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Near Zero NO_x Emissions and High Fuel Efficiency Diesel Engine: the NADI™ Concept Using Dual Mode Combustion

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Résumé — Un moteur Diesel avec quasiment zéro émissions de NO_x et un rendement élevé : le concept NADI™ utilisant deux modes de combustion — Grâce à leur bon rendement thermique et à leurs faibles émissions de dioxyde de carbone (CO_2), les moteurs Diesel verront leur part de marché dans les transports croître, à condition que leurs émissions d'oxydes d'azote (NO_x) et de particules soient réduites. Aujourd'hui, des systèmes de post-traitement adéquat des NO_x et des particules sont en passe d'être industrialisés. Cependant, des inconvénients subsistent encore en termes de consommation de carburant, de fiabilité, de sensibilité au soufre présent dans le carburant et de coût lié à leurs stratégies de fonctionnement complexes et sophistiquées.

De nouveaux procédés de combustion comme la combustion homogène de type HCCI (*homogeneous charge compression ignition*) sont actuellement à l'étude en raison de leur très fort potentiel à atteindre des émissions de NO_x et de particules quasi nulles. Leurs principaux inconvénients sont des émissions d'hydrocarbures imbrûlés (HC) et de monoxyde de carbone (CO) trop élevées, un contrôle de la combustion difficile à forte charge et une puissance limitée.

En réponse au défi que doit relever le moteur Diesel, l'*IFP* a développé un système de combustion capable d'atteindre des émissions de NO_x et de particules quasiment nulles tout en conservant les standards de performances des moteurs Diesel actuels. Ce concept de moteur "bi-mode" appelé NADI™ *(Narrow Angle Direct Injection)* utilise la combustion homogène à charge partielle et bascule en combustion traditionnelle pour répondre aux exigences de pleine charge.

En charges partielles (incluant les cycles MVEG européen et FTP américain), la combustion HCCI permet des émissions de NO_x et de particules quasi nulles en maintenant un très bon rendement, proche d'un moteur Diesel Euro III. À 1500 et 2500 tr/min, le concept NADI™ atteint respectivement 0,6 et 0,9 MPa (6 et 9 bar) de pression moyenne indiquée (PMI) avec des émissions de NO_x et de particules inférieures à 0,05 g/kWh. Cela représente respectivement 100 et 10 fois moins qu'un moteur Diesel traditionnel.

À pleine charge, le système NADI™ est en phase avec les futurs standards de puissance au litre. À 4000 tr/min, 50 à 55 kW/l ont été atteints avec des conditions limites de fonctionnement et des réglages moteur conventionnels.

L'utilisation de technologies moteur avancées comme la prochaine génération de systèmes d'injection common rail, les moteurs à distribution variable (VVA : *variable valve actuation*), les moteurs à taux de compression variable (VCR : *variable compression ratio*) ou les turbo-compresseurs assistés électriquement sera très utile pour les futures étapes de développement du concept, qui sont bien identifiées.

***Abstract* — Near Zero NO_x Emissions and High Fuel Efficiency Diesel Engine: the NADI^{TM} Concept Using Dual Mode Combustion** — *The part of Diesel engines on the transport market should increase within the years to come thanks to their high thermal efficiency coupled with low carbon dioxide (CO_2) emissions, provided their nitrogen oxides (NO_x) and particulate emissions are reduced. At present, adequate after-treatments, NO_x and particulates matter (PM) traps are developed or industrialized with still concerns about fuel economy, robustness, sensitivity to fuel sulfur and cost because of their complex and sophisticated control strategy.*

New combustion process such as homogeneous charge compression ignition are investigated for their potential to achieve near zero particulate and NO_x emissions. Their main drawbacks are too high unburned hydrocarbons (HC) and carbon monoxide (CO) emissions, combustion control at high load and then limited operating range and power output.

As an answer for challenges the Diesel engine is facing, IFP *has developed a combustion system able to reach near zero particulate and NO_x emissions while maintaining performance standards of the DI Diesel engines. This "dual mode" engine application called NADI^{TM} (Narrow Angle Direct Injection) applies homogeneous charge compression ignition at part load and switches to conventional Diesel combustion to reach full load requirements.*

At part load (including Motor Vehicle Emissions Group—MVEG—and Federal Test Procedure—FTP—cycles), HCCI combustion mode allows near zero particulate and NO_x emissions and maintains very good fuel efficiency close to an Euro III Diesel engine. At 1500 and 2500 rpm, NADI^{TM} reaches 0.6 and 0.9 MPa (6 and 9 bar) of indicated mean effective pressure (IMEP) with emissions of NO_x and particulate under 0.05 g/kWh. That means respectively 100 and 10 times lower than a conventional Diesel engine.

At full load, NADI^{TM} system is consistent with future Diesel engine power density standard. At 4000 rpm, 50 to 55 kW/l has been reached with conventional limiting factors and engine parameters settings.

Advanced engine technology such as further generation of common rail fuel injection system, Variable Valve Actuation (VVA), Variable Compression Ratio (VCR) engine or electric assisted turbocharger will be useful for the well identified next development steps of the concept.

NOTATIONS

Gas

CO_2	Carbon Dioxide
CO	Carbon Monoxide
HC	Unburned Hydrocarbons
NO_x	Nitrogen Oxides
O_2	Oxygen
PM	Particulates Matter.

Combustion Concepts

HCCI	Homogeneous Charge Compression Ignition
HCDC	Homogeneous Charge Diesel Combustion
HiMiCS	Homogeneous charge intelligent Multiple injection Combustion System
MK concept	Modulated Kinetics Concept
NADI^{TM}	Narrow Angle Direct Injection
PCI	Premixed Compressed Ignited Combustion
PREDIC, MULDIC	PREmixed lean and MULtiple stage DIesel Combustion
UNIBUS	UNIform BUckly Combustion System.

Cycle and Norms

EURO III	Emission Norm for Europe (2000-2005)
FTP	Federal Test Procedure
MVEG	Motor Vehicle Emissions Group
SULEV	Super Ultra Low Emission Vehicle.

