

Influence of the Valve-lift Strategy in a CAI™ Engine using Exhaust Gas Re-Breathing – Part 2: Optical Diagnostics and 3D CFD Results

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ABSTRACT

Among the existing concepts that help to improve the efficiency of spark ignition engines at part load, Controlled Auto-Ignition™ (CAI™) is an effective way to lower both fuel consumption and pollutant emissions without major modifications of the engine design. The CAI™ concept is based on the auto-ignition of a fuel mixture highly diluted with burnt gases in order to achieve high indicated efficiency and low pollutant emissions through low temperature combustion.

Large amounts of burnt gases can be trapped in the cylinder by re-breathing them through the exhaust ports during the intake stroke. For that, a 2-step exhaust valve-lift profile is used. The interaction between the intake and exhaust flows during the intake stroke was identified as a key parameter to control the subsequent combustion in a CAI™ port fuel injected (PFI) engine. Not only does this interaction directly act on the mean in-cylinder conditions (temperature, dilution), but it can also induce some mixture stratification.

As identified in part 1, the phasing and the duration of the second exhaust valve event have a very significant influence on combustion. Nevertheless, the combustion variations that were observed cannot only be explained by the mean in-cylinder conditions. A difference in mixture stratification was proposed as an explanation. To confirm this hypothesis, optical diagnostics and 3D CFD calculations were used. Using acetone-LIF (Laser-Induced Fluorescence), optical diagnostics provided information about the stratification between fresh and burnt gases. Those results were related with the auto-ignition locations which were observed using direct visualisation of the combustion. 3D CFD results explained the origin of such mixture stratification and

balanced the species stratification with the temperature stratification. The combination of both tools allowed the CFD results to be validated and provided complementary information on the combustion process.

INTRODUCTION

The fuel consumption penalty for spark ignition engines is mainly due to controlling the load through throttling. Among the existing concepts to help improve the efficiency of spark ignition engines on low load operating points, Controlled Auto-Ignition™ is an effective way to lower both fuel consumption and pollutant emissions at part load without major modifications of the engine design [1-3]. The drawback of spark ignition engines can therefore be reduced whilst keeping the engine simple and relatively cheap to build and run.

The CAI™ (Controlled Auto-Ignition™) concept is founded on the auto-ignition of a highly diluted gasoline-based mixture in order to reach high indicated efficiency and low pollutant emissions through a low temperature combustion. The recycling of large amounts of hot burnt gases aims to ensure auto-ignition despite the gasoline high octane number. Burnt gases originating from previous combustion process are trapped in the cylinder to bring thermal energy to the air-fuel mixture and to reach appropriate conditions at the end of the compression stroke.

This operating mode allows significant fuel consumption reduction together with limitation of NO_x emissions (and the need for dedicated after-treatment). These benefits are related to the large dilution rates and the unthrottled operation (or at least, "less-throttled" operation). While conventional gasoline engines suffer from significant

cycle-to-cycle IMEP (Indicated Mean Effective Pressure) fluctuations when operated with large exhaust gas recirculation (EGR) rates, CAI combustion allows both highly-diluted mixture and stable operation which implies additional fuel consumption gains.

In a CAI engine, the whole combustion process is controlled by the chemical kinetics. Therefore, the quantities acting on the kinetics (dilution rate, temperature, pressure, fuel/air equivalence ratio,...) control the auto-ignition timing, the combustion spreading and the consequent noise emissions [4-5]. Since most of these quantities are influenced by the presence of the trapped burnt gases from the previous engine cycle, the CAI combustion is characterised by a large cycle-to-cycle coupling of the combustion processes.

On the other hand, the predominant influence of the chemical kinetics also induces a lack of direct actuator on the start of combustion (with opposition to the spark timing in conventional SI operation). The combustion control must therefore be indirect which is more complex and necessitates more precision.

