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ScienceDirect

Energy Procedia 81 (2015) 846 – 855

Energy Procedia

69th Conference of the Italian Thermal Engineering Association, ATI 2014

Numerical investigation on the effects of bore reduction in a high performance turbocharged GDI engine. 3D investigation of knock tendency

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Abstract

Downsizing is a must for current high performance turbocharged SI engines. This is often achieved through the reduction of cylinder number, while keeping constant unit displacement and increasing boost pressure. However, the ensuing higher loads strongly increases the risk of abnormal combustion and thermo-mechanical failures. An alternative path to downsizing is the reduction of cylinder bore: this approach is more expensive, requiring a brand new design of the combustion system, but it also provides some advantages.

The goal of the present paper is to explore the potential of bore reduction for achieving a challenging downsizing target, while preserving the engine knock safety margins.

A current V8 GDI turbocharged sporting engine is taken as a reference, and a preliminary CFD-3D analysis is carried out in order to define the most suitable bore-to-stroke ratio. On this basis, bore is reduced by 11% at constant stroke, thus obtaining a reduction of about 20% on the engine displacement.

In order to achieve the same peak power target, both engine boost and spark advance are adjusted until the knock safety margin of the original engine is met. 3D CFD tools, accurately calibrated on the reference engine, are used to address engine design and the calibration of the operating parameters.

Peer-review under responsibility of the Scientific Committee of ATI 2014

Keywords: : knock, downsizing, bore, 3D-CFD

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Peer-review under responsibility of the Scientific Committee of ATI 2014
doi:10.1016/j.egypro.2015.12.094

1. Introduction

Turbocharging, variable valve timing and direct injection are the most widespread technologies for enabling engine downsizing in high performance petrol cars. BMEP of current production engines has already passed the limit of 23 bar, and this threshold is going to be pushed further. The most important challenges to be addressed in highly-downsized / highly-boosted engines are: air supply at low engine speed and knock. Concerning air supply at low speed, conventional turbocharging systems (typically, one turbocharger per bank of cylinders) present stalling issues. In fact, the maximum compressor flow rate must be about the same of a full size engine, in order to deliver the same airflow at maximum power. However, at low speed, boost pressure should be higher, to compensate the displacement reduction, and the operating points on the compressor maps may go close, or cross, the surge line. The adoption of a two-stage turbocharger system may be a solution to the low speed air supply problem. In particular, the triple turbocharger layout, described in reference [1], may be successfully adopted. Concerning knock, the increase of in-cylinder pressure levels in SI engines has always been limited by the arising of abnormal combustion phenomena. In particular, the autoignition of gasoline-like fuel in the periphery of the combustion chamber induces a variety of damaging mechanisms that eventually leads to severe engine failures, so that knock still remains nowadays a fundamental performance limiters in SI engines [2,3]. As a countermeasure to the increased risk on knock onset, a possible solution may be the reduction of the cylinder displacement by adopting smaller bores. Smaller bores require a completely new design of the combustion chamber, including valve ports, injector and spark plug(s): as bore decreases, design constraints are generally tighter, so that it's more difficult to keep constant the geometrical ratios. On the other hand, a smaller bore yields the following advantages: a) reduced flame path, thus less risk of knocking at the same boost; b) smaller heat transfer area, thus higher thermal efficiencies at a given load and speed; c) smaller and lighter valves, then possibility to adopt more "aggressive" cam profiles; d) smaller and lighter pistons, then less vibrations; e) smaller dimensions and weight of the powertrain. If the stroke does not change from the full size engine, also in-cylinder turbulence intensity remains constant. The paper reports a comprehensive numerical activity aiming at assessing the potential of engine downsizing through the reduction of the cylinder bore. The issues related to combustion development, knock onset, and knock-limited brake specific performance are addressed in the present paper by means of detailed 3D-CFD analyses of a current production V8 turbocharged GDI engine, used as a reference, and of a "virtual" downsized engine. In order to maintain the same overall engine performance over the entire range of engine speeds, the boosting system is completely redesigned, substituting the conventional single-stage system adopted on the reference engine with a triple-turbocharger system, which is described and optimized in a parallel paper [1].

In order to compare the two engines, some assumptions are made:

- equivalent flow losses from ambient to the compressor inlet, and from the last turbine outlet to ambient;
- identical intake plenums (one for each bank) and throttle bodies (one for each bank);
- same correlation between fmep, mean piston speed and in-cylinder peak pressure;
- same geometry of intake and exhaust ports, as well as of the combustion chamber (dimensions are scaled by 10% for the downsized engine);
- same valve discharge coefficients as a function of non-dimensional lift and constraints on profiles;
- the rotational speed range is the same (from 1000 to 7000 rpm).

