

Increasing Power Density in HSDI Engines as an Approach for Engine Downsizing

2010-01-1472

Published
05/05/2010Maria Thirouard and Pierre Pacaud
IFP

Copyright © 2010 SAE International

ABSTRACT

In the context of CO₂ emission regulations and increase of energy prices the downsizing of engine displacement is a widely discussed solution that allows a reduction of fuel consumption. However, high power density is required in order to maintain the power output and a good drivability.

This study demonstrates the potential to strongly increase the specific power of High Speed Diesel Injection (HSDI) Diesel engines. It includes the technological requirements to achieve high specific power and the optimal combination of engine settings to maximize specific power. The results are based on experimental work performed with a prototype single-cylinder engine (compression ratio of 14). Tests were conducted at full load, 4000 rpm. Part load requirements are also taken into account in the engine definition to be compatible with the targets of new emission standards. The influence of boost pressure, peak in-cylinder pressure, injection pressure and nozzle flow rate on full load limitations (exhaust temperature and smoke level) is studied to define the technological requirements for a power density of 80, 85 or 90 kW/l.

INTRODUCTION

Reducing fuel consumption is a main issue in all current engine developments, whether for complying new CO₂ emission legislations or meeting consumer's expectations regarding cost or ecological aspects. Due to their advantage in fuel consumption compared to gasoline engines, the market share of Diesel engines in Europe have seen a continuous increase over the past years: Diesel engines help reducing the CO₂ fleet consumption of new passenger car engines and help to achieve the CO₂ emission legislations. An important improvement in the specific power and torque output over the years has contributed to this success [1].

In this context of growing concerns about global warming different approaches are being studied by the scientific community to reduce fuel consumption: hybridization, alternative fuels, downsizing, vehicle weight reduction, alternative transmissions. Among these, downsizing is a widely discussed approach to reduce fuel consumption within passengers cars [1,2]. Downsizing, or the reduction of the engine displacement, requires high power density in order to keep a constant power output and guarantee the drivability customers are used to. However, increasing the power density of Diesel engines requires a strong engineering effort in many fields of the engine development. Some of them are:

- 1). Engine architecture: As a result of higher peak in-cylinder pressures and heat flux density, the engine has to be capable of resisting higher levels of thermo mechanical stresses. Some of the most sensitive parts are the cylinder head, the piston and the crankshaft. A great improvement may be achieved by optimizing the design and the cooling system (oil cooling in the piston and water cooling in the cooling jacket of the cylinder head) to reduce the stress levels [3,4]. Other ways to improve the resistance of the engine are the use of new alloys with new surface treatments that improve mechanical characteristics of the materials or the improvement of the fabrication process.
- 2). Injection system: In order to increase the power density, higher levels of fuel flow are used. The fuel injection should be as fast as possible in order to guarantee good combustion efficiency and low exhaust temperature. The potential of high injection pressure on the power output has already been studied [5,6,7,8]. The injection system should be capable of providing fuel pressures above 200 MPa. It is clear that the injection system needs to be flexible and precise since the range of required fuel is increased. At the same time, part load requirements limit the nozzle hole size to assure a good mixing process.

3). Combustion system: Since the reduction of the engine displacement shifts the operating points towards higher loads, the combustion system needs to be adapted for high load operating points in order to meet the emission legislation.

4). Boost system: High levels of boost pressure are required in order to achieve high power density. The boost system needs to be upgraded and improved charging systems are needed.

This study will discuss the potential to achieve high power density with an optimized engine architecture and a prototype injection system that allows a maximum fuel pressure of 250 MPa. The combustion system has been optimized for high power results at full load without neglecting part load constraints. To guarantee satisfactory part load results, the defined combustion system uses a reference nozzle flow rate of 0.64 l/min @ 10 MPa which is a low value for a high power density engine. Tests were performed at full load, 4000 rpm, in a wide range of boost pressure, maximum in-cylinder pressure, fuel pressure and nozzle flow rate to follow up the work started in [5]. An optimization of these parameters allows to define the best technical solutions in order to achieve different levels of specific power output. Finally, this work presents results of high specific power (94 kW/l) obtained with this prototype engine.

EXPERIMENTAL SETUP

TEST ENGINE

Tests were performed with an IFP single cylinder engine. The technical data of the engine are presented in Table 1. Low compression ratio of 14 is adopted as it is fitted to performance enhancement and emission control [9]. The bowl shape geometry (H, W, Htip, Wlip) has been optimized at full load and part load (Figure 1).

