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A REVIEW OF PRECHAMBER IGNITION SYSTEMS APPLIED IN SI ENGINES

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ABSTRACT

Social-environmental issues have been neglected for a long time in internal combustion engines development. In recent years, due to increased awareness of its users and manufacturers, reductions in fuel consumption and pollutant emission have become essential requirements for the production and commercialization of engines. A viable solution to achieve these reductions is the use of lean burn engines. However, its low combustion stability is a recurrent preoccupation. The use of a prechamber combustion system rises as an efficient alternative, providing a highly stable combustion. Highlighting the importance of this system, this paper reviews the main advances and limitations on the use of prechamber combustion in spark ignition engines as well as expose its influences over combustion and pollutants emissions.

Keywords: Prechamber, engine, torch,

NOMENCLATURE

BMEP	Brake Mean Effective Pressure
BPI	Bowl-Prechamber-Ignition
BSFC	Brake specific fuel consumption
CA	Crank angle
CFR	Cooperative fuel research
CO	Carbon monoxide
COV_{IMEP}	Coefficient of variance of IMEP
CR	Compression ratio
EGR	Exhaust Gas Recirculation
HAJI	Hydrogen assisted jet ignition
HC	Hydrocarbons
HRR	Heat Release Rate
IMEP	Indicated mean effective pressure
ISCO	Indicated Specific Carbon monoxide
ISFC	Indicated Specific Fuel Consumption
ISHC	Indicated Specific Hydrocarbons
ISNO	Indicated Specific Nitrogen oxides
LIF	Laser-induced Fluorescence
LPG	Liquefied petroleum gas
MAP	Manifold absolute pressure
MBT	Maximum Brake Torque
MBT ST	Minimum ignition advance for best torque

MC	Main chamber
MFB	Mass Fraction Burned
NO_x	Nitrogen oxides
PC	Prechamber
PFJ	Pulsed Flame Jet
PLIF	Planar Laser-Induced Fluorescence
RCM	Rapid Compression Machine
SI	Spark ignition
THC	Total hydrocarbons
TJI	Turbulent Jet Ignition

Greek symbols

λ	Air to fuel equivalence ratio
ϵ	Maximum specific energy provided by the prechamber

INTRODUCTION

Researchers have been trying to improve the conventional spark ignition engine in order to meet new regulations and society demands such as reducing emissions and increasing fuel economy. One way to achieve these goals is using lean or ultra-lean air-fuel ratios (A/F). However, it generates major concerns, as the mixture ignition, since it requires greater energy to start combustion due to low fuel concentration in the cylinder [1]. With this question, many different high energy methods have been studied, as plasma jet igniters, rail plugs, photochemical devices, lasers, microwave concepts, torch cells, divided chamber stratified charge engines, flame jet igniters, combustion jet concepts and exhaust gas recirculation systems [2]. In this review, the main advances and limitations on the use of prechamber combustion strategies are contemplated.

Prechamber combustion strategies can be applied with homogeneous or stratified charges [1]. Stratified prechamber charges consists in create regions with different A/F inside the main combustion chamber (MC), facilitating the ignition process. Thus, near the spark plug the mixture is richer (there is an excess of fuel). In a prechamber engine with homogeneous charge, air-fuel mixture in MC fills the prechamber (PC) by interconnection orifices. Therefore, parameters as prechamber geometry, orifices diameters and spark plug location are extremely important for this system operation. In a prechamber engine with stratified mixture, the PC is still filled with the MC mixture, although there is an additional fueling system that supplies more fuel in PC than in MC, promoting charge stratification [3].

Use of stratification in PC shows potential to great improvement, mainly for its capacity to work with overall lean or ultra-lean mixtures. So, different configurations of engines with these methods have been studied since Ricardo [4] proposed the first engine with this technology. Other researchers have followed this path of study, as Gussak and Karpov [5] in Russia, which reported detailed studies of the concept. The Russians described the combined use of these technologies as a conventional spark ignition engine, with a small PC equipped with the spark plug and a third valve, supplied with a considerably richer A/F mixture [6].

Target of many researches in recent years, different PC ignition systems settings and strategies were used in order to achieve the desired results. Among prechamber studies, variations in size, geometry, placement, presence or not of valves or spark plugs in PC, and how the air flows within it, were explored [7]. Table 1 shows distinct models, with different nomenclatures, all with similar goals and the same main principle.

This paper presents a technical review about spark ignition engines with combustion prechamber, contemplating geometric and combustion aspects and investigating impacts on pollutant emissions. It aims to highlight the potential of this technology over the conventional spark ignition engine. Are observed better combustion properties [19, 20], reduction of nitrogen oxides emissions [21], reduction of specific fuel without losing too much power [22], energy delivered more efficiently [23], more turbulence generation inside the combustion chamber [24], and reduction of combustion temperature, being a great solution for a more efficient and sustainable engine.

GEOMETRICAL PARAMETERS

Performance of engines equipped with PC ignition systems is highly dependent on conception factors, such as geometry and volume of PC, nozzle diameter and number of orifices in the nozzle. Several researchers used fixed values of volume and orifice size, without analyzing their effects in order to optimize the systems. Others made numerical simulations to analyze combustion parameters, but still with fixed parameters and no variation of them. This section contains a brief review of the geometric factors influence in PC systems.

Table 1. Different nomenclatures for prechamber spark ignition engines.

CVCC	Compound Vortex Controlled Combustion	Date *et. al*, 1974 [8]
LAG	Ignition by Avalanche Activation	Gussak *et al.*, 1975 [5]
SKS	Stable Kernel of Combustion	Garret, 1975 [9]
TI	Torch Ignition	Adams, 1979 [10]
PJI	Plasma Jet Ignition	Giardini, 1980 [11]
SCSP	Swirl-Chamber Spark Plug	Latsch, 1984 [12]
TC	Torch Cell	Schaub, 1994 [13]
APIR	Self-ignition Triggered by Radical Injection	Robinet *et al.*, 1999 [14]
BPI	Bowl Prechamber Ignition	Kettner and Rothe, 2005 [15]
HAJI	Hydrogen Assisted Jet Ignition	Toulson *et al.*, 2007 [16, 17]
TJI	Turbulent Jet Ignition	Attard *et al.*, 2010 [18]
FJI	Flame Jet Ignition	Bunce *et al.*, 2014 [19]

Prechamber volume

Adams [20] said that the nozzle orifice size and auxiliary chamber, i.e. prechamber, were chosen in a way that the gases rushing out of the PC generates sufficient turbulence in MC to cause total burning of the mixture in an acceptable period of crank angle, not quenching the flame. He mentions that the most designs incorporate in PC, 8 to 15% of volume from the total clearance volume of the engine. The author proposes a methodology to calculate the nozzle diameter and PC volume as a function of the turbulence intensity and turbulence generation. His calculations showed that, for a 5L engine, the optimum PC volume is 12% of the MC volume.

Wall and Heywood [7] experimented two PC designs, one with 8.7% of MC volume and other with 12%. Results showed that specific nitrogen oxides (NO) emissions decreases, while hydrocarbons (HC) and carbon monoxides (CO) emissions are little affected with the increase of PC volume. As the A/F was maintained constant, a larger volume of the PC produce a leaner mixture, which explains the lower NO levels.

Nozzle diameter

Sakai et al. [21] investigated the influence of the nozzle's diameter in the performance of a PC in a single cylinder engine with 10% volume of the MC. The engine was tested under partial load at 1200rpm and full load at 1600rpm. All tests were done at Maximum Brake Torque (MBT) condition and the nozzle diameter was varied from 4mm to 13mm. It was found that, as the diameter decreases, the ignition timing is retarded and combustion speed increases. Under partial load conditions, this increase in combustion speed tends to lower exhaust temperatures and improve the indicated specific fuel consumption (ISFC). However, under full load conditions, the effect was the opposite: the results showed an increase on knocking phenomena and throttling losses as the nozzle diameter decreases, making ISFC worse.

Wall and Heywood [7] also tested three different orifice sizes and evaluated combustion parameters as well as emissions. They found that decreasing the orifice diameter, therefore the sectional area of the nozzle, a significant drop in peak pressure difference between the PC and the MC, which leads to an increase in cycle-to-cycle variation. This trend is attributed to the decrease in kinetic energy of the gases leaving the PC as the sectional area of the nozzle gets bigger. Also associated with the kinetic energy, authors found that smaller orifices produce much faster combustions, reducing ignition delay and Mass Fraction Burned (MFB), and require less advance in ignition timing.

Ryu and Asanuma [22] investigated the effects of nozzle diameter in a single cylinder 297cc engine, equipped with a PC of 7% from the displacement volume. Authors tested three nozzle diameters for different air/fuel ratios: 6, 11 and 14mm. Attention was made to the fact that the engine was not able to run smoothly with a PC nozzle of 6mm and A/F of 15 and 17, probably because the mixture were too lean to ignite due to poor scavenging. They found that the heat release rate (HRR) of the combustion increases inversely proportional to nozzle diameter. However, the ignition delay and combustion duration were smaller for orifice diameter of 11mm, followed by the 6mm and then 14mm with much slower combustion than the other two nozzles. This trend was confirmed by the analysis of the maximum gas temperature of the combustion and the temperature-rise rate, which demonstrated the same pattern.

Murase and Hanada [23] observed two-dimensional images produced with Planar Laser-Induced Fluorescence (PLIF) technique to study the effects of A/F ratio, cavity volume and orifice diameter on combustion parameter of Pulsed Flame Jet (PFJ) ignition systems. Peak pressure of combustion was 3ms after spark discharge in all cases, regardless of cavity volume and nozzle size. Although, the highest mean pressure was attained with the smaller orifice and the bigger cavity.

Ukawa [24] made a numerical simulation, using the Honda CVCC engine as a base, to investigate the NO_x emission and fuel consumption reduction mechanisms. Author varied some design factor, as PC volume and nozzle sectional area. The values were 0.0248cm², 0.1791cm² and 0.8151cm² for nozzle sectional area and 2.167cm³, 3.954cm³ and 8.667cm³ for PC volume. Author concludes that there is an optimum nozzle diameter (0.677 cm) that must be chosen in a way that the torch-jet penetration length grows to the full length of the piston bore, as fast as possible.

Rothlisberger [25] analyzed the effects of PC geometric factors with numerical simulation and experimental tests. Were tested different PC volumes, shapes and nozzle diameters. Author found that the magnitude of gas-flow velocity depends essentially on the PC shape and the reduction of nozzle diameter increases significantly the flow velocity from the PC to the MC. Also, the turbulence generation depends highly on the number of orifices and their size. High turbulence is expected to generate fast combustions and in most cases the prechamber ignition system generated values much higher than convention SI engines. However, it also provokes a disturbance of the ignition process and initial kernel growth which can increase the probability of ignition failure. Accordingly to the numerical simulations, the mixture temperature is not significantly affected by the PC shape, but the heat transfer to the walls is. A decrease in nozzle diameter increases significantly the pressure drop between the PC and the MC, which increase turbulence intensity and affects negatively in the PC filling.

Cruz et al. [26] developed a zero-dimensional computational model with the objective of analyze the combustion of a torch ignition engine. Computational model was capable to perform a parametric study of PC geometric design factor, being able to modify and test alternative parameters and to find the optimum value for PC volume and nozzle size for the engine used.

Alvarez et al. [27] proposed a one-dimensional mathematical model to suppress the lack of methodologies for calculation of PC volume and nozzle diameter. Their model uses Adams [20] model as a basis but extend its use to different engines, conditions and fuels. From desired combustions parameters, the model performs numerical simulations varying parameters as PC volume, combustion duration, nozzle diameter and maximum specific energy provided by the prechamber (ϵ). A total of 17010 different arrangements were simulated. Authors mentioned that the maximum specific energy provided by the prechamber and maximum engine speed are highly affected by the orifice diameter. Therefore, considering a maximum speed of 6000 rpm and maximum ϵ presented in various arrangements was concluded that the optimum PC volume is about 7% of the MC's volume and the nozzle size should be of 6mm.

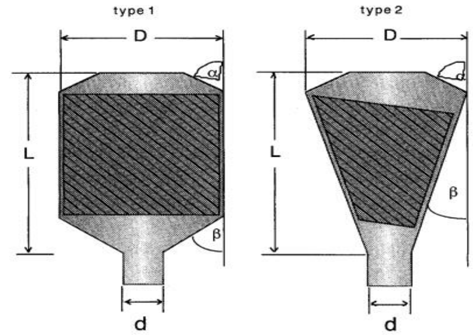
Spark plug position

Wallesten and Chomiak [28] experimented different spark plug positions inside the prechamber. Cylindrical PC with 1% of the MC volume and a high length/diameter ratio were chosen to enhance the effects of spark plug positions. The nozzle had 5mm of diameter. It was tested three different spark plug positions: the first one close to the opposite wall from the PC nozzle, the second one in the middle section and the third position close to the orifice that connects PC to MC. Study concluded that the spark plug position had great influence on the jet-generated turbulence and on ignition lean limit. The first position presented the highest turbulence and increase in reaction rate. The second generated the strongest jet prior to the flame due to the largest flame area inside the PC, but had the smallest improvement of lean limit. Authors claimed that the third position was the best spark plug position as it showed the best improvement in the ignition ability and had the lower turbulence generation effects.

Pre-chamber shape and position

Wolff [29] experimented two different prechamber shapes, one cylindrical and another conical, as shown in Figure 1. Both PC's are made of quartz and connected to the main chamber via a 7mm diameter, 19mm long nozzle. By means of a laser-induced fluorescence (LIF) method, the author investigated the mixture formation inside the pre-chambers and compared them. Special attention was made to mixture homogeneity. Author claims that LIF images at various crank angles shows that mixture formation is better for cylindrical pre-chambers, compared to conical, due to early recirculation flows, especially near the nozzle. Almost no recirculation of gases appeared in the conical shaped pre-chamber, which can be explained by the smaller angle near the nozzle entry. Also, fuel concentration images indicate that great areas of extremely rich or lean mixtures appears at conical pre-chamber, representing up to 30% of total PC volume, in contrast to cylindrical PC which presented values lower than 6%. Those regions are above or lower than ignition limit, thus representing a problem as mixture will not ignite, causing an increase in cycle-to-cycle variations and hydrocarbons emissions.

