

System Approach for Compliance with Full Load Targets on a Wall Guided Diesel Combustion System

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ABSTRACT

Low temperature combustion concept as HCCI is one of the most promising research ways to comply future emission regulations of Diesel passenger vehicles. IFP promoted this concept with NADI™ (Narrow Angle Direct Injection) combustion design whose original approach lies on a fuel spray guided by the bowl central tip to the re-entrant. For full load operating range, one of the key issue for success is to use as much as possible available air in the combustion chamber in order to reach low value of air fuel ratio, and therefore high value of specific power and specific torque. In this study, engine tests on a single cylinder engine with NADI™ concept are performed at full load ; 3-D calculations as well as air/fuel mixing process visualizations in a constant volume vessel with optical access allowed to establish criteria for helping future combustion system design for full load operation. Accordingly, at low engine speed, a large bowl with a thin re-entrant is recommended because this configuration implies a weak interaction between fuel jets and the bowl central tip. At high engine speed, a large bowl with a thin re-entrant as well as a dedicated configuration to guarantee a strong interaction between fuel jets and the bowl central tip ; this design promotes the fuel spray momentum conservation in order to get a well homogenized fuel vapor within piston bowl but also a good use of the air available in the squish area.

INTRODUCTION

In Europe, passengers cars powered by diesel engine have known a great success over the last decade as they offer a low fuel consumption together with high density power and torque. Nevertheless, research efforts have to be continuously lead to comply with the diesel engine emissions according to the more and more stringent emission regulation, particularly regarding NOx and PM release.

The up-to-date concept to achieve near zero NOx and PM emissions lays on low temperature combustion process. LTC or HCCI combustion process consists in preparing a high dilution by burned gases air fuel mixture so as to reach a well premixed mixture before the start of combustion.

Various concepts have recently emerged as the MK concepts of Nissan [1], Toyota smokeless System [2], HCLI (Homogeneous Charge Late Injection) or HPLI (Highly Premixed Late Injection) [3] commonly based on using a great EGR rate associated with sometimes late injection timing.

First investigations about near zero NOx and PM emissions focused on thoroughly premixed mixture which implied very early injection timing. However applying such injection strategies causes some drawbacks as risks of fuel impingement on the cylinder liner leading by the way to a high level of fuel dilution in the lubricant. To raise this key issue, IFP has developed a dedicated early injection timing combustion system NADI™ [4]. This combustion system is mainly composed of a bowl with a central tip as well as a narrow spray cone angle. Nevertheless, as HCCI combustion is not able to cover the whole operating range, studies have been performed to make the NADI™ concept consistent with using it at full load and therefore reaching the output power and torque requests.

Key issues at full load lays on a well optimized air utilization so as to reach the best efficiency and the greatest fuel air equivalence ratio facing soot emission level. Tests at full load on a single cylinder engine showed that optimal NADI™ design with regard to the interaction between fuel jets and the central tip of the bowl at high speed (4000 rpm) and low speed (1500 rpm) are opposed. Accordingly, further investigations including 3D calculations with IFP-C3D code were performed to bring out relevant explanations. Besides, in order to better understand physical phenomena controlling efficiency of combustion and in order to define the optimal combustion chamber design, additional tests were carried out on a single cylinder engine and in an optically-accessible vessel to correlate engine output performance and fuel vapor distribution inside the piston bowl. To visualize the concentration of fuel vapor in the combustion chamber, the laser induced exciplex fluorescence (LIEF) technique was used.

As a result of all these tests, head guidelines to optimize an overall NADI™ design for full load operation including width of piston bowl, bowl width to lip width ratio, height

of bowl central tip, spray cone angle and nozzle tip protusion (NTP) have been finally defined.

NUMERICAL SET UP

DESCRIPTION OF THE 3D CFD CODE

IFP-C3D is a 3D CFD parallel code developed at IFP solving the Navier-Stokes equations using an ALE-extended (Arbitrary Lagrangian Eulerian) finite volume method on unstructured and conformal hexahedral meshes [5]. For turbulent combustion, the 3-Zone Extended Coherent Flame Model (ECFM3Z) shortly described below is used [6]. The auto-ignition prediction and the influence of chemistry on reaction rates are provided by the TKI auto-ignition model [7], also shortly introduced below. To obtain a realistic spray modeling during the injector needle motion, the Transient Injection Conditions (TIC) model [8] is used.