Over the past ten years numerous works have been conducted at IFP to better understand CAI combustion [1, 4-8]. Among them, the effects of intake configuration (one or two valves), burnt gas trapping (negative valve overlap versus exhaust gas re-breathing) or injection strategies (port-fuel injection versus direct injection; single or multiple injections) were specifically studied. In particular, the influence of the species and temperature fields as well as their interactions are now better understood. Selecting the operating strategies to control these fields allows the engine efficiency and emissions to be optimised and enables a wider operating range to be obtained.

In a PFI CAI engine, most of the control on the in-cylinder conditions is obtained with the valve-lift strategy. This strategy has a strong influence on the mean in-cylinder quantities such as dilution rate and temperature but also on the stratification of the species and temperature fields. To be able to test all desired valve-lift strategies, a RAVTS (Research Active Valve Train System) from Lotus has been adapted to a single-cylinder engine. This RAVTS allows the valve-lift profiles to be fully controlled in terms of shape, timing and duration. Valve-lift profiles with double opening during one 4-stroke engine cycle can also be achieved so that an exhaust gas re-breathing strategy can be considered.

The potential of engine operation control with the RAVTS is detailed in part 1 [9]. The engine operation analysis shows that some phenomena cannot be explained based on the mean in-cylinder quantities. Therefore, hypotheses about mixture stratification are proposed to explain the observed trends. The objective of the present part of the paper (part 2) is to confirm these hypotheses based on optical diagnostics and 3D CFD.

VALVE-LIFT STRATEGIES AND TESTED OPERATING POINTS

The CAI combustion mode relies on the availability of a high rate of trapped burnt gases. These may basically be obtained with two different valve-lift strategies:

- the Negative Valve Overlap (NVO) strategy is based on a truncated exhaust stroke to trap the needed amount of burnt gases
- the Exhaust Gas Re-Breathing (EGRB) strategy is based on a conventional exhaust stroke to first expel the burnt gases and a second exhaust valve event to re-breathe the required amount of burnt gases.

In the present study, NVO strategy was only used to compare results obtained from the all-metal engine with results from the optical engine. A brief comparison of the CAI combustion progression in those two engines can be found in Appendix 1.

All other tests were performed using the EGRB strategy. The EGRB strategy was chosen as it is better adapted to extend the CAI operating range towards higher loads. Compared to the NVO strategy, burnt gases trapped in the cylinder are cooler in the EGRB strategy [5]. The EGRB strategy is therefore more efficient to slow down combustion when reaching the higher load range.

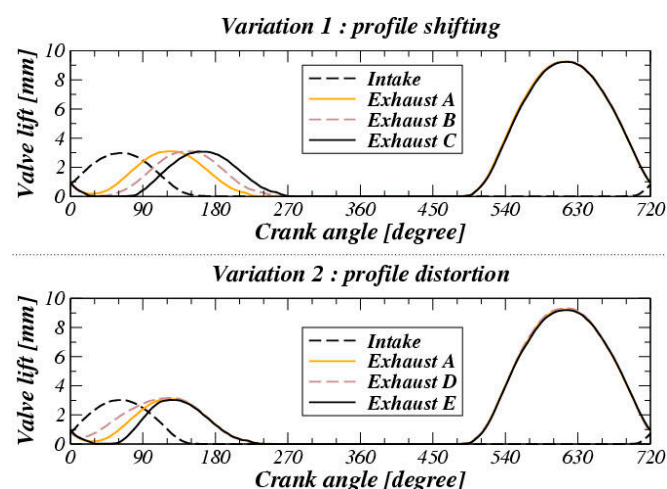


Figure 1: Valve-lift strategies. Top: profile shifting. Bottom: profile distortion (360 CAD is combustion TDC)

Variation 1 : variation of the timing of the second exhaust valve-lift			
	A	B	C
Optical engine	X	X	
3D CFD	X		X

Variation 2 : variation of the opening timing of the 2 nd exhaust valve-lift			
	D	A	E
Optical engine		X	X
3D CFD	X	X	

Table 1: Allocation of the test cases between optical diagnostics and 3D CFD

Two kinds of valve-lift strategy variations are detailed in the present paper (see Figure 1 and Table 1). First, the influence of the timing of the second exhaust valve-lift is tested. Then, the influence of the opening timing of the second exhaust valve-lift for a fixed closure timing (lift profile distortion) is tested.