Table 1. Characteristics of single-cylinder engines

Bore [mm] x Stroke [mm]	85x88
Displacement [cc]	499
Compression ratio	14
Bowl W/H ratio	3,8
Nozzle flow rate [l/min @10 MPa]	0.56 to 0.85
Number of nozzle holes	6

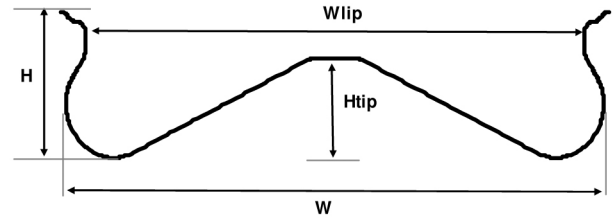


Figure 1. Bowl shape

PROTOTYPE INJECTION SYSTEM

Tests were carried out with a prototype common rail injection system capable of very high fuel injection pressure (250 MPa). The injector is designed and supplied by Delphi based on the DFII generation which uses a fast balanced solenoid valve concept. This injector technology also provides a low fuel back leakage, which contributes to a very good hydraulic efficiency and leads to a very high pressure capability [10]. Various nozzle definitions (number of holes, spray cone angle and flow rate) were tested for the combustion system optimization.

MEASURING EQUIPMENT

Test bench equipment includes standard control systems to regulate air, fuel, oil, and water temperature feeding the engine so that test conditions could be controlled to be as similar to those of a four-cylinder engine. Intake air flow rate was controlled by a sonic flow nozzle. Intake and exhaust air pressure were independently controlled. Raw emissions were measured by a standard 5 gas analyzer. Fuel / air equivalence ratio was calculated from the exhaust gas analysis. Soot is measured by a smokemeter AVL 415. The engine was equipped with a water cooled in-cylinder pressure sensor AVL QC43D. The cylinder head was specially machined to place the pressure sensor as close as possible to the central axis of the combustion chamber. Heat release rate was calculated from in-cylinder pressure trace using an IFP internal code.

EXPERIMENTAL CONDITIONS

Tests were conducted at full load, 4000 rpm. Test consisted in a variation of fuel/air equivalence ratio by progressively increasing the amount of fuel injected. Start of injection (SOI) was fixed to reach a set peak in-cylinder pressure. No EGR was used.

The range of variation of the main parameters studied is presented in Table 2 :

(See Table 2 after last section of paper)

Full load tests have two main limitations: the soot level and the exhaust temperature. The soot level limitation of 3 FSN (Filter smoke number) was chosen regarding the particulate

filter loading and car manufacturer's requirements. Exhaust gas temperature is limited by thermo-mechanical properties of engine's components, including turbocharging devices. Increasing this limit is only possible by improving the properties of the materials and the engine design which can bring penalties in turbocharger integration, reliability and cost. Exhaust gas temperature in a single-cylinder engine is lower than in a 4-cylinder engine because of exhaust gas flow reduction and differences in pipe connections length. Equivalence between exhaust gas temperature (T_e) in a 4-cylinder engine and in a single-cylinder engine was established out of previous tests experiences:

$$T_{e \text{ 4-cylinder engine}} = T_{e \text{ 1-cylinder engine}} + 40^\circ\text{C}$$

Two exhaust temperature limits (equivalent to a 4-cylinder engine) were considered in the tests : 840 °C and 880°C.

Since friction losses in a single-cylinder engine are not similar to those in a 4-cylinder engine, an equivalence was established for the friction mean pressure (FMP) as follows:

$$\text{FMP} = 2.5 \text{ bar at 4000 rpm full load.}$$

In this way, a break mean effective pressure equivalent to a 4-cylinder engine (BMEP) could be calculated taking into account the 4-cylinder engine friction losses. Friction losses as considered here take into account the increase of losses due to higher levels of load and in cylinder peak pressure. However, a possible increase of friction losses due to the demand of additional torque needed to reach a fuel injection pressure above 200 MPa was not considered.

All the results presented in this paper represent the values that may be obtained in a 4-cylinder engine with these hypotheses of exhaust temperature and friction losses.

The fuel used in the tests is an EN590 Diesel fuel. This fuel corresponds to a conventional European Diesel fuel (LHV= 42.8 MJ/kg, density= 835 kg/m³ at 15°C) with a low sulfur content (< 10 ppm).