The downsized engine is at first optimized through 1D-CFD simulations, with the following degrees of freedom:

- intake and exhaust valve profiles and timings;
- intake runners main cross section and length, and exhaust manifolds layout;
- high and low pressure compressor and turbine sizes;
- boost pressure and relative air to fuel ratio (Lambda);
- spark timing.

The goals of the activity are:

- maximum brake torque of 650 Nm at 2000 rpm (corresponding to a bmep of 27.2 bar);
- maximum power higher than 400 kW at a speed lower or equal to 7000 rpm.

The constraints are:

- same or lower level of knocking, in comparison to the full bore engine;
- minimum clearance among piston crown and valves: 1 mm;
- maximum gas average temperature at the turbine inlet: 1000 °C.

2. Methodology

Combustion analyses are developed in the framework of Star-CD 4.20, licensed by CD-adapco. The combustion model adopted for the analyses is the ECFM-3Z, whose well-known suitability for premixed and diffusive and auto-igniting combustion regimes is a fundamental requirement for the present study. In particular, knock is taken into account thanks to the previously described hybrid user-coded/standard formulation. The fuel is injected by means of a 7-hole injector placed between the intake valves. The lagrangian phase is modeled by means of a user coded routine for primary breakup [4], the Reitz model for secondary break-up and the Bai-Gosman approach for droplet-wall interaction. In order to achieve a converged solution, several preliminary RANS cycles are performed based on periodic boundary conditions from a 1D model of the whole engines. Once a cyclic-converged cycle is obtained, this is taken as the baseline for the subsequent analyses. Thanks to the symmetry, the computational grid is halved.
The study is focused on engine performance and knock tendency. Particular care is devoted to achieve an optimal trade-off between accuracy and computational cost: a knock model implemented by the authors [5-7] and based on a tabulated-chemistry approach for auto-ignition is used. The model considers the mass fraction of an intermediate fictitious species for autoignition called Y_{IG}, as defined by Lafossas et al. [8]:

$$\frac{dY_{IG}}{dt} = Y_{TF} \cdot \frac{\sqrt{\tau^2 + 4(1-\tau)^{Y_{IG}}/_{Y_{TF}}}}{\tau} \tag{1}$$

A numerical criterion is then adopted to discern the occurrence of knock, based upon the equality between the mass fraction Y_{IG} and the mass fraction of fuel tracer Y_{TF}, the latter being defined as the "non-reacting" fuel. Since both Y_{IG} and Y_{TF} are transported scalars, it is then straightforward to define a third scalar variable, called Knock Tolerance (*KT*), as the difference between the two species:

$$KT(\vec{x}, t) = Y_{TF}(\vec{x}, t) - Y_{IG}(\vec{x}, t) \tag{2}$$

3. Preliminary investigations

To evaluate the influence of downsizing on knock inception, two preliminary sets of analyses are carried out:

- Cylinder displacement reduction at constant Bore-to-Stroke (B/S) ratio;
- Bore reduction at constant cylinder displacement.

3.1 Displacement reduction at constant B/S

Three different cases are generated by scaling the combustion chamber geometry of the real engine and changing the stroke accordingly. A few simplifications are made: the bowl-in-the-piston is eliminated (flat piston) and a premixed charge at fixed lambda=0.9 is imposed. In this way, the influence of fuel injection, mixing and stratification on the combustion development is neutralized. The same boundary conditions (in terms of time-dependent pressure and temperature profiles) are applied at the intake and exhaust ports. The peak power operating condition is considered.

CASE	BIG	MEDIUM	SMALL
DISPLACEMENT [cm3]	495	430	370
BORE [mm]	90.8	86.5	82.2
STROKE [mm]	76.4	73.1	69.8
SQUISH HEIGHT [mm]	0.8	0.8	0.8
COMPRESSION RATIO	10.38	10.38	10.38
BORE/STROKE	1.18	1.18	1.18

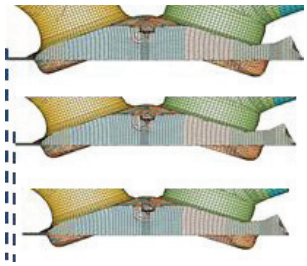

Fig. 1. Displacement reduction at constant B/S.