Variation 1 is split into a single timing shift on the optical engine and a single timing shift with 3D CFD. For variation 2, profile distortion with earlier valve re-opening is considered with 3D CFD while profile distortion with later valve re-opening is measured on the optical engine. A reference operating point (point A) is available both with optical diagnostics and 3D CFD to allow comparison. Note that the comparison will only be qualitative and not quantitative as the computed operating conditions are derived from the all-metal engine and not from the optical engine (see Table 2).

Tests performed with the optical engine were carried out with the same IMEP as that obtained with the all-metal engine for a similar valve-lift strategy. Nevertheless, the engine speeds are different (1800 rpm for the optical engine and 2500 rpm with the all-metal engine) and the engine materials induce hotter wall temperatures for the optical engine. Consequently, the combustion timing and efficiency are different in the all-metal engine and the optical engine which means that the same operating conditions cannot be used. As the fuelling is maintained stoichiometric, the IMEP was adjusted through intake pressure alteration. An additional difference is the use of commercial gasoline for the all-metal engine (and consequently 3D CFD) and a Primary Reference Fuel (PRF) for the optical engine to allow LIF measurements.

	CFD	Optical engine
Fuel	RON 95	PRF 65
Load (IMEP)	3.5 bar	3.5 bar
Speed	2500 rpm	1800 rpm

Table 2: Operating conditions specific to each engine

ENGINE CONFIGURATION

The study is performed with two single-cylinder engines: an all-metal engine and an engine with optical accesses. The optical engine geometry is identical to that of the all-metal engine, except for the addition of optical accesses in the cylinder head and a slightly less deep lens-shaped piston bowl to compensate the additional volumes due to these accesses (identical compression ratio for both engines).

The main characteristics of the engines are summarised in Table 3. All tests are performed at stoichiometry with a constant backpressure (absolute mean exhaust pressure of 1.06 bar). The engine is equipped with an AVL noise-meter (model AO1) and pressure transducers (water-cooled AVL 43D in the exhaust pipe, water-cooled AVL 33 D in the cylinder and Kistler XT-123B-190-50A in the intake pipe). Intake air-flow is controlled via a sonic orifice.

Number of cylinders	1
Number of valves	4
Displacement	500 cm ³
Bore	82.7 mm
Stroke	93.0 mm
Compression ratio	12.0:1
Injection	PFI, one injector per intake duct
Cylinder-head roof angle	40°

Table 3: Engine specifications

FUEL SPECIFICATIONS

The all-metal engine is operated with commercial unleaded RON95 gasoline while the optical engine is operated with a Primary Reference Fuel (PRF).

A non-fluorescent fuel is needed to perform LIF diagnostics which means that the optical engine cannot be operated with commercial gasoline. The PRF 65 (65% isooctane, 35% n-heptane in volume) is used as fuel for its non-fluorescing properties together with its auto-ignition characteristics that are similar to the commercial gasoline in the CAI operating range [6].

Moreover, for the needs of LIF diagnostic, acetone is homogeneously seeded into the intake air, about 2m upstream of the tranquillisation volume. It was estimated (measuring the lambda offset) that seeded acetone quantity was about 5% (in volume) of the total fuel amount. As PFI configuration was used, the air/fuel mixture was very homogeneous. Therefore, it can be considered from a combustion point of view that fuel contains 5% acetone. The influence of 5% acetone on the combustion results is analysed in Appendix 2.

OPTICAL DIAGNOSTICS TOOLS

Two types of optical diagnostics are applied. Acetone-LIF is used to visualise the fresh gases. Burnt gases are revealed as negative by reduction of the LIF signal. The stratification between fresh and burnt gases can therefore be analysed. The second diagnostic consists in a direct visualisation of the combustion chemiluminescence to locate the auto-ignition sites. The combination of both diagnostics allows the mixture stratifications to be correlated with the first auto-ignition sites.

