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Energy Procedia 126 (201709) 1003–1010

Energy Procedia

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72nd Conference of the Italian Thermal Machines Engineering Association, ATI2017, 6-8 September 2017, Lecce, Italy

Combustion System Development of an Opposed Piston 2-Stroke Diesel Engine

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Abstract

Today, the interest towards 2-stroke, opposed-piston compression-ignition engines is higher than ever, after the announcement of imminent production of a 2.7L 3-cylinder light truck engine by Achates Powers. In comparison to other 2-stroke designs, the advantages in terms of scavenge and thermal efficiency are indisputable: a perfect "uniflow" scavenge mode can be achieved with inexpensive and efficient piston controlled ports, while heat losses are strongly reduced by the relatively small transfer area. Unfortunately, the design of the combustion system is completely different from a 4-stroke DI Diesel engine, since the injectors must be installed on the cylinder liners: however, this challenge can be converted into a further opportunity to improve fuel efficiency, adopting advanced combustion concepts.

This paper is based on a previous study, where the main geometric parameters of an opposed piston engine rated at 270 kW (3200 rpm) were defined with the support of CFD 1D-3D simulations. The current work will focus on the influence of an innovative combustion system, developed by the authors by means of further CFD-3D analyses, holding constant the boundary conditions of the scavenging process.

The numerical study eventually demonstrates that an optimized 2-S OP Diesel engine can achieve a 10% improvement on brake efficiency at full load, in comparison to an equivalent conventional 4-stroke engine, while reducing in-cylinder peak pressures and turbine inlet temperatures.

Peer-review under responsibility of the scientific committee of the 72nd Conference of the Italian Thermal Machines Engineering Association

Keywords: Opposed Piston; 2-Stroke; Diesel engine; CFD; combustion

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Peer-review under responsibility of the scientific committee of the 72nd Conference of the Italian Thermal Machines Engineering Association
10.1016/j.egypro.2017.08.268

1. Introduction

2-Stroke Diesel engines are very promising in the automotive field for their intrinsic down-speeding and/or downsizing potential, due to high torque density, as well as for their enhanced fuel efficiency [1, 2].

The opposed-piston design (figure 1) is one of the most interesting propositions, since it combines the advantages of piston controlled ports (no poppet valves, no camshafts) to the efficiency of asymmetric uniflow scavenging [3-9].

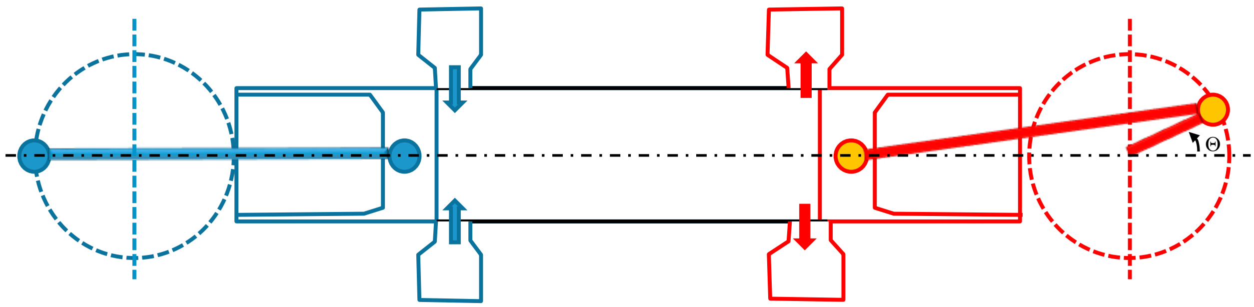

Fig. 1. Sketch of an Opposed Piston 2-stroke engine

With a proper offset of the exhaust crankshaft (depicted in red, in figure 1) from the intake crankshaft (depicted in blue), as well as with an optimized design of scavenge and exhaust ports, trapping efficiency can be dramatically enhanced, in comparison to other 2-Stroke designs. Another fundamental advantage of using opposed piston is the elimination of cylinder head, thus reducing the surface area of the combustion chamber, which in turn results in a reduction of engine heat rejection to the coolant. The after-treatment and charging devices can benefit from this additional heat available in the exhaust. Moreover, a strong turbulence can be generated within the cylinder, helping the combustion process.

