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Fine-tuning of a two stoke engine in full power configuration provided with a Low Pressure Direct Injection system

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Abstract

The main drawbacks of two stroke (2S) engines consist in poor engine efficiency and high level of pollutant emissions. The contemporary opening of transfer and exhaust ports during the scavenging process causes the short circuit of fresh air-fuel mixture in case of indirect injection or carbureted engines. Despite the intrinsic strengths such as high power density, simplicity, compactness, lightweight and low production costs, 2S engines have been substituted by four stroke (4S) engines in many applications. Direct injection represents an effective solution to reduce the short circuit of fuel in 2S engines. Usually it is carried out by adopting high-pressure systems but the related increase of complexity and costs is inevitable. In order to maintain the intrinsic simplicity of a 2S engine, the most suitable solution is represented by a Low Pressure Direct Injection (LPDI) system. 2S LPDI engines are characterized by the presence of one or two injectors, working at 5 bar, installed on the cylinder wall. Previous works of the authors have shown the effectiveness of an LPDI system applied to a 300cc single cylinder engine in underpowered version with different ports timing and exhaust system with respect to the full power configuration. In the present paper, the authors show the fine-tuning of a 2S engine in full power configuration provided with two injectors installed on the cylinder and directed towards the exhaust port; the injector nozzles were located above the scavenge ports in order to guarantee the maximum interaction between injected fuel and inlet air flow. The engine has been deeply tested and analyzed at the test bench. Particular attention was paid to definition of the optimal injection timing in order to guarantee the best compromise between performance, efficiency and emissions. The experimental setup and the calibration methodology are discussed in detail. The results show the advantages of the LPDI system in terms of increased engine efficiency and emissions reduction with respect to the original carbureted engine maintaining the same level of performance.

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Keywords: 2Stroke Engine; LPDI; Low Emissions; High Efficiency.

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1. Introduction

The conventional small crankcase-scavenged two stoke engine is cheap, reliable and provides a high power-to-weight ratio. However, it is affected by high specific fuel consumption and pollutant emissions. These issues are mainly related to the symmetrical port timing combined with the carburetor or PFI (Port Fuel Injection) system. A significant amount of the fresh fuel-air mixture tends to short-circuit across the cylinder during the scavenging process, thus flowing directly through the exhaust port before its closing. A direct injection system allows the introduction of the fuel independently from the scavenging air, minimizing the fuel short-circuit and substantially improving the potential of two stroke engines. With an appropriate choice of both the positioning of the injectors' nozzles and the injection timing, only air can be lost through the exhaust port during the scavenging process. The whole amount of injected fuel is trapped in the cylinder and takes part in the combustion process, thus improving the engine efficiency and reducing the pollutant emissions.

The present study focuses on the low pressure direct injection system that, conversely to the high pressure system, preserves the simplicity of the design, the low weight and costs that are typical characteristics required by small 2S engines [1, 2, 3, 4, 5, 6, 7].

Previous works of the authors have shown the advantages in terms of efficiency increase and emission reduction of a first prototype equipped with a LPDI system applied to an underpowered version of the Betamotor 300cc naturally aspirated two-stroke cross-scavenged engine. In more detail, the first engine prototype has been designed with port timing and exhaust system different with respect to the full power configuration in order to limit the maximum torque and power. In the present paper, the authors show the fine-tuning of the same engine in full power configuration provided with the LPDI system. Moreover, the full power engine configuration is provided with an exhaust port valve, which raises and lowers the top of the exhaust port.

Two injectors working at 5bar of injection pressure ensure the requested fuel flow rate at both low and high loads. The best injectors' positioning inside the cylinder was configured using computational fluid dynamics (CFD) [8]. The injection nozzles are located in the liner wall at the opposite side of the exhaust port. The injectors are positioned above all of the cylinder ports and, consequently, the injection timing is only limited to the piston motion. The injectors' axes are oriented towards the piston at the BDC in order to enhance the atomization of the liquid droplets by means of the interaction with the incoming fresh air. Standard low cost components from the automotive market were used, without a significant increase of weight.

