

Development of Specific Tools for Analysis and Quantification of Pre-ignition in a Boosted SI Engine

Jean-Marc Zaccardi, Laurent Duval and Alexandre Pagot
IFP

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ABSTRACT

Recent developments on highly downsized spark ignition engines have been focused on knocking behaviour improvement and the most advanced technologies combination can face up to 2.5 MPa IMEP while maintaining acceptable fuel consumption. Unfortunately, knocking is not the only limit that strongly downsized engines have to confront. The improvement of low-end torque is limited by another abnormal combustion which appears as a random pre-ignition. This violent phenomenon which emits a sharp metallic noise is unacceptable even on modern supercharged gasoline engines because of the great pressure rise that it causes in the cylinder (up to 20 MPa).

The phases of this abnormal combustion have been analysed and a global mechanism has been identified consisting of a local ignition before the spark, followed by a propagating phase and ended by a massive auto-ignition. This last step finally causes a steep pressure rise and pressure oscillations.

One of our objectives was to evaluate the sensitivity of an engine to pre-ignition regarding its design and settings. Therefore, in addition to our comprehension work, we have developed a first methodology based upon robust statistics to define new reliable and repeatable criteria to quantify this stochastic pre-ignition but also to detect each of its occurrence, suggesting the possibility of an on-line detection during steady state and transient operation as well. The statistical approach also showed that the distributions of well chosen combustion indicators are strongly altered by pre-ignitions. A second

methodology was then defined to evaluate the influence of different parameters on pre-ignition by quantifying this alteration. This analysis notably gives the opportunity to achieve a deeper analysis of pre-ignition during engine development on test bench.

INTRODUCTION

Spark ignition engines used today offer the great advantage of limiting local pollutants emissions (HC, CO & NO_x) thanks to the excellent adequacy between their operating mode (lambda one) and their simple and low cost after-treatment system. Despite this fundamental asset, those engines are badly considered concerning the greenhouse gases emissions because competitor Diesel engines reach an average CO₂ emissions 20 % lower thanks to an overall good efficiency.

Fuel consumption of SI engines has now to be reduced to meet future CO₂ legal emission agreements but also environmentally-friendly customer demands. In light of this, the combination of air charging and downsizing is one of the most promising solution because of its low technological cost and because it helps to reduce pumping and friction losses.

Unfortunately, the combustion mechanism of SI engines can be disturbed by abnormal phenomena [1]. Two major kinds of uncontrolled combustion can appear.

- Surface ignition : in this case, the ignition of fresh mixture is not controlled by a spark at a predetermined crank angle. This ignition takes place at variable timing on an overheated component (like

a valve or a spark plug) because of the unbalanced thermal transfer between the energy brought by the combustion and the thermal dissipation within the engine. The main feature of this kind of pre-ignition is that it progressively degenerates since the overheated component is getting warmer and warmer cycle after cycle as a consequence of the early timing combustion (hence it is generally called a "run away pre-ignition"). This first abnormal combustion can be fought with an optimised combustion chamber using for example an adapted thermal range for the spark plug and improving the cooling circuit to remove potential hot surfaces.

- Auto-ignition of fresh charge : this self ignition is a consequence of appropriate thermodynamic conditions inside the combustion chamber. Indeed, if the fresh mixture is strongly compressed and heated up the auto-ignition delay can be reduced. Knocking and CAI combustion modes (Controlled Auto Ignition) are the best known examples of auto-ignition.

More recently, developments of highly charged gasoline engines showed that a violent form of sporadic pre-ignition can appear under high loads at low engine speeds. Because of its occasional characteristic, this new kind of uncontrolled combustion does certainly not correspond to a classical surface pre-ignition.

This phenomenon has already been mentioned and described in numerous previous papers. Our aim is not to discuss the potential causes of this pre-ignition. In fact, many hypotheses have already been formulated to explain this phenomenon but no one has really been confirmed and it is possible that several factors could be combined to provoke this dangerous combustion as well [2] [3] [4] [5] [6] [7] [8] [9] [10] [11] [12] [13].