RESULTS

Four main parameters were studied: boost pressure, peak in-cylinder pressure, fuel injection pressure and nozzle flow rate. The impact of these parameters on specific power, smoke level and exhaust temperature is first studied at a constant fuel air equivalence ratio to help understand the involved phenomena. In a second part, the potential of combined optimizations of these settings to improve specific power is discussed. The work was conducted within the following smoke and exhaust temperature limitations :

- Two levels of maximum exhaust temperature : 840 and 880°C.

IMPACT OF BOOST PRESSURE

It is well known that rising boost pressure increases the power output: when working at constant fuel air equivalence ratio the amount of fuel injected is significantly higher ([Figure 2](#)).

(See [Figure 2](#) after last section of paper)

At high boost pressure, the SOI is delayed to guarantee a constant peak in-cylinder pressure, so that combustion phasing is delayed. Additionally, the amount of burnt fuel is higher at high boost pressure but the efficiency is slightly impacted.

These results may be explained by the combustion process shown on [Figure 3](#) :

(See [Figure 3](#) after last section of paper)

1. For high boost pressure, start of combustion is closer to TDC since peak in-cylinder pressure constraint imposes a delayed SOI. The amount of fuel burning in premixed-type combustion is significantly lower since thermodynamic conditions are favorable to shorter ignition delay.
2. The mixing controlled phase as defined here begins after the premixed phase and ends with the end of injection. The increase of boost pressure helps increasing the heat release rate in the mixing controlled phase. [Figure 4](#) shows the evolution of global fuel air equivalence ratio in the cylinder as fuel is injected: even though final equivalence ratio is constant for both test, at high boost pressure the combustion is globally leaner throughout most of the combustion process (and specially throughout the mixing controlled phase). Higher combustion speed may be achieved thanks to the increased fresh air availability within the combustion chamber. The increase in combustion speed explains that the amount of burnt fuel in this phase is significantly increased (both in absolute value and in percentage) compared to the low boost pressure. Since combustion phasing is delayed at high boost pressure due to the peak in-cylinder pressure limit, an improvement of efficiency might be expected if peak in-cylinder pressure is optimized at high boost pressure.
3. The late combustion phase starts with the end of injection and ends when combustion is completed. The amount of fuel burning in the late combustion phase is higher at high boost pressure. End of combustion is delayed.

- 3 FSN maximum smoke level at full load

Table 3. Effect of boost pressure - Fuel/air eq. ratio=0.87
 - Nozzle flow 0.64 l/min - Pmax= 19 MPa - FP= 200MPa

	0.25 MPa	0.34 MPa
SOI [CAD BTDC]	26.0	17.0
10% BMF [CAD]	348.2	356.7
50% BMF [CAD]	368.0	384.4
90% BMF [CAD]	418.9	427.2
90% - 10% BMF [CAD]	70.7	70.5

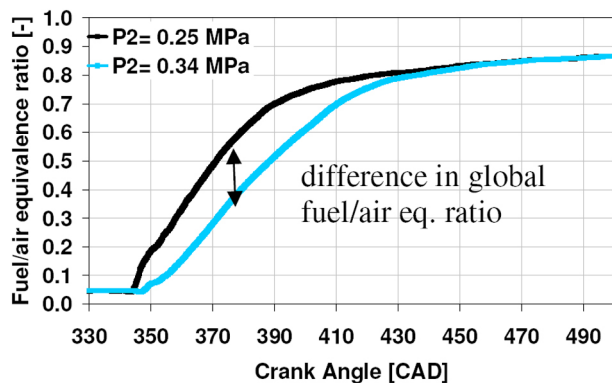


Figure 4. Effect of boost pressure on combustion at constant fuel/air equivalence ratio 0.87 - Nozzle flow 0.64 l/min - Pmax= 19 MPa - FP= 200 MPa

Figure 5 shows the influence of boost pressure on both full load limitations: soot level and exhaust temperature. Final soot level at full load is the result of local phenomena regarding the fuel air mixing process, the resulting local fuel / air equivalence ratio and the post-oxidation duration. It is commonly accepted that, at high speed full load, the later the combustion phasing is (or the end of combustion, 90% burnt mass fraction), the higher the final soot level since post oxidation duration is reduced.