In the present paper, the results of the experimental activity are reported. A direct comparison in terms of maximum performance between the engine equipped with the LPDI system and the original carbureted engine is carried out. The experimental setup and the fine-tuning methodology are discussed in detail. The LPDI system was tuned in order to reach the lowest fuel consumption and pollutant emissions and, at the same time, to guarantee analogous torque and power outputs with respect to the conventional configuration.

Nomenclature

ATDC	After Top Dead Centre
BDC	Bottom Dead Centre
BMEP	Brake Mean Effective Pressure
BSFC	Brake Specific Fuel Consumption
CA	Crank Angle
CFD	Computational Fluid Dynamic
ECU	Electronic Control Unit
IA	Ignition Advance
LPDI	Low Pressure Direct injection
PFI	Port Fuel Injection
SOI	Start of Injection
TDC	Top Dead Centre
WOT	Whole Opening Throttle

2. Engine Design

In the present paper, the authors have analyzed and tested a 2S LPDI engine in full power configuration. With respect to the underpowered engine presented in previous works [8, 9, 10], in case of full power version two critical aspects should be taken into account: at wide-open-throttle (WOT) the engine traps a higher quantity of air and reaches an higher maximum engine speed; as a direct consequence, the required fuel mass flow at full load increases. At the same time, due to the higher position of the cylinder ports with respect to the BDC, the time interval in which all the ports are closed decreases with an increased risk of short-circuit. The engine development can be divided into three main phases:

- CFD optimization studies for the analysis of the interaction between the injected fuel and the fresh scavenging air. During this phase, the injectors' position was defined in order to find the best compromise between short-circuit reduction and fuel evaporation and homogenization.
- CAD design and manufacturing of the new cylinder with two injectors installed on the cylinder wall over the scavenge ports and directed towards the exhaust port.
- Experimental analysis of the engine at the test bench, in order to fine-tune the engine mapping taking into account performance, emissions and efficiency.

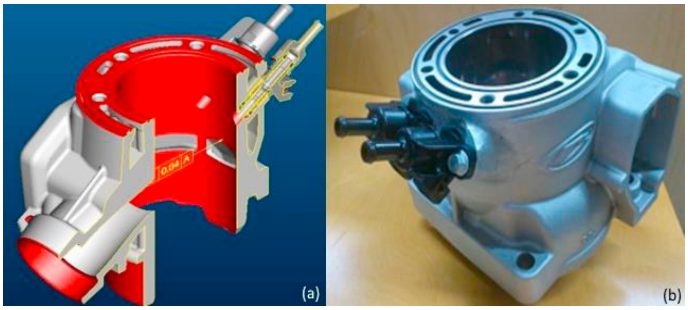

Fig. 1. New LPDI cylinder, CAD model (a) and first prototype (b).

The LPDI engine is developed on the base of 2S 300cc single cylinder carbureted engine produced by Betamotor, in Table1 are summarized the main features.

Table 1. 2S LPDI engine main features.

Type	Single Cylinder, 2-Stroke, Spark-Ignition, Liquid Cooled
Displaced volume	293.1 cc
Stroke	72 mm
Bore	72 mm
Connecting Rod	125 mm
Compression Ratio	12:1
Exhaust Valve	No
Engine Oil	SAE 10W/40
Starter	Electric Starter
Spark Plug	NGK BR7ES
Induction System	Induction Reed
Fuel System (Original)	Carburetor (Keihin PXK 36)
Fuel System (LPDI)	Synerject Continental PFI injectors
Oil injection (LPDI)	DellOrto reciprocating pump

Fig. 1 shows a section view of the CAD model of the 300cc cylinder in the LPDI configuration, in which the injectors' installation is highlighted. The injectors are model Deka VII, supplied by Synerject working at 5 bar. Their positioning was defined thanks to a dedicated CFD analysis [8, 9]: the injectors distance from the cylinder top and their inclination are fixed taking into account both fluid-dynamic (reduction of the short-circuit and air-fuel homogenization) and mechanical-design aspects. The nozzle axes are parallel and directed towards the piston crown at the BDC; the only constraint on the injection timing is related to the piston motion that covers the injectors at 282° CA (Crank Angle degree) (TDC equal to 0° CA). Considering their positioning, during the scavenge process, the injected fuel collides with the fresh air upcoming from the transfer ports and goes under spark plug.