We focus here on new methodologies of analysis, quantification and control of this violent combustion. The aim is of course to minimize its occurrence frequency and its intensity in order to reach safely higher low end torque and to maximise the consumption reduction with a highly downsized engine.

The first part of this paper gives a description of pre-ignition based upon cylinder pressure analysis and direct in-cylinder imaging.

The second part introduces the statistical data processing that has been used to characterize this phenomenon.

Then, the third and fourth parts describe two strategies usable on board and on test bench to achieve a real-time detection and control of pre-ignition.

EXPERIMENTAL APPARATUS

ENGINE

Results shown in this paper were obtained on a single cylinder engine used at IFP for combustion studies and especially for highly charged SI engines. The main features of our selected technical definition are summarized in Table 1 and a simplified CAD view of the combustion chamber is given in Figure 1.

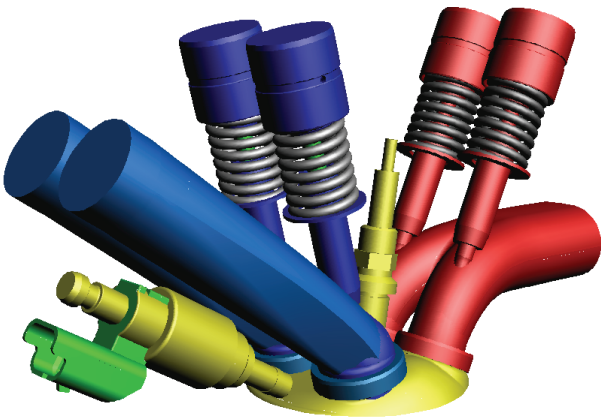


Figure 1 : CAD view of the pent roof cylinder head

Type	Pent roof shape 4 valves
Bore (mm)	69
Stroke (mm)	80.5
Displacement (cm ³)	301
Compression Ratio	9.5:1
Connecting rod length (mm)	139
Piston shape	Flat
Intake duration (CAD at 1 mm)	190
Inlet Valve Opening (CAD at 1 mm)	10 ATDC
Maximum intake valve lift (mm)	7.3
Exhaust duration (CAD)	210
Exhaust Valve Closure (CAD at 1 mm)	10 BTDC
Maximum exhaust valve lift (mm)	8.2

Table 1 : Engine specifications

Manifold Port Fuel Injection and Direct Injection (DI) modes were used on this engine. The gasoline direct injection was performed with a multi-hole injector placed under the intake ducts with a special spray targeting. The characteristics of this injector are summarized in Table 2.

Injector type	Multi-hole – 6 holes Oval spray
Injection pressure	6 – 20 MPa
Static flow	720 g/min @ 10 MPa (n-heptane)
Single beam angle	16 °

Table 2 : Multi-hole injector specifications

The ignition was realised with a high energy DELPHI ignition coil (110 mJ) combined with a centrally located NGK spark plug which had an iridium centre electrode and a heat rating number of 8 (NGK rating).

Rotating and reciprocating forces were balanced to achieve safely our full load objectives of 3.0 MPa IMEP at 1000 rpm and 2.5 MPa IMEP at 5500 rpm (115 kW/L).

TEST BENCH

This study was realised on a special test bench designed for single cylinder engines. Oil, coolant and fuel were supplied by electrically driven pumps. Several temperature and pressure measurements were made on the coolant, oil, fuel, intake and exhaust circuits to settle the different operating points and to supervise the engine.

Intake pressure was recorded with an absolute pressure transducer (KULITE XT-123B-190-50A). Exhaust and in-cylinder pressures were recorded with water-cooled relative pressure transducers (AVL QC43D & KISTLER 6043A60).

There was no turbocharger on our engine, but the test bench was equipped with a compressor for the intake line (sonic flow meter up to 0.35 MPa abs) and a flap in the exhaust line to simulate the backpressure of a turbine.