Engine Parameters

A/F ratio	Air/Fuel ratio
ATDC	After Top Dead Centre
BDC	Bottom Dead Centre
BMEP	Brake Mean Effective Pressure
BMF	Burned Mass Fraction
BSFC	Brake Specific Fuel Consumption
BTDC	Before Top Dead Centre
CA	Crank Angle
CFD	Computational Fluid Dynamic
DI	Direct Injection
EGR	Exhaust-Gas Recirculation
FMEP	Friction Mean Effective Pressure
FSN	Filter Smoke Number
IMEP	Indicated Mean Effective Pressure

ISFC	Indicated Specific Fuel Consumption
rpm	Rotation per minute
TDC	Top Dead Centre.

Engine Technology

FIS	Fuel Injection System
VCR/VVA	Variable Compression Ratio/Variable Valve Actuation.

INTRODUCTION

HCCI combustion engines constitute alternative or complementary answers to the sophisticated and complex after-treatments strategy, which seems to be compulsory for classical Diesel engine. Their principle consists in preparing a highly diluted by burned gases air/fuel mixture, in achieving its simultaneous ignition in the whole space of the combustion chamber and in precisely controlling such combustion for the best performance in terms of efficiency and pollutant emissions. As the combustion takes place in a homogeneous way throughout the bulk of the mixture, the thermal NO_x formation and the soot production are known to be much lower than with the typical conventional Diesel combustion diffusion flame.

There are three families of HCCI concepts: with port injection, with in-cylinder injection and concepts using both port and in-cylinder injections.

Port Injection

These engines are developed by the *University of Wisconsin*, Madison [1], the *Southwest Research Institute (SWRI)* [2-4], the *Lund Institute of Technology (LIT)* [5-12, 14, 15], and the *Tokyo Gas Corporation* [13].

There are two types of engine using port-injection, one working with liquid fuel and other working with gaseous fuel. They generally use an electronically driven gasoline injector in the intake pipe, and in some cases, two injectors with two different fuels in 2 intake pipes (LIT). The *Tokyo Gas Corporation* uses a mixing chamber before the intake pipe, to promote gas and air mixing.

In-Cylinder Injection

This concept uses one or many direct injectors in the main chamber. Two different ways of injection timing are possible: an early injection is the most common way, but *Nissan* has developed a new concept with a late injection.

Early in-cylinder injection with one injector

- *Himics (Homogeneous charge intelligent multiple injection combustion system).*

This concept was developed in 1997 by *Hino* motors. The injection system used is a common rail (with an injection pressure of 90 MPa), and one single centre injector (6 holes with a diameter of 0.23 mm and a spray cone angle of 125° or 30 holes diameter 0.1 mm and three spray cone angles: $12 \times 155°$, $12 \times 105°$, $6 \times 55°$).

The injection strategy is divided into four injections: early injection (called E), pilot injection (P), main injection (M) and late stage injection (A) [16, 17].

- *Unibus: Uniform buckly combustion system (Toyota) and PCI: Premixed compressed ignited combustion (Mitsubishi).*

These two concepts are very close to each other as they use one single centre injector: a piezzo injector with a hollow cone spray (60°) and a spray guide tip *(Unibus)* and an Impinged Spray Nozzle to limit penetration (PCI), and in both cases, a Common Rail system. The injection timing is very early: till 120° BTDC [18-21].

Early in-cylinder injection with two or more injectors: *Predic* and *Muldic*

- *Predic and Muldic mean premixed lean and multiple stage Diesel combustion.*

In this concept, developed by the *New Ace Institute* in 1996, the direct in-cylinder injection is realised thanks to three injectors, two on each side of the cylinder, and one in the centre. The injection strategy is not simple: two injection pressures: 150 and 250 MPa, each injected quantity is controlled, and *Predic* uses two injection timings: an early injection: 80° BTDC (with side injectors) and a late injection: 40° BTDC (with centre injector), whereas *Muldic* adds a second stage injection with the centre injector (2° BTDC to 30° ATDC) [22-29].

Late in-cylinder injection: MK Concept

- *MK Concept means modulated kinetics concept and was developed by* Nissan *in 1998.*

This concept uses a common rail injection system with one single injector (5 holes diameter 0.22 mm).

The injection strategy is based on the concept: “low temperature, premixed combustion”. So they use a high EGR rate (O_2 15 %), a vigorous swirl (3 to 5) and a late injection (7° BTDC to 3° ATDC) so as to have a long ignition delay and no diffusion combustion; the goal is to inject all the fuel before the combustion starts [30-34].

Dual Fuel Introduction: HCDC Concept

HCDC means Homogeneous Charge Diesel Combustion and was developed in 1997 by the *Traffic Safety and Nuisance Research Institute.*

The injection system consists in a gasoline injector in the intake pipe (injection pressure 5 MPa) and a DI Injector (4 holes diameter 0.21 or 0.14 mm with an injection pressure of 18 MPa) in the combustion chamber.

The strategy is to use premixed injection and direct injection (15° BTDC) to ignite the mixture and so to vary the premixed fuel ratio [35-38].

Comparison between the Concepts

The three next tables (Tables 1 to 3) give a summary of the main characteristics of the HCCI concepts studied here.

Through these tables, it appears to the authors that one of the best configuration seems to be the *Unibus*/PCI concept, with really low NO_x and particulates emissions, and an

TABLE 1

Comparison between HCCI concepts: emissions

		NO_x	Smoke	HC/CO	BSFC
Port Injection	SWRI, Lund, UWM	Very low (/100)	Low (–27%)	High	Very high (+28%)
Early In-Cylinder Injection	**HiMiCS**	Low (1/3 or < 200 ppm)	Low (1/3)	High (3000-8000 ppm)	Improved in a limited area
	UNIBUS PCI	**Very low (1/100 and 90%)**	**Very low (near 0)**	**Acceptable (2000 ppm)**	**High (+10%)**
	PREDIC, MULDIC	Very low (< 20 ppm)	Improved (1/2)	High	High (+15%)
Late In-Cylinder Injection	MK Concept	Very low (–90%)	Very low (<1 BSU)	Low (< 1000 ppm)	No change/DI
Dual Fuel Introduction	HCDC	Very low (–75%)	Low (–40%)	High (6000 pmc)	No change/DI

