

A Study of Methods to Lower HC and CO Emissions in Diesel HCCI

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ABSTRACT

The dramatic change in terms of pollutant constraints for diesel engines, with future Euro-6 regulations for example, will probably require the improvement of alternative combustion modes such as homogeneous combustion (Homogenous Charge Compression Ignition - HCCI). These new concepts allow the reduction of NO_x and particulate emissions to very low levels for low loads thanks to a high level of external Exhaust Gas Recirculation (EGR) while maintaining CO₂ emission advantage of diesel engines.

Nevertheless, due to a resultant low combustion temperature, HC and CO emissions rise significantly, especially at low load when the catalyst bed temperature is not sufficient for their aftertreatment. This paper describes three considered ways to potentially overcome this barrier thanks to HCCI combustion improvement. After a preliminary optimization of the combustion system performed on a single cylinder, the work was operated on a mid-size multicylinder engine to reduce engine out HC–CO levels.

Firstly, it focused on the optimization of multi-injection strategies in HCCI combustion: the Heat Release Rate could be split up with a direct impact on the noise level and a reduction of the EGR rate to modify the trade-off with NO_x emissions in favor of HC and CO emissions.

Secondly, to facilitate their oxidation in the combustion chamber, two concepts were also examined with the aim of a higher in-cylinder temperature. Whereas the study of a short high-pressure-loop EGR without cooler shows that it could be a simple attractive low-cost solution, the evaluation of a Variable Valve Lift system highlights the

potential of an exhaust secondary lift by using Internal Gas Residual (IGR) in replacement of an EGR part.

Finally, combination of adapted injection strategies and an increased global burnt gas temperature allows extending their respective benefits with an HC–CO emissions reduction of 60% for identical low NO_x emissions level compared to standard HCCI results. Further development would be necessary to deal with NVH and transient control.

INTRODUCTION

The performance and exhaust emissions of Direct Injection Diesel engines have been improved significantly in recent years (figure 1). Whereas the major challenge for gasoline engines is reducing fuel consumption, diesel engines have to fulfill stricter regulations by reducing emissions to zero-impact levels while minimizing CO₂.

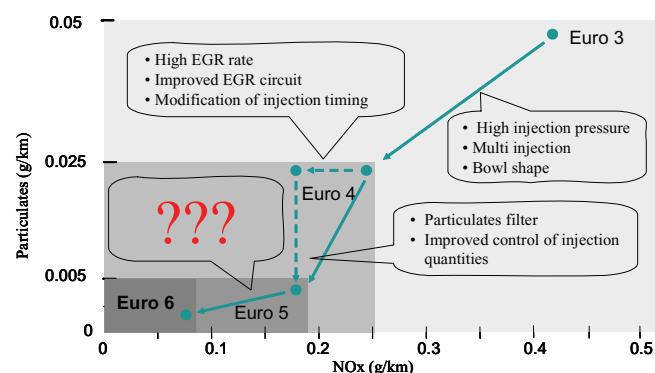


Figure 1: Diesel engine evolution from Euro III to considered Euro VI

Actually, specific exhaust aftertreatment systems such as SCR or Lean NO_x Trap are attractive solutions for reducing NO_x emissions at intermediate and high load thanks to rich gases and a sufficient exhaust temperature level. However these complex and relatively expensive solutions are not so efficient at very low load.

To avoid losing part of diesel advantages, new combustion developments are key issues in this field and HCCI has shown promising results in terms of near zero NO_x and PM. However, this concept presents some drawbacks especially on the raw HC and CO emissions at low load.

In a diffusion combustion mode, even at low load, these emissions are insignificant due to a high localized temperature during the combustion and the proximity of oxygen enabling their oxidation (figure 2).

In premixed combustion mode, EGR allows the combustion process to be controlled by reducing the local temperature and by decreasing the oxygen partial pressure in the cylinder: NO_x emissions and particles are dramatically lessened. In return, complete post oxidation of HC and CO is then difficult to achieve.

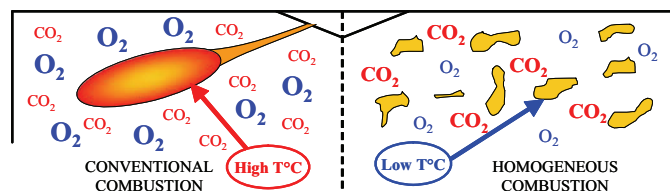


Figure 2: Comparison between CI and HCCI combustion

This paper deals with three different solutions to reduce HC–CO emissions at low load in HCCI (figure 3):

1. Develop new specific injection strategies in order to find a better trade-off between NO_x emissions and HC–CO emissions for an identical noise level.
2. Use of high-pressure-loop EGR without cooler to increase the burnt gases temperature.
3. Use of VVL system to substitute some EGR by IGR [1] and [2].