The oil for the lubrication, as for PFI engine, is injected through a nozzle located on the intake manifold downstream the throttle body by a very small reciprocating oil pump supplied by DellOrto. The oil mixes with air in the intake manifold, and flows into the cylinder through the crankcase. The pump is controlled by the ECU on the base of a dedicated map in order to manage the percentage of injected oil. The amount of injected oil was optimized for each engine operating point in order to minimize the oil consumption with strong benefits in terms of pollutant emissions, Fig. 2 (a), Fig. 2 (b).

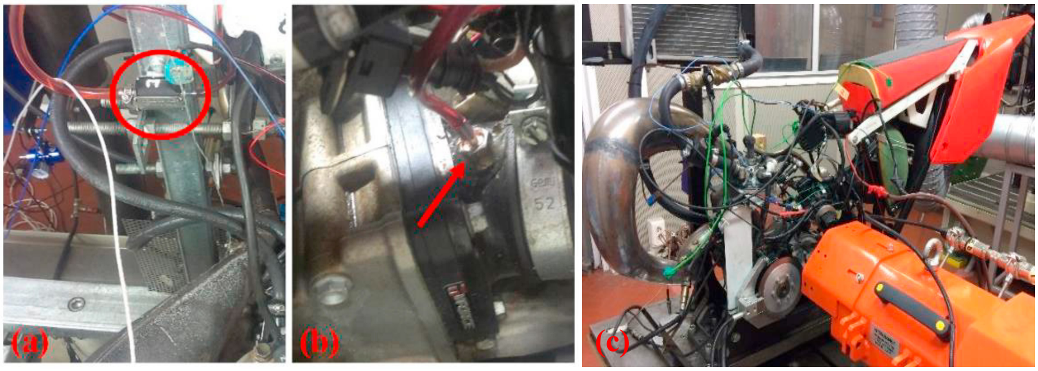

Fig. 2. Engine lubricating system, (a) oil pump, (b) oil injection nozzle. (c) Engine at the test bench.

3. Experimental setup and mapping process

The engine calibration was done at the test bench of the Department of Industrial Engineering of the University of Florence. The cell is provided whit a Dynamic test bench, APICOM mod. 12361-13- EDF (Power 80 kW, Nominal Max Torque 200 Nm, Max Speed 4000 rpm). The mapping process of the LPDI engine aims to obtain the same maximum performance (torque and power) of the carbureted engine with a strong improvement of efficiency and raw emissions reduction in the whole operating range, taking advantage of the fine-tuning of the direct injection system.

The engine installed on the test bench is depicted in Fig. 2 (c). In order to investigate in deep the engine behavior several operating parameters have been acquired during the test:

- Torque and Power of the engine
- Fuel consumption: an AVL SORE PLU 110 fuel mass flow is placed along the fuel line to measure fuel flow and brake specific fuel consumption (BSFC); a static pressure sensor is adopted to monitor the correct functioning of the fuel pump.
- Exhaust raw emission: CO, CO2, NOX, HC raw emissions were monitored using AVL gas analyzer mod. DIGAS 4000 NOx.
- Air to fuel ratio: an O2 sensor is installed on the exhaust duct in order to measure the A/R during the combustion process.

- Dynamic pressures: high frequency piezo-resistive and piezo-electric sensors are used to measure the pressure inside the intake and exhaust lines and into the cylinder.
- Crankshaft angular position: an optical encoder, AVL 365X, is installed keyed on the crankshaft.
- Temperature: thermocouples K and T typology were used to measure the temperature of the crack-case, exhaust manifold and liquid.
- Cell temperature and pressure.