There was no ECU associated to this engine. Injection and ignition were manually controlled with a specific software developed at IFP. This way, we could control the engine as we want for steady state running conditions even with multiple injection and ignition timings during the same cycle.

All the measurements were recorded and analysed with an in-house software.

PRE-IGNITION PROCESS

DESCRIPTION BASED UPON CYLINDER PRESSURE

We propose here to sum up the description of the abnormal combustion process based upon cylinder

pressure analysis. A full description is already available in a previous paper [2].

The observation of this uncontrolled combustion on test bench is actually mainly made on in-cylinder pressure traces. A simple example with a violent pre-ignition compared to a normal combustion is given in Figure 2. Cycles n° 8 and 10 show a normal combustion with an initiation at the spark plug and a flame propagation through the combustion chamber. In this two cycles, we can observe the late ignition and hence combustion timing necessary to cope with the knocking phenomenon. Cycle n° 9 is an example of pre-ignition. The pressure difference between cycles n° 9 on the first hand and n° 8 and 10 on the other hand just before the spark is an obvious evidence that combustion began too early.

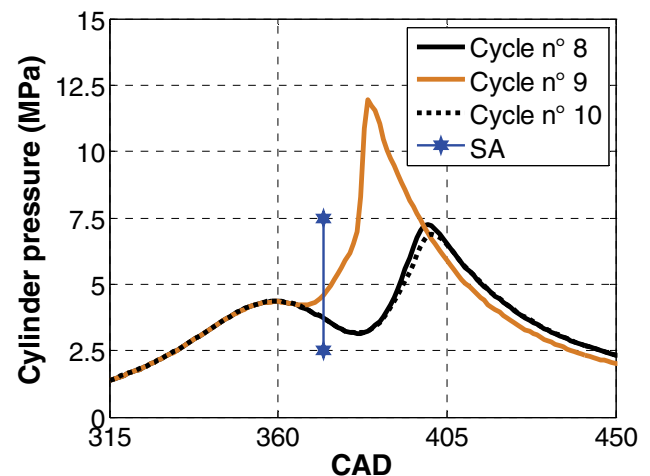


Figure 2 : Pre-ignition example

This kind of pre-ignition can be so violent that a single occurrence can definitively damage the moving parts (especially the connecting rod). This abnormal combustion is all the more dangerous that it is impossible to fight against it simply by setting differently the spark advance or the air fuel ratio contrary to a classical knocking combustion.

A simplified rate of heat release ROHR can be calculated from the cylinder pressure and combustion chamber volume (see Figure 3).

$$ROHR = \frac{1}{V_{cyl}} * \left(\frac{1}{\gamma-1} * V * dP + \frac{\gamma}{\gamma-1} * P * dV \right)$$

γ : polytropic coefficient

P : cylinder pressure (MPa)

V : combustion chamber volume (dm³)

V_{cyl} : displacement (dm³)

Then, based upon the cylinder pressure and this simplified ROHR analyses, it is possible to propose a three steps general mechanism for pre-ignition [2].

1. The ignition takes place before the spark occurrence with a variable timing for each pre-ignition and anywhere in the combustion chamber (as far as we know using only one in-cylinder pressure transducer).
2. Disregarding the early timing, the second part of the combustion seems to be quite normal with a heat release which is comparable to a normal combustion.
3. The third and last part of combustion is clearly faster than the second part. The pre-ignition somewhere in the combustion chamber combined with the compression at high load provoke such an increase of temperature and pressure that a bulk auto-ignition occurs.

