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Benefits and Drawbacks of Compression Ratio Reduction in PCCI Combustion Application in an Advanced LD Diesel Engine

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ABSTRACT

The present paper describes an experimental study on the effect of the compression ratio on the performance of a LD diesel engine operating with a PCCI calibration, near the estimated EURO 6/Tier2 Bin5 NO_x emission limits.

The research activity is the result of a collaborative project between Istituto Motori and Centro Ricerche Fiat aimed to carry out an exhaustive analysis of the compression ratio (CR) influence on the performance of a LD diesel engine. Starting from a reference engine configuration the CR was reduced in two steps sequentially. Each CR value was characterized under PCCI operation mode and, under conventional diesel operating mode, at maximum torque.

The exploration of the PCCI application in the NEDC operating area was performed prefixing limits on maximum fuel consumption, maximum pressure rise and maximum tolerable smoke.

The main result was that no significant increment in PCCI application area reducing the CR was possible without overcoming the limits. But thanks to a drastic drop in smoke emission for the low CR values, a further enlargement of near EURO 6 PCCI application area was possible by means of engine recalibration.

The results allowed the definition of the optimum range values of the CR able to minimize the NO_x-particulate trade-off coupled with acceptable increment in unburned compound emissions (HCs+CO) and FC, helping to the definition of the line-guides for the future EURO 6/Tier2 Bin5 engine design.

INTRODUCTION

A large field of the internal combustion engine research is today dedicated to the investigation on the new concept of Premixed Combustion Compression Ignition (PCCI) operation mode. Despite the interesting advantages offered by PCCI condition in terms of very low NO_x and soot emissions, several problems still remain for its large application to the modern diesel engines.

At the moment, one of the main issues, is the limitation of its use in the low and medium load/speed range of the whole engine operating map, due to the difficulties in controlling simultaneously both the premixed air-fuel charge and the ignition timing. The extension to higher speed and load ranges, in fact, is coupled with problems of combustion noise control and excessive fuel consumption increment [1]. At the same time, of course, it is essential to enlarge the PCCI applicability in the whole NEDC cycle, to take advantage of its well known potentiality in reaching the very low NO_x emissions in view of the future EURO 6 regulation. However even if a further development in PCCI combustion does not guarantee the fulfilment EURO 6 limit, requiring a foreseeable application of complex after-treatment systems as $DeNO_x$ plus DPF, PCCI use will help to reduce the “load factor” on the exhaust catalytic systems, reducing the fuel consumption increment required for system regeneration.

Many engine operating and geometrical parameters have a strong influence on both the quality of the premixed charge and ignition timing and as a consequence on combustion characteristics of PCCI in the diesel engines [2]. Some operating parameters are quickly and precisely controlled by the adoption of technologically advanced

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accessories as swirl throttle valves, sophisticated turbo and EGR systems, variable valve actuation system and so on. However, among all, the most effective parameter in controlling the ignition delay time, by means of the controlled pressure and temperature values at TDC, remains the compression ratio (CR) [3, 4, 5, 6]. A lot of technological solutions for the realization of the real variable compression diesel engine (VCR) are under development, but up to now they are still in the prototype version. Therefore, in order to extend the PCCI applicability to the modern light duty engines to the greatest possible engine operating area, the compression ratio has to be chosen carefully. In fact, a too low CR value, if permits a long ignition delay period and so high premixing air-fuel level [5], it suffers for cold engine startability and reduced engine efficiency, mainly at high load. On the contrary, an high CR gives no problems in cold starting and permits good performance at high load but limits the premixing air-fuel time and the EGR tolerance thus reducing the benefits of PCCI combustion.

To this aim, an extensive research program was planned between Istituto Motori and Centro Ricerche Fiat. The experimental program lasted two years including both simulation and experimental activities. At the end of the activities a complete database on the performance of the FIAT LD diesel engine versus compression ratio value was carried out. Some of these results were already presented in a previous paper [6].

The present paper describes a part of the research activities performed within the project and in particular the results collected testing three different compression ratio values in a reference four-cylinder engine running in conventional diesel and PCCI combustion mode. The explored CR values were chosen in cooperation with Centro Ricerche Fiat on the basis of a preliminary analysis in a range of possible real solutions for the next generation of LD engines, taking into account also the current worldwide trends in diesel engine CR reduction.