TABLE 2

Comparison between HCCI concepts: combustion control and HCCI

		Control start of reaction	Knock at high load	Range	Test speed (rpm)	EGR rate
Port Injection	SWRI, Lund, UWM	No	High	Large (IMEP 16 bar)	1000-1500	High (65%)
Early In-Cylinder Injection	**HiMiCS**	Possible	No	Large	1000-1600	Low (15%)
	UNIBUS	Possible	High	Small (IMEP 3 bar)	1000	?
	PCI	Possible	High	Large with supercharge	1000	No but possible
	PREDIC, MULDIC	Possible	High	Very limited ($\lambda > 2.5$)	1000	No but possible
Late In-Cylinder Injection	MK Concept	Possible	No	Limited ($\lambda > 1.3$)	1200-2000	High (45%)
Dual Fuel Introduction	HCDC	Possible	No	Large	1500	Low (30%)

TABLE 3

Comparison between HCCI concepts: design

		Conception	Walls Impingement	Compression ratio	Displacement (cc)
Port injection	SWRI, Lund, UWM	Simple	Yes	Variable (10-28)	702 1600-2000
Early In-cylinder Injection	**HiMiCS**	Simple	Yes	18	2147
	UNIBUS	Simple	Yes	Variable (12-21)	915
	PCI	Simple	Limited	12	2000
	PREDIC, MULDIC	Very complicated	Yes	16.5	2004
Late In-cylinder Injection	MK Concept	Simple	No	18 and 16	488 622
Dual Fuel Introduction	HCDC	Very complicated	Yes	20.4	522

acceptable level of HC and CO and of BSFC. Note that the compression ratio is a major parameter of the engine, and in many cases, a variable compression ratio seems to be necessary.

HCCI Challenges

This study of the main HCCI concepts shows that applying HCCI combustion to engines raises some difficulties:
– Mixture preparation: the main problems are to avoid fuel wall impingement and to promote fuel vaporisation and air mixing, so as to limit particulate and HC emissions, and to prevent oil dilution.
– Operating range: when the A/F ratio approaches stoichiometric values, combustion stability degrades, heat release increases and knock appears.
– Control of combustion: it is the key point in HCCI combustion, especially to extend the operating range and power output.

The ambitious goal of this work is to develop a combustion system able to reach near zero particulate and NO_x emissions (by using HCCI combustion) while maintaining the higher and higher performance standards of the DI Diesel engines. The main concerns in HCCI mode are to limit fuel wall wetting and to have new effective ways to control the combustion, so as to extend the operating range.

1 APPROACH DESCRIPTION

1.1 Concept Overview

To overcome limitations in power output, we have developed a “dual mode” engine, using HCCI combustion at low and medium loads and conventional Diesel combustion at high loads (with injection close to TDC), which is called NADI^{TM} (Narrow Angle Direct Injection). This means that the combustion system should be able to switch between the two combustion modes.

1.1.1 General Architecture

At an early development stage, we have decided to keep the general architecture of a conventional Diesel combustion system:
– direct injection;
– flat cylinder head;
– combustion piston bowl.

The idea was to use the fuel wall interaction to mix the air with the fuel well at high and full load operation, using conventional combustion mode.

1.1.2 Fuel Injection System

Thanks to the study of the different HCCI concepts, we decided to use an early injection strategy. The problem with such a configuration is to avoid fuel wall impingement. That’s why the key point of this concept is to use a narrow spray cone angle (less than 100 degrees) to limit fuel wall impingement and to promote fuel air mixture, while having a great flexibility in terms of injection.

A common rail fuel injection system has been selected due to its continuously increasing flexibility, especially in terms of injection events.

1.1.3 Compression Ratio

Concerning the compression ratio, a lot of works pointed out the advantage to have a moderate compression ratio in order to control the start of combustion better, especially to extend the engine HCCI operating range. In the present study, the engine geometric compression ratio has been set to 16:1. 1D simulations have shown that mechanical VVA hardware could be sufficient to solve the cold starting problem: the early closing of the intake valve(s), around BDC, during engine cranking, allows an increase of effective compression ratio by one unit. At the same time, it will be possible to optimise the injection events, without wall wetting drawbacks thanks to the selected narrow injection angle. Moreover, VVA could be used to adjust the effective compression ratio for better combustion control. This will be developed later.

Figure 1 gives an overview of the NADI^{TM} combustion system concept whose main features can be listed as:
– conventional flat cylinder head;
– narrow spray cone angle (lower than 100°);
– reduced geometric compression ratio (16:1);
– multi stage injection (common rail FIS);
– VVA hardware.

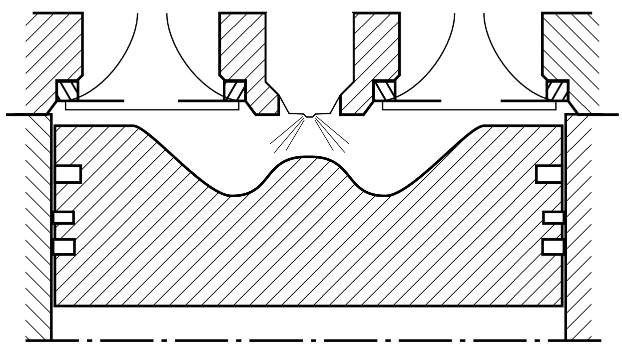

Figure 1

Overview of the NADI^{TM} combustion system concept.

2 COMBUSTION SYSTEM DEFINITION

Our first objective was to find a combustion chamber well adapted to a narrow spray cone angle in order to reach a good conventional Diesel combustion behaviour, especially at full load. The approach is based on the use of CFD tools combined with development work done with an operating engine. A brief description of the CFD tools is given in Appendix.