The main engine parameters on which the engine calibration has been focused are: injection quantity, injection timing and ignition advance (IA). The ECU M3C, made by Synerject, ensures the complete control of all engine mapping parameters. The ECU allows managing one or more injectors independently and the oil pump for the separate lubrication. The engine rotating speed and the crank angle position are acquired by the ECU using the phonic well and the pick-up sensor. The pick-up is a hall-effect sensor and the specific flywheel has 24–2 pattern.

The mapping procedure for the definition of the injection quantity (MFF map) follows the logic flow diagram described in Fig. 3(a). It consist in an iterative loop in which the fuel quantity per stroke is defined in order to obtain the desired air-fuel (A/F) ratio at the exhaust. Even though for a two-stroke engine, the measurement of A/F ratio by using an O_2 sensor is not correct, because of the presence of air short circuit at the exhaust port [11, 12, 13], it represents a comparative term that is indicative of the combustion mixture. Both injectors are mapped with the same MFF and SOI maps. Once defined the total amount of fuel per stroke, strong attention was paid to the tuning of the start of injection (SOI). Fuel can be injected in the injectors uncovering CA range (between 80CA and 280CA) and only after the in-cylinder pressure is decreased under the fuel pressure delivery (5 bar), Fig. 3(b). The SOI map has a fundamental role in the direct injection engine and it was therefore fine-tuned in order to match the best compromise between BSFC and HC emission.

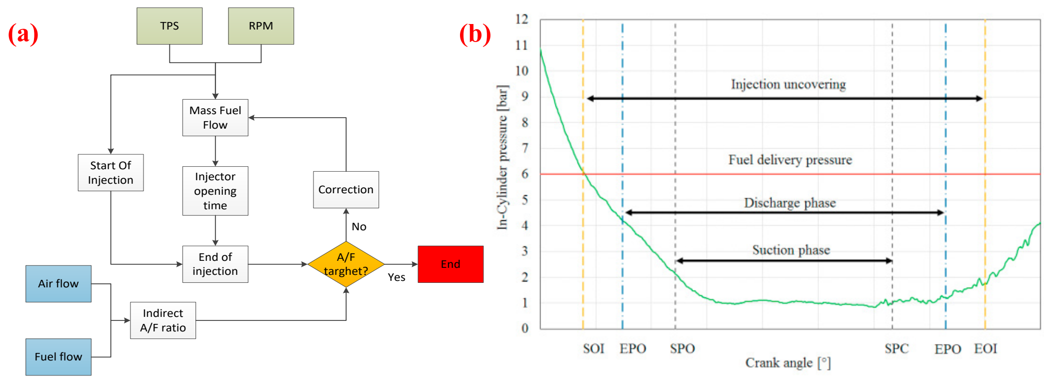

Fig. 3. (a) Mapping procedure. (b) Port timing, in-cylinder pressure and fuel injection pressure.

The trend of HC emission as a function of SOI at 3500 rpm and 40% of load is depicted in Fig. 4(a); it is possible to notice that an excessive advance of SOI causes a rise of HC emission because the injected fuel is not involved in the scavenge process and goes directly out through exhaust port. The trend of IMEP as a function of SOI at the same engine operating condition is reported in Fig. 4(b). When the SOI timing is too retarded, a reduction of IMEP occurs due to the poor air fuel mixing in the combustion chamber and low level of fuel vaporization, while a too advanced SOI causes an IMEP reduction due to fuel short circuit. The best compromise between the two curves is around 160° ATDC that ensures a strong reduction of HC emission without compromising the IMEP and consequently the BSFC. In other words, the injection timing has to be set in order to find the best compromise between the short-circuit reduction (injection close to the TDC) and the complete fuel evaporation (injection far away from the IA). After the definition of the mixture strength and the injection timing, the IA map was adjusted in order to maximize the area of the thermodynamic cycle (IMEP) and then to minimize the BSFC in the whole engine operating range.