In comparison to a 4-stroke engine, or to different types of uniflow scavenged two strokes [10-14], the opposed piston concept needs a second crankshaft, as well as a mechanical device to merge the power of the two axes. This transmission must be carefully designed, otherwise the intrinsic advantage of the elimination of the poppet valves could be completely lost.

The goal of this study is to assess the influence of an efficient combustion system, specifically developed for the opposed piston engine presented in a previous publication [15]. The main features of the analyzed engine are reviewed in table 1.

The compression ratio presented in table 1 is defined as the ratio of maximum to minimum cylinder volume, without any offset between the exhaust and the inlet crankshaft. When an offset is applied, as in this case, the cylinder time-volume law changes, and the compression ratio tends to decrease a little bit (from 18.0:1 to 17.3:1). However, if the compression ratio is calculated as the ratio of the cylinder volume at exhaust port closing to the minimum cylinder volume, this parameter is almost not affected by the crankshaft offset. Much more significant is the influence on ports timing: as the offset increases, the exhaust port opening advance from the inlet crankshaft bottom dead center (bdc) goes up, while the closing retard decreases of the same quantity, see figure 2. Ports timing controls the scavenging process, and it has a fundamental impact also on combustion. The CFD calculations reported in [15] show that a positive increment of the offset angle makes the trapped charge increase (lower cylinder pressure at scavenge ports opening, thus higher delivery ratio; early closure of the exhaust, thus better trapping efficiency). Due to the reduced retard from bdc of the exhaust port closure, as well as for the bigger trapped mass, the compression stroke yields higher pressure values around top dead center (tdc), with an ensuing reduction of the auto-ignition retard. On the other hand, the early opening of the exhaust ports has a very little influence on the completion of combustion, but it tends to reduce the expansion work, thus the cycle efficiency. The numerical results obtained in the previous project suggest to select an offset angle of 10°, as the best trade-off among the above mentioned conflicting requirements.

Table 1. Main features of analyzed opposed-piston Diesel engine

Parameters	Values
Bore x Stroke [mm]	84.3 x 109.5
Number of Cylinder/Pistons	3/6
Total Displacement [cc]	3665
Air Metering system	Turbocharger + supercharger + intercooler
Fuel Metering system	Common Rail, piezo-injectors (2000 bar)
Target Brake Power [kW] @3200rpm	270
Target Peak Torque [Nm] @1600rpm	930
Number/height of scavenge ports [#/mm]	12/12.5
Number/height of exhaust ports [#/mm]	8/24.2
Crankshafts offset [c.a. deg.]	10
Compression Ratio at offset=0°	18:1

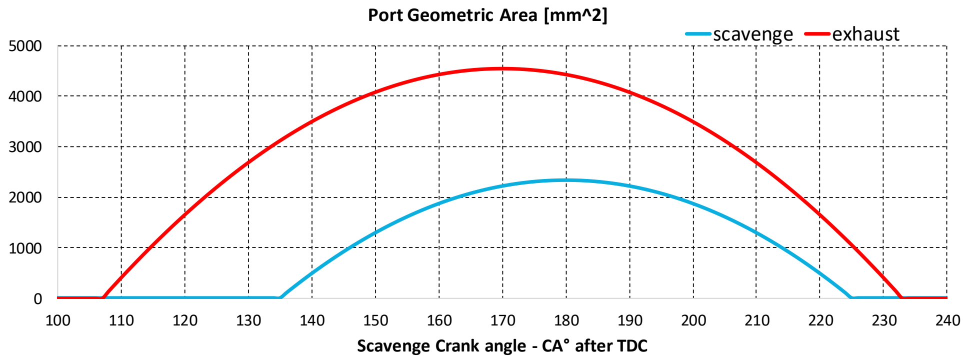

Fig. 2. Scavenge and exhaust ports area, plotted as a function of the inlet (or scavenge) crankshaft angle

The most important constraints considered in the design of the combustion system are: peak cylinder pressure: 160 bar; maximum injection pressure: 2000 bar; turbine inlet temperature upper limit: 800 °C; minimum trapped air-fuel ratio at full load: 18; back-pressure at the turbine outlet, at maximum speed: 1.9 bar.