2.1 Combustion Chamber Optimisation

The first challenge was to make a proper transport and mixing with air of the fuel injected inside the piston bowl. Several combustion chamber shapes associated with nozzle geometry and swirl motion variations have been computed. In Figure 2, a graph giving the computed output power against burned mass fraction (BMF) just before the exhaust valve opening, the optimisation way is represented in red. This iteration process allows to shift from about 73 % BMF and 72 % power to about 98 % BMF and 120 % power. This improvement is the result of a work on different parameters of the bowl. The bowl dome was modified so as to have more space under the spray and to reduce the auto-ignition delay. Moreover, a main characteristic of this design is to promote fuel vapour progression along the bowl shape. The fuel has to go out of the bowl so as to reach the air in the squish area. So, a work on the total length of the bowl and its out-section allows an increase of the "extraction" speed of the vapour from the bowl, and finally improves the power and efficiency of this concept. After a few iterations, some combustion chambers, associated with nozzle and swirl definitions, adapted to narrow spray cone angle, have been defined.

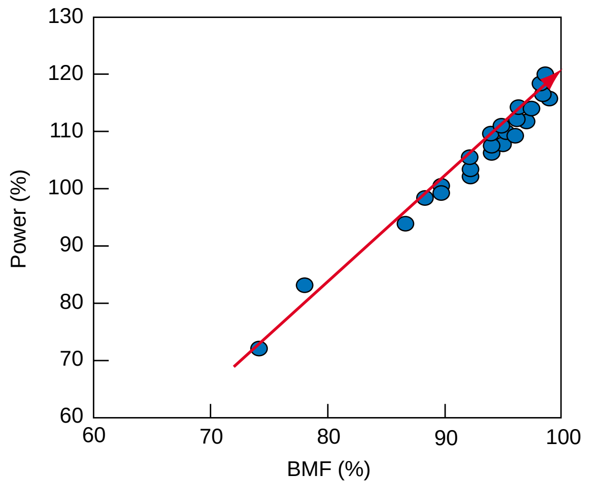

Figure 2

CFD results at 4000 rpm, full load.

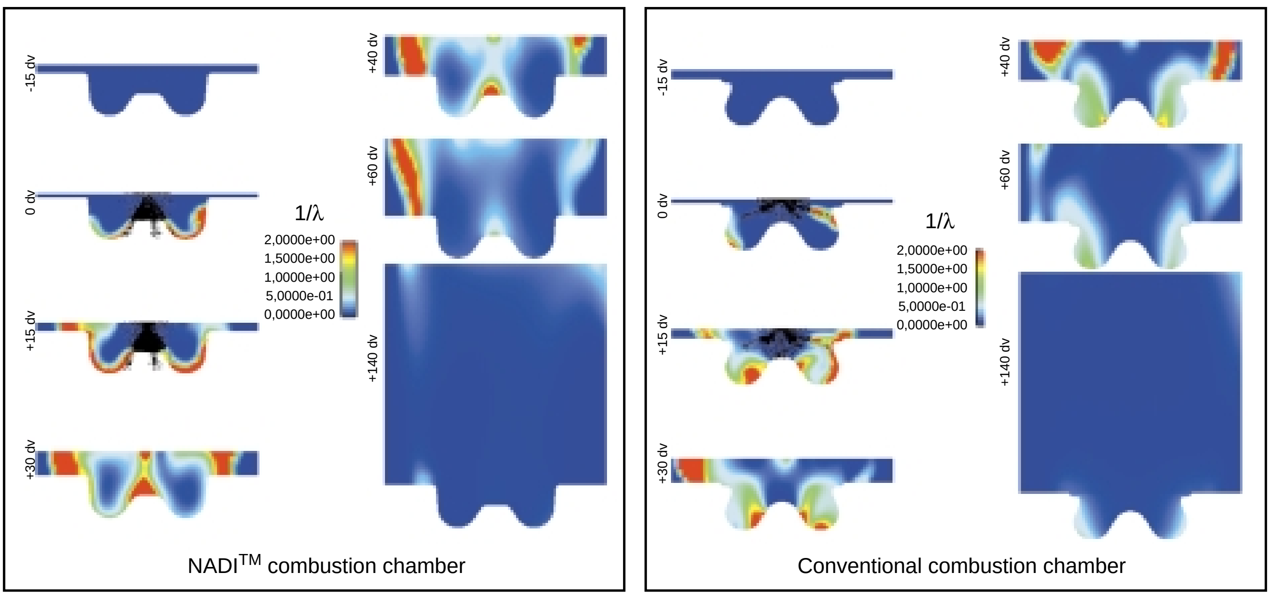

Figure 3

Computed combustion process at 4000 rpm full load.

2.2 Best Configuration

CFD computations helped to define the best fuel injection and combustion process. Figure 3 shows the combustion process at 4000 RPM, full load, with a conventional nozzle and combustion chamber geometry and with the NADI™ concept. The fuel air mixture is represented in a colour that depends on the lambda value: the fuel droplets in black, the fuel vapour in red, the air in blue. With conventional combustion system, the fuel is injected towards the bowl periphery. Due to fuel/wall interaction, the majority of the fuel is sent to the centre of the bowl, mixes with air and burns. Some fuel mixes with air and burns in the squish area. With the NADI™ concept, the fuel is injected at the centre of the combustion piston bowl. Due to fuel/wall interaction, fuel is transported to the piston bowl periphery, mixes with air and burns.

3 NADI™ ENGINE DESCRIPTION

The experiments were performed on a single-cylinder engine.

The main characteristics are:

- bore: 78.3 mm;
- stroke: 86.4 mm;
- displacement: 416 cm^3;
- compression ratio: 16:1.

The intake ducts have been modified in order to adapt the swirl motion. All the results shown in this paper, including full load conditions have been obtained with the same swirl number. A first generation production Bosch common rail fuel injection system (maximum injection pressure: 135 MPa) has been modified in order to allow up to eight injections per cycle. The engine was externally boosted and the exhaust pressure was controlled by a throttle valve in accordance with intake pressure and EGR values.

4 FULL LOAD TESTS

The first target was to validate the concept on a single cylinder engine with conventional combustion mode at full load.

4.1 Reference

For the concept development program, the first objective was to find the same performance, in terms of power, as for the initial conventional combustion system. To have a fair comparison basis at full load, the original combustion piston bowl was modified in order to reduce the compression ratio to 16:1. With the next typical Diesel engine parameters settings and limiting factors, which are used in most of the actual European Diesel engines, a power of 45 kW/l[1] was achieved at 4000 rpm, full load:

- maximum in-cylinder pressure: 15 MPa;
- exhaust temperature: 750°C (equivalent to 710°C on a single-cylinder);
- smoke: < 3 FSN;
- intake pressure: 205 kPa;
- exhaust pressure: 315 kPa;
- intake temperature: 60°C.