In the following, the methodology used for the design of the piston geometries, the engine test bench characteristics, the adopted testing methodology and the results analysis are described in detail.

The effect of compression ratio on LD diesel engines was explored in various previous experimental works; some of them are reported in literature [5, 7, 8, 9, 10, 11] and regard studies mainly performed on single-cylinder research engines. However a critical approach to the exploitation of low CR values for emission reduction on both conventional and PCCI combustion is not present in literature until now. Following this aim, the peculiarity of the present paper is the analysis of the effect of CR reduction on the performance of a four-cylinder engine, at the state of the art of the current technology, and with the imposition of realistic limits in smoke emission, combustion noise and fuel consumption increment with respect to the current standards. This testing methodology will be carefully described in a dedicated section.

THE MULTI-CYLIDER ENGINE

The engine selected for the research is the FIAT 1.9 MultiJet. Its displacement represents the most popular size of diesel engine for passenger car in the European market. The main characteristics of the engine in its EURO 4 version are listed in Table 1.

Table 1: Main features of the multi-cylinder engine

Engine type	In-line 4 cylinders
Bore [mm]	82.0
Stroke [mm]	90.0
Compression Ratio	17.5
Displacement [cm3]	1910
Valves per cylinder	4
Injection system	Bosch Common Rail 2nd generation
Injector	(1600 bar of maximum injection pressure)
Turbocharger	Bosch CRIP 2.2, Centred 7 holes microsac
Swirl control	Single stage VGT, model Garrett GT1749MV
Catalyst system	Throttle valve on intake duct electronically controlled
Maximum Power	Closed-couple DOC plus underfloor DOC
Maximum Torque	112 kW @ 4200 rpm

The engine is fully instrumented for indicated signal measurements in each cylinder (cylinder pressure, injection pressure, energizing injector current). At the engine exhaust, smoke was measured by a high-resolution (0.01 FSN) smoke meter (AVL415S), while gaseous emissions were measured upstream and downstream the oxidation catalyst by means of an analysis test bench (AVL-CEB-2).

An overview of the engine layout scheme with the main sensors positions is displayed in Figure 1.

The fuel employed for the tests is a commercial low sulphur diesel fuel, sold in Italy by AGIP Petroli with the name of “BluDiesel”, whose characteristics respect the EN 590 standards.

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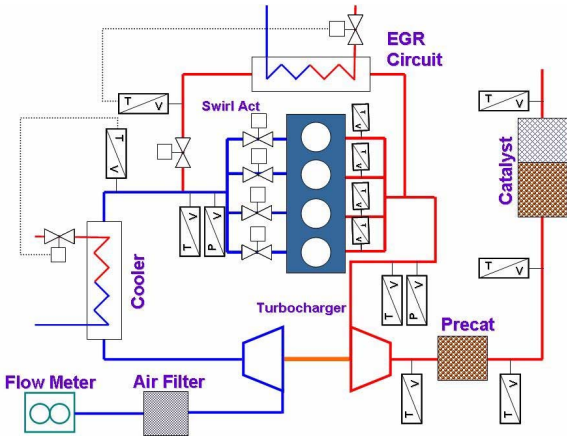


Figure 1: The FIAT four-cylinder engine layout.

COMPRESSION RATIO VALUE SELECTION AND PISTON BOWL DESIGN

The current CR standard of most of the EURO 4 engines is in the range 17.0-18.0. The reference engine in its EURO 4 production version is 17.5 (see Table 1). However a brief survey on the most advanced LD PCCI engines reported in literature, points out that the modified engines have a CR value in the range between 17 and 12 approximately [6, 12, 13, 14, 15, 10, 16, 17, 18, 19]. Also the last trends in diesel engine development indicate a tendency in compression ratio reduction in a range between 16.5 and 15.5 [6, 20].

Looking at the results of the papers dedicated to the analysis of the CR effect on PCCI performance [3, 4, 5, 8, 9, 10] it appears clear that the reduction of CR in a range of 14-16, with respect to the current value of EURO 4 engine, permits the increment of upper engine load limit curve of PCCI applicability, without deterioration of BSFC. However, as a general overview and as clarified in the introduction, it is unclear which are the real limits in CR reduction, considering some practical constrains as BSFC and noise increment that have to be maintained within limit values.

