TRABAJO FIN DE GRADO

EVALUATION OF A PCCI COMBUSTION IN EURO VI DIESEL ENGINES

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Llegados a este punto es momento de recordar el camino que nos ha llevado hasta aquí. Quiero agradecer a toda persona que directa o indirectamente ha contribuido a la redacción de este proyecto.

No hubiera sido posible llegar a este punto sin mi padre, mi madre y mi tía que siempre me apoyaron y me dieron la educación y cariño que me han convertido en la persona que soy. Ellos siempre han sido y serán fuente de inspiración y sacrificio.

También quería recordar a todo aquel que en algún momento de este camino me acompañó y compartió suerte conmigo, y en especial a mis amigos con los que he compartido este viaje y me han dado ánimos en los momentos más difíciles.

ABSTRACT

Currently, the world is facing two crises from the point of view environmental: the scarcity of fossil fuels and environmental degradation. According to projections by 2020 the use of vehicles would triple and also increase demand for fossil fuel and therefore emissions contaminants.

Compared with the gasoline engine, diesel engine has the advantage of being a more efficient engine and therefore emits less CO2. However, produce high levels of NOx and particulates. To address these difficulties, we have proposed different actions. One solution, constitute the advanced concepts of combustion Diesel. These concepts allow to reduce levels of NOx and particulates.

However, have the disadvantage of producing high levels of noise combustion, the early use of injections, which cause a more fuel is burned in premixed conditions. However, NOx emissions increased in most operating conditions.

On that way, various technologies have been introduced to reduce emissions from diesel engines, the in-cylinder reduction techniques of PM and NOx like low temperature combustion (LTC which is an important field of research and development of modern diesel engines. Furthermore, increasing prices and question over the availability of diesel fuel derived from crude oil have introduced a growing interest. Hence it is most likely that future diesel engines will be operated on pure biodiesel and/or blends of biodiesel and crude oil-based diesel. Being a significant technology to reduce emissions, LTC deserves a critical analysis of emission characteristics for both diesel and biodiesel.

We can divide LTC in two categories, field of our study. Those in which the combustion phasing is decoupled from the injection timing and the kinetics of the chemical reactions dominate the combustion, are in the first category which is known as HCCI mode. In the second category, combustion phasing is closely coupled to the fuel injection event which is termed as PCCI mode. PCCI combustion seeks to obtain a fully premixed charge before the start-of-combustion, which will result in fully premixed combustion. By injecting very early in the cycle, the air and fuel mix thoroughly such that, upon combustion of the mixture, there are no locally rich regions and little PM is formed.

PCCI combustion differs slightly from pure HCCI combustion in that the direct fuel injection results in minor air-fuel mixture gradients, and thus the mixture is not truly homogeneous. Start of combustion is initiated by auto-ignition of the mixture when a sufficiently high cylinder temperature is attained during the compression stroke. When the auto-ignition occurs, the combustion takes place nearly instantaneously throughout the cylinder.

The application of PCCI combustion suffers from several practical problems. First, because the mixture combusts almost instantaneously, the heat release is very rapid. If the auto-ignition occurs too far Before Top Dead Center (BTDC), this rapid heat release results in very high cylinder pressure rise rates, and high peak cylinder pressures. In addition, this rapid heat

release tends to expose the in-cylinder nitrogen and oxygen to prolonged high temperatures, which can lead to high NOx formation rates. Second, early injection can result in spray impingement on combustion chamber wall surfaces, since the spray penetration is increased at the low gas densities in the chamber early in the compression stroke. However, perhaps the biggest challenge with PCCI combustion is control. This makes emissions control, and overall engine control, very difficult.

In conclusion, the aim of this project is to open a field of knowledge of one of the most advanced new combustion concepts in low-temperature combustion, the PCCI combustion technology, which is already an important implantation via research and development in the field of the automotive industry.

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1. Chapter I: Introduction

1.1. Justification

Since the diesel engine was finally presented in 1898 has been in continuous development and through the years has been gaining greater market penetration, especially in Europe and Asian, because of their good fuel economy. His favoritism was mainly due to improvements in performance, comfort in driving vehicles equipped with these engines, and the ability that offer to meet the stringent environmental regulations to which they are subjected.