4.2 Comparison with NADI™ Combustion System

The conventional combustion system and the first version of the NADI™ concept were compared on the same single-cylinder engine at 4000 rpm, full load. Figure 4 shows the exhaust temperature and the smoke evolution during a fuel injected quantity increase. NADI™ combustion system shows good conventional Diesel combustion behaviour, reaching almost the same power (only 1% less) with more smoke, but still below the limit. This first engine evaluation had confirmed the possibility to find a combustion chamber adapted to narrow spray cone angle. Nevertheless, some drawbacks were pointed out at this initial development stage. We have observed an increase of the ignition delay and a smaller reaction rate. So, these preliminary results have been obtained with an advanced injection timing, with some drawbacks on maximum in-cylinder pressure (+ 0.5 MPa),

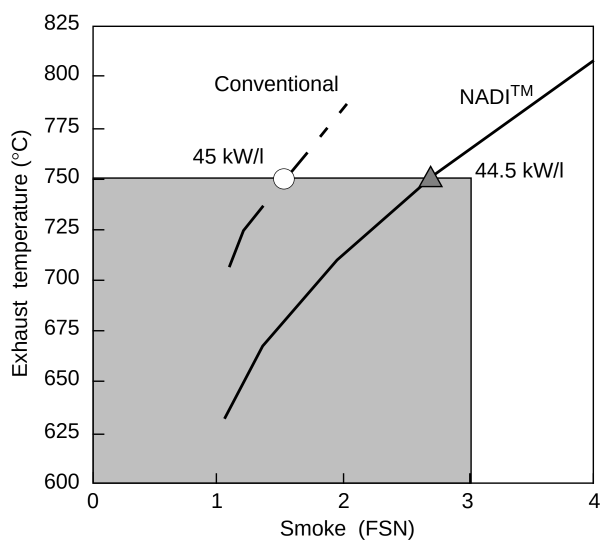

Figure 4

First results at 4000 rpm, full load.

(1) Calculated from IMEP, with a friction mean effective pressure (FMEP) or 2 bar.

on fuel consumption (+10%) and on combustion noise (+2 dB). As it will be discussed later, improvements have been done to avoid these drawbacks.

4.3 Improvment at Full Load

After the first evaluation of the concept, the combustion system has been improved to obtain better results at high load, using conventional combustion mode. Then, this improved concept has been evaluated at part load using HCCI combustion mode.

The future Diesel engine requirements in terms of power density (up to 60 kW/l) will lead to higher intake pressure and in-cylinder pressure. In this context, it becomes obvious that a too high sensitivity to injection timing could be a critical point. The combustion bowl shape has been modified in order to reduce the ignition delay and to improve the combustion rate by promoting a better air/fuel mixing. As seen for Figure 2, the “extraction” speed of the bowl was increased by a reduction of the out-section and thanks to a work on the total length of the bowl and on bowl dome shape.

Figure 5 shows the in-cylinder pressure and the burned fraction with the initial and optimised geometries with an intake pressure of 260 kPa and maximum cylinder pressure of 15 MPa. The ignition delay is reduced by 3° CA and it is possible to achieve 6.3% more power in the same conditions.

With this improved design, the combustion behaviour is less sensitive to injection timing variations and it is possible to reach the future Diesel engine requirements in terms of power density. Figure 6 shows the improvements: the intake pressure has been increased up to 300 kPa at a constant maximum in-cylinder pressure of 16 MPa, and with less than 3 FSN for the smoke and 710°C for the exhaust temperature.

5 RESULTS IN HCCI MODE

5.1 Combustion Control

As said in the Introduction, combustion control is the key point in HCCI mode.

We have developed many ways to have a good control of combustion, using EGR, the effective compression ratio, gas temperature and injection timing.

5.1.1 EGR

External EGR is the main way used to control the ignition of the combustion, with a given compression ratio. EGR rate is defined as:

$$\mathrm{EGR}(\%) = 100 \times \frac{\mathrm{CO_2 intake}(\%) - \mathrm{CO_2 air}(\%)}{\mathrm{CO_2 exhaust}(\%) - \mathrm{CO_2 air}(\%)}$$

As Figure 7 shows, air dilution by exhaust gas increases the ignition delay and allows good combustion timing control. However, the increase of EGR decreases the air/fuel ratio and could, above a certain limit, increase the unburned fuel with a negative impact on fuel consumption: high EGR rates are not totally compatible with high load, and in these

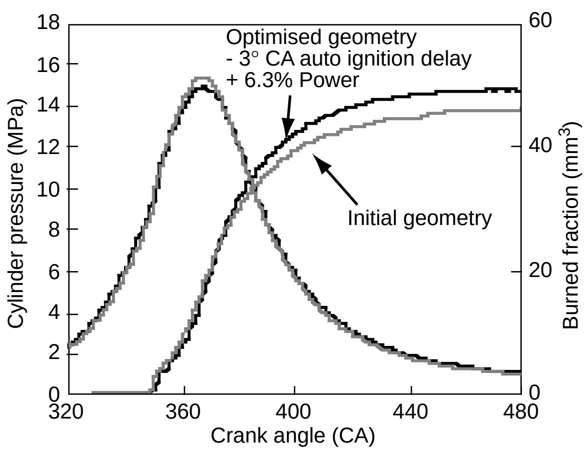

Figure 5

Comparison between initial and optimised geometries.