On the basis of the above considerations, three scaled CR values were selected for the experimental program: 16.5, 15.5 and 14.5.

The designs of the three piston bowls, corresponding to the three CR values, were carefully performed with the 3D simulation by Centro Ricerche Fiat.

Starting from the design of the EURO 4 piston shape, the following guide-lines for piston shape design were met:

- To maintain the same air flow structure at the end of compression stroke in order to assure about the same swirl and turbulence characteristics versus CR;
- To avoid an increase in the squish height and keep constant the internal diameter, in order to increase the k-factor (see in Table 2 the definition) and to keep the same squish flow inside the bowl during compression stroke;
- To keep the same lip profile of the bowl in order to guarantee the same structural robustness at rated power.

In this way, all the characteristics of air-flow mixing process (spray/wall interaction, spray dispersion etc.) and combustion products oxidation of the designed bowl shape of the production piston should be preserved. Furthermore, the difference carried out by experiments will be ascribed mainly to the CR variation.

At the end of the work the three shapes were licensed as plotted in the Figure 2. In the Figure 3, it is also reported the scheme of the piston geometry indicating some main geometrical parameters.

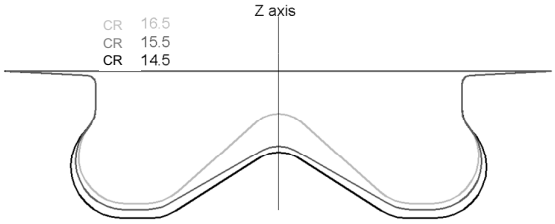


Figure 2: Piston bowl shape for the three selected CR values. In the figure is reported also the Z axis assumed as the central axis of the cylinder.

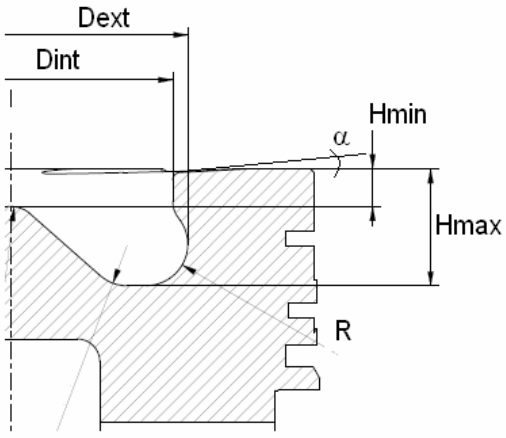


Figure 3: Main geometrical parameters of a bowl diesel engine piston geometry.

The well known geometrical factors of the bowl design, RR, K-factor and AR, corresponding to the three CR values, are reported in Table 2 together with the definition of each factor. The significance and the importance of each factor on the diesel combustion can

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be easily found in literature and also described by Cipolla *et al*. [21].

Table 2: Geometrical factors of the bowl geometry for the three CR values

Parameter	Formula	CR 16.5	CR 15.5	CR 14.5
K-factor [%]	Bowl volume/Combustion chamber volume	75.2	76.8	78.4
RR [%]	(Dext-Dint)/Dint	9.3	10.9	13.5
AR [%]	Hmax/Dext	33.0	34.0	35.2

Table 2 indicates that all the factors increase with CR reduction. K-factor increases as desired, the others show a progressive increase versus CR precisely due to the bowl enlargement keeping the same geometrical structure.

The air flow patterns for the three CRs 20 crank angle degrees before the TDC is plotted in Figure 4. The plots are the result of an engine cycle simulation without injection performed at Istituto Motori with KIVA3V code.

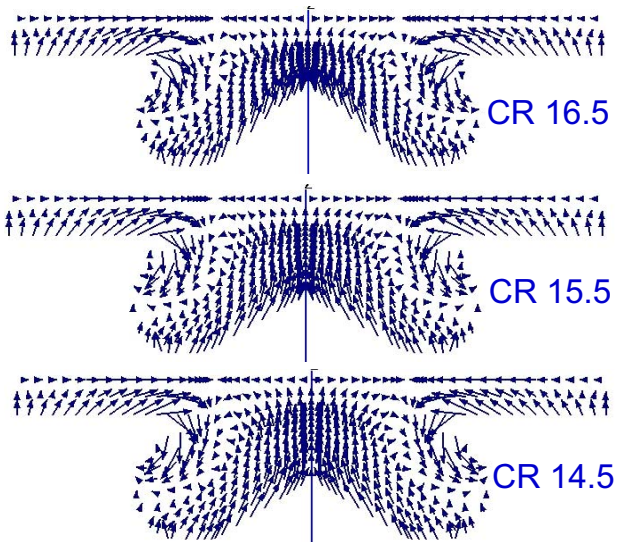


Figure 4: Air flow patterns for the three selected CR values 20 crank angle degrees before the Top Dead Center. The arrows are plotted with the same scale for all CRs.

As expected, from the Figure 2 the main air flow patterns appear very similar, with the main difference for CR 16.5 in the central part of the bowl. In terms of swirl ratio behavior versus CR, Figure 5 reports the three curves for the three CRs. The slight increase of swirl ratio versus CR decrease depends on the friction loss reduction due to the surface-volume ratio reduction as well as from the decrease of moment of inertia. In fact, following the swirl ratio formula [22]:

$$SR = \frac{M_z}{I_z \cdot \pi \cdot \frac{rpm}{30}}$$

lowering the CR, the moment of inertia I_z is reduced and the momentum M_z decrement is less effective due to the lowered friction loss as can be observed in the graphs of Figures 6 and 7 respectively.

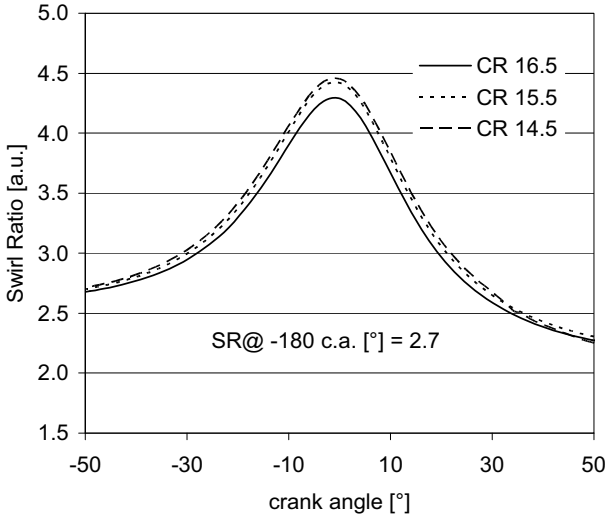


Figure 5: Swirl ratio (SR) versus crank angle for the three CR values.

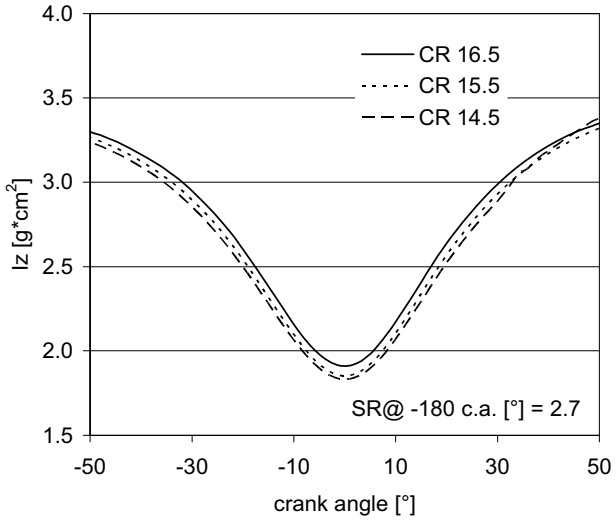


Figure 6: Moment of inertia versus crank angle for the three CR values.

The differences for the three bowls are small and so the differences in terms of ignition delay time as well as

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combustion evolution can be ascribed mainly to the thermodynamic status at the start of injection timing (SOI) derived by CR variation.

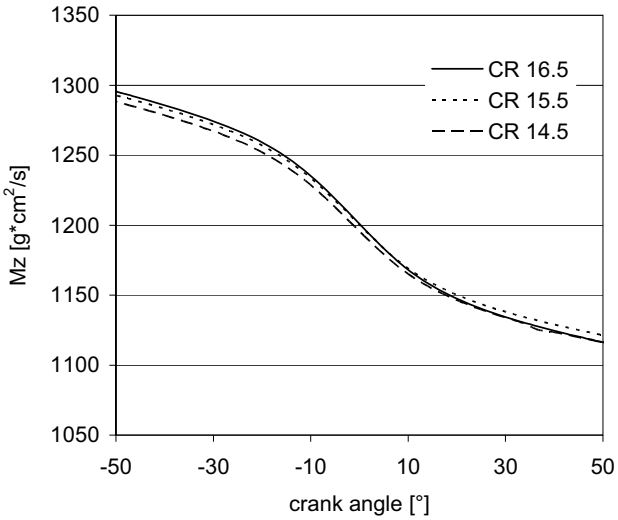


Figure 7: Momentum versus crank angle for the three CR values.

TEST METHODOLGY AND SELECTED OPERATING POINTS

In general, when experimental engine tests are performed in order to evaluate the influence of geometrical and/or operating parameter on combustion performance, the analyzed parameter is varied with respect to a reference well known condition. This is often observable in a lot of works on single cylinder research engines.

However, with this methodology, even if it is possible to put into evidence the parameter influence, very important from a scientific and technical point of view, it is difficult to evaluate the transferability of the results or trends to a real four cylinder engine. The reason is that hardware or software constrains present in the real engine configuration are not carefully considered in the single cylinder experimental tests or are difficultly reproducible. So, the single cylinder research engine data are useful to analyze the effect of parameter variation but not for the quantitative evaluation of four cylinder engine improvement.

Furthermore, often, the tests are carried out in some operating steady state points that are poorly representative of the engine performance in the whole NEDC cycle that includes also transient conditions. On the contrary, the analysis of the CR reduction effect on engine performance under NEDC cycle or transient conditions requires an ad hoc engine calibration for each CR step.

On the basis of these considerations, in order to point out the CR influence on the PCCI engine performance, taking into account both the practical limits of the real engine configuration and a test point matrix reasonable representative of the engine behavior during the NEDC procedure, the following test methodology was chosen.

Starting from an engine configuration with CR equal to 16.5, the CR was progressively reduced to 15.5 and to 14.5.

Starting from a first whole engine calibration supplied by Centro Ricerche Fiat relative to a CR equal to 16.5, the engine operating points were tested for each CR value in an engine speed range from 1400 rpm to 3000 rpm and increasing the engine load from idle until one of the following limits were reached:

- 5% of fuel consumption increment with respect to the EURO 4 calibration value;
- 80 bar/ms of maximum pressure rise (dp/dt_{max}) for each engine condition preserving the current comfort standards;
- 1.5 FSN of exhaust smoke value as estimated limit in order to avoid excessive DPF system loading, when adopted.

The choice of 5% of maximum FC increment with respect to EURO 4 standards can appear too high preliminarily considering the current trends strongly focused on FC reduction. However, its choice depends on the fact that the engine has an EURO 4 configuration, so a further contribution to FC improvement by PCCI use are expected from EURO 5 and EURO 6 technology. Foreseeing the PCCI application to EURO 5 and to EURO 6 engines, a current 3% of FC increment could be recovered by the next technology upgrade. The difference of 2% of the accepted FC increment is due to the injection pressure control system that has the pressure regulator placed on the rail. Therefore, at same condition, the injection pump sends to the rail the total inlet fuel flow. On the contrary, the reference FC data of EURO 4 calibration refers to an engine scheme with a Bosch CP1h high-pressure injection pump that works only on the fuel flow injected in the cylinder (pressure regulator placed on the inlet channel of the pump) reducing the friction power required by the injection pump. The difference in FC penalty for the present injection system configuration as a whole with respect to EURO 4 arrangement was estimated of about 2%.

In this way, the exploration of PCCI combustion mode permits the identification of the maximum load curve versus rotating speed, and then of the relative applicable engine operating area, for each CR value. This also allows to the identification of the factors limiting or improving the PCCI application to diesel engines.

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