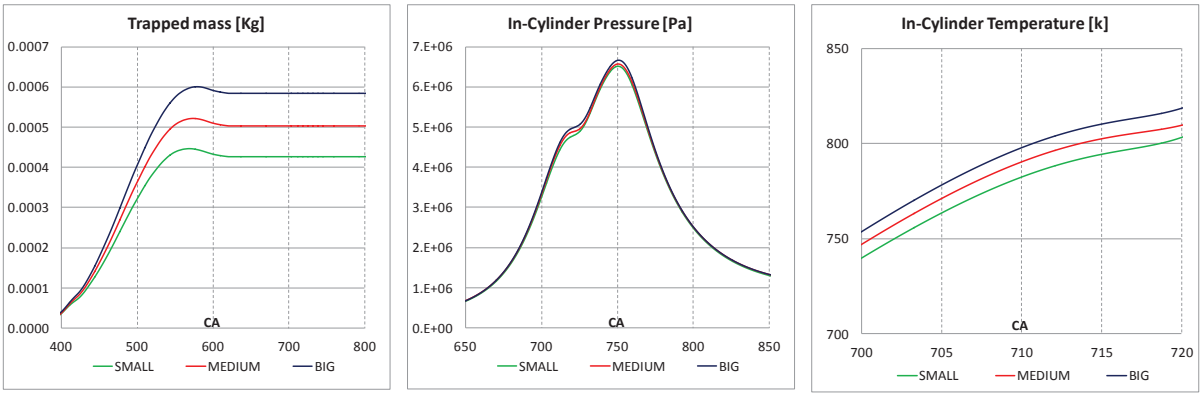

Fig. 2. In-cylinder trapped mass, pressure and temperature at constant B/S.

As visible from Fig. 2, the air trapped mass decreases with cylinder displacement, causing a reduction in engine performance at equal spark advance. The reduced trapped mass corresponds to a drop of in-cylinder charge temperature during the compression stroke, thus reducing the end-gas auto-ignition reactivity, as confirmed by Fig. 3 below, where the knock tolerance function is depicted 20°CA BFTDC.

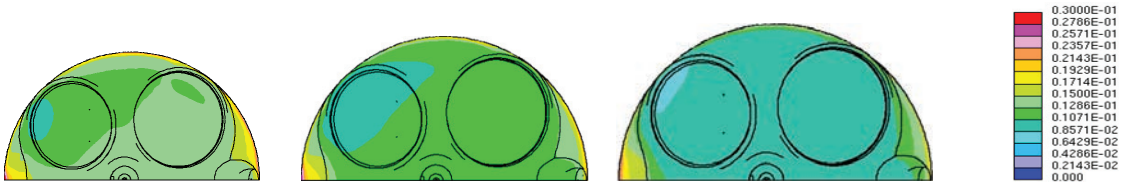

Fig 3. Knock tolerance at constant B/S @ 20°CA BFTDC (blue: low tolerance, red: high).

It is then possible to adjust the spark advance (SA) for each case in order to compare the engines under the assumption of equal knock tendency. In particular, a threshold value equal to that of the spark discharge is imposed for the knock autoignition heat release rate, and SA is varied until such a threshold is met.
As visible, the reduced end-gas reactivity of the smallest cylinder displacement allows the engine to tolerate 3°CA of further advance, which is able to partly compensate the performance loss due to the lower amount of charge. Fig. 5 reports some indicated parameters for the three cases as a function of SA.

The increase in SA is not able to completely compensate the reduced displacement, and a reduction in the overall engine performance is visible for the smallest bore. Nevertheless, in terms of specific performance, the reduction in cylinder displacement proves to be beneficial for a given knock tolerance, as highlighted in the comparison between the two extreme cases reported in Table 1.

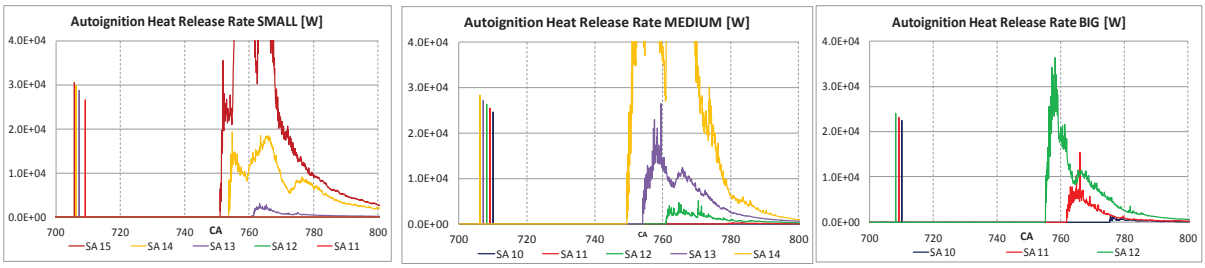

Fig. 4. Autoignition Heat Release Rate at constant B/S, different SAs.

