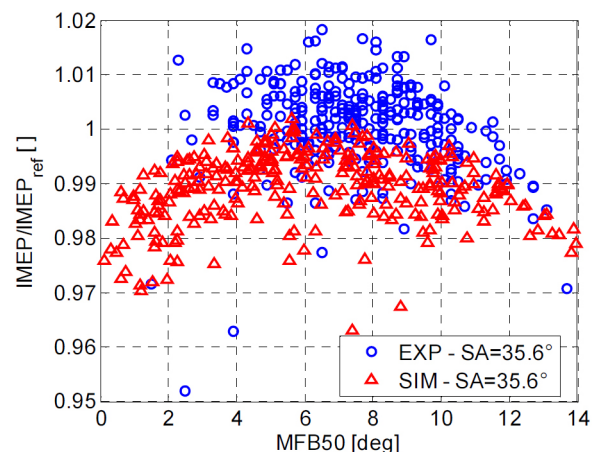


Figure 4. experimental and simulated IMEP-MFB50 scatters

The reconstruction of the scatter IMEP vs MFB50 for all the simulated cycles is the first evidence of the results. Fig. 4 shows the comparison between the experimental and simulated engine cycles: the typical distribution of parabola-like patterns is well gathered. Experimental and simulated MFB50 data can be fitted with normal distributions with respectively [7.10-7.65] and [6.52-7.25] mean (95% confidence intervals). The corresponding standard deviation intervals are [2.07-2.46] and [2.72-3.25] degrees. The GOV of IMEP