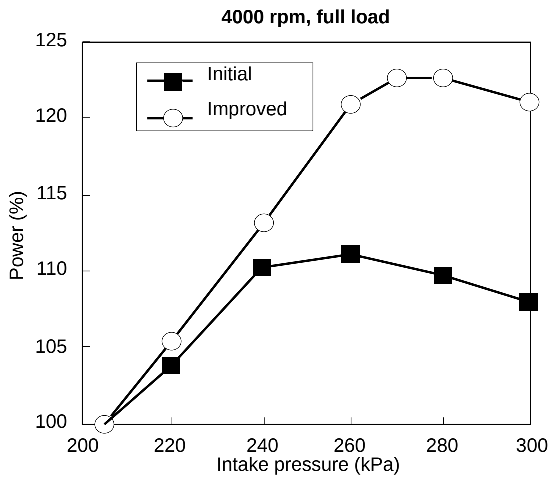

Figure 6

Engine power output *versus* intake pressure (in-cylinder pressure < 16 MPa).

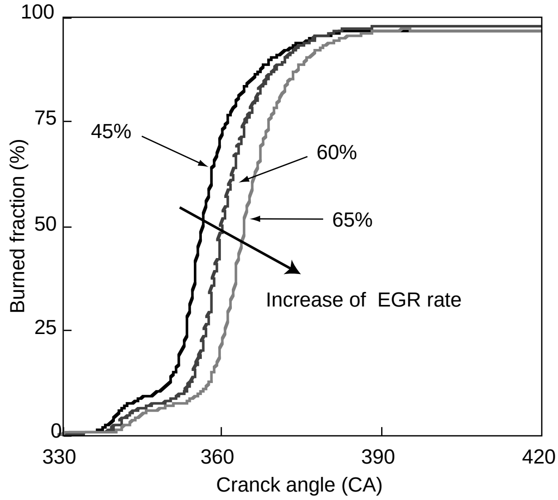

Figure 7

Influence of EGR on combustion timing.

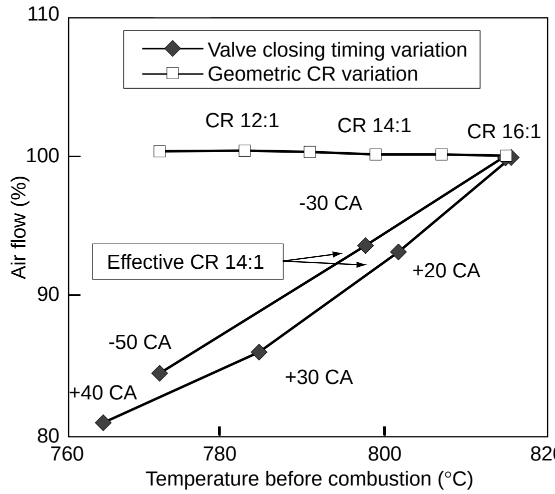

Figure 8

Influence of valve closing timing on effective compression ratio.

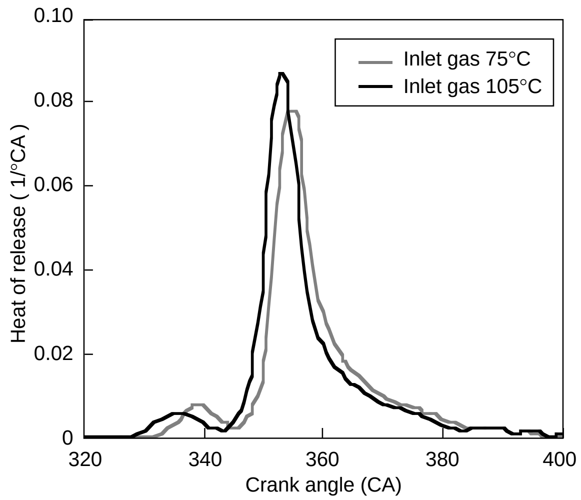

Figure 9

Effect of intake gas temperature on the combustion.

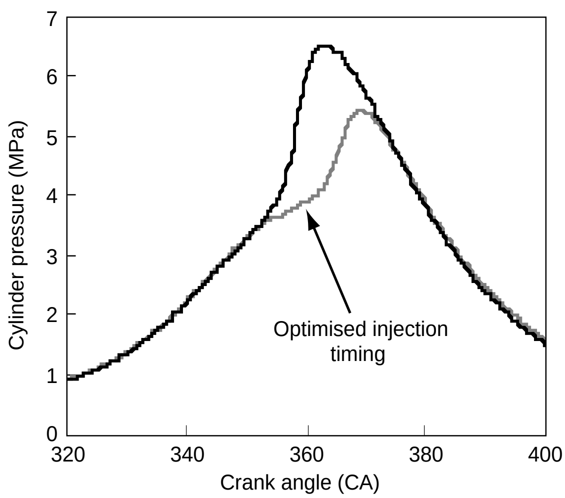

Figure 10

Effect of injection timing.

cases, a decrease of the effective compression ratio is necessary to have the same ignition timing with a smaller EGR rate.

5.1.2 Compression Ratio

At this initial stage, two geometric compression ratios (16:1 and 14:1) have been used to easily evaluate its effect on combustion ignition. In the meantime, 1D calculations have shown that it is possible to adjust pressure and temperature evolution against crank angle by varying the intake valve closing at constant geometric compression ratio of 16:1. As shown in Figure 8, an earlier valve closing of 30° CA or a later valve closing of 20° CA gives the same ignition characteristics as a geometric compression ratio of 14:1 with a small loss of airflow.

Combined with high EGR rate, the decrease of the effective compression ratio allows to increase the maximum load.

5.1.3 Intake Gas Temperature

Another parameter to control the combustion timing is the intake gas temperature: the higher the temperature, the sooner the ignition. To control this temperature, EGR cooling is necessary, especially with high EGR rates. Nonetheless, EGR does not have to be too cooled as problems of condensation of unburned hydrocarbons appear with a temperature below 70°C. Figure 9 shows the impact of a decrease of the intake gas temperature (mix of air and EGR in intake manifold) on the combustion ignition timing. Note that there is an effect on the low temperature oxidation phase as well as on the main combustion.

5.1.4 Injection Timing

A fourth way to control the ignition timing is linked to the injection strategy. As shown in Figure 10, injection timing could be optimised in order to change fuel stratification and to interact with auto-ignition sites.

5.2 Engine Results

Two engine speeds have been selected to perform our first HCCI mode engine evaluation (1500 rpm and 2500 rpm). The results obtained in HCCI mode with the NADI^{TM} concept geometry are compared to results with conventional combustion mode with standard geometry using a compression ratio of 18:1, and parameters settings consistent with Euro III emissions standards.