In order to put the 2-S engine performance in context, a comparison is made with a state-of-the-art 4-Stroke automotive engine, whose features are reviewed in table 2. This engine is analyzed by using GT-Power, and the CFD-1D model is strictly derived from an experimentally calibrated model, presented in a previous project [16]. The reference 4-stroke model includes the experimental values of: friction mean effective pressures, intake/exhaust valves discharge coefficients, turbine/compressor parameters, burn rates and air-fuel ratio limits corresponding to a Euro VI compliant calibration.

2. Numerical optimization of the combustion system

The numerical study is divided into two phases: the first one focuses on the combustion process, considering only the cylinder, after ports closing (no manifolds, imposed initial flow field and charge composition); in the second step, a whole engine cycle is simulated several times, keeping the same boundary conditions and changing the initial mass, from a cycle to the following one, until reaching steady conditions. In both cases, the operating point under investigation is the most critical, i.e. the one at rated power (3200 rpm, relative air-fuel ratio: 1.2, about 20% of residuals in the trapped charge)

The tool adopted for CFD-3D simulation is a customized version of the KIVA-3V code, developed and calibrated in the last years through a number of similar research projects [14, 16]. Engine performance is predicted by means of a GT-Power model, incorporating all the detailed information provided by the parallel 3-D analyses. In particular, this information includes the scavenge patterns (i.e. the correlation between the amount of residuals trapped within the cylinder and the fraction of exhaust gas leaving the cylinder through the exhaust port), the effective time-area diagrams of ports (effective port area as a function of crank angle), and of course the burn rates (instantaneous fraction of burnt fuel, referred to the total injected fuel mass).

In the first phase, the influence of the number of injection points, the injector nozzle geometry and its orientation is investigated by KIVA, along with the injection strategy; initial conditions are provided by a GT-Power model.

In the second phase, the most promising design is analyzed, considering different injection timings. In this case, GT-Power provides the boundary conditions, while the scavenge quantities are calculated at each cycle.

The preliminary CFD-3D analyses enabled to find an optimum design for the combustion system, that cannot be disclosed in this paper.

The injection strategy is further optimized by means of the full cycle analyses (second phase). A view of the computational grid created for this purpose is presented in figure 3.

Table 2. Main features of the reference 4-S HSDI Diesel engine

Parameters	Values
Bore x Stroke [mm]	95 x 100
Compression Ratio	16:1
Layout	V6-60°
Total Displacement [cc]	4251
Air Metering system	1 VGT Turbocharger+ intercooler
Injection system	Common Rail, piezo-injectors (2000 bar)
Top Brake Power [kW] @3600rpm	285
Top Brake Torque [Nm] @1600rpm	930
EGR	High Pressure + Low Pressure

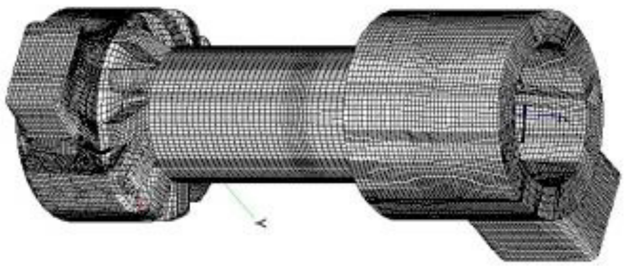

Fig. 3. Computational grid built for the full cycle analysis

The best result is found for a single-shot injection at 1300 bar, starting at 10 cad before TDC. For this setting, combustion is complete (efficiency 100%), despite the high amount of residuals (22%) and the relatively low air-fuel ratio (17). Figure 4 shows the calculated traces of cylinder pressure, temperature, rate of heat release and cumulative released heat.