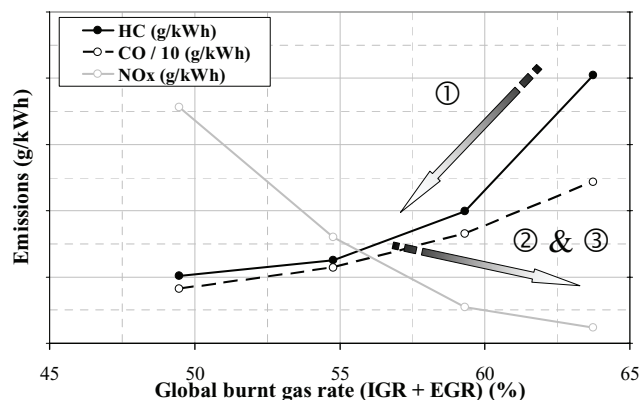


Figure 3: Example of HC–CO and NO_x evolution with the global burnt gas rate (IGR + EGR)

EXPERIMENTAL CONDITIONS

The design of the specific hardware for this study was experimentally tested on a single cylinder engine, especially to define the best combustion system (compression ratio, injector flow, bowl shape, swirl...). It was then integrated on a Renault mid size multicylinder engine.

Two different air loop systems were tested:

1. Short high pressure EGR loop without cooling in addition to low pressure EGR loop (figure 4).
2. VVA system with double event at the exhaust cam in addition to low pressure EGR loop (figure 5).

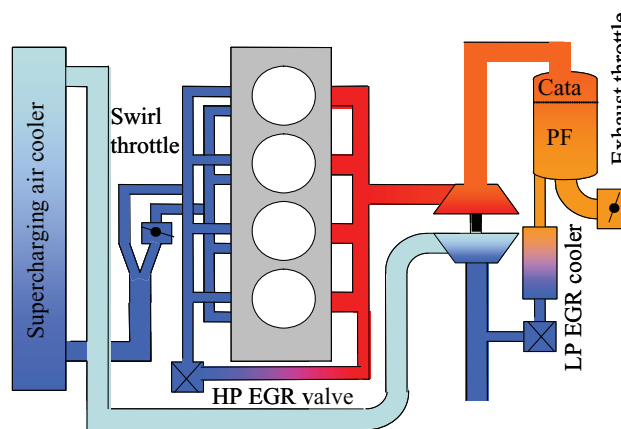


Figure 4: Renault engine concept with LP / hot HP EGR

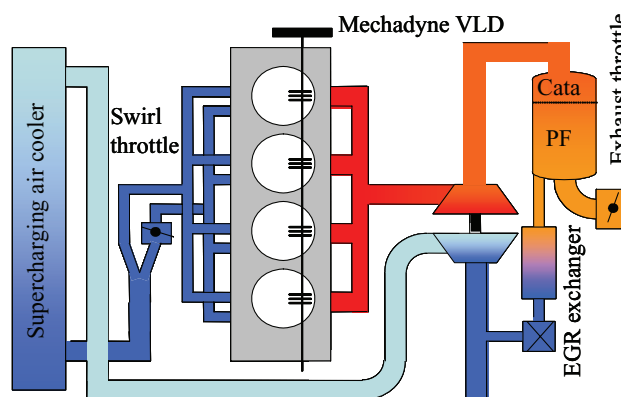


Figure 5: Renault engine concept with VVA + LP EGR

The low pressure EGR concept allows for efficient homogenization, a good mixture distribution on each cylinder and prevents any problem of fouling in the intake side, thanks to a necessary particulate filter [3]. Hot high pressure EGR or a double event VVL system allows an increase in the global burnt gas temperature and consequently better HC and CO emissions. The combination HP EGR or VVL system with LP EGR allows the regulation of the burnt gas temperature.

ADVANCED COMMON RAIL INJECTION SYSTEM

The injection system used is a Delphi Direct Acting System technology [4] which relies on a novel injector operating principle, where the nozzle needle is directly driven by a piezo actuator (Figure 6).

