Effect of prechamber on engine fuelled with gas/liquid fuel for emission reduction

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Abstract

The aim of the study was the optimization of the gasoline combustion process by means of a passive prechamber. The improvement of the engine efficiency in lean-burn operation condition is an opportunity to give further use of Spark Ignition (SI) engine. A commercial small SI engine was modified with a proper designed passive prechamber. Engine performance in terms of indicated Mean effective pressure, heat release rate and fuel Consumption were evaluated as well as gaseous emissions. Particulate Mass, Number and Size Distributions were measured. Several engine operative conditions were investigated at full load varying the engine speeds for stoichiometric and lean conditions. The results were compared with those obtained with the engine equipped with the standard spark plug. The results indicated that both performance and emissions were strongly influenced by the prechamber.

Introduction

The engine efficiency and the simultaneous pollutant emissions reduction is mandatory for next years to the aim of improvement of quality of life of us and of future generations. Engines with gasoline direct injection (GDI) systems allow to achieve improved efficiency and higher level of power. The injection of the fuel directly into the cylinder increases the knock resistance and the volume efficiency because of the improved air cooling due to fuel evaporation. Moreover, the fuel amount can be more precisely controlled with a better operation in part load and during transient operation mode. The main drawback of the GDI technology is the increase of unburned hydrocarbons (UHC), CO and particle emissions due to the short time for fuel evaporation and mixture formation as well as the fuel impingement on piston head and cylinder wall . The application of lean stratified combustion can reduce UHC during the cold start emissions and the CO_2 emissions [1-4], but the engine stability is worsened. The Cycle to Cycle variations (CCV) create engine vibrations and noise that reduce the power output meanwhile the reduction of the CCV results in the increase in power output for the same fuel consumption. The sources of the CCV in a spark ignition engine are due to the turbulence intensity of the flow field in the cylinder, the variations in the fuel–air ratio, the amount of residual or recirculated exhaust gases in the cylinder, the spatial inhomogeneity of the mixture composition especially close to the spark plug, the spark discharge characteristics and the flame kernel development play an important role on the CCV worsening. The increase of the burning speed results in a reduction of the indicated mean effective pressure coefficient of variation, IMEP CoV [5]. Several solutions can be applied to improve the flame front propagation. The improvement of engine efficiency can be obtained with an optimized prechamber that permits to increase the flame speed and the turbulence improving the flame front propagation and then

increasing the combustion stability especially in lean condition. Moreover, the prechamber permits to obtain also some advantages for the GDI engines without its drawbacks because it can be improved both the stratified charge combustion and the very lean mixture combustion, saving the fuel economy as for the DI engine. The application of prechamber leads to a fuel stratification considering the entire engine because the fuel injected into the prechamber stays in this volume during the compression stroke and only a small part of the overall mixture has to be enriched to help ignition, in combination with the benefit of using conventional injection and ignition systems. The faster combustion permits the increase of the compression ratio and consequently the engine efficiency without any effect on the PM emissions.

The aim of this paper is the study of the effect of prechamber ignition on the engine stability and efficiency in stoichiometric and lean-burn operation conditions. The prechamber was optimized to increase the burning speed. Its design and the preliminary analysis of the combustion using the prechamber was performed by means the optical diagnostic performed into the combustion chamber [6].

The experimental activity reported in this study was carried out on a small displacement single cylinder Spark Ignition (SI) engine representative of the most used gasoline engine for automotive application. The tests were performed at 2000, 3000, 4000 and 5000 rpm WOT (Wide Open Throttle) both in stoichiometric and in lean conditions. These engine speeds are representative of some typical conditions considered in the New European Driving Cycle (NEDC). The engine was operated in lean condition in order to increase the engine efficiency. The analysis of the combustion process was performed by indicated data, moreover gaseous regulated emissions and particle size distribution were measured too.