However, despite the great advances that have had internal combustion engines since its inception and the many benefits that have provided mankind, these propulsion systems are not without drawbacks, the main problems of high levels of emissions, the use of petroleum fuels and high levels of noise produced. In fact, diesel engines emit more soot and nitrogen oxide that gasoline engines, such that reducing these emissions while maintaining a high level of efficiency presents an urgent and major challenge.

The main emissions from a diesel internal combustion are carbon monoxide (CO), nitrogen oxides (NOx), particulate matters (PM) and unburned hydrocarbons (UHC). Driven by the debate on potential adverse health effects of particulates and the environmental impact of nitric oxides(NOx) emitted by conventional compression ignition engines, legal emission limit values for road transport vehicles are continuing to decrease. Since current diesel combustion technology results in NOx and soot levels much higher than the limits imposed by the current (Euro V) and forthcoming (Euro VI) emissions legislation[5], after treatment systems based generally on Selective Catalytic Reduction and Diesel Particulate Filtration technologies have to be used for the reduction of NOx and soot, respectively. These systems are expensive, require maintenance and introduce a fuel consumption penalty due to the associated higher back pressure and active regeneration cycle. However, decreases in NOx emission often cause increases in PM emission, and vice versa; reduced NOx and reduced PM are difficult to achieve simultaneously through combustion improvement.

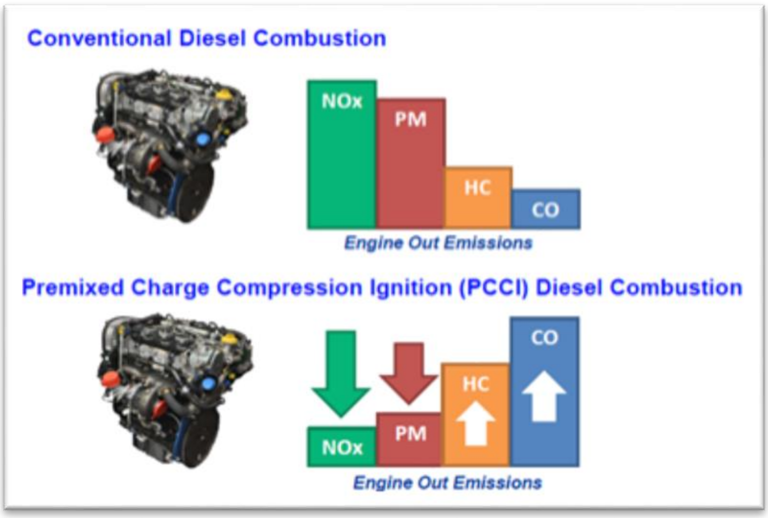

Figure 1. Advanced diesel engine combustion techniques present different emissions challenges.

In order to overcome these problems , advanced combustion strategies have attempted to find an in-cylinder approach to either meet these emission standards fully and thus avoiding the need to use after treatment or at the very least, to lower the performance demands required from after treatment systems and thus reducing their cost and complexity. One of these strategies is PCCI (Premixed Charge Compression Ignition) [1]. This strategy has the potential to reduce NOx and PM emission, ensuring practically the same efficiency than conventional diesel combustion, as it is showed on Figure 1.

In summary, the development of a project which allows to expand the knowledge about this promising combustion technique of diesel engines is justified in an industry in continuous expansion and with so many challenges to face in the next non-dependency fuel-years, with environmental concern growing and the arrival from Europe of more restrictive emissions regulation.

1.2. Object

The object of this project is focused in the study of a Premixed Charge Compression Ignition combustion. First of all, a preliminary overview of the different low temperature combustion (LTC) systems, their actually state and challenges and a wide development of the cited PCCI strategy. On the other hand, there is a practical application of the studied concepts in a FTP diesel engine in collaboration with Polytechnic of Torino within an Erasmus+ program, where it will be studied several instrumentation and a test bench in the study of a PCCI combustion
To specify this object of study, these are the main objects:

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- Characterize the impact of LTC strategies on diesel engine emission.
- Deeply study of a PCCI combustion.
- Define from an objective point of view the main affecting components on emission levels in a diesel engine.
- Analysis of the effects of several parameters on the performance of a premixed charge compression ignition (PCCI) diesel engine and its effect on the engine performance.
- Experimental instrumentation review on the tuning of a FTP industrial diesel engine.

1.3. Development

In this section will be described in detail each of the chapters that have been carried out for the development of this project.