COMBUSTION MODELING

For compression ignited internal combustion engines, the combustion modeling relies on the combination of two models. The ECFM3Z model describes the turbulent mixing in each of the mesh cells and provides the local thermodynamical conditions and species concentrations to the TKI auto-ignition model which represents the chemical kinetics related to the combustion process.

In order to correctly represent the in-cell mixing process, the ECFM3Z model [6] includes a description of the local mixture stratification by considering three mixing regions (see Figure 1): 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 turbulently mix to form region M. When the auto-ignition delay is reached, combustion starts in region M, thereby dividing it into two zones: a fresh (or unburned) gas zone and a burnt gas zone. As the mixed fresh gases in region M are oxidized by auto-ignition, the formed hot products are added to the burnt gas part of the mixing region.

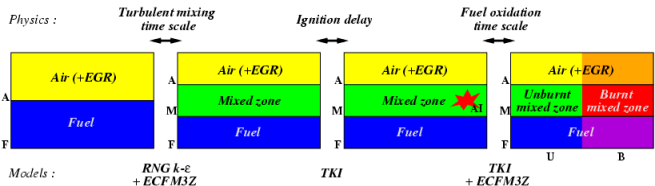


Figure 1 - Scheme of the ECFM3Z combustion model

The Tabulated Kinetics of Ignition (TKI) auto-ignition model [7] relies on tabulated auto-ignition parameters deduced from detailed chemistry calculations performed with the Senkin code. The 3D model parameters are calculated once and for all for a large range of initial thermodynamic conditions and are stored in lookup tables. When the thermodynamic conditions lead to the

absence of the cool flame effect, the tabulated parameters are the auto-ignition delay τ and the reaction rates ω_{ck} describing the combustion progress. If there is a cool flame effect, an additional parameter C_1 representing the portion of fuel consumed by the cool flame is considered. To take into account the evolution of the auto-ignition delay depending on the in-cell conditions, an intermediate species is considered which concentration 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.

COMBUSTION SYSTEM DESIGN AND SIMULATED EXPERIMENTAL CONDITIONS

3D calculations have been performed to provide consistent explanations to single cylinder engine results, especially in regard to the influence of the interaction between the fuel jets and the bowl central tip wall on the global engine performance.

The features of the simulated engine, the combustion chamber and injector nozzle used to carry out 3D calculations are described in Table 1. Simulated experimental conditions are also included.

Table 1 – Engine support and testing conditions for numerical applications

Bowl shape	
D / H	3.1
Re-entrant	12%
Tip angle	70°
H tip (10 ⁻³ m)	12.5
Injector nozzle - CRI 2 Bosch	
Number of holes	6
Spray cone angle	60°
Hydraulic flow (10 ⁻³ L/30s/10 MPa)	450
Experimental conditions	
Inlet pressure at 4000 rpm (MPa)	0,24
Inlet pressure at 2000 rpm (MPa)	0,23
Equivalence ratio at 4000 rpm	0,75
Equivalence ratio at 2000 rpm	0,85
Rail Pressure at 4000 rpm (Mpa)	160
Rail Pressure at 2000 rpm (Mpa)	135

Figure 2 presents a schematic view of a NADI™ piston bowl. The main defining terms currently used in this study to characterize a combustion design are reported.

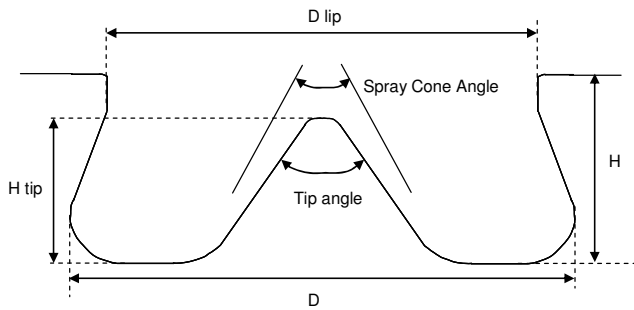


Figure 2 – Scheme of a NADI™ piston bowl

Re entrant is defined as following :
$$Re\ entrant = \frac{D - D_{lip}}{D_{lip}}$$