VISUALISATION OF THE STRATIFICATION OF FRESH AND BURNT GASES

Planar Laser Induced Fluorescence is used to image the species distribution in the combustion chamber. Several studies [6, 10] concluded that acetone is an appropriate fuel tracer because of its good resistance to thermal decomposition and the limited oxygen quenching of acetone fluorescence in CAI running conditions.

Depending on the thermodynamic conditions and the excitation wavelength, the fluorescence emission of

acetone may exhibit a significant temperature dependence [10]. Consequently, spatial fluctuations in the fluorescence signal may be attributed to composition and/or temperature variations in the mixture. In the present application, the recycled burnt gases being significantly hotter than the air, both effects combine and the heterogeneities between fresh and burnt gases are very probably amplified.

Excitation is realised with the fourth harmonic of a Nd-YAG laser (266 nm). A laser sheet formed with a 70 mJ laser pulse passes through the cylinder head side windows along the pentroof axis of the combustion chamber (see Figure 2). PLIF images are captured through the transparent piston crown with an intensified CCD camera (Princeton 16-bit Pentamax) and an 80 mm / f# 1.4 Nikkor lens.

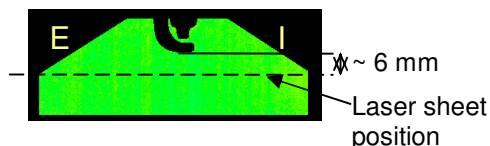
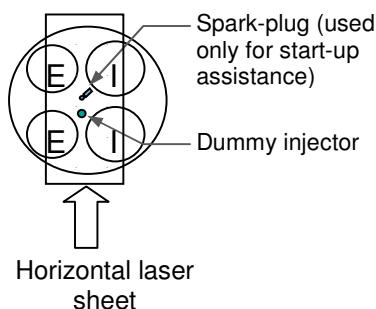
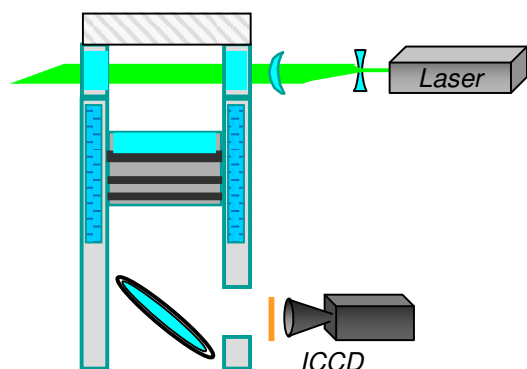


Figure 2: Scheme of the experimental arrangement for LIF visualisation

COMBUSTION VISUALISATION

Direct visualisation of the combustion chemiluminescence is used to detect the auto-ignition sites. This diagnostic is sensitive to the natural chemiluminescence of the flame. It has been demonstrated that the chemiluminescence intensity is approximately proportional to the local reaction rate [6]. An intensified CCD camera (Princeton 16-bit Pentamax) and an 80 mm / f# 1.4 Nikkor lens are used to visualise

combustion through the piston window (same configuration as in Figure 2, but without the laser). As this camera is able to take only one image per cycle, statistical analyses of images are performed in order to correctly capture the auto-ignition process (which is very short in time). Series of 500 images are acquired, at a few crank angles around the one where the first signal traces appeared on average. An automatic image post-processing is set up to detect the auto-ignition sites and to display them on a synthesis image showing the distribution of all the detected auto-ignition sites (Figure 3).

The image post-processing is based on the following automatic procedure. First, each pixel is tested based on the light intensity and on a defined threshold to detect the burning zones while discriminating signal from noise. Then, a test on contiguous pixels is performed to determine the extension of each burning zone. A burning zone is finally classified as an auto-ignition zone if its area is between two predefined bounds. The location of the auto-ignition site is computed as the barycentre of its pixels. One of the input images and the statistical distribution of the auto-ignition sites are depicted in Figure 3.