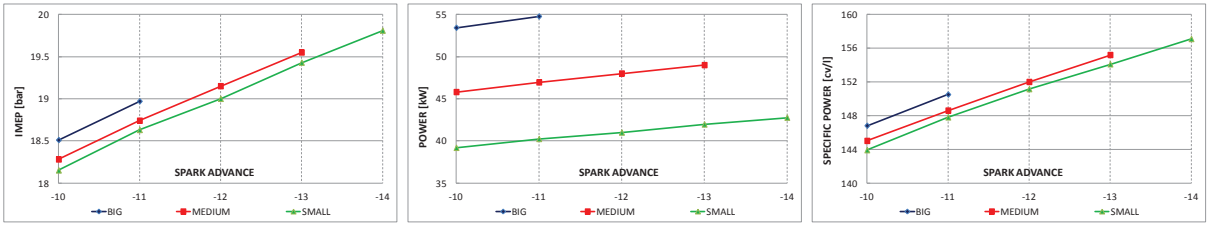

Fig. 5. Imep, Power and Specific Power function of Spark Advance.

Table 1. Comparison between the two extreme cases at constant B/S.

@KNOCK ONSET	BIG	SMALL	DELTA%
DISPLACEMENT [cm^3]	495	370.2	-25.2
TRAPPED AIR [mg]	1035.5	756.71	-26.9
FUEL [mg]	80.2	58.65	-26.9
IMEP [bar]	18.97	19.81	4.4
POWER [kW]	54.78	42.77	-21.9
SPECIFIC POWER [kW/l]	110.7	115.5	4.4
ISFC [g/kWh]	307	288	-6.2

3.2 B/S reduction at constant cylinder displacement

The following analyses aim at evaluating the effect of B/S reduction on knock tendency, for a given cylinder displacement. Starting once again form the original engine geometry, three different cases are generated. The piston bowl depth is adapted in order to keep the same compression ratio and the same squish height at TDC. To eliminate the influence of the bowl geometry on the mean and turbulent flow field, all the simulations start at the beginning of the combustion with equal initial conditions (in terms of pressure, temperature, velocity field and turbulent quantities). As for the previous set of analyses, the SA is varied in order to compare the geometries under the assumption of equal knock tendency. Details of the investigated cases are listed in Fig. 6.

CASE	BIG	MEDIUM	SMALL
DISPLACEMENT [cm^3]	376	376	376
BORE [mm]	80.00	77.00	74.00
STROKE [mm]	74.85	80.80	87.50
SQUISH HEIGHT [mm]	1	1	1
COMPRESSION RATIO	9.6	9.6	9.6
BORE/STROKE	1.069	0.953	0.846

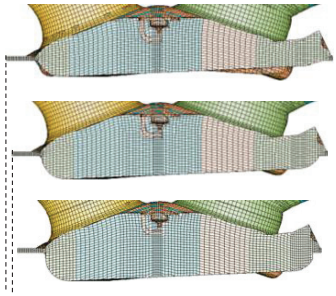

Fig. 6. B/S reduction at constant cylinder displacement.

Results reported in Fig. 7 clearly indicate that the best case is the one with the smallest bore size, whose IMEP increases up to 26.37 bar starting from 25.37 bar of the biggest bore, since it tolerates a more advanced SA for given auto-ignition tendency. Being initial conditions equal, the lower knock tendency can only be attributed to the reduced flame path from the spark plug to the cylinder walls. Fig. 8 confirms the above statement.