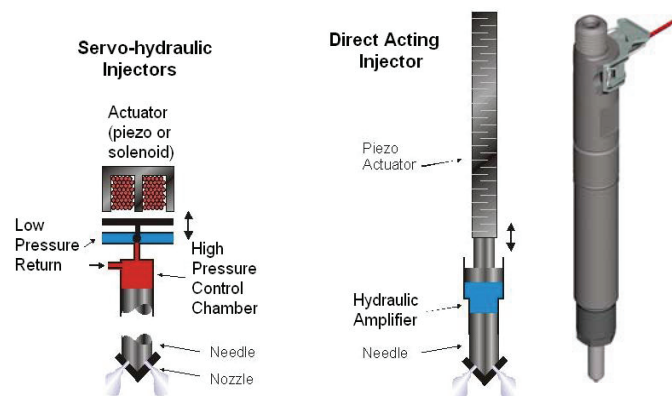


Figure 6: Delphi Direct Acting Injector (DFI3)

This operating principle significantly accelerates and simplifies the operation of the injector (Figure 6). In a conventional servo hydraulic injector, a multi stage physical process is needed to initiate an injection: actuation of a piezo or solenoid actuator, leading to the opening of a servo hydraulic valve, initiating orifice flows leading to a rather sensitive pressure balance in a control chamber that leads to an opening of the needle. Compared to this complex process, the direct actuation is simple: a large diameter piston and a smaller diameter needle are part of a control chamber. The actuation of the piston by a piezo actuator leads to a stroke amplified motion of the needle. The direct acting operating principle addresses a target conflict that every servo hydraulic injector suffers from: good minimum quantity capability requires a low needle opening speed, whereas for high effective pressures, i.e. a square rate of injection, and thus good emission performance, a fast needle opening speed is required. The removal of the servo-hydraulic flow processes leads to such improvement in controllability and stability that the direct acting injector can both be very fast in needle speed, and at the same time deliver small pilot quantities in good accuracy with very low shot to shot variation.

This new technology concept allows both fast and controlled opening and closing of the nozzle needle, with an accurate control of injection quantities. The needle approaches a maximum speed of 3m/s compared to conventional Common Rail servo-systems solenoid or piezo which are closer to 1m/s. With such speeds the injection rate literally approaches a square wave (Figure 7).

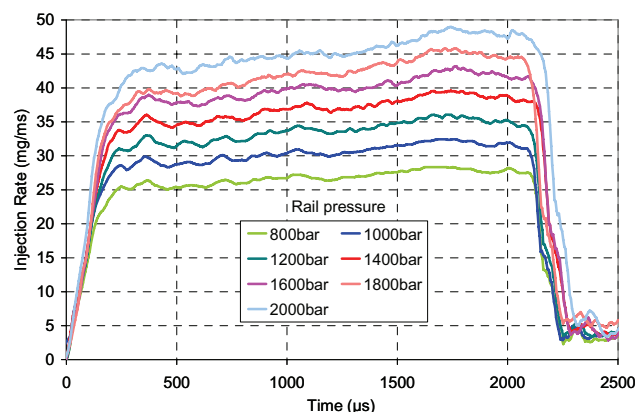


Figure 7: Injection rate measurement for different rail pressures

This type of injector leads to an excellent multiple injection behaviour and has the capability of operating at very high pressures up to 2000bar. Besides the fast needle motion, the injector literally contains a small rail supply of fuel within it, eliminating the pressure wave effect. The result is an injection separation specification of 0μsec. Figures 8 & 9 show the fuel delivery stability with the Direct Acting Injector compared to the conventional servo controlled one. The direct acting injector improves the multiple injection control capability.

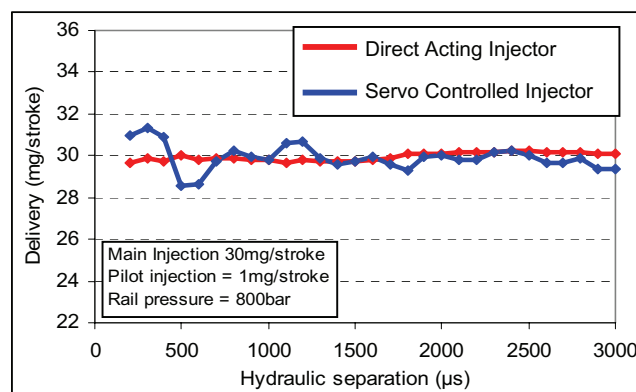


Figure 8: Influence of hydraulic separation between pilot and main injections, on the main injection delivery (no compensation)

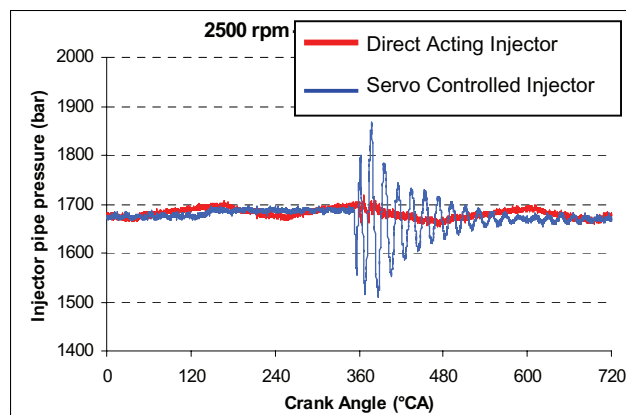


Figure 9: Pressure oscillation in injector pipe before and after the main injection.