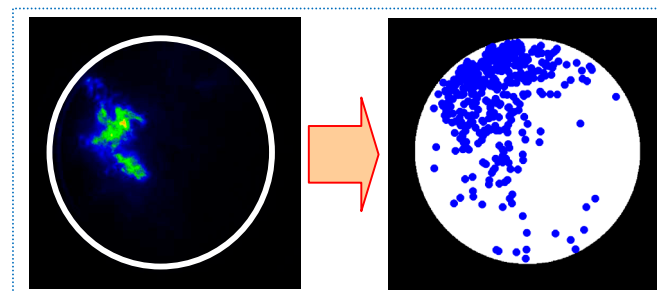


Figure 3: Direct visualisation of chemiluminescence at an auto-ignition site (left) and distribution of the auto-ignition sites detected on 500 images (right)

COMPUTATIONAL FLUID DYNAMICS TOOL

IFP-C3D [11] is a 3D CFD parallel code solving the Navier-Stokes equations using an ALE-extended (Arbitrary Lagrangian Eulerian) finite volume method on unstructured and conformal hexahedral meshes. For turbulent combustion, the 3-Zone Extended Coherent Flame Model (ECFM3Z) is used [12]. The auto-ignition prediction and the influence of chemistry on reaction rates are provided by the TKI auto-ignition model [8].

The ECFM-3Z combustion model

The principle of the ECFM3Z model, which blends the ECFM and CFM3Z approaches into a unique combustion model, is detailed in Figure 4 for compression ignition operating conditions. A more complete description is available in [12].

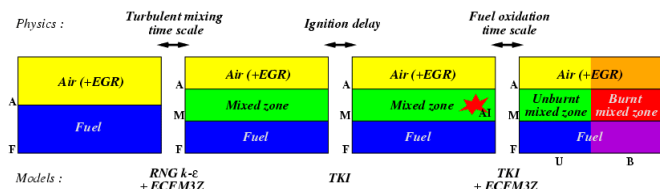


Figure 4: Description of the ECFM3Z model and its interaction with other models for each mesh cell

In order to correctly represent the local mixing process, the ECFM3Z model includes a description of the local stratification by considering three regions of mixing: one region contains only pure fuel (region F), the second region contains fresh air plus any additional EGR (region A) and the third region contains a mixture of the latter two (region M). During injection, the evaporation of the spray droplets leads to a source of mass in region F. The fuel and air (+EGR) from regions F and A mix with the help of turbulence to form region M. The mixture in region M may then auto-ignite after a delay depending on the local thermodynamic conditions. When the auto-ignition delay is reached, combustion in region M starts, thereby dividing it into two sub-zones: a fresh (or unburned) gas zone and a burnt gas zone. As the mixed fresh gases in region M are oxidised following auto-ignition, the hot products formed are added to the burnt gas side of the mixed region.

The TKI auto-ignition model

To numerically study the CAI combustion mode, an auto-ignition model is needed which has to provide the auto-ignition delay and an accurate description of the subsequent chemistry reactions. Indeed, a precise knowledge of the reaction rates is needed to correctly predict the pressure rise during combustion since this is a limiting factor for the higher loads in the CAI operating range. The latest version of the TKI auto-ignition model [8] meets all these requirements.

The Tabulated Kinetics of Ignition (TKI) auto-ignition model [8] relies on tabulated auto-ignition variables deduced from detailed chemistry calculations performed with the Senkin code, which is part of the Chemkin package. The detailed mechanism has been generated by the Département de Chimie Physique des Réactions (Nancy, CNRS) for mixtures of n-heptane, isooctane and toluene [13-14]. This mechanism contains about 2500 reactions between circa 500 species. The 3D model variables are calculated once and for all for 22800 different initial thermodynamic conditions and are stored in look-up tables. The fresh (or unburned) gas quantities that are used as input parameters of the auto-ignition lookup tables are temperature, pressure, Fuel/Air equivalence ratio and dilution rate.