In order to put the previous results in a more general context, a comparison is made among the non-dimensional combustion rates of different Diesel engine types at rated power. The selected references are the 4-stroke automotive turbocharged engine (see Table 2) and a 2-stroke loop scavenged automotive prototype [15]. In all the cases, fuel injection is made up of a single shot at high pressure. The curves are shifted in order to have 50% of fuel burned at the same crank angle (32° in the plot). It may be observed that combustion in the OP engine is remarkably similar to that found in the reference 4-stroke engine. Conversely, the loop scavenged engine reaches the maximum speed at the very beginning of the process, then the curve becomes similar to the others.

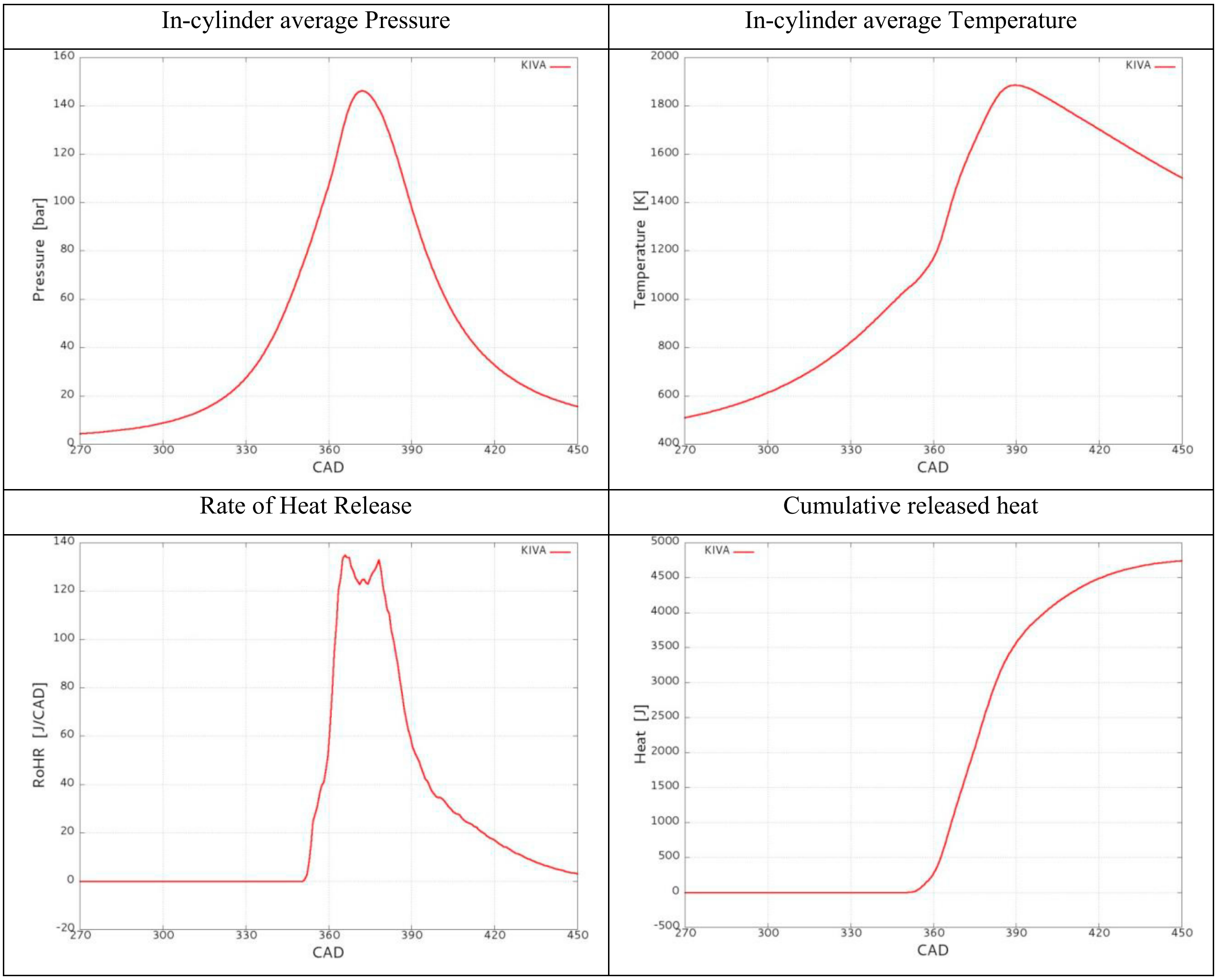

Fig. 4. CFD-3D cycle analysis results at 3200 rpm, full load (Opposed Piston engine)

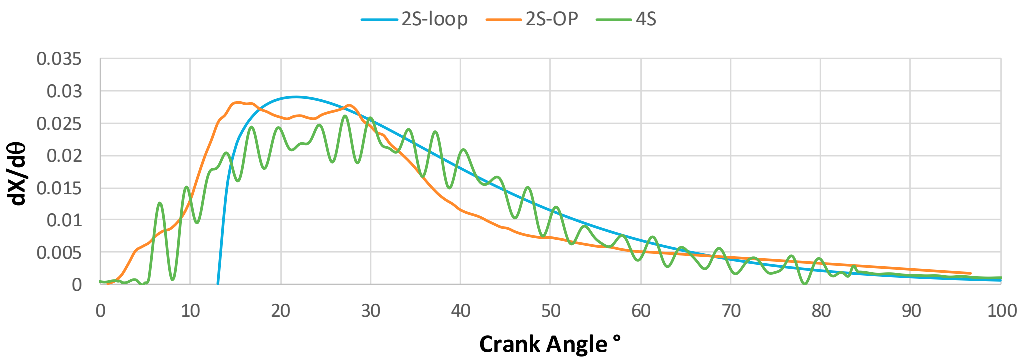

Fig. 5. Comparison of non-dimensional combustion rates for 2 and 4-stroke engines at rated power.

3. Comparison to the 4-S reference engine

The combustion rates calculated by KIVA-3V at some engine speeds, full load, are entered in the GT-Power engine model, in order to compare the Opposed Piston 2-S engine equipped with the new combustion system to the reference 4-Stroke engine. Analyzing figure 6, it may be observed that the two engines have identical brake performance (torque and power, fig. 6a). However, in the 2-stroke engine the ratio of trapped air to fuel is 20% higher, enabling a reduction of soot (fig. 6b). Also NOx emissions are expected to be lower, since the fresh charge is always diluted by burnt gas (15% on average, see fig. 6b). Obviously, in the OP engine the heat rejected through the cylinder walls is strongly reduced (-20%, on average, fig. 6c), but also the Turbine Inlet Temperature (TIT) is more than 100 degrees lower, an important advantage when adopting a variable geometry turbine.