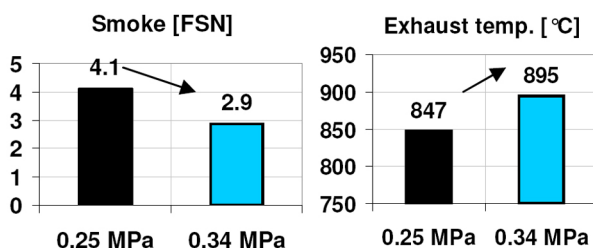


Figure 5. Effect of boost pressure on soot level and exhaust temp.-Fuel/air eq. ratio=0.87 - Nozzle flow 0.64 l/min - Pmax= 19 MPa - FP= 200MPa

Nevertheless, present results show an important reduction of smoke level at high boost pressure even though combustion is delayed. As seen on Figure 4, global fuel / air equivalence

ratio is lower when the fuel is being injected which improves the mixing process. To illustrate this behavior, Figure 6 shows the evolution, during the combustion process, of the difference between the mass of injected fuel in the combustion chamber and the mass of burnt fuel. Results are plotted from start of combustion so that both curves may be compared easily. These curves have three well differentiated parts that correspond to each combustion phase:

1. First, the difference between injected fuel and burnt fuel has a very steep slope that corresponds to the ignition delay. Here, fuel is being injected and is accumulating in the combustion chamber as combustion has not started yet. At the end of this steep slope, the curve has an inflexion point that corresponds to the peak of premixed combustion phase.
2. In the second part, the slope of the curve is lower. This part corresponds to the mixing controlled phase of combustion where diffusion flames are well established. During this phase the fuel is being both injected and burned in diffusion-type flames. This part of the curve would be horizontal if the fuel injection rate was equal to the fuel burning rate. In reality, it appears that the combustion process and in particular the air / fuel mixing is not fast enough for the fuel to be consumed at a rate equal to the injection rate. This phenomena may be referred to as the saturation of combustion. The end of this part is a peak that corresponds to the end of injection.
3. Finally, the curve goes down after a maximum corresponding to end of injection. This part corresponds to the late combustion phase. The difference between injected fuel and burnt fuel goes down to zero when complete combustion is achieved.

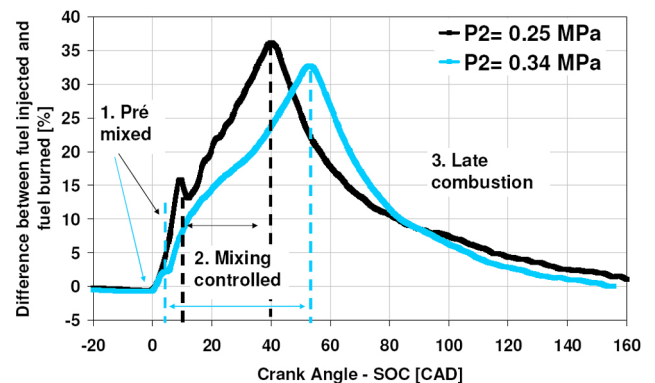


Figure 6. Difference between fuel injected and fuel burned at constant fuel/air eq. ratio = 0.87, Nozzle flow 0.64 l/min - Pmax= 19 MPa - FP= 200 MPa

The combustion saturation is more severe at low boost pressure where combustion speed is clearly limited by the amount of air available in the cylinder. Accordingly, it can be seen that, at low boost pressure, the maximum amount of accumulated fuel in the chamber is higher. The unburned fuel

accumulated in the mixing controlled phase is consumed during the late combustion phase. When the amount of unburned accumulated fuel increases, the end of combustion is achieved later during the cycle and the post oxidation duration is reduced. As a consequence soot level is increased and becomes the limiting factor. As increasing boost pressure allows for increased air availability and thus a faster air and fuel mixing, it induces a reduction of the saturation. However, even at high boost pressure (0.34 MPa) saturation was observed: the fuel injection rate was higher than the fuel burning rate.

The end of combustion also has an impact on the exhaust temperature, which is the other full load limiting factor. However, exhaust gas temperature is strongly dependent on the mass of burnt fuel during each cycle and on the fuel / air equivalence ratio. Keeping fuel air equivalence ratio constant the amount of injected fuel at high boost pressure is significantly higher than in the low boost pressure case. Accordingly, exhaust temperatures achieved at high boost pressure are increased. From this analysis it can be deduced that low boost pressure tests are limited by soot level limitation while high boost pressure tests are limited by exhaust temperature limitation. The boost pressure at which one limitation becomes predominant over the other one depends on the values of the specified maximum smoke level and exhaust temperature.