Figure 5 shows an example of the temperature profile predicted by the TKI model and a comparison with the temperature profile computed with the detailed chemistry mechanisms. In this example, the thermodynamic conditions lead to a first part of the combustion controlled by the cool flame. To represent this phenomenon, two

auto-ignition variables are defined: the auto-ignition delay τ and the portion C_1 of fuel consumed by the cool flame. Then, main ignition is modelled based on reaction rates ω_{c_k} tabulated as a function of the progress variable. When the thermodynamic conditions lead to the absence of the cool flame effect, C_1 is simply zero and the combustion is entirely described with reaction rates.

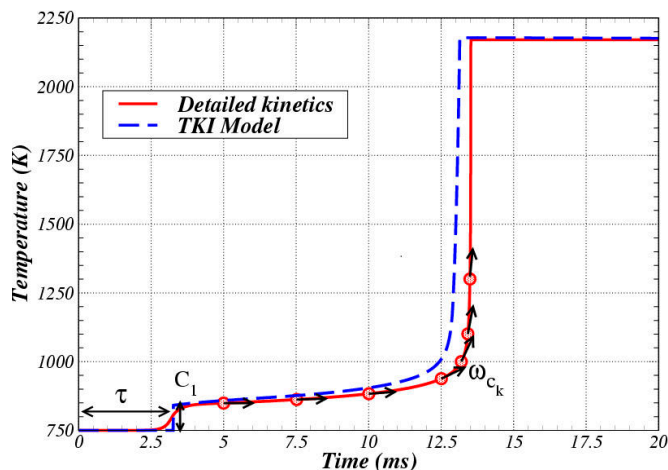


Figure 5: Comparison of temperature evolution from detailed kinetics and the TKI model for a constant volume reactor

As, in an internal combustion engine, thermodynamic conditions evolve with the piston motion, all model variables (and among them the delay) extracted from the database vary on a per cell basis. To take this evolution into account, an intermediate species is considered for the in-cell auto-ignition delay, the concentration of which has a growth rate proportional to the local fuel tracer concentration and to the relevant auto-ignition delay. The actual auto-ignition occurs when the intermediate species concentration reaches that of the fuel tracer.

To represent the behaviour of a commercial unleaded RON95 gasoline, a ternary fuel mixture composed of isooctane, n-heptane and toluene is used. Such a mixture has the advantage over the Primary Reference Fuels to be representative of the real gasoline for all conditions leading to an auto-ignition occurrence (CAI combustion mode and knocking conditions).

RESULTS AND DISCUSSION

VARIATION 1: VALVE-LIFT PROFILE SHIFTING

As presented in part 1 [9], a timing shift for the second exhaust valve event does induce a large change in dilution rate and combustion timing. Nevertheless, the combustion timing shift is not linearly dependent on either the valve timing or the dilution rate. As depicted in Figure 6, for late second exhaust valve opening, the combustion timing does not widely vary despite the large variation in dilution rate. On the contrary, for early second

exhaust valve opening, the slightest change in dilution rate implies a large shift in combustion timing.

	A	B	C
2 nd opening	55	75	95
2 nd closure	195	215	235

Table 4: Opening and closure timings for the second exhaust valve event (timings at 0.7 mm valve-lift)

As the mean in-cylinder conditions cannot explain such a trend, a hypothesis about some changing in in-cylinder mixture heterogeneity has been proposed. To verify this hypothesis, 3 operating points (see Table 4) were tested with the optical engine and 3D CFD.

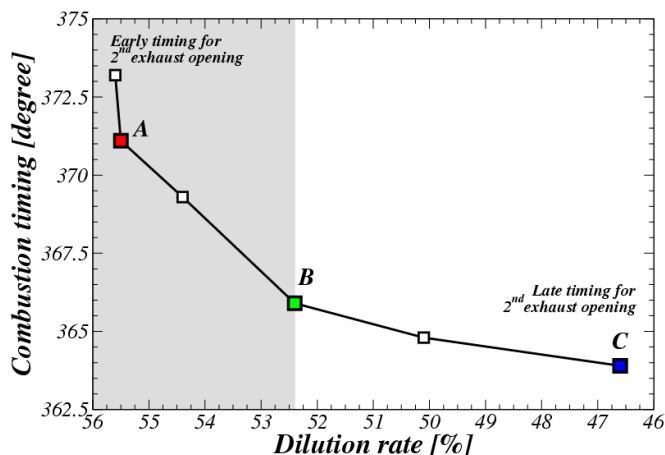


Figure 6: Evolution of combustion timing with dilution rate for the all-metal engine (each square represents a 10 cad shift in valve timing)