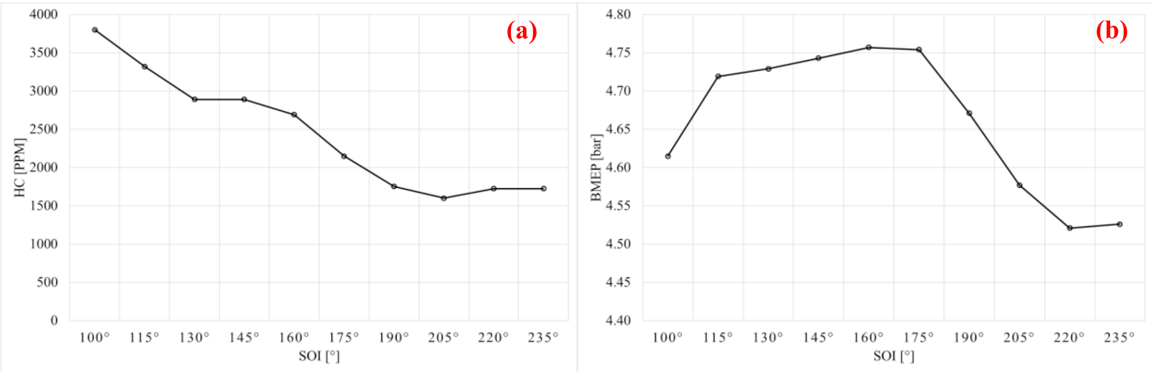

Fig. 4. (a) HC emission as a function of SOI at 3500 rpm and 40% of load. (b) IMEP as a function of SOI at 3500 rpm and 40% of load.

4. Results

The potential of the proposed solution was assessed by comparing the performance of the original carbureted engine with that of the new engine equipped with LPDI system. The challenge is to guarantee both almost the same maximum torque and power of the carbureted engine with a higher efficiency and lower pollutant emission in the whole operating range. As highlighted in the previous paragraph, the mapping process consists first in the definition of the injected fuel mass in order to obtain the desired air-fuel (A/F) ratio maximizing the brake mean effective pressure (BMEP), then the SOI is advanced as much as possible in order to maximize the fuel vaporization taking into account the fuel short-circuit reduction. The correct choice of the SOI is more critical for WOT condition and in particular for high engine speed, in which the time per stroke available for the injection is limited and the amount of injected fuel per stroke is very high. The direct comparison of torque and power curves of carbureted engine and LPDI engine at WOT is presented in Fig. 5(a).

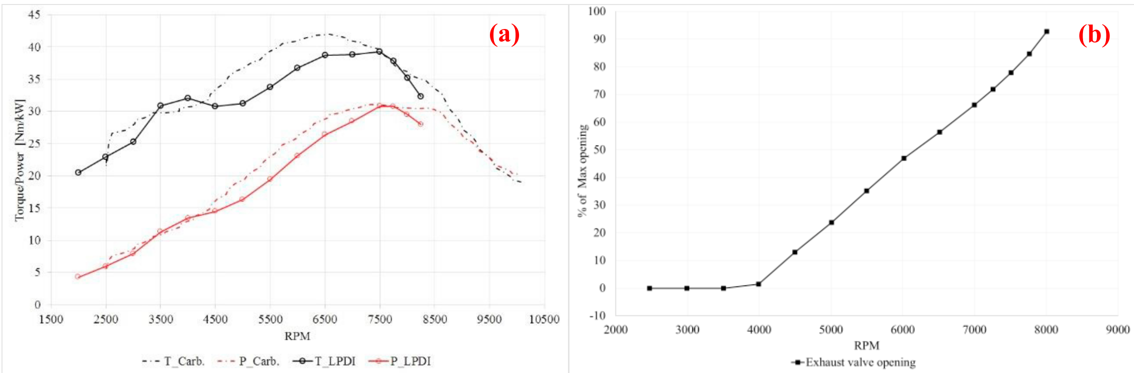

Fig. 5. (a) LPDI engine VS Carbureted engine, torque and power curves. (b) Exhaust-valve opening curve.