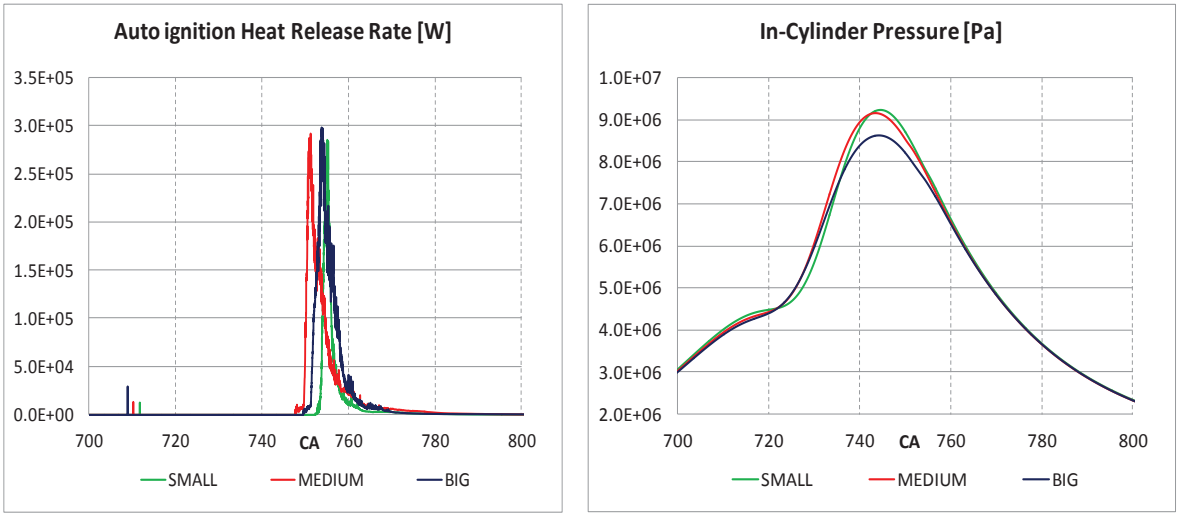

Fig. 7. Auto Ignition Heat Release rate and In Cylinder Pressure at constant displacement.

It may be noticed that, for any given amount of burnt fuel, the flame front is much closer to the cylinder walls in the smallest bore case. Therefore, the amount of charge undergoing knocking decreases as bore becomes smaller. As a consequence, when adjusting SA in order to achieve an equal auto-ignition heat release rate (see Fig. 7), the global amount of energy released by knock is reduced up to 20% (see Fig. 9, case “SMALL SA 9”). Furthermore, assuming equal auto-ignition energy release, the SA can be slightly advanced, leading to a further improvement of the combustion performance and IMEP (case “SMALL SA 9.5”).

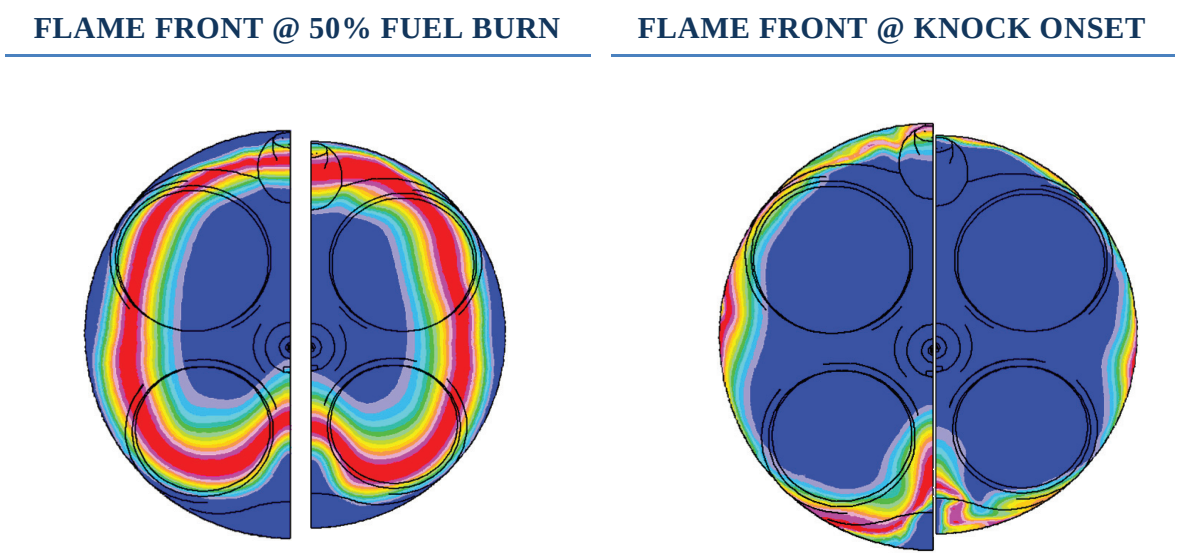

Fig. 8. Flame front at 50% of fuel burned and at knock onset: comparison between the two extreme cases at constant displacement.