5.2.1 Results at 1500 rpm

The Figures 11 to 13 show fuel consumption and pollutant emissions *versus* engine load. The boost- and back-pressures used for these tests were very close from values used on the production engines. Indeed, for the 1500 rpm tests, only a light boost pressure has been used above 0.5 MPa of IMEP with the lowest compression ratio.

Operating Range and ISFC (Fig. 11).

With the geometric compression ratio of 16:1, it is possible to achieve 0.4 MPa of IMEP using HCCI mode. The fuel consumption increase is due to the unburned fuel at the exhaust (HC and CO), which could represent 10% of the injected fuel at 0.4 MPa. With the geometric compression ratio of 14:1, it is possible to achieve 0.6 MPa of IMEP using HCCI mode. The fuel consumption is better because less EGR is needed to control the combustion timing well with a net benefit on the unburned fuel which represents only 2% of the injected fuel quantity at 0.5 MPa of IMEP.

Emissions

For NO_x and particulate emissions, as Figure 12 shows, the levels are near zero (always below 0.03 g/kWh) and consistent with SULEV standard levels. NO_x emissions are more than 100 times lower than conventional Diesel engine and particulate emissions are more than 10 times lower. The HC and CO emissions are not too high with levels in the same range or lower than direct gasoline engine ones (*see Fig. 13*). However, as the exhaust temperature is low, especially at low load, improvements of oxidation catalyst performances will be required.

5.2.2 Results at 2500 rpm

Figures 14 to 16 show fuel consumption and pollutant emissions against engine load. At high load, the EGR flow has been cooled to maintain the recycled gas temperature at about 90°C. The intake pressure has been progressively

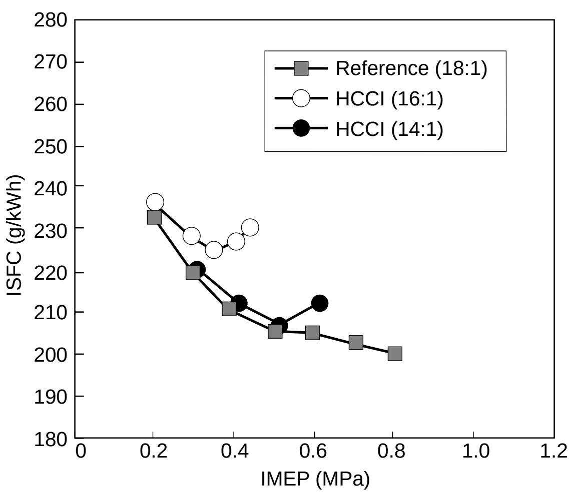

Figure 11

ISFC *vs* IMEP at 1500 rpm.

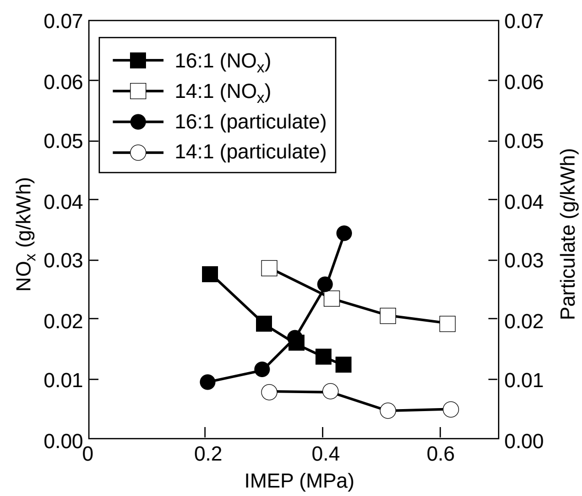

Figure 12

Emissions *vs* engine load at 1500 rpm.

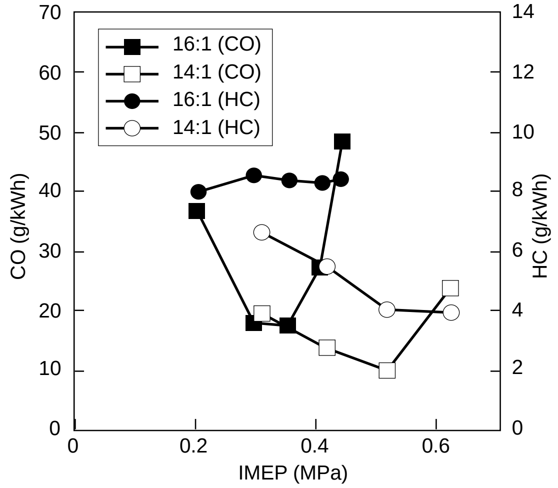

Figure 13

Emissions *vs* engine load at 1500 rpm.

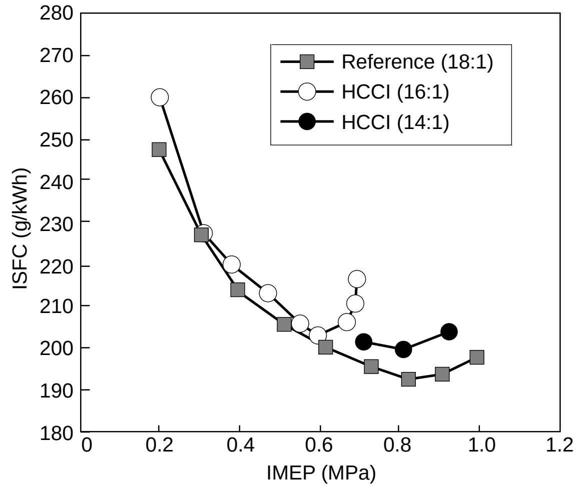

Figure 14

ISFC *vs* IMEP at 2500 rpm.