In both cases, the maximum power is about 31kW, reached at 7500rpm, but two critical aspects arise from this comparison:

- Notwithstanding a comparable power output up to 4500rpm, the range from 5000 to 7000 rpm is characterized by a loss of torque for LPDI engine.
- The LPDI engine does not reach the same maximum speed of the carbureted engine.

From the analysis of the graphs, it can be deduced that at low engine speed the LPDI system guarantees a suitable vaporization and air-fuel mixing, since the same performance of the carbureted engine were obtained. For medium engine speed, being the area of maximum torque and maximum trapping efficiency and consequently maximum request of injected fuel, an incomplete vaporization of the fuel and a poor air-fuel mixing is likely to occur, thus affecting the combustion efficiency and the power output. Conversely, at 7500-7750 rpm, the increase in speed and therefore in turbulence promotes the correct mixing of air and fuel despite the decrease in available time. Beyond 7750 rpm, air and fuel do not mix properly again because of the short time available for the thermodynamic cycle, consequently power and torque decrease with respect to the original engine. Starting from 8250 rpm, the arise of high level of misfires causes a sudden drop of performance. It is important to point out that the torque trend between 4500 and 7000rpm of the considered 2S engine is strongly influenced by the behavior of the mechanical exhaust port valve, which raises and lowers the top edge of the exhaust port as a function of engine speed taking advantage of a mechanical actuation system. The operation of the valve, which mainly acts between 4500 and 7000rpm (Fig. 5(b)), was appropriately calibrated for the carbureted engine and it was not modified for the LPDI configuration. An increase of the LPDI engine performance is expected by means of a proper calibration of the valve also in the LPDI configuration.

Notwithstanding the slight reduction of performance at full load, the most relevant advantage is represented by an average increase of 35% in terms of efficiency (Fig. 6(a)). At WOT there are consistent benefits at low engine speed where, for the right air-fuel mixing, not too advanced SOI can be adopted with a strong reduction of short-circuit. At high engine speed the unavoidable small amount of fuel short-circuit combined whit a lower combustion efficiency causes a reduction of the benefits of the LPDI solution. Nevertheless, the BSFC reduction is about 25%.

It can be concluded that the LPDI system represents an effective solution for the reduction of fuel consumption; the maximum benefits are achieved at low and medium load for which the BSFC reduction is about 50% with respect to the conventional engine, (Fig. 6(b)). At low and medium loads, the fuel can be injected whit conservative SOI values in order to minimize the fuel short circuit.

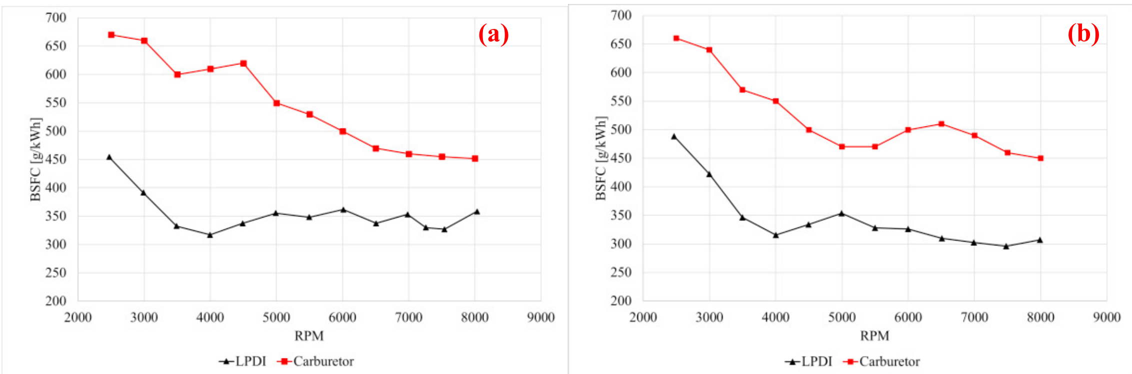

Fig. 6. Comparison of BSFC trend between LPDI engine and carbureted engine: (a) Full load, about 35% of BSFC reduction for LPDI engine. (b) Half load, about 50% of BSFC reduction for LPDI engine.