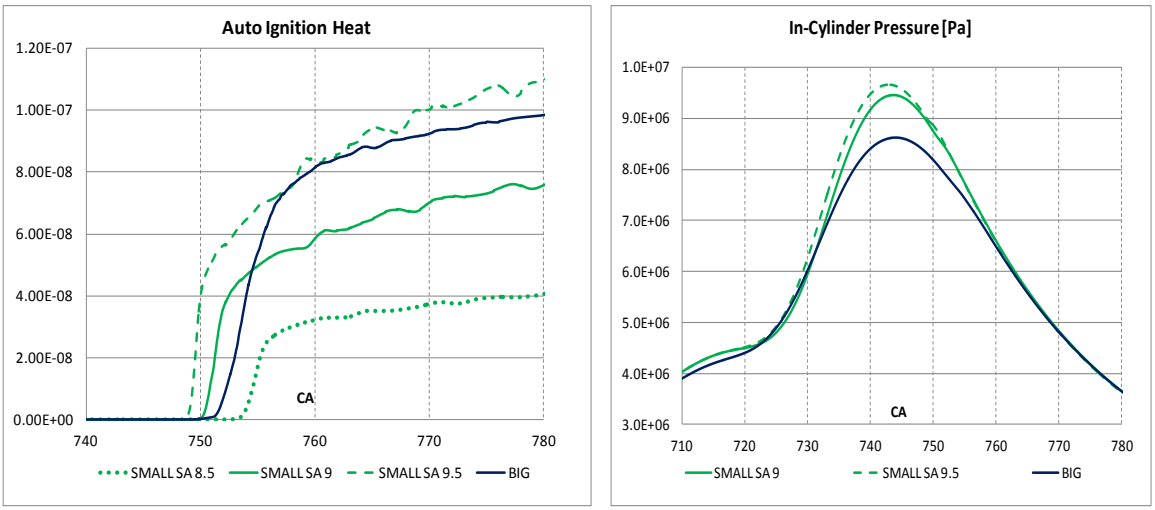

Fig. 9. Auto ignition heat and in cylinder: comparison between the two extreme cases at constant displacement and for several spark advances.

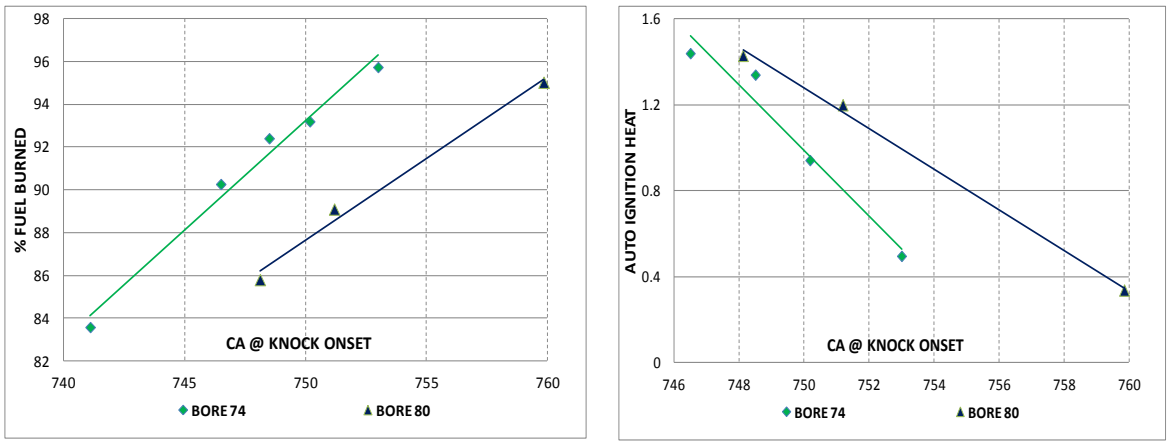

Fig. 10. Comparison between the extreme cases in terms of fuel burned fraction and heat release by knock as a function of crank angle of knock onset (FTDC=720).

Finally, Fig. 10 compares the two extreme cases in terms of fuel burned fraction and heat released by knock at knock onset, further confirming the advantages of bore reduction in terms on performance/knock trade-off.

4. Analysis of the downsized engine

Given the potential advantages of bore reduction presented earlier, the analysis is now applied to the modification of an existing currently made V8 GDI engine, whose bore is reduced by 11%, thus achieving a reduction in cylinder displacement of 21%. In order to achieve the same peak power and low-end torque over the same engine speed range, the turbocharging system of the downsized engine is completely revised, as described in [1], and the compression ratio is slightly reduced. CFD-1D results obtained by [1] are applied as boundary conditions to the 3D-CFD analyses here presented. Two engine speeds are investigated, at W.O.T.: the first is peak power, 7000 rpm, while the second is low-end torque, 2000 rpm. For both cases, the 3D CFD model is firstly validated against

experiments for the actual engine, as visible in Fig. 11, and both engine operations are confirmed to be very slightly knocking towards the tail of the combustion process.