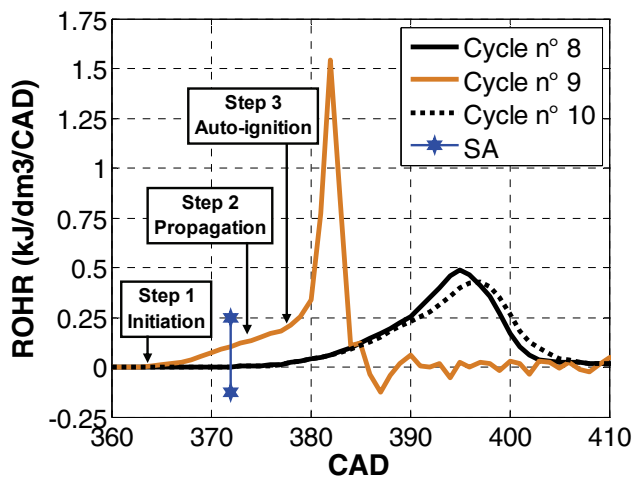


Figure 3 : Pre-ignition general mechanism

The final bulk auto-ignition looks like a knocking combustion except that the fresh mixture quantity concerned by this auto-ignition is far greater than for a classical knocking phenomenon (sometimes more than 50 % of the total fresh mixture quantity).

Pressure oscillations are not clearly visible in the previous example because cylinder pressure was recorded at each crank angle degree. A ten times faster data acquisition rate can illustrate those pressure oscillations (see Figure 4). Besides, our analysis showed that those oscillations have roughly the same frequency content of a classical knocking combustion obtained on the same engine and evaluated beforehand from Bessel functions. Moreover, parasitic mechanical resonance frequencies can also appear in the frequency content due to the violence of the phenomenon.

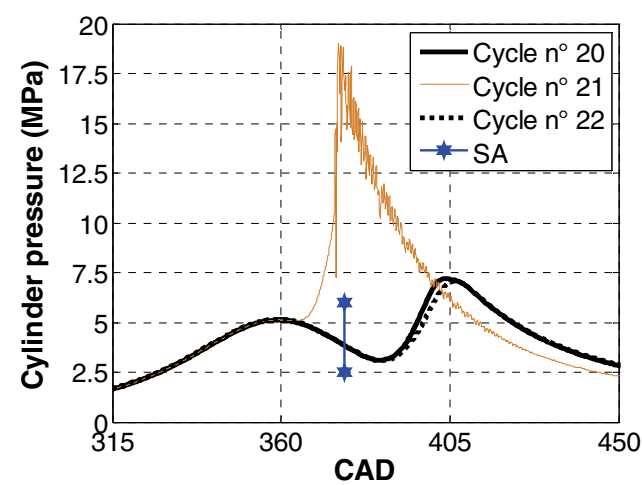


Figure 4 : Final bulk auto-ignition with associated pressure oscillations

This result shows that we can use not only the same sensor to detect knock and pre-ignition, but also the same kind of methodology for the data analysis and treatment.

DESCRIPTION BASED UPON IMAGING

Imaging system

Numerous tests were made to find a special operating point with a high frequency of pre-ignitions but always with limited intensities. The selected settings are summarized here below in Table 3 (reference for ignition and injection timings is taken on the combustion TDC).

Engine speed (rpm)	1000
IMEP (MPa)	≈ 1.8
$T_{coolant} = T_{oil}$ (°C)	95
Spark Advance	from KL to KL -10 CAD
Start of Injection (CAD BTDC)	from 220 to 260 CAD
Intake temperature (°C)	from 30 to 50
Boost pressure (MPa)	≈ 0.18

Table 3 : Operating point for direct pre-ignition visualisations

A non-intensified colour Charge-Coupled Device (CCD) camera was used to visualise pre-ignition in the combustion chamber (see Table 4). We decided not to use any additional light source relying only upon the light that the combustion would emit itself.

Type	IDS uEye UI-2210-M VGA Camera – CCD Sensor
Resolution	640*480
Colour	Colour
Start timing (CAD)	4.5 CAD before Spark Advance (SA)
Opening duration (CAD)	9

Table 4 : Main characteristics and settings of the camera

The camera was mounted on an air-cooled endoscope placed on the flywheel side of the cylinder head. The field of view with such an assembly is given in Figure 5.