In addition, from the analysis of the efficiency map of the LPDI engine in Fig. 7(a), it can be readily noticed that also at low BMEP and low engine speed the engine is characterized by good efficiency (about 21-23%). In particular, for the operating points at medium engine speed and medium BMEP, typical area of amateur use, the engine efficiency is higher than 26% for a significant region. In conclusion, the great advantage of the 2S LPDI engine is that the area characterized by high global efficiency extends over a wide operating range.

The direct consequence of the fuel short-circuit reduction is represented by a low level of HC raw emission. Fig. 7(b) shows the trends of HC raw emission of the LPDI engine as a function of the rotating speed for different engine loads. Common values of HC emissions for a 2S engine equipped with carburetor or PFI system are about 8500-10000 ppm [14]. Thanks to the LPDI fuel supply system, the HC emissions are strongly reduced, about four times with respect to the conventional 2S engine, achieving a level comparable to that of a 4S engine of the same category.

For very low load and low speed, HC emissions slightly increase because of the arise of misfires, while at low load and high engine speed the relative increase of HCs is due to the necessary advanced values of SOI.

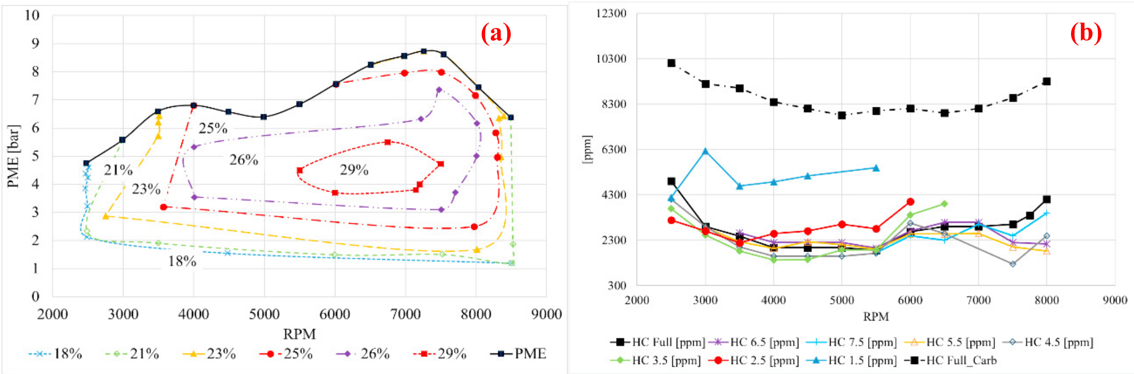

Fig. 7. (a) LPDI engine efficiency map; (b) LPDI engine HC emissions and HC emission of carbureted engine at full load.

5. Conclusions

In order to overcome the typical issues of two stroke engines, i.e. low engine efficiency and high pollutant emissions, the authors have developed a LPDI system for the injection of fuel. The proposed solution prevents the fuel short circuit during the scavenging process because fuel is injected directly into the cylinder through two injectors. In the present paper, the authors have fine-tuned a LPDI system on a 2S single cylinder engine in full power configuration, starting from previous works carried out on the same engine in underpowered configuration. The activity aims to investigate the potentiality of LPDI system also in case of high quantity of fuel injected per stroke and high engine speeds. The results show a direct comparison of the same engine equipped with both a carburetor and LPDI system. The proposed solution seems to be promising, in fact the LPDI engine reach almost the same maximum performance of the original one with a significant increase in engine efficiency and reduction in HC emission.

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