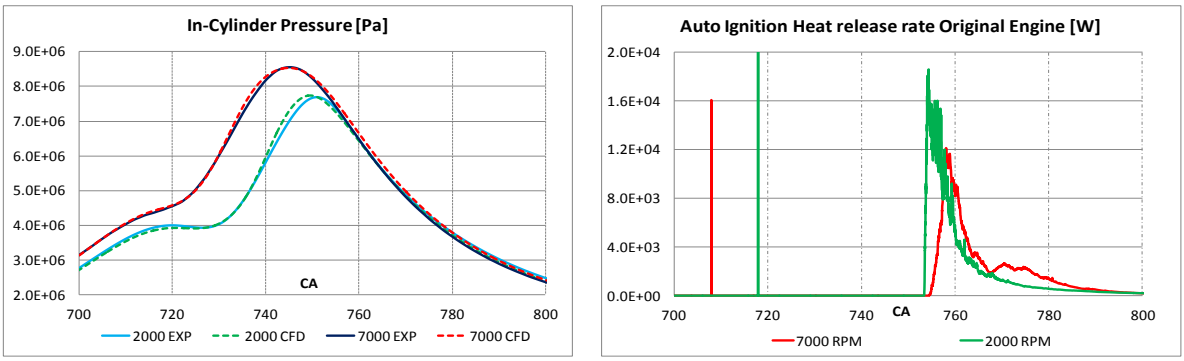

Fig. 11. Comparison between experimental and CFD data at 7000 and 2000 rpm, and heat release by knock in the original engine.

The comparison between the original and the downsized engine is reported below in terms of in-cylinder flow characteristics (both mean and turbulent), mixing and combustion development. In particular, Fig. 12 shows that the two investigated cases are characterized by similar flow structures in terms of both temporal evolution and intensity, with a slightly more intense flow field for the original case.

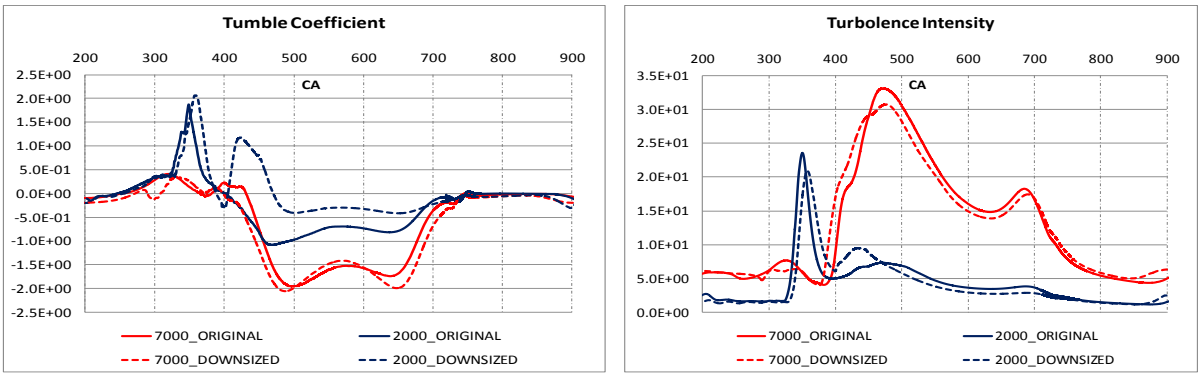

Fig. 12. Comparison between original and downsized engine in terms of Tumble coefficient and turbulence intensity at 2000 and 7000 rpm.

Despite the higher tumble and turbulent intensity fields, mixing seems to be less effective for the original geometry, since just before SA a wider charge stratification can be noticed, see Fig. 13. Given the more favorable mixture conditions at the beginning of combustion and the reduced flame path, the downsized engine allows more advanced SA values, which in turn result in a faster combustion and higher pressure levels within the combustion chamber for equal knock tendency, as visible in Fig. 14.