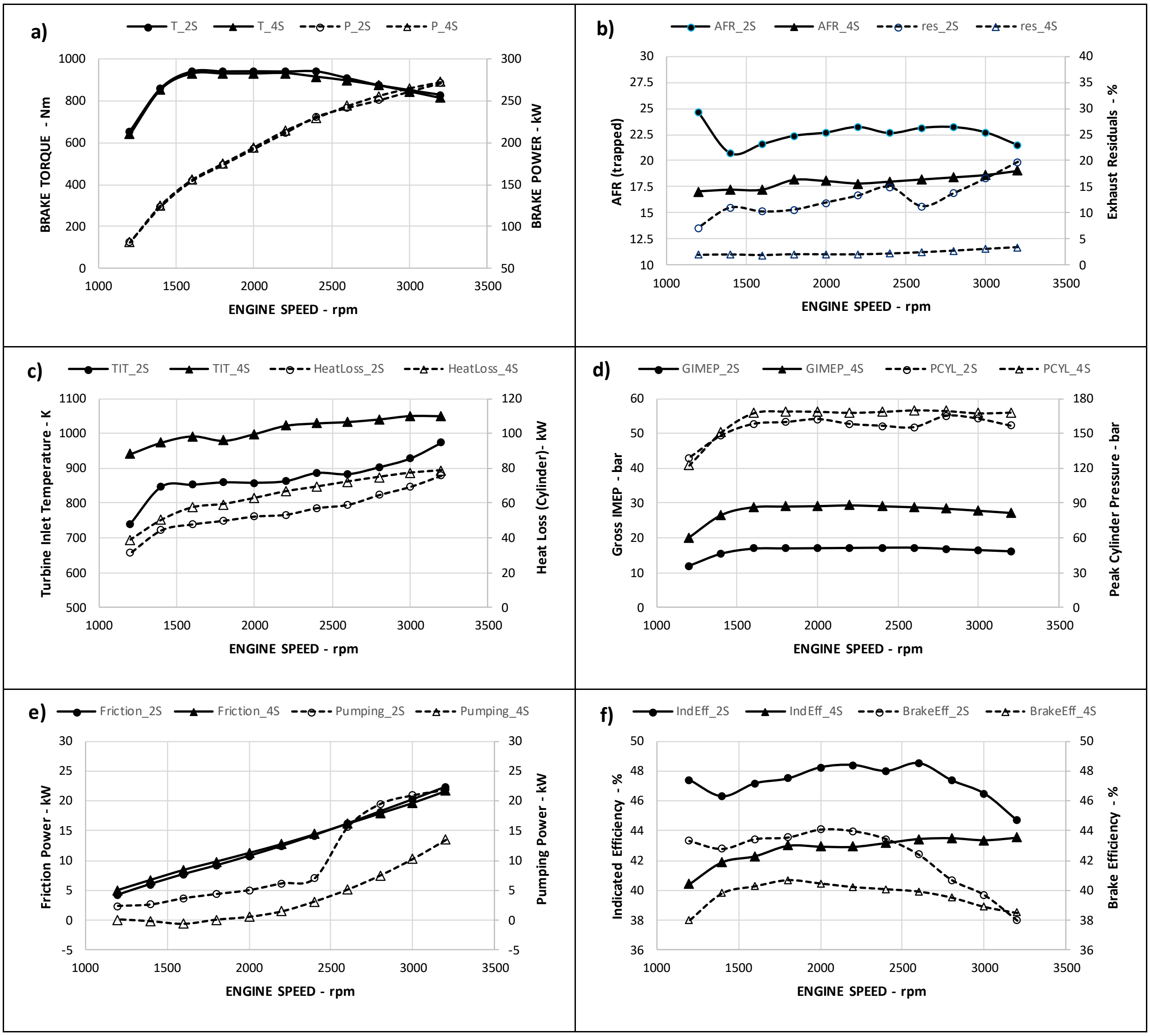

Fig. 6. Comparison between the 2S engine equipped by the new combustion system and the reference 4-S engine of Table 2. Full load operations analyzed by GT-Power.

The values of gross mean indicated pressure (the specific work transmitted from gas to piston during the compression-expansion strokes, fig. 6d) are almost double in the 4-S engine: as a result, more aggressive and efficient combustion strategies can be adopted in the 2-Stroke engine, while complying with the constraint of maximum cylinder pressure (which remains about 10 bar lower than in the reference 4-S engine). The smaller amount of energy rejected by the engine (in terms of heat losses and exhaust enthalpy), as well as the more efficient combustion strategies, yield a better gross indicated efficiency (+15%, on average, fig. 6f). This advantage is slightly reduced when considering the brake efficiency, figure 6f, since the 2-Stroke engine requires more energy for sweeping the cylinder: as visible in figure 6e, at 3200 rpm the pumping power is 22 kW against 14 kW (however, brake power is almost 20 times larger, so the influence of this loss is quite low). The difference in terms of friction power, figure 6e, is negligible: the intrinsic loss due to the double crankshaft and the higher mean piston speeds are compensated by the lack of a valve-train. The best brake efficiency of the 2-S engine is obtained at 2000 rpm (44.0%), while the best point for the 4-S power unit is at 1800 rpm (40.7%). On average, the advantage of the 2-Stroke engine is about 10%.