Trend reproduction with the optical engine

The trend we wish to reproduce has been obtained with the all-metal engine in part 1 [9]. Before detailing the optical engine measurements, we need to verify that this engine does induce the same global trends. The operating points measured with the optical engine are points A and B.

First, to maintain an identical engine load (here IMEP), the intake pressure has to be lowered when a later valve timing is considered (see Table 5). This is coherent with the measurements on the all-metal engine and tends to confirm the dilution rate reduction with later timings.

In terms of combustion timing, the shift is similar to that observed with the all-metal engine despite the globally later combustion (see Figure 7). This difference in combustion timing is a consequence of the different engine speed, different heat transfers due to the change in wall material and the different intake pressure necessary to obtain the same load.

Global results with the optical engine

As shown by the in-cylinder pressure trace (Figure 7), a larger mass of gases is trapped in the cylinder in case A (see the end of the compression stroke). As volumetric

efficiency is similar in both operating conditions, the amount of exhaust gases trapped in the cylinder is higher in case A.

With a later exhaust valve re-opening, point B has a lower dilution rate than point A. As a consequence, the in-cylinder temperature is lower during compression but higher during combustion and expansion.

The lower mixture temperature during compression is overcompensated by the reduced dilution rate which explains the earlier combustion timing of point B.

The higher combustion temperature induces larger NO_x emissions but also reduced emissions of unburned hydrocarbons and carbon monoxide. The latter probably indicates less local kinetics freezing thanks to a higher mixture temperature.

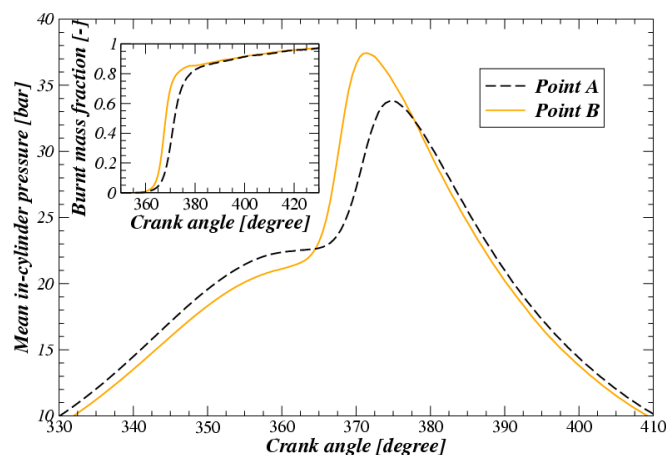


Figure 7: Comparison of mean in-cylinder pressure traces and burnt mass fraction histories of points A and B for the optical engine

	A	B
Fuel/Air equivalence ratio [-]	1.0	1.0
Volumetric efficiency [-]	0.277	0.280
Intake pressure [bar]	0.891	0.806
IMEP [bar]	3.61	3.56
High pressure IMEP [bar]	3.90	3.99
Low pressure IMEP [bar]	-0.29	-0.43
Covariance of IMEP [%]	1.5	1.6
Noise [dB]	84	86
ISFC [g/kWh]	218	226
HP ISFC -HC-CO/4 [g/kWh]	195	195
Peak of RoHR [1/CAD]	0.129	0.170
CA5 [CAD]	365.7	363.7
CA50 [CAD]	371.7	368.2
Exhaust Temperature [K]	727	747
uHC [g/kWh]	5.5	5.2
CO [g/kWh]	4.8	3.8
NOx [g/kWh]	0.2	0.6

Table 5: Global engine results for points A and B for the optical engine