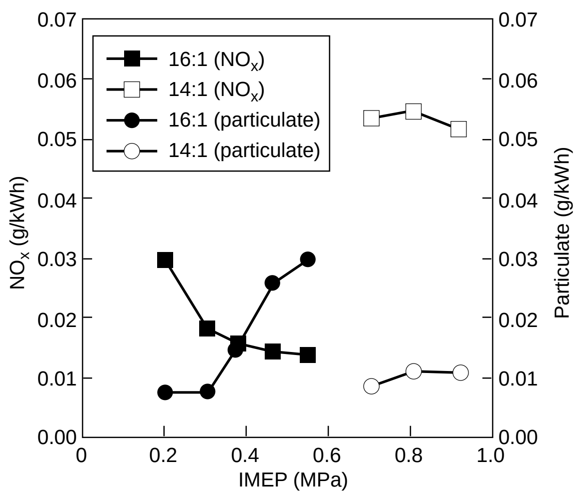

Figure 15

Emissions *vs* engine load at 2500 rpm.

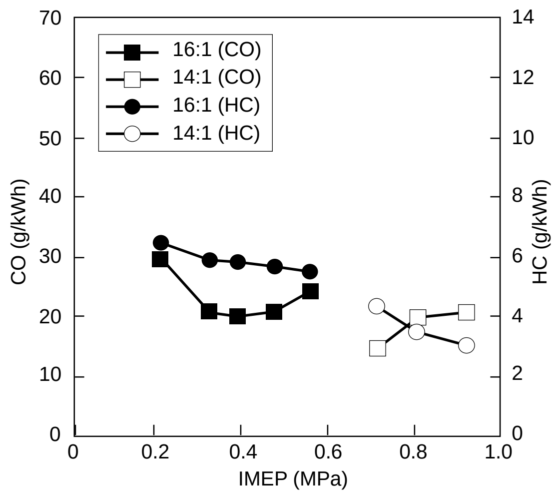

Figure 16

Emissions *vs* engine load at 2500 rpm.

increased for IMEP higher than 0.5 MPa, but remains lower than conventional value in order to take into account the fewer energy available on the turbine, due to HCCI combustion mode.

Operating Range and ISFC (Fig. 14)

Due to higher engine speed, it is possible to achieve good HCCI combustion mode at higher load. It is possible to achieve 0.6 MPa of IMEP with a compression ratio of 16:1, and 0.9 MPa of IMEP with a compression ratio of 14:1 using HCCI mode.

Emissions

Once again, near zero NO_x and particulate emissions are achieved as shown in Figure 15.

The HC and CO emissions are in the same range as direct gasoline engine ones *(Fig. 16)*. At this engine speed, the exhaust temperature is high enough to oxidise more easily the HC and CO by a catalyst. Depending on engine parameters settings such as EGR or injection timings, it is possible to find different trade-off between the four pollutants. The results shown with the lowest compression ratio illustrate this point.

6 PERSPECTIVES

The NADI™ concept has already achieved interesting results. 50 kW/l has been reached with conventional limiting factors and engine parameters setting (in-cylinder pressure below 15 MPa, exhaust temperature below 750°C, smoke less than 3, intake pressure of 230 kPa). In order to fulfill the future DI engine standard, directions of progress are identified.

To improve the conventional combustion mode, the increase of injection pressure (from 135 MPa to 160 MPa or higher) will promote the air/fuel mixing. The nozzle configuration (number and diameter of the holes) could be optimised by taking into account the completely different trade-off between pollutant emissions at low and medium load and power at full load.

To improve the HCCI combustion mode, the control parameters settings such as the split of injection, the EGR or the boost pressure has to be deeply optimised. The impact of new technologies suchlike as piezzo-electric common rail FIS to increase the injection events, electric assisted turbo charger to enlarge the operating range or, for advanced development, fully VVA system, has to be precised.

Due to the complexity of the processes involved, *IFP* has already planned to use sophisticated scientific tools, such as optical access engines and associated diagnostics and 3D CFD tools, to acquire the necessary knowledge to support the NADI™ concept development process to the industrial application.

CONCLUSION

This paper has presented the main design features and preliminary results of the NADI™ new combustion system applied to an existing DI Diesel engine. It was shown that this concept could be an answer to the Diesel engine challenges. This “dual mode” engine has reached near zero NO_x and particulate emissions at part load (including MVEG operating range) by using some homogeneous charge compression ignition combustion and is compatible with further specifications concerning power density. In the HCCI mode, a lot of development studies have been done on the control of combustion with the use of:

– high EGR rate;
– variable effective compression ratio thanks to valve timing variations;
– intake gas temperature (cooled EGR);
– injection timing.

Some points have to be improved and further development works are already planned:

– CO and HC emissions have to be improved, especially at low engine speed and engine load;
– transition strategies between the two modes have to be defined, with a special attention to EGR control;
– HCCI engine load domain has to be extended a little.

Nevertheless, the use of advanced engine technologies such as VVA, low temperature oxidation catalysts light off, electric assisted turbocharger or further generation of common rail fuel injection system make practical applications of NADI™ engine realistic in near future production powertrains

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APPENDIX

The KMB code, which stands for Kiva-Multi-Block, is based on the KIVA-II code originally from *Los Alamos Laboratory* [A1]. It makes possible computation of fully 3 dimensional, transient, compressible, chemically reactive, turbulent flows in moving geometries, with sprays. The formulation of the equations, the averaged Navier-Stokes' equations coupled with turbulence equations, is an arbitrary Lagrangian Eulerian finite volume formulation, resolved on hexahedrons in structured domains.

A multi-block version developed at *IFP* [A2], allows computation in complex geometries. Each part of a geometry, for example the cylinder and the intake port, can be meshed and logically structured independently. Some overlap allows physical data to be transmitted from one block to the other. This method drastically reduces the number of ghost cells with respect to a single block containing all the geometry.

A remapping algorithm is also used at valve closing, to transfer all the computed fields from a grid with valve used for intake phase, to a grid without valve used for compression and combustion phases. This avoids very thin and distorted cells near the engine head during these two last phases. This more homogeneous grid allows larger time steps and better accuracy during injection and combustion. These abilities are described in [A3].

The calculations presented here use the following models to describe the physical processes occurring within the combustion chamber: a standard κ–ε model for turbulence, a classical logarithmic law of the wall, two models for Diesel combustion (a kinetic combustion model for self-ignition developed at *IFP* [A4] and the Magnussen model [A5] for turbulent combustion), and the Diwaker model [A6] for heat transfer.

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