THE MECHADYNE VLD SYSTEM

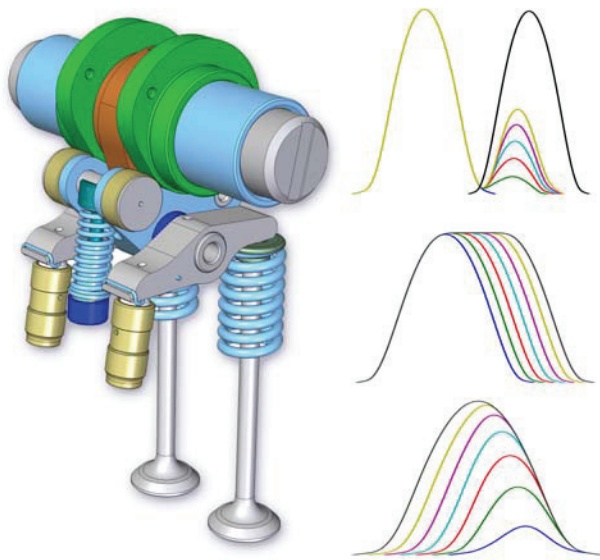


Figure 10: The VLD Mechanism and associated Lifts

VLD is Mechadyne's camshaft operated variable lift system, a particular feature of which is its ability to produce controlled secondary valve openings.

A key feature of VLD is its modularity and the cylinder head for this project was designed so that it can be upgraded from a conventional finger follower valvetrain to VLD by simple part substitution.

The engine cylinder head has been designed to take advantage of this modularity. As such, the engine can run with either a standard finger follower valvetrain or VLD on both intake and exhaust.

The VLD system used to obtain the test results for this paper produces a variable exhaust re-opening (see the top curve in figure 10). This proves to be a highly efficient way of introducing hot exhaust gasses back into the combustion chamber.

How does THE VLD Mechanism work?

The VLD system works by adding together two cam profiles. A summing rocker that pivots in-between the two lower finger followers performs this addition.

Altering the relative phase between the green and brown cam profiles (see figure 10) changes the sum and therefore controls the valve lift.

The Camshaft

The camshaft uses Mechadyne's concentric camshaft technology. It has an inner shaft and outer tube assembly. The green cams in figure 10 are fixed to the tube whilst the brown cams are pinned through slots in the tube to the shaft.

A simple hydraulically controlled rotary actuator can then be used to alter the relative phase of the shaft and tube and hence the cams. The benefit of this approach is that the control for the system is integrated within the camshaft. No external shafts are required.

The Cylinder Head and Cover



Figure 11: Shows the open cylinder head structure.

Because the VLD system takes up the space that is usually assigned to the cam bearings, a new cylinder head and cover assembly was used where the cover contains all the cam bearings. This produces a very simple, modular, cylinder head and cover structure as shown by figures 11 and 12.

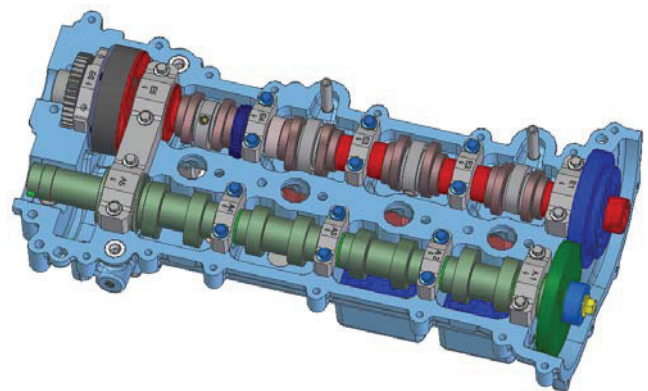


Figure 12: Shows the cover assembly with integral cam bearings.

More detail relating to the design of the cylinder head and valvetrain, including dynamic rig test results from this engine, is included in [5].