Experimental apparatus

Prechamber

The prechamber was properly designed to fit with the single cylinder SI engine head used for the experimental activities. In order to not modify the engine head the prechamber was designed to be housed inside the spark plug hole. The main challenge was to adapt the new component to an existing very small geometry. The prechamber is composed of two parts so that it can be made only with mechanical machining: the spark plug / injector housing (green part in Figure 1) and the combustion prechamber (grey part in Figure 1). In Figure 1 is shown a detail of the prechamber tip with the holes for the injection of plasma into the combustion chamber. Different number of holes and diameter was designed and tested, in this paper are reported only the result obtained with 4 holes characterized by a diameter of 1mm. The original head spark plug housing has a maximum diameter of 24 mm. Due to the limited available space, the spark plug and the injector could not be fixed parallel to each other, therefore, the injector housing has an axis parallel to that of the original spark plug and instead the spark plug housing axis has an angle of 14 ° with respect to the injector axis. The volume of the prechamber, obviously, affects the compression ratio, and if this volume is very large (approximately >2.5% of the dead volume) the benefits of combustion do not exceed the disadvantages of the compression ratio reduction.

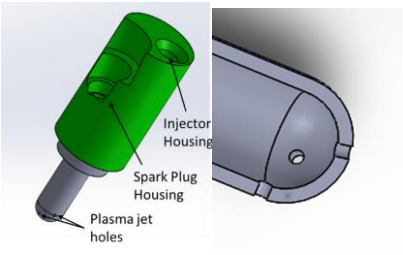

Figure 1. a) Prechamber image, spark plug / injector housing and combustion pre-chamber (grey), b) detail of the pre-chamber tip.

In this paper, in order to show the effect of the prechamber on the combustion evolution, it was used a prechamber, not optimized for the performance, with a volume of 2.2 cm^3, 9,4 % of the dead volume, with a reduction of the compression ratio from 11.5:1 to 10.5:1. In order to overcome the compression ratio decrease, for each engine speed the prechamber WOT condition was compared with a partialized standard ignition condition. In order to compare similar test engine condition, the throttle was set in order to have at each engine speed the same motored peak pressure. In Figure 2 are shown a section of the engine head equipped with the prechamber and the image of the prechamber mounted in the spark plug location of the engine head.

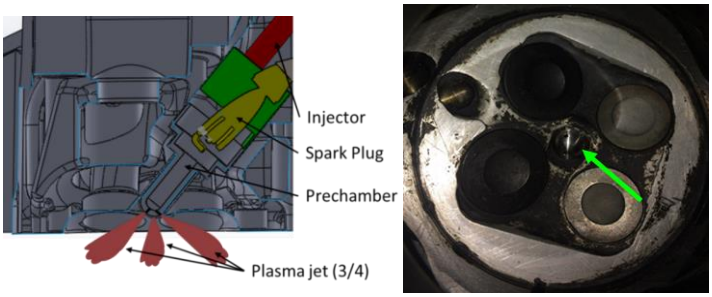

Figure 2. a) Section of the engine head equipped with prechamber, b) image of the prechamber tip in the central location of the head.

The effect of the prechamber on the combustion ignition was previously analyzed in an optical engine with the same characteristics of the real engine used in this paper. In Figure 3 are reported two sequence of images, acquired in the optical engine, that represent the evolutions of the combustion process from the start of spark until the flame front reaches the optical limit/cylinder wall, for both ignition configurations at 2000 rpm in stoichiometric mixture condition. For standard ignition the flame kernel moved from the spark plug with a radial like behaviour reaching the cylinder wall in more than 30 crank angle degrees (CAD ASOS), instead using the prechamber four flames spread from its with a jet behaviour. These four flames, much brighter than the standard flame front, spread until the cylinder wall in 5 CAD after the first evidence of flame in the combustion chamber.

Engine

The investigation was carried out on a 4-stroke single cylinder SI engine equipped with the cylinder head of a naturally aspirated PFI engine. All specifications are reported in [6].

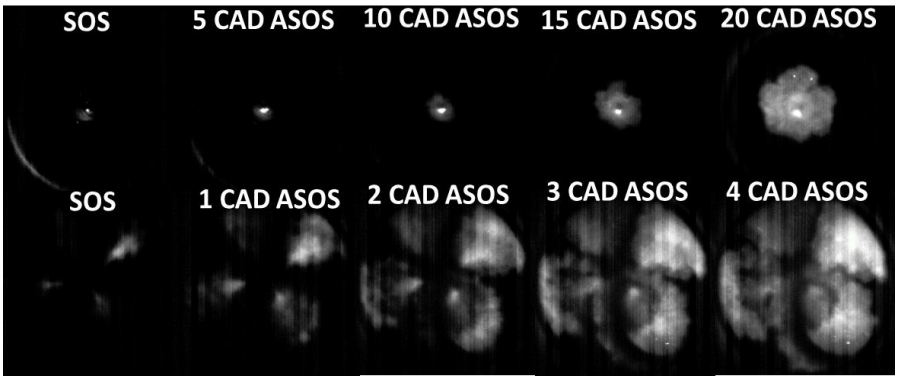

Figure 3. 2D chemiluminescence measurements detected at typical crank angles of the combustion process for standard (up) and active prechamber (down) ignition at 2000 rpm in stoichiometric condition [6].

The in-cylinder pressure was measured by means of a quartz pressure transducer flush-mounted in the region between the intake and exhaust valves. The in-cylinder pressure, the rate of chemical energy release and the related parameters were evaluated on an individual cycle basis and/or averaged on 500 consecutive cycles [5]. A lambda sensor was installed at the engine exhaust for the measurement of the air/fuel ratio. The injection timing, as well as the ignition timing, was controlled by the Engine Timing Unit (ETU) multi-channel system. The engine was equipped alternatively with the standard spark plug and with a properly designed prechamber equipped with a direct injector (not used in this paper), used to inject the fuel into the prechamber, and a spark plug used to ignite the mixture. The engine was fueled with gasoline. The gasoline was port fuel injected at 3 bar.

Emissions Measurement Systems

HC, CO and CO_2 emissions were measured by means of a NDIR analyser. NOx emissions were measured by means of chemical sensors. Particle number concentration and size were measured by means of a TSI® engine exhaust particle sizer (EEPS) in the range from 5.6 to 560 nm. For particle number measurements, the exhaust was sampled and diluted by means of the Dekati® engine exhaust diluter (DEED), a Particle Measurement Program (PMP)-compliant engine exhaust conditioning system. The dilution ratio was fixed at 1:79.

Results and discussion

Engine operating conditions

All the experimental investigations were carried out at engine speeds of 2000, 3000, 4000 and 5000 rpm at wide open throttle for prechamber and partialized for standard ignition condition, in order to have the same peak pressure in motored condition. The intake air temperature was fixed at 298 K and the cooling water temperature was set at 333 K.
The end of fuel injection (EOI) and the start of spark were fixed in order to minimize the IMEP CoV and the duration of injection (DOI) were fixed in order to reach the stoichiometric equivalence ratio and the leanest combustion with a reasonable IMEP CoV,

which varied according to the engine point and the injection configuration. For both configurations, the end of fuel injection in the manifold was fixed at 0 CAD (TDC firing). For all the test cases, the electronic spark timing was fixed to operate at the maximum brake torque. The details of engine performance at the selected operating conditions are reported in [7].

Indicated data analysis

Using the prechamber, despite the lower compression ratio, the in-cylinder pressure is higher with respect to that obtained with the traditional spark plug for all the conditions because of the faster combustion due to the faster flame front propagation. In particular, at 2000 rpm for both stoichiometric and lean conditions the quantity of energy available in the combustion chamber is similar but slightly higher for prechamber configuration due to the air partialization of the standard spark plug condition, instead the engine efficiency is much higher in prechamber condition with respect to standard ignition one. Moreover at 3000, 4000 and 5000 rpm both stoichiometric and lean conditions despite the quantity of fuel injected in the combustion chamber is slightly higher for standard ignition condition the IMEP and the efficiency are always higher in prechamber mode. In all the configuration the fuel was injected in the same time and with the same pressure, there are no differences on the mixture formation and therefore on lambda distribution into the combustion chamber at the SOS and during the flame front propagation, the main difference is the flame front propagation due to the prechamber. At 2000, 3000 and 4000 rpm, for both lambda conditions, analyzing the prechamber combustion it is evident that ROHR curves sharply increase indicating a faster combustion with respect to the standard configuration resulting in higher in-cylinder pressure. The combustion acceleration for the prechamber configuration can be ascribed to the faster flame front propagation obtained with the outgoing plasma jets from the prechamber. Moreover, at the increase of the lambda value, the combustion speed is slowed down by the lower flame speed for lean mixture both for prechamber and standard ignition configurations. In Figure 4 the effect of the prechamber on the pressure cycles and ROHR at 5000 rpm WOT is reported. In standard ignition condition the combustion is faster and advanced as larger the fuel content, as evidenced by the ROHR curves. These results can be ascribed to the properties of rich mixture, such as the higher flame speed. With the prechamber ignition, combustion acceleration result in higher peak of the ROHR and a higher slope with respect to the standard ignition. Moreover, in lean condition, the combustion evolves closer to TDC in standard ignition mode due to the advanced start of spark improving the combustion efficiency and producing a higher effective pressure, as it can be seen in [7]. It is important noting, that despite the IMEP increase, in lean condition the efficiency is higher for the prechamber configuration due to the shorter injection duration with respect to the standard ignition. In stoichiometric condition at 5000 rpm the combustion shows similar injection duration, the prechamber strongly improves the stability of the combustion and consequently causes the increase of the Indicated Mean Effective Pressure (IMEP) averaged on 500 consecutive cycle. The values of the IMEP, the related coefficients of variance (CoV) and the lambda, for all the conditions are reported in [7]. A slight decrease of IMEP values for the prechamber conditions was observed only for 5000 rpm lean condition because of the worse mixture formation into the prechamber. In all the conditions the benefits of prechamber combustion exceed the disadvantages of the compression ratio reduction. The

CoV of IMEP is a marker of cyclic variability derived from pressure data, it is should be less than 3% to indicate that the combustion is very stable in each condition. The CoV of IMEP is influenced by the lambda and by the flame kernel. In particular, the CoV IMEP is lower for prechamber configuration suggesting a more stable engine combustion except for 5000 rpm lean condition, probably because at the increase of engine speed the prechamber scavenging and the subsequent filling are not perfect resulting in a mixture far from the stoichiometric value that causes a weak ignition. The reduction of the cycle-by-cycle variations can be ascribed to the increased turbulence flame speed rather than at the laminar flame speed that depends on lambda value.

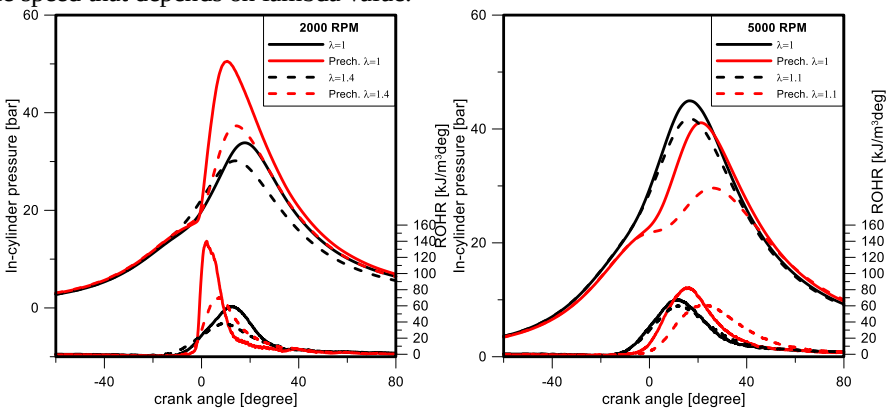

Figure 4. Comparison between the in-cylinder pressure and the ROHR for both ignitions type in stoichiometric and lean configurations at 2000 and 5000 rpm.

The effect of ignition strategy and lambda value on combustion is also evident in terms of Indicated Specific Fuel Consumption (ISFC), as it possible to observe from the figure 5 the ISFC is obviously always lower for lean condition in both ignition configurations. For the prechamber ignition the ISFC is much lower than standard ignition except at 5000 rpm where the prechamber scavenging and mixture formation needs to be improved.

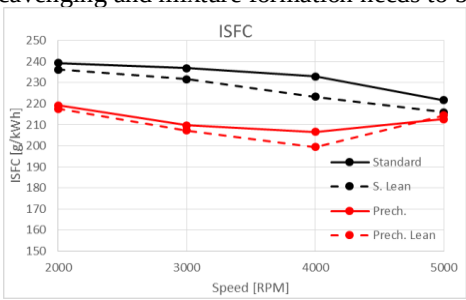

Figure 5. Comparison between the ISFC for both ignitions type in stoichiometric and lean configurations at several engine speeds

Exhaust gas emission data analysis

CO_2 and CO emissions are depicted in Figure 6 The CO emissions are lower in prechamber configuration despite for the larger fuel quantity than the standard ignition, as in this case the engine runs in partial load (PL), indicating a more efficient combustion, except at high

speed where the efficiency of the prechamber combustion decreases with respect to the lower engine speed, this data corroborates the indicated data analysis at 5000 rpm. Looking at the indicate data it is observed in this case a more advanced and faster combustion for the standard ignition despite the delayed SOS.CO_2 emissions are quite higher for the prechamber not only because of the larger fuel amount but because of the efficient combustion, as suggested by the CO results. In lean condition CO_2 emissions in prechamber condition are always lower with respect to the standard configuration, in this case the benefit of low fuel quantity injected in prechamber the combustion exceed the disadvantages of the higher CO_2 production due to the efficiency rise.

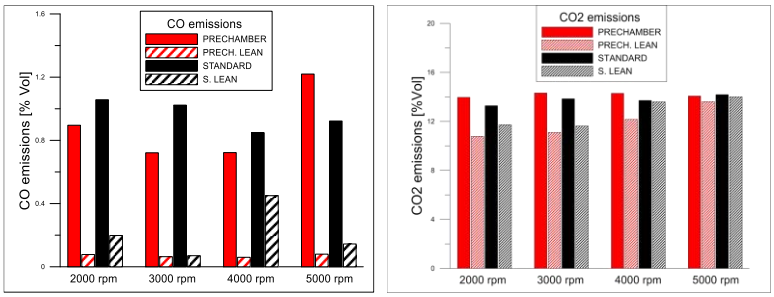

Figure 6. CO and CO_2 emissions for all the tested conditions.

The HC emissions are shown in Figure 7. The more efficient combustion is also evidenced by the much lower HC emissions for the prechamber configuration, for which a faster and more advanced combustion results in a higher in-cylinder temperature that enhances the fuel evaporation. Another aspect to consider is the lower quenching distance due to the faster combustion and then lower thermal dispersion. At 5000 rpm despite the high in-cylinder pressure/temperature is observed a larger emission of HC for the standard ignition. In this case the combustion is quite faster in the early phase but after slows down, see ROHR curve, losing the beneficial effect on the quenching of the flame.

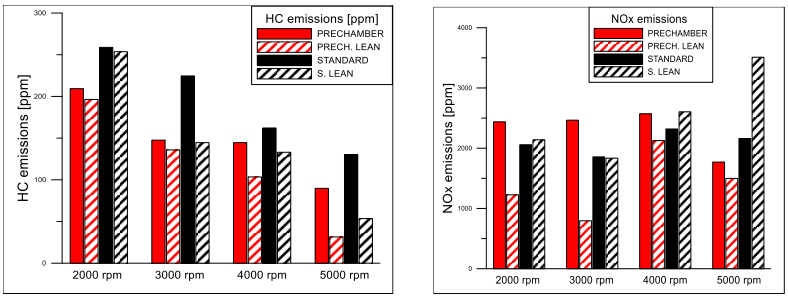

Figure 7. HC and NOx emissions for all the tested conditions.

NOx emissions are strongly affected by the in-cylinder temperature. They are larger for the prechamber configuration in stoichiometric condition, the faster and more advanced combustion results in higher in-cylinder temperature, this effect is more evident at low engine speeds. At 5000 rpm, instead, there is an inversion of the trend as a higher temperature is measured for the standard ignition configuration. In lean condition, instead, the NOx emissions are much lower for the prechamber configuration due to the low

differences between the pressure peaks of the two ignitions mode and probably because of the different turbulence in the main chamber caused by the flame jets outcoming from the prechamber.

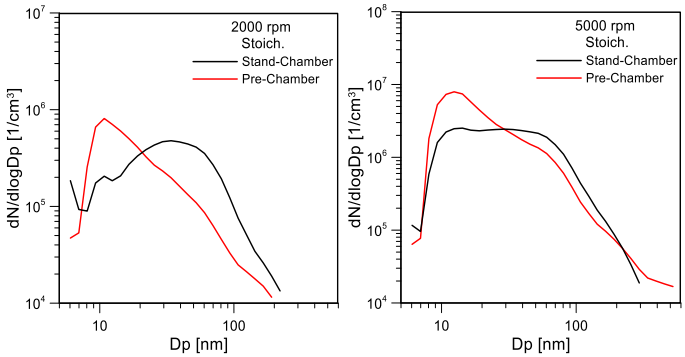

Figure 8. PSDFs measured in prechamber and standard ignition configurations at some engine speeds in stoichiometric condition.

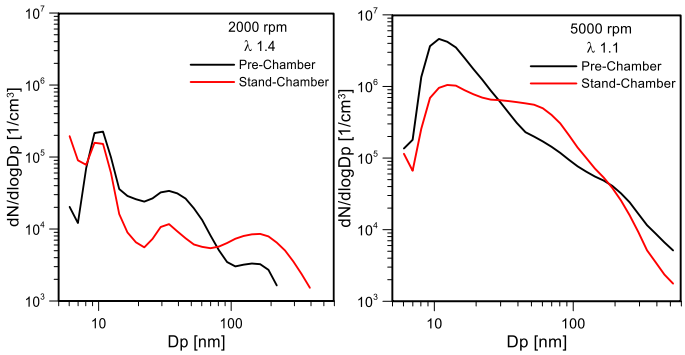

Figure 9. PSDFs measured in prechamber and standard ignition configurations at some engine speeds in lean conditions

Figure 8 depicts the particle size distribution functions (PSDF) at all tested stoichiometric engine operating conditions. The particle emissions are strongly affected by the engine configuration both in terms of number and size. A large number of nuclei particles are measured in the prechamber configuration at each engine speed. On the other hand, the accumulation particles are larger for the standard ignition configuration. The faster combustion due to the prechamber results in higher in-cylinder temperature that enhances the fuel evaporation across the valves, the main source of particles in the PFI engine. This effect is more evident at low engine speed where combustion is more influenced by the presence of the prechamber, as reported in indicated data. Another aspect to consider is that in standard configuration the engine works al partial load resulting in lower turbulence so despite the injected fuel is lower the particle formation is enhanced because of the lower turbulence. The high concentration of nuclei particles can be ascribed to the larger volatile. Despite the lower HC emissions, the more advanced combustion, due to the advanced spark and faster flame propagation, does not allow the complete combustion of volatiles that can nucleate. Figure 9 depicts the particle size distribution functions (PSDF) at some of the tested lean engine operating conditions. For lean condition similar results are observed. In

this case it is worth noting that particle emissions at 2000 rpm and 3000 rpm are very low and outcomes cannot be drawn as the Signal Noise Ratio is too high.

Conclusions

The analysis of the effect of prechamber ignition on the combustion stability and efficiency in stoichiometric and lean-burn operation condition in a small SI engine fueled with gasoline and equipped with a prechamber prototype is reported. It is properly designed to fit with the single cylinder SI commercial engine head. The engine is equipped alternatively with the standard spark plug and with the prechamber equipped with a spark plug used to ignite the mixture.

Using the prechamber, despite the lower compression ratio, the in-cylinder pressure is higher than that obtained with the traditional spark plug for all the conditions except for 5000 rpm because of the faster combustion due to the faster flame front propagation. At 5000 rpm the much advanced SOS in standard ignition condition causes a higher-pressure peak but a lower value of the CoV IMEP with respect to prechamber mode. It is a more stable engine combustion except for 5000 rpm lean condition, probably because at the increase of engine speed the prechamber scavenging and the subsequent filling are not perfect resulting in a mixture far from the stoichiometric value which causes weak ignition.

The effect of ignition strategy and lambda value on combustion is also evident in terms of indicated specific fuel consumption (ISFC). The ISFC is obviously always lower for lean condition in both ignition modes. For prechamber ignition the ISFC is much lower than standard ignition with except of 5000 rpm were the prechamber scavenging and mixture formation needs to be improved.
The CO emissions are generally lower in prechamber configuration despite for larger fuel quantity with respect to the standard ignition, as in this case the engine runs in partial load, indicating a more efficient combustion. In lean condition CO_2 emission in prechamber condition is always lower than standard configuration, in this case the benefit of low fuel quantity injected in prechamber combustion exceed the disadvantages of the higher CO_2 production due to the efficiency rise. The more efficient combustion is also evidenced by the much lower HC emissions for which a faster and more advanced combustion results in higher in-cylinder temperature that enhances the fuel evaporation. NOx emissions are much lower due to the low differences between the pressure peaks of the two ignitions mode and probably because of the different turbulence in the main chamber caused by the flame jets outcoming from the prechamber. A large number of nuclei particles are measured in the prechamber configuration at each engine speed. On the other hand, the accumulation particles are larger for the standard ignition configuration.
Prechamber offers the possibility to enhance the engine efficiency and the lean operating limit of engines, whilst ensuring stable operation at high excess air ratios. At high engine speed, probably it is necessary a fueled prechamber in order to improve the mixture formation and consequently the engine stability.

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