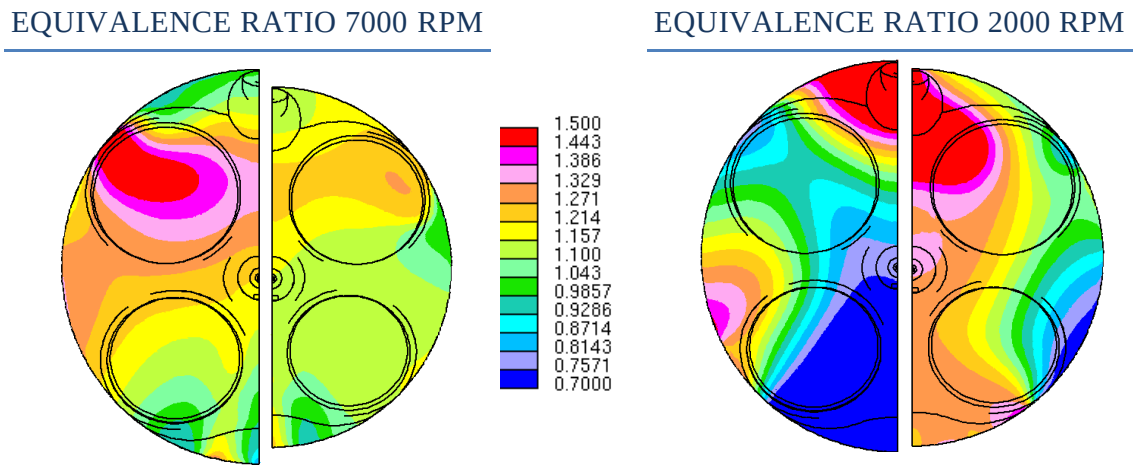

Fig. 13. Equivalence ratio @ 20°CA BFTDC, comparison between original and downsized engine at 7000 and 2000 rpm.

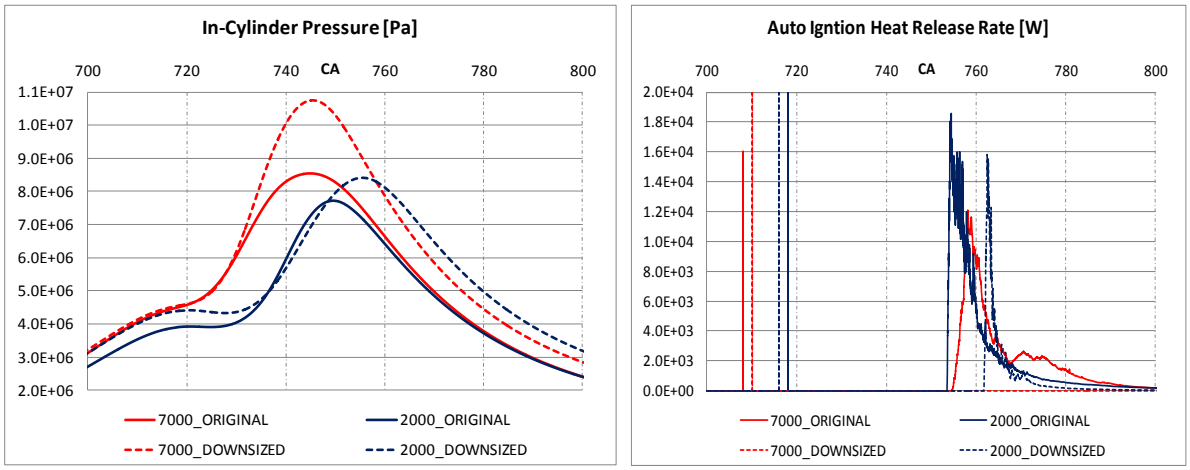

Fig. 14. In Cylinder pressure and heat release rate by knock at 7000 and 2000 rpm, downsized vs. original engine.

5. Conclusions

A numerical study is performed in order to assess the benefits of bore reduction for engine downsizing. In particular, three different sets of simulations are reported. First, a comparison is carried out on a simplified combustion chamber to analyze the effects of cylinder displacement reduction at constant B/S. Second, B/S is varied at constant cylinder displacement. In both analyses, the bore size reduction allows the engine to achieve higher specific performance for constant boundary conditions. Finally, a third set of simulations is performed, comparing a current V8 turbocharged GDI engine to a virtual engine, obtained by scaling the original bore by 11% and keeping stroke constant (-21% of displacement). The downsized engine has been numerically optimized in order to recover the displacement reduction. Thanks to the beneficial effects of bore reduction on flame propagation and knock inception, the virtual engine is able to achieve the same peak power and low-end torque of the original one. Table 2 summarizes the effects of the proposed downsizing on engine performance, at peak power and low end torque conditions.

Table2. Downsized vs. original engine.

	7000 rpm		
	ORIGINAL	**DOWNSIZED**	**DELTA%**
IMEP [bar]	20.07	27.64	38
POWER [kW]	55.6	60.6	9
SPECIFIC POWER [kW/l]	117.0	161.6	38
INJECTED FUEL [mg]	87.5	75.4	-14
ISFC [g/kWh]	329.9	261.3	-21
	2000 rpm		
IMEP [bar]	23.30	29.14	25
POWER [kW]	18.4	18.3	-1
SPECIFIC POWER [kW/l]	38.8	48.7	25
INJECTED FUEL [mg]	76.5	92.5	21
ISFC [g/kWh]	249.5	314.4	26

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