The opening duration and start timing of the camera imply that during a normal combustion only a small bright point corresponding to the spark between the spark plug electrodes is visible in the middle of the combustion chamber (the location of the spark initiation point is surrounded by a white dashed circle in Figure 5).

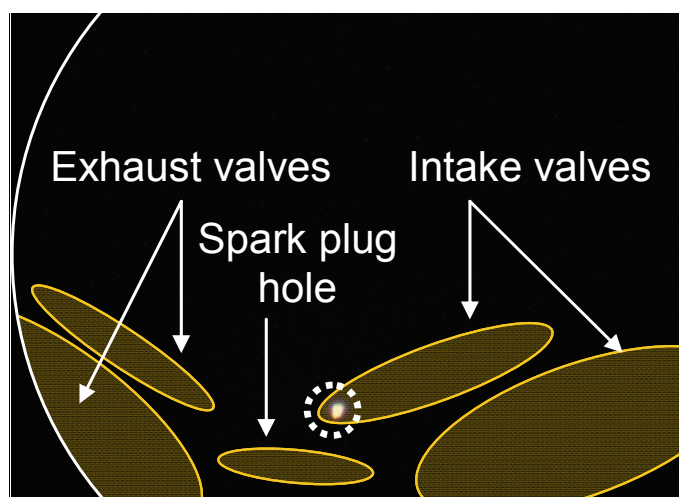


Figure 5 : Field of view in the combustion chamber with spark position indicated by dashed circle

Image processing

Image processing is relatively involved due to acquisition settings : the light sources are essentially combustion related, leading to extreme contrast conditions (dark and bright areas), combined with a number of non-linear disturbances than should be removed before image enhancement, to avoid noise amplification. Figure 6 displays a detail of an image (top) along with its histogram (bottom) for each red, green and blue colour plane. Colour intensities are scaled between 0 (dark) and 1 (bright) and represented on the horizontal axis, while the colour proportions, between 0 (rare) and 1 (frequent) are represented on the vertical axis with a logarithmic scale. Dark intensities are thus prominent (left end), with

a fairly low proportion of intermediate intensities, as indicated by “orders of magnitude” decreases in the central part of the histogram. Let us precise that this image contains a pre-ignition even if it is rather invisible at this time.

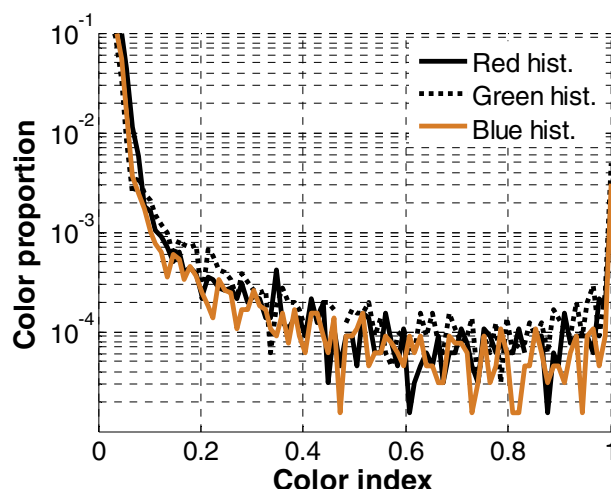


Figure 6 : Image example (top) and its color histogram (bottom)

As a consequence, standard contrast enhancement methods based on histograms may fail in providing insight to the non visible parts of the image.

The proposed processing method is based on non-linear image denoising and enhancement based on the discrete wavelet transform (we refer to [14] for a detailed exposition on wavelet transforms). The latter constitutes an alternative to the Fourier transform which is better suited to local features (e.g. contours) preservation. It consists in iteratively filtering low and high frequencies of the image rows and columns. The result of this decomposition on two levels is given in Figure 7. The top left sub-image corresponds to a filtered and sub-sampled approximation of the image in Figure 6 (top), while other sub-images contain mid-range and high-frequency