Figure 7 compares the two engines in terms of brake specific fuel consumption and specific NOx. The last parameter is calculated by GT-Power using a model experimentally calibrated on the 4-Stroke engine, thus the results are only qualitative. The graph clearly shows the enhancement of fuel efficiency provided by the optimized 2-S engine, while, in terms of NOx emissions, an improvement can be observed only at medium-high speeds (>2000 rpm), thanks to the internal EGR. A further reduction of NOx is probably possible adopting less "aggressive" injection strategies.

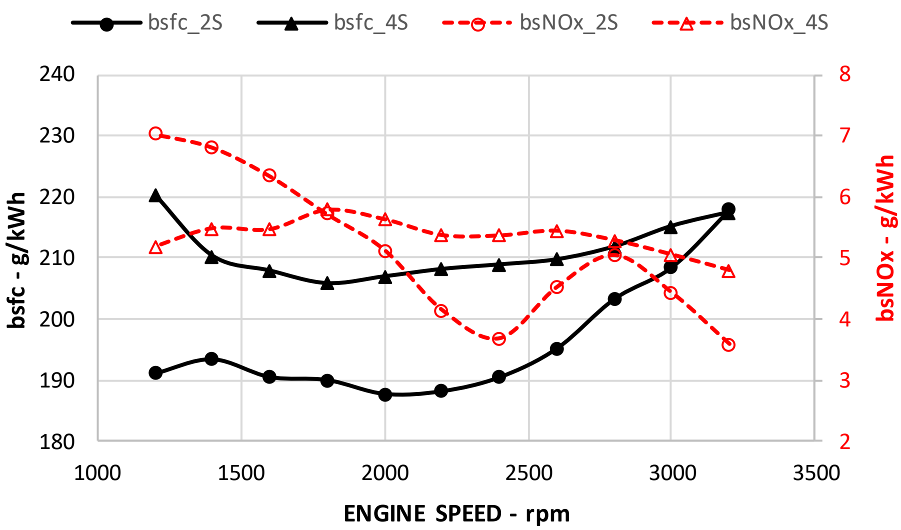

Fig. 7. Comparison between the 2S engine equipped by the new combustion system and the reference 4-S engine of Table 2: brake specific fuel consumption and NOx emissions are plotted at full load

Conclusions

The paper analyzes the influence of an optimized combustion system on the performance of a new 3-cylinder, opposed-piston two stroke Diesel engine, total displacement 3665 cc, rated at 270 kW (3200 rpm). This engine is compared to a conventional 4-stroke, V6, 4225 cc turbocharged Diesel engine, Euro VI compliant. The two engines have the same brake performances.

The new combustion system enables a strong improvement of indicated efficiency at full load (+15%), thanks to a complete and fast combustion (similar to a 4-stroke engine), coupled to a strong reduction of heat losses through the walls, due to a reduction of heat transfer areas (the opposed piston engine has no cylinder heads). Exhaust gas temperatures are reduced too, because of the higher air-fuel ratios and the higher amount of exhaust residuals.

The advantage in terms of indicated efficiency is slightly reduced when considering brake efficiency, because of the higher pumping losses (the 2-S engine needs an additional supercharger). However, the 2-S engine enables a 10% reduction of fuel consumption, on average.

Soot emissions at full load are expected to decrease in the 2-S engine, because of the higher air-fuel ratios (+20%). A reduction of specific NOx emissions is predicted at medium-high speeds, thanks to the internal EGR. At low speed, NOx are higher, because of the fast combustion, but they could be reduced adopting a different injection strategy.

The values of gross indicated mean effective pressure are almost one half of the equivalent 4-stroke engine: this is a fundamental advantage for injection calibration, since it enables the implementation of innovative combustion concepts, requiring more freedom from cylinder pressure constraints.

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