IMPACT OF PEAK IN-CYLINDER PRESSURE

The peak in-cylinder pressure directly impacts the efficiency since it allows or limits the optimization of the combustion phasing. The maximum in-cylinder pressure sustained by the engine is defined during the engine design. The observed peak in-cylinder pressure depends, among other factors, on the compression ratio of the engine and the boost pressure used during the intake stroke. For commonly used boost pressure values and a CR of 18, increase in maximum peak in-cylinder pressure usually results in increased specific power (Figure 7). When the CR is reduced to 15, the SOI and the combustion phasing can be advanced which results in improved efficiency. At CR 15, the increase in maximum in-cylinder pressure provides a moderate improvement in specific power as the combustion phasing gets closer to the optimum. Further advancing combustion would reduce the efficiency as the combustion phasing would be too early in the cycle.

As seen before, downsized engines use high boost pressure levels to improve power density. Usually the compression ratio is reduced to limit peak in-cylinder pressure and to allow to approach optimal combustion phasing on the efficiency standpoint. However, the impact of the peak in-cylinder pressure on the power density is reduced when the compression ratio is lowered.

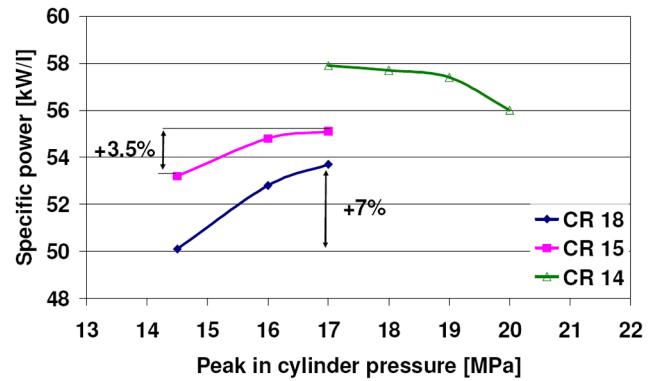


**Figure 7. Impact of peak in cylinder pressure in the specific power at constant fuel air equivalence ratio 0.77-
 $Q_{air} = 275 \text{ kg/h/l} - FP = 200 \text{ MPa}$**

The importance of associating high values of peak in-cylinder pressure with high values of boost pressure has already been studied and discussed [5]. This is specially important when a reduced compression ratio is used. It can be seen in Figure 8 that increasing peak in-cylinder pressure provides an improvement of power density only at high boost pressure in the case of a reduced CR (14) engine. The peak in-cylinder pressure showed no direct effect on the full load limitations (exhaust temperature or soot level).

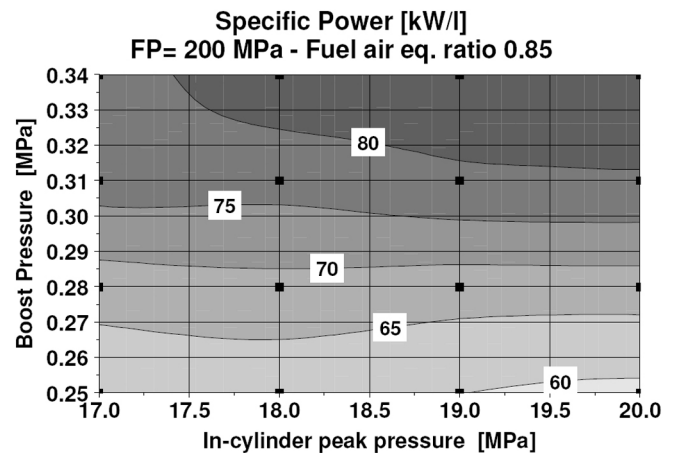


Figure 8. Specific power at $FP = 200 \text{ MPa}$ - Fuel air eq ratio 0.85 - Nozzle flow: 0.64 l/min

IMPACT OF FUEL FLOW RATE

The potential to increase power density by using high fuel injection pressure was previously discussed [5,6,7]. It was shown that increasing fuel flow rate by increasing fuel pressure produces better results in terms of power density than increasing fuel flow rate by increasing the nozzle hole diameter [5]. In this section the potential of combination of high fuel pressure and high nozzle flow rate is discussed. The number of nozzle holes was kept constant at 6. As seen in the previous section, an optimized combination of boost pressure