OPTICAL VESSEL SET UP

HIGH PRESSURE CELL FOR SPRAY / WALL VISUALIZATION

In order to account for the results of the influence of the re-entrant bowl shape obtained on a single cylinder engine (engine 2 described in the section below), visualization of the fuel vapor distribution within piston bowls has been performed in a high temperature, high pressure, optically-accessible vessel under thermodynamic conditions similar to those encountered in the single cylinder engine at specified injection timing. In order to heat and pressurize the gases in the vessel prior to the fuel injection process, a technique of pre-combustion was used. The pre-combustion consists in burning a mixture of He, C₂H₂ and O₂. The initial gaseous composition prior to pre-combustion is calculated in order to get, after the pre-combustion, a gaseous composition that is as close as possible as the in-cylinder gaseous mixture before injection (taking EGR into account). The start of injection (SOI) takes place at a specified timing after the precombustion ignition. The SOI timing is determined so that the injection takes place when temperature and pressure in the cell have reached the desired values. As the experimental technique to visualize the fuel vapor is laser induced exciplex fluorescence (LIEF), stoichiometric conditions are required for pre-combustion. Indeed, LIEF signal is strongly quenched by O₂. This requires no O₂ left after pre-combustion. In comparison to the in-cylinder gaseous mixture, oxygen has been replaced by nitrogen in the high pressure vessel gaseous mixture at SOI. However, similar in-cylinder gaseous density is reproduced. More details about this high pressure cell working can be found in [11, 14].

The vessel is equipped with a system able to hold in position a 80° section of a full piston and cylinder liner under an injector placed on the top face of the cell. In order to correctly reproduce the NTP (Nozzle Tip Protrusion) as well as the distribution of fuel vapor close to the squish area, a horizontal plate have been implemented to simulate the presence of the cylinder head. The distance between the injector nozzle and the upper face of the piston can be varied so as to simulate

different injection timings (piston positions). An overview of the experimental system is displayed in Figure 3.

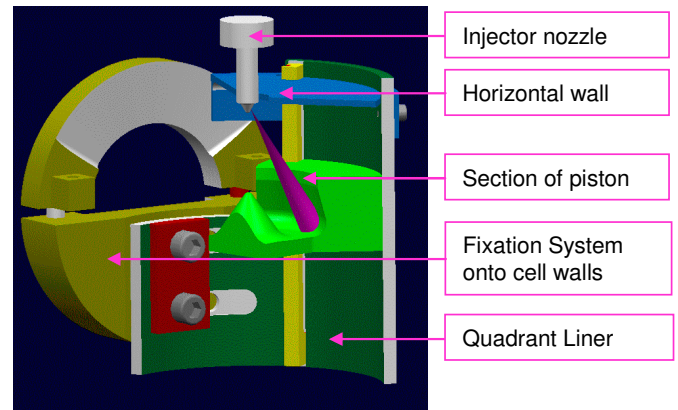


Figure 3 – Overview of the experimental system for visualization of fuel vapor concentration within a piston bowl

MEASUREMENT TECHNIQUE

The LIEF technique was used to visualize the fuel distribution in the constant volume chamber. This technique is able to separately visualize liquid and vapor phase of the fuel according to frequency separated optical filters used for collecting the fluorescence signal. Further details concerning the principle of this technique are available in refs. [11, 12].

LIEF technique was performed using the specific tracer TMPD (and α -methyl-naphtalen as a partner) added to a mono component fuel (decane) and excited with a frequency-tripled Nd:YAG laser at 355nm.

The LIEF experimental set up is presented in Figure 4. The lighting of the piston bowl is produced by a laser beam (100 mJ) diluted by a divergent spherical lens. An intensified CCD camera (Princeton Instruments, 12b) coupled to a 80mm f/1.4 objective was used to record the fluorescence signal through an optical high pass filter ($\lambda > 475$ nm) to visualize the liquid phase and a band pass filter ($\lambda = 400$ nm, $\Delta\lambda = 25$ nm) to visualize the vapor phase of the fuel.

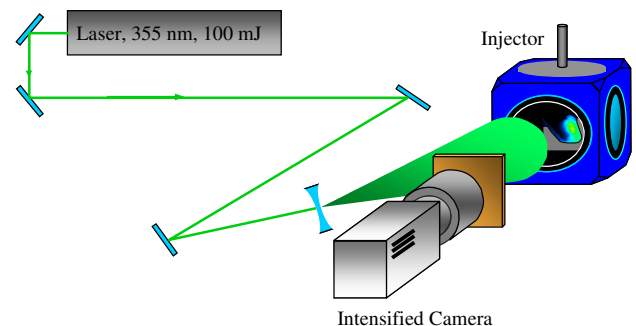


Figure 4 – LIEF experimental set up



SINGLE CYLINDER EXPERIMENTAL SET UP

Tests on two different single cylinder engines have been performed. With the first one, the influence of the interaction between fuel jets and the central tip of the bowl on output power and torque have been highlighted. With the second one, the combustion system designs used in the high pressure cell were tested in the engine.