After the overall justification of a study of a PCCI combustion, first of all, it is made a global exposition of the situation of diesel engine, its drawbacks and future challenges. On chapter 2, on the one hand, new concepts of combustion and a deep development of the low temperature combustion concretely the premixed mode, on the other hand, a review of the main emission under PCCI combustion mode, in chapter 3 the most relevant information exposed the revision bibliographic about the main affecting parameters on the development of the PCCI strategy, their consequences on the attainment of the strategy and a graphic review. On the last chapter, it is exposed a review of the main experimental apparatus which are necessary on the tuning of a diesel engine in order to the implementation of a PCCI strategy.
The main themes of search, according to the objectives of the project were:

- Description of new combustion concepts, highlighting their advantages and disadvantages, particularly as it refers to the production of emissions and performance.
- Exploration in depth of the attainment of the PCCI strategy, highlighting its effect on pollutant emissions, fuel consumption and efficiency.
- Present the main governing parameters in the development of the PCCI strategy and achieve optimum operating points.

2. Chapter II: Advanced combustion techniques

2.1. Introduction

Diesel engines have traditionally been used in heavy vehicles and naval applications, but over time, these are being used increasingly in vehicles automotive and unconventional applications such as in the aviation industry. The improvements in performance, fuel consumption, comfort in the conduct and capacity offered to meet the restrictive regulations to which they are subjected, is what makes diesel engines have an advantage with respect to engines gasoline . In recent years, the world and especially the European continent, has seen the Diesel direct injection (DI) engines equipped with systems injection high pressure (common rail) has increased its position in the automotive sector.

Diesel engines consume 30% less fuel and emit 25% less CO2 on average than gasoline engines. Among the major disadvantages of diesel engines are high levels of NOx and particles, the use of fuels derived from petroleum, and high levels of noise emitted.
Development of diesel engines are based on simultaneous reduction of nitrogen oxides (NOx) and particulate matter (PM) emissions. There were been several researches to reduce diesel engines emissions, but the in-cylinder reduction techniques of NOx and PM like low temperature combustion (LTC) will continue to be an important field in research and development of modern diesel engines.
In order to face these difficulties, last years, it has been developed new combustion concepts, such as LTC, and new blends of fuel. For this reason, one of the objectives of this chapter is summarize the existing knowledge of the new combustion systems based on LTC mode.

First of all, we describe the current situation of the diesel engine and the forthcoming restrictive normative that must face in the imminent future from Euro VI.

Then, we made a briefly description of the main new combustion concepts in diesel engines.

2.2. Drawbacks of conventional diesel engines

Diesel engine is the most efficient type of internal combustion engine, offering good fuel economy and low carbon dioxide (CO2) emission. Unfortunately, is also a source of particulate matter (PM) and nitrogen oxides (NOx), both are now subjected to legislative limits because of their adverse effects on the environmental and human health. In the last few years, the levels of NOx and particles have been seriously reduced almost by an order of magnitude. A review of the historic normative it is showed on Figure 2.

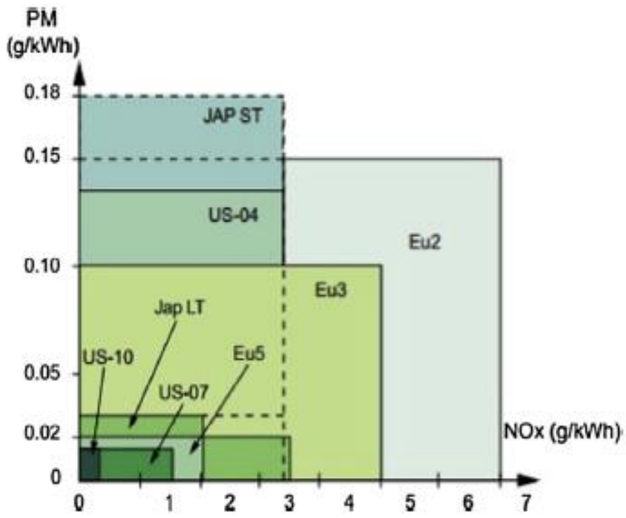

Figure 2. Decreasing limit of NOx and PM.

Unfortunately, the simultaneous reduction of NOx and particles from diesel engines is not an easy task, since both contaminants have opposing developments; this is usually known as the "