Moreira [30] says that spherical shape are widely used in early PC systems due to its easy adaptability, since they were commonly used in compression ignited engines. However, spherical PC concentrate more easily the intensity of combustion within itself, which harms the propagation of the flame into the main combustion chamber. Also, sphere shaped PC makes more difficult to produce a homogeneous mixture, increasing cycle-to-cycle fluctuations. Therefore, cylindrical are more adequate to achieve better combustion characteristics.



	D [mm]	d [mm]	L [mm]	α [deg]	β [deg]	V [cm3]
type 1	28	7	42	60	45	21
type 2	28	7	40	60	16.5	11

Figure 1. Cylindrical and conical prechambers tested by Wolff [29]

Ryu et al. [31] studied the influence of nozzle direction relative to piston head plane. As shown in Figure 2, four positions were analyzed and engine performance parameters recorded. Nozzle direction of 90° presented better results for all parameters. Notably, combustion is faster with vertical oriented nozzle producing a heat release rate much steeper than those for other directions. This fact is explained by the high turbulence created by the torch jet impingement onto piston head. Moreover, vertical nozzle allows decrease in advance of ignition timing, reduction of the mass burned fraction 10-90% and of specific fuel consumption. Similar results were obtained with directions of 60° and 120°. Worst values of break mean effective pressure (BMEP) are presented for the horizontal oriented nozzle (180°), caused mainly by spark position far from main chamber center.

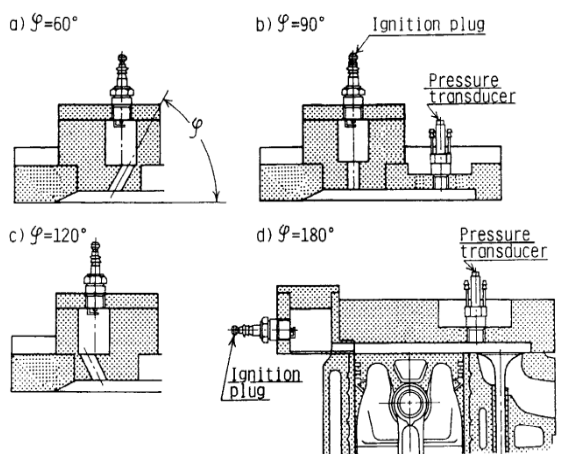


Figure 2 . Nozzle orientations tested by Ryu [31]

INFLUENCES ON COMBUSTION

Many authors [14, 32-35] identified the necessity of provide more energy to start combustion, and the low flame propagation speed, as the main problems of working with lean mixtures. Implementation of a PC ignition system provides a remarkable improvement in lean mixtures ignition due to better lean limit, spark timing, start of combustion, flame propagation speed and heat release rate, as presented and discussed by these authors.

Lean limit

Implementation of stratified ignition system with PC is able to maintain combustion quality and repeatability under stoichiometric mixtures conditions, when working with lean mixtures on the main chamber. Further, systems with PC obtain high reduction in fuel consumption and low nitrogen emissions [32]. These conclusions were reinforced in a study of a Cooperative Fuel Research (CFR) engine performed by Wimmer [36], and in a divided chamber bomb studied by Yamaguchi et al. [37].

Toulson et al. [38] evaluated the lean limit of a Hydrogen Assisted Jet Ignition (HAJI) system in a CFR engine with gasoline mixture in MC for different gaseous fuels in PC. Results indicate that hydrogen (H_2) extended the MC lean limit to an air to fuel equivalence ratio (λ) equal to 2.5, while Liquefied Petroleum Gas (LPG), Compressed Natural Gas (CNG) and Carbon monoxide (CO) extended the lean limit to λ = 2.35, λ = 2.25 and λ = 2.15, respectively. This range of engine lean stable operating limit indicates that the level of PC fuel ignition enhancement depends on several factors, including flame speed propagation and generation of chemically active combustion products, and not only on the amount of energy contained in the fuel [16].

Subsequently, Toulson et al. [39] investigated the use of LPG in the MC and PC, which would allow a single fuel alternative system. The effects of have LPG and gasoline separately in the MC and H_2 and LPG in the PC were analyzed to determine the fuel influence over the mixture lean limit, the emissions levels and in the combustion characteristics. Figure 3 shows lean limits for different fuel combinations, being LPG-H_2 combination the one with higher lean limit, followed by H_2-gasoline combination. LPG-LPG and LPG-gasoline cases were very similar, and the difference between them was within the experimental error, thus were not considered significant.

Toulson et al. [39] reported that at low manifold absolute pressures (MAP), lean limits variations were much higher than at high MAP, and in all fuel combinations the lean limit was extended to slight higher than λ =2.5. At low MAP, the lean limit extension differed over 0.5 among considered fuel combination. Overall, using H_2 as PC fuel extended the lean limit even more than when using LPG. However, there was no significant difference in lean limit extension whether GPL or gasoline was used as MC fuel.

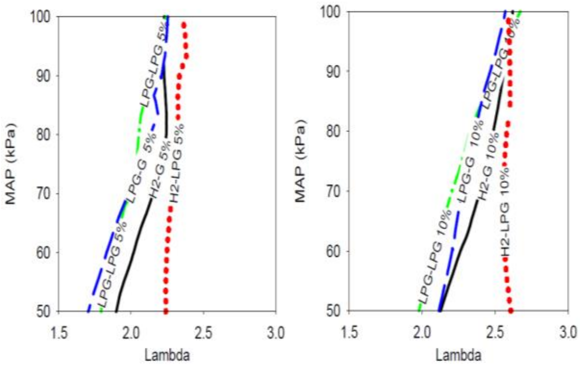


Figure 3. Comparison at 5% (left) and 10% (Right) COVIMEP of the PC-MC fuel combinations with varying λ and MAP (1200 rev/min, CR=11 and MBT ST) [39]

Attard et al. [40] compared ignition and combustion behavior in an optical single cylinder engine using a stratified Turbulent Jet Ignition system (TJI) [41], burning natural gas in the MC in order to provide information on the ignition process. The lean limit was extended from λ = 1.3, present in the baseline engine, to λ =1.8 as the TJI system was implemented, achieving an acceptable combustion stability. In this study was found that the use of TJI with leaner mixtures produced a brighter flame with more intense blue color than in the baseline engine. The authors believed that it was due to the increase of heat release rate, indicating a more stable combustion.

Spark timing

In general, PC ignition systems presented a decrease in ignition advancement which can be explained by an improvement in flame propagation speed. Since the first three valve system with auxiliary carburetor [42-44], it was shown that combustion speed was significantly increased by the PC implementation, where the ignition advancement for maximum power were 2 to 3 times lower than in an engine without PC.

PC effects in spark timing of a homogeneous ignition system were studied by Ryu [31], which correlated it with the intake jet direction for MBT condition. Author concluded that the smallest spark timing angle occurred when the angle between the jet and the piston horizontal face was 90°. Robinet [14], with an stratified prechamber ignition system denominated APIR, from French definition 'Auto-inflammation Pilotée par Injection de Radicaux', also made a detailed comparison between the operation of a conventional SI engine and a system with APIR device. Figure 4 shows that spark timing for the conventional burning case was optimized for MBT at 32° BTDC and it was drastically reduced to 10° BTDC with APIR system.

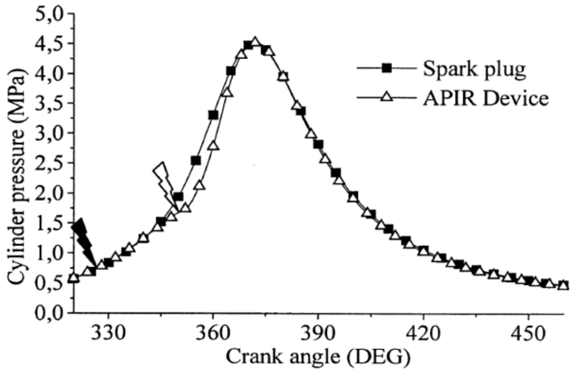


Figure 4. Mean pressure vs. each crank angle degree over 200 consecutive cycles [14]

One of major consequences of the spark timing delay observed by Roethlisberger [41] was the significant decrease in combustion temperature. This behavior was also documented by Pischinger [19], however in this study was found that the spark timing reduction had a negative impact over the Brake Mean Effective Pressure (BMEP) and Brake Specific Fuel Consumption (BSFC).

Figure 5 shows the spark timing behavior of recent studies about a single cylinder engine with TJI system. It shows the effects of ignition variations on Indicated Mean Effective Pressure (IMEP) and on combustion stability at 1500 rpm, with λ = 1.8, using a TJI system with constant air and fuel flow [18].

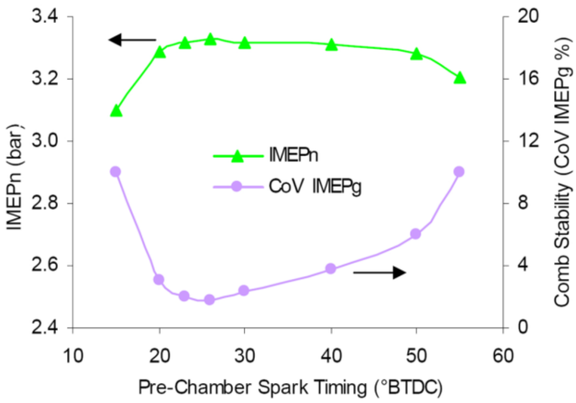


Figure 5. IMEP and combustion stability for prechamber spark timing variation with TJI. 1500 rpm, constant air flow and fuel flow, λ = 1.8 [18]

Results highlight the insensitivity of TJI combustion system to ignition variations when compared to conventional SI engine. Figure 5 shows an IMEP variation about 1% over a range of 30° on spark timing and combustion stability over a range of 40° on spark timing.

Start of combustion

Sakai [44] defined ignition delay as “the time duration expressed in crank angle degrees (CA) from the spark timing to the point when the pressure rise due to combustion is detected”. Author identified that the ignition delay in PC ignition system was 11°CA over all ranges of A/F, while the ignition delay adopted in conventional engine was longer and increased gradually as the A/F becomes larger. Ryu [22] reached similar results.

Another way to evaluate start of combustion is analyzing the Mass Fraction Burned (MFB). It is a fraction of energy released from combustion by the fuel to the total energy at the end of the combustion process. MFB is determined from cylinder pressure analysis. Rate of combustion is a very important parameter that affects thermal efficiency, peak cycle temperature and pressure, and exhaust emissions. Combustion rate is usually quantified in gasoline engines by the calculation of burn angles, which are the crank angles in which the MFB reaches a specified value [26, 45].

Toulson et al. [38] analyzed the influence of HAJI system in combustion start in a range of 0-2.5% MFB duration. Results indicated that flame initiation does not increase substantially with increasing A/F equivalence ratio. In addition, combustion start is faster with H_2 than with other fuels. This effect was most likely due to the increased laminar flame speed and high levels of chemically active species present in the combusting H_2 jet. Attard et al. [18] showed in his work that ignition system with PC presented a faster ignition due to the near constant mixture composition in PC, coupled with the distributed ignition sites provided by the jets and the high levels of chemically active species present in the combusting jets. For the engine without PC, it takes more time to initiate and stabilize the flame kernel after the spark discharge due to the reduced kernel growth associated to the diluted mixture. Small reduction in MFB duration with lean mixtures in a stratified PC system was also reported by Kettner et al. [15].

Gentz et al. [46] used the method presented in [3] to compare 0-10% burn rate of baseline SI with TJI system performance in a Rapid Compression Machine (RCM). It was observed that 0-10% MFB for SI system is slower than almost all TJI tested cases, indicating that TJI improved SI combustion start. As shown in Figure 6, TJI system had a 0-10% burn duration longer than SI system when working with $\lambda = 1.65$ and with larger nozzle diameters.

Results from Figure 6 can be assigned to the jet speed reduction and combustion instability, due to mixture impoverishment. Authors analyzed the images of turbulent jet flame structure and observed that the jet development for the smaller orifices resulted in increased turbulence relative to the produced jet by the largest orifice.

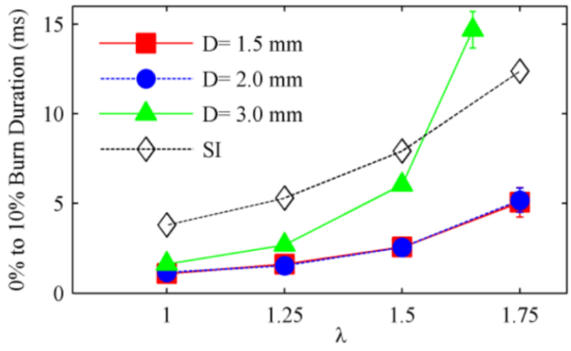


Figure 6. Variation in 0-10% burn duration with λ for all TJI nozzle diameters an SI [46]

Flame propagation speed

Sakai et al.[44] performed experiments on combustion duration with PC ignition systems and noticed that combustion becomes longer as mixtures become leaner in both ignition systems, with and without PC. It was also shown that burn duration was greater in engines with PC system for all ranges of A/F when compared to conventional engines. This behavior was not found or ratified by other authors. To quantify and estimate flame propagation speed in a RCM, Gentz et al. [46, 47] used 10-90% burn duration as evaluation parameter. Figure 7 shows the results for tests described in [46].

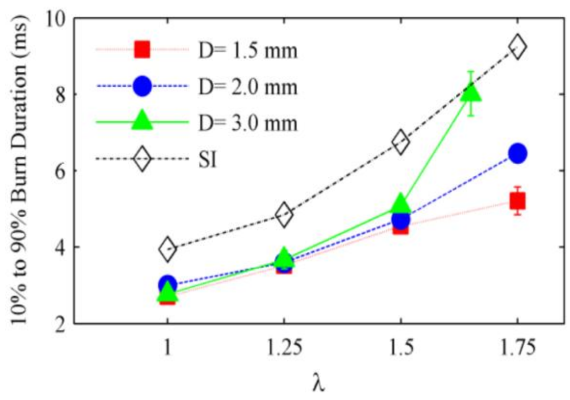


Figure 7. Variation in 10-90% burn duration with λ for all TJI nozzle diameters and SI [46]

Authors identified that 10-90% burn duration for TJI system are very similar for different nozzle diameters until $\lambda = 1.5$, where the smallest nozzle had lowest burn duration while the largest nozzle had greatest duration. These results means that for conditions near stoichiometric, nozzle diameter had a reduced effect on flame propagation speed on the main chamber. However, when λ is bigger than 1.5, smaller nozzle diameters were advantageous for combustion. This was probably due to turbulence increase provided by a turbulent jet from the smaller nozzles, which generated more active radicals to initiate main chamber combustion. 10-90% burn duration was slower for conventional system than for TJI system in all cases, indicating that the flame propagation speed is greater with TJI system [46].

To determine the better fuel combination in MC and in PC, Toulson et al. [39] used 2.5-90% MFB duration and noticed that this parameter was very similar for all fuel combinations, indicating that following the flame initiation process, the PC fuel no longer has a large effect on burn duration, suggesting similar jet penetration. In addition, flame propagation and main energy release phase was very similar whether gasoline or LPG was used as MC fuel. For all fuel combinations, the increased 2.5-90 % MFB duration that occurs as the mixture becomes leaner was due to the decrease in combustion temperatures and related decrease in flame speed.

Attard et al. [18] agrees that there was an increase in flame propagation speed and discuss that the effect of jet is less pronounced during the flame propagation in MC. The 10-90% burn data is more an indication of the multiple flame fronts propagating through the MC. However, PC fuel continues to play a role during the main fuel combustion, as is evidenced by the shorter burn duration that occurs with jet ignition relative to spark ignition. This was attributed to enhanced combustion provided by the prechamber fuel and the high levels of active radicals produced in the combusting jet being maintained in the propagating flame. Some authors identify the increase of flame propagation speed as a knock control agent [34, 36, 44, 48].

Heat release rate

The heat release rate is defined as the rate at which the chemical energy of the fuel is released by the combustion process. It is calculated from cylinder pressure versus crank angle as the energy release required creating a measuring pressure system [44, 49, 50]. Therefore, many authors analyzed the heat release rate as a parameter on combustion process study. One of the first studies about this subject was made by Sakai et al [44] as shown in Figure 8. It was observed from a PC system a rapid increase in heat release rate at initial stage of combustion process, and a decrease after a short stable period. Moreover, in the conventional engine the heat release rate shows a simple bell-shaped form.

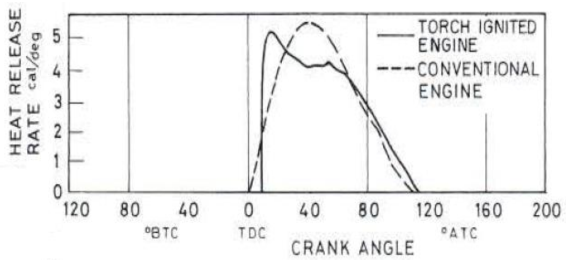


Figure 8. Comparison of combustion characteristics at constant indicated specific fuel consumption. Adapted from [44]

Sakai said that the first heat release rate peak in PC system, showed in Figure 8, was affected by A/F ratio inside the PC, thus for rich mixtures in PC the initial heat release rate showed a sharp peak and decreased steadily afterward. However, as PC mixture became leaner the peak value of heat release rate gradually decreased, being nearly the same as the conventional engine, and the peculiar peak in the pressure diagram disappeared [44].

Ryu et al. [17] related the same behavior of heat release rate to diameters of interconnection holes between PC and MC. For smaller diameters, heat release constant patch found by Sakai was no longer detected, and the combustion was so fast that only the peak from combustion’s first stage was identified. Later, Ryu found that the angle in which PC flames enters in MC also affected the heat release rate. To vertical torch nozzle, the crank angle of combustion finish was much earlier, and the rising gradient of heat release rate was steeper than for other torch nozzle directions, due to violent turbulence caused by torch jet impingement vertically onto the piston head. As shown in Figure 9, Kawabata et al. [51] noticed that a larger number of nozzles presents a shorter duration of heat release. This fact was due to the promoted ignition and mixture combustion in MC.

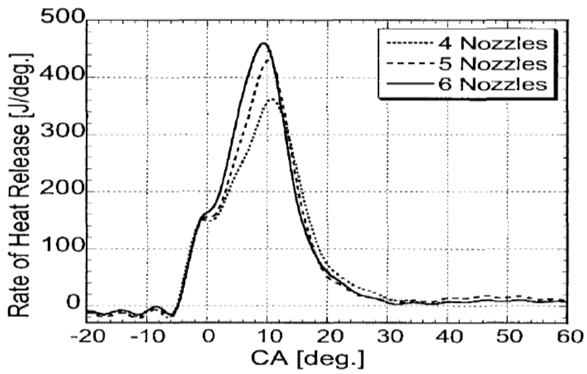


Figure 9. Comparison of Rate of Heat Release in the main chamber [51]

This section review shows that almost all problems about burning lean mixtures can be mitigated by the implementation of an ignition system with a combustion PC. Based in what has been showed, the main chamber combustion is affected by:

- Increased lean limit;
- Decreased Spark Timing for MBT;
- Decreased time for Combustion Start;
- Increased Flame Propagation Speed;
- Decreased knock;
- Increased Heat Release Rate.

INFLUENCES ON EMISSIONS

The increase number of vehicle and related emissions has led to environmental problems such as global warming and acid rain, resulting in adverse effects on human health. The major toxic components in the exhaust gas of combustion engines are carbon monoxide (CO), hydrocarbons (HC) and nitrogen oxides (NOx), being the NO_x emissions the most difficult to be reduced [52].

Ignition systems with combustion prechamber holds an attractive potential for reducing exhaust emissions relative to conventional spark-ignition engines [53]. The mixing of MC air/mixture with rich combustion products from PC results in burned-gas temperatures that not support NO_x formation while maintaining low CO and HC emissions [36].

Emissions in homogeneous prechamber engines

In homogeneous prechamber engines, turbulence accelerate the burning speed in PC and this mixture accelerates the burning speed in MC. Both eddies and jets seems to increase the cooling loss from the chamber wall, decreasing the gas temperature. Burned gas from PC has the highest temperature, while the burned gas in MC has the lowest. This temperature gradient, and the cooling loss, tends to reduce the NO_x formation [54, 55].

Jarosinski et al. [56] studied a homogeneous prechamber engine with a catalytic insert, and found that the CO and unburned HC emissions levels have tendency to increase with the rise of the air-fuel ratio, while the NO_x emissions has the opposite tendency.

Roethlisberger and Favrat [57] investigated the size, number, distribution and orientation of nozzle orifices, PC internal volume and its impacts, on NOx, HC and CO emissions. Authors found that the reduction of the total nozzle orifice cross sectional area leads to similar CO and Total Hydrocarbons (THC) emissions. They attribute this similarity to two conflicting effects: the deeper penetration of the gas jets induced by a smaller nozzle orifice cross sectional area tends to reduce the amount of unburned mixture compressed in the combustion chamber crevices. However, the intensification of combustion process by stronger gas jets increases the cylinder pressure, tending to increase the quantity of unburned mixture flowing into the crevices. Results indicated the existence of an optimal cross sectional area for nozzle orifices, of 18.85 mm2, which achieves the lowest NO_x emissions.

Furthermore, the transition from 6 to 4 orifices resulted in a reduction of approximately 5% and 9% of the CO and THC emissions, respectively, with low NO_x emissions. The orientation of nozzle orifices towards the squish region simultaneously reduced CO and THC emissions, particularly when increasing the relative A/F ratio to decrease NO_x emissions. Reduction of PC internal volume yields higher CO and THC emissions at constant NO_x emissions. This tends indicate that the increase in CO and THC emissions mainly results from the mixture in combustion chamber crevices [57].

Gomes [58] tested a 4-cylinder engine provided with homogeneous PC in three different settings, the first with four individual ducts from the top and a nozzle with inclination to direct the flow to the intake and exhaust valves, the second with a 6mm diameter central duct and four nozzles with the targeted primary chamber to the intake and exhaust valves, and the third in which the central duct was directly interconnected with the main chamber. The results of the influence of the setting variation on CO, HC and NO_x emissions can be verified in Figure 10, Figure 11 and Figure 12, respectively.

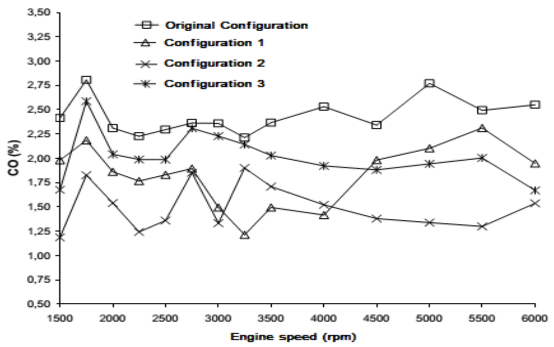


Figure 10. Variation on CO emissions [58]

Author described that the CO emissions to the engine equipped with PC ignition system, are less than the index presented by the engine equipped with original ignition system for all rotations, indicating improved combustion due to the generation of greater turbulence [58].

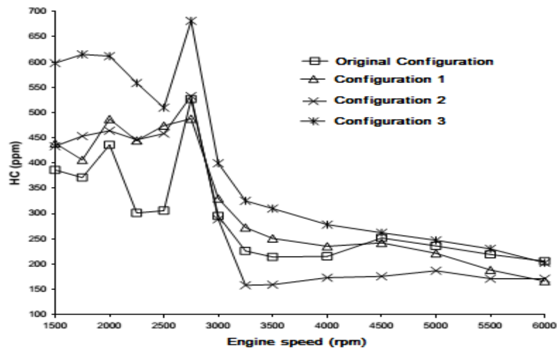


Figure 11. Variation on HC emissions [58]

Figure 11 presents that the hydrocarbon emissions indexes showed the same trend behavior for all systems, presenting the highest value in the maximum torque rotation [58] .

Contrary to the most studies about PC ignition systems, Gomes [58] found that NO_x emissions for the engine with PC was greater than that obtained with the original engine. Author explained that the high NO_x content occurred due to the high temperature in PC and MC, requiring improvements in his project design to promote the effective cooling of prechamber.

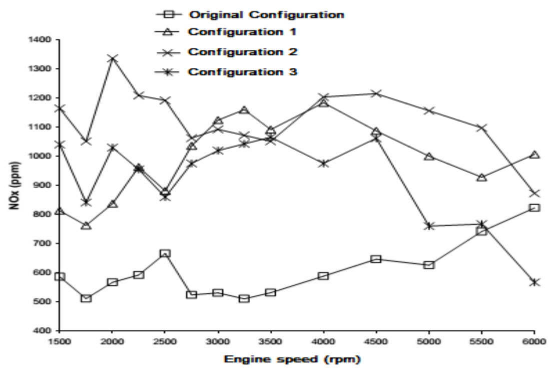


Figure 12. Variation on NO_x emissions [58]

Roubad et al. [59] investigated the use of a synthetic biogas in a homogeneous PC engine and compared the emissions with the resultant in use of natural gas for the same engine. With a rated brake power output of 150 kW and the same NO_x emissions, the CO and HC emissions were 15% and 8% lower, respectively.

Moreira [60] in his research with a 16 valves, 4-cylinder engine found a reduction of 92.25% on CO emissions when operating with a

homogeneous PC charge. On the other hand, HC percentage was usually higher, except when operating at $\lambda = 1.20$. Author explained that this occurred due to the nozzle configuration, which had only one central jet, generating some blind zones where the flame can be quenched, being also favored by the effect of the liquid wall in HC production.

Emissions in stratified prechamber engines

In engines with stratified prechamber, combustion starts in the fuel-rich PC with small formation of NO due to lack of oxygen. As burning charge reaches the main chamber, the relatively cool mixture contained there promotes the rapid quenching of NO formation reactions. Furthermore, the fast rate of energy release permits the use of retarded spark timing, contributing further to reduction of NO_x emissions [53].

Several authors [2, 5-7, 21, 33, 44, 61-64] studied the influence of A/F ratio on emissions in stratifies PC engines. Wimmer and Lee [36] showed that for $\lambda = 1.2$, the NO_x emissions have been reduced by half, reaching values less than 1 gram per indicated horsepower-hour (g/ihp.h) for $\lambda = 1.8$. HC and CO emissions were higher for both lean limits, but the values were still so low that were not considered a problem for emissions control at the time of the study was conducted.

When each engine studied by Wimmer and Lee [36] was operated for minimum ISFC, the prechamber engine produced an average NO_x emission lower by 83%, an average CO emission lower by 82%, while average HC emission was higher by 83%. This increase in hydrocarbon emissions was attributed to quenching of the oxidation reactions in the combustion chamber wall region. Moreover, fuel caught in engine crevices has contributes to the emission of unburned HC [36, 53]. Lawrence and Watson [65] found that active radicals help to decrease the level of quench and piston crevice sourced HC.

The influence of PC geometrical aspects on emissions is another factor that has been extensively studied. Sectional area of the nozzle is one of them, and has a great effect on HC emissions, although not shown trends in CO and NO_x emissions [20, 44]. Other aspects that distinctly influence HC emissions are the number of injection nozzles and the position of spark plug in the PC [2].

Wall and Heywood [62] by varying the diameter of the nozzle from 8 to 12 mm have found that the indicated specific NO, HC and CO emissions tends to increase with increasing orifice size, they attribute the lower values with smaller orifice diameters as a result from more complete mixing between quench zone and bulk gases during the combustion process. Similar results were found by Dimick et al. [63]

Adams [10] investigated the influence of nozzle orientation on emissions and found that the nozzle orientation changes the HC and NO_x emissions approximately 300%, reaching the minimum values at the 8:30 position.

About prechamber volume variation, Wall and Heywood [62] found that the indicated specific NO emissions are seen to decrease as the PC volume increases, although this variation has little effect on specific HC and CO emissions [62]. However, Dimick et al. [63] investigated the percent of PC volume variation over a range of 4% to 12% and found that the 8% volume design gave the best compromise between HC and NO_x emissions. Nakazono and Natsume [8] explain that when the PC volume is larger, the combustion period is shorter so NO_x emission is higher.

Lumsdem and Watson [66], with the HAJI system, found over than 98% of reduction in raw NO_x emissions and 60% in raw CO emissions compared with the baseline operation, while HC emissions had a considerable increase. Authors justified that those emissions can be reduced or maintained using an oxidation catalyst, compared with the same engine running with $\lambda = 1.0$ using a three-way catalyst.

With the same HAJI system, Toulson et al. [35] investigated the effects of hot and cool Exhaust Gas Recirculation (EGR) on Indicated Specific Hydrocarbons (ISHC) and $ISNO_x$ emissions. The tests showed a general trend of decreasing HC emissions with increasing load, this was attributed to higher in-cylinder and exhaust temperatures, which promoted HC burnup. Increasing the fraction of cool EGR. HC emissions increases, while increasing the hot fraction reduced them. NO_x emissions decline substantially with decreasing load but is difficult to discern any trends due to EGR. At higher loads, however, $ISNO_x$ emissions decreased significantly with increasing of cool EGR as a result of the lower combustion temperatures. Increasing percentages of hot EGR led to slight increases in NO_x emissions, this was attributed to the higher quantities of H_2 required to sustain combustion stability, which leads to higher combustion temperatures.

Hamori [67] realized an investigation into the use of the HAJI system with supercharging, to simultaneously reduce engine out NO_x and improve the thermal efficiency. Results showed that for all load points, the system operating with gasoline increased thermal efficiency by up to 41%, reduced CO by 90% and increased HC emissions by up to 3.5 times while maintaining an almost zero NOx. The same tests performed with a hydrogen fuel supply, increased thermal efficiency by up to 10% over its spark ignition hydrogen counterpart and reduced CO, HC and NO_x emissions to near zero levels.

Another interesting research about HAJI system was made about the effect of prechamber fuel variation on emissions [38, 39, 68]. Figure 13 shows the effects of the PC fuels on indicated specific NO_x emissions. Once an air to fuel equivalence ratio of 1.9 is reached the $ISNO_x$ emissions become negligible, and those emissions appear to slightly increase above λ = 2.2, which was due to the decreasing value of IMEP, and hence thermal efficiency. The NO_x values for hydrogen are higher than those of the other fuels until λ = 1.9 due to the higher peak temperatures. Otherwise, the NO_x emissions do not appear to be strongly affected by the PC fuel [38].

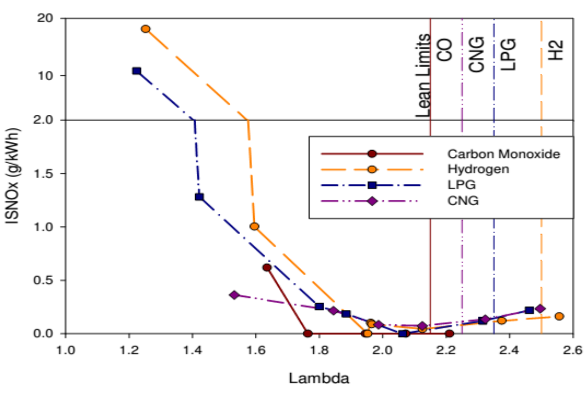


Figure 13. Effect of PC fuel on $ISNO_x$ emissions [38]

Figure 14 shows that ISHC emissions increased substantially as the lean limit was reached with each PC fuel. This was primarily a result of lower combustion temperatures, which led to less HC burn-up during the expansion and exhaust strokes. HC emissions increased further as slow and partial burning began to occur [38].

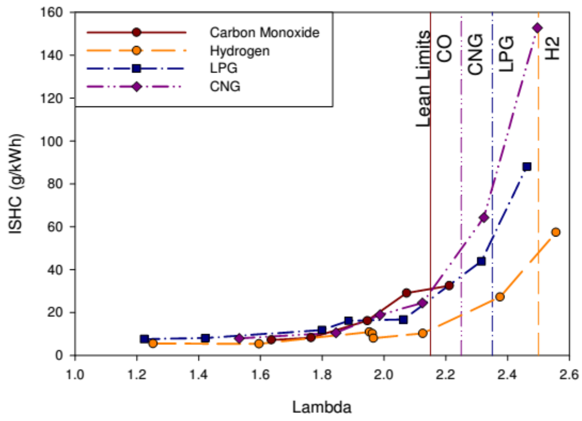


Figure 14. Effect of PC fuel on ISHC emissions [38]

Figure 15 shows the results about Indicated Specific Carbon monoxide (ISCO) emissions. Authors attributed the initially very low carbon monoxide emissions to the excess of oxygen and to the majority of the CO have been oxidized to CO2. An important consideration was made about the employ of CO as the PC fuel. There was no noticeably increase in ISCO emissions in comparison to other fuels, indicating that CO was completely combusted as it propagated out of PC. Further, the choice of PC fuel had a considerable effect on ISCO and ISHC emissions [38].

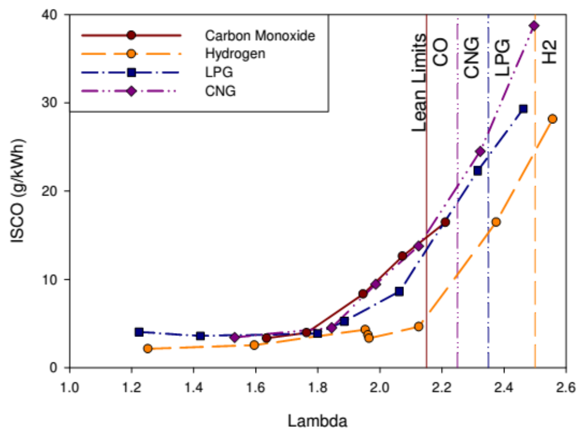


Figure 15. Effect of PC fuel on ISCO emissions [38]

Kettner et al. [15] compared NO emissions between an engine with a homogeneous charge and other with a stratified charge, using direct injection and a prechamber, this last system has been called BPI. As can be seen in Figure 16, there was a significant reduction in fuel consumption in comparison to homogeneous engine and a reduction in ISNO emissions in comparison to spray guided DI engine.

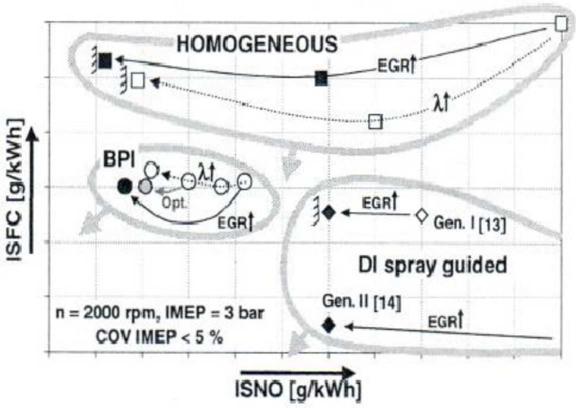


Figure 16. ISNO-ISFC trade-off [15]

Rodrigues Filho [30] developed and analyzed a prototype stratified prechamber ignition system operating with E25, for different levels of mixture stratification, operation loads and engine speeds, studding the effects on specific emissions. It was observed a reduction of 8.21% for CO2, 71.58% for CO and 49.51% for NOx. An average percentage of 32.9% increase in unburned hydrocarbon emissions was detected. The author attributed this increase to the formation of a fluid fuel film on the prechamber.

A great part of recent efforts involving stratified PC engines were made by MAHLE Powertrain [18]. They compared the emissions of a conventional SI engine working with gasoline and a jet ignition combustion system with gasoline in MC and propane in PC, at the fixed speed/load worldwide mapping point of 1500 rpm and 2.62 bar BMEP.

As showed in Figure 17, NO_x emissions in jet ignition system were reduced to almost zero levels (<10 ppm) for exhaust air to fuel equivalence ratio values greater than 1.8, offering the possibility of control NO_x emissions without after treatment systems [18].

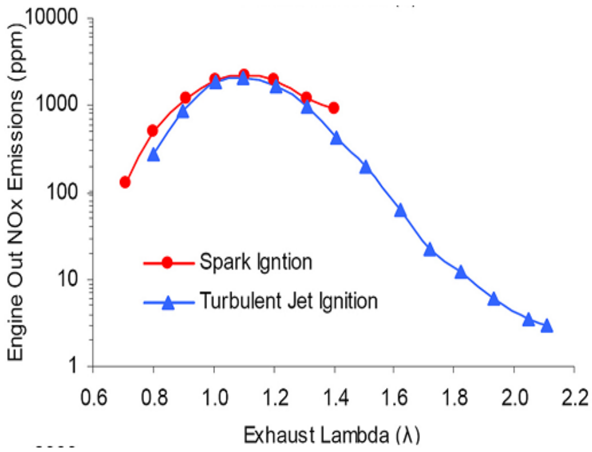


Figure 17. Engine out NOx emissions [18]

Figure 18 and Figure 19 shows CO and HC emission comparisons, respectively, with significant increases as the rich or lean dilution limit is reached for each combustion system. Authors explain that the higher CO levels near the lean dilution limit may be due to partial oxidation of the increasing quench, crevice and oil layer HC, which emerge during the expansion and exhaust strokes [18].

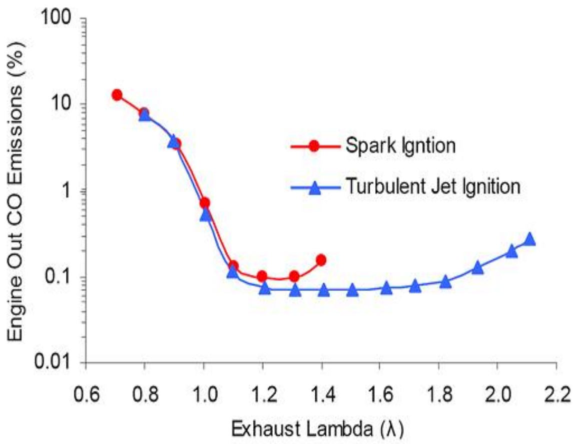


Figure 18. Engine out CO emissions [18]

As showed in Figure 19, the minimum values of HC emissions for both systems in the 1.1-1.2 air to fuel equivalence ratio region, increased for both leaner and richer mixtures. For richer mixtures, the increase in HC emissions was due mainly to incomplete combustion and for lean mixtures, HC emissions rise was due to the lower combustion temperatures which reduced wall temperatures and led to less HC burn-up during the expansion and exhaust strokes [18].

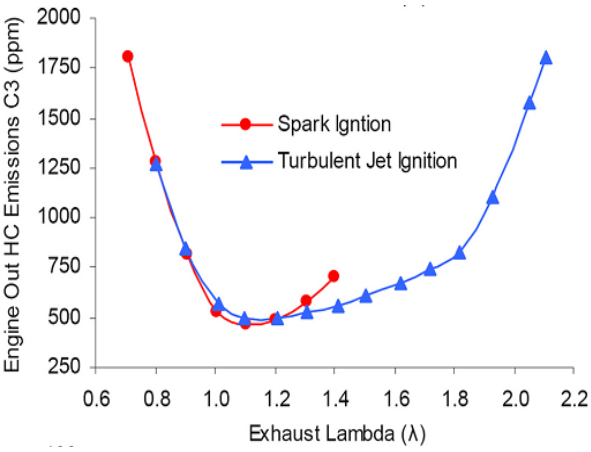


Figure 19. Engine out HC emissions [18]

Based in these results the authors suggested a emission control strategy using a 3-way catalyst operates as an oxidizer for HC and CO emission control, with NO_x levels already controlled with the combustion system [18].

Recent publication about stratified prechamber engines was published by Rodrigues Filho et al. [69]. An average reduction of 8.21% in CO2 specific emissions was achieved by the STI engine in comparison with the baseline engine.

CONCLUSIONS

About the influences of ignition systems with combustion prechamber on the combustion process, was found in this review work that its implementation promotes an increased lean limit, flame propagation speed, and heat release rate. Furthermore, the use of PC systems decreases the spark timing for MBT, the time of start of combustion and tends to decrease the knock during the combustion process.

Regarding to pollutant emissions, the use of combustion prechamber in spark ignition engines has proven to be a promising alternative, especially when used in conjunction with other technologies such as EGR and Supercharging.

The main reductions were found in the nitrogen oxide concentrations, which reached almost zero levels. Three factors that have most effect on formation of NO_x emissions were the peak combustion temperature, the residence time at high temperatures, and the oxygen availability. Reductions in carbon monoxide concentrations were due to sufficient oxygen available to complete the CO reactions.

In most studies about prechamber engines, the HC emissions were higher than the conventional engine. It can be attributed to the increased crevice area caused by inclusion of an additional combustion chamber, but further studies are needed in order to identify the true cause of this increase and to treat or control such emissions.

There is a lack of methodologies to determine prechamber geometry and dimensions. Recently, two models proposed showed good potential to calculate optimum nozzle diameter and prechamber volume to a given engine. Investigations suggest that smaller orifices produce fast combustions and requires less advance in ignition timing. Compromise must be made, as prechamber filling and scavenging are affected negatively as nozzle diameter decreases. Further studies of how geometric factors influence prechamber systems performance are still required.

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