ENGINE TESTS

Geometric characteristics of both engines are described in Table 2.

Table 2 – Geometric characteristics of the two single cylinder engines

	Engine 1	Engine 2
Bore (10^{-3} m) x Stroke (10^{-3} m)	87 x 92	85 x 88
Compression ratio	15 : 1	14 : 1
Capacity (10^{-6} m ³)	547	499
Head cylinder	production - Renault G9T 4 valves	production - PSA DW12 4 valves
Head cylinder swirl	1.3	1.1
Injection System	Bosch CRI 2.2 6 holes	Bosch CRI 3 6 holes

For each engine, intake air flow was controlled via a sonic flow nozzle. Air was dehumidified, and a high air capacity was implemented to damp air pulses. An air heater was also used to control the inlet air temperature and produce experimental conditions similar to those encountered with multicylinder engines. Exhaust gas were collected by a production exhaust manifold. A damping capacity was set up downstream on the exhaust line and exhaust pressure was adjusted by a throttle. The engine was managed by an electric generator able to absorb a maximal power of 50 kW. In addition, the engine test bench was equipped with a system controlling the pressures and temperatures of fuel, lubricant and water feeding the engine.

MEASURING EQUIPMENT

The cylinder was equipped with a water cooled pressure transducer. Data acquisitions were performed by an intern IFP product KlepatV3 in which a precise method to set up the top dead center is implemented. Fuel consumption was given by an AVL 733 s. Exhaust emissions were analysed by a six gases analysis device. Fuel air ratio was measured by an oxygen sensor in the exhaust line and it was also calculated from the gas analysis. The EGR rate was computed from intake and exhaust CO₂ rates measures.

EXPERIMENTAL CONDITIONS

The experimental conditions used for the two engines are described in Table 3 below.

Table 3 – Experimental conditions for the two engines

	Engine 1	Engine 2
Inlet pressure at 4000 rpm (MPa)	0,29	0,25
Inlet pressure at 1500 / 1750 rpm (MPa)	0,21	0,28
Exhaust pressure at 4000 rpm (MPa)	0,34	0,32
Exhaust pressure at 1500 / 1750 rpm (MPa)	0,26	0,28
Rail Pressure at 4000 rpm (MPa)	1600	1800
Rail Pressure at 1500 / 1750 rpm (MPa)	1600	1800

The tests consisted in increasing the fuel air equivalence ratio (by progressively increasing the amount of injected fuel quantity) at a constant engine speed. SOI (start of injection) was adjusted to maintain the same maximum in-cylinder pressure.

COMBUSTION SYSTEM DESIGNS

With the engine 1, four combustion system designs (bowls geometries) which can be grouped into two distinct classes were studied. A description of these combustion chambers is given in Table 4. The first class is composed of two bowls with a central tip angle of 70°. The analysis resulting from the comparison of these two geometries aims at defining the influence of the height of the bowl central tip. The second class is defined by two bowls with a larger tip angle. Their comparison aims at pointing out the best spray cone angle suited to a given bowl central tip angle.

Table 4 – Description of combustion system designs used with engine 1

	D/H	Re entrant	Distance between piston surface and tip	Tip Angle	Spray cone angle
BOWL A1	3	15%	4,3	70°	60°
BOWL B1	2,5	without	1,5	70°	60°
BOWL A2	3,3	6%	4,3	110°	100°
BOWL B2	3,4	6%	4,3	120°	100°

No comparison between the piston bowls from class 1 and class 2 is given in this paper.

With the second engine, were also studied four piston bowl designs to notably underline the influence of the re-entrant ratio. The characteristics of these piston bowls are displayed in Table 5.

The bowl C0 is defined as the reference. The bowl C1 offers a higher re-entrant ratio than reference while the bowl C2 presents a lower re-entrant ratio. The bowl C3 is characterized by a lower re-entrant ratio as well as a circular-bottomed bowl as opposed to the flat-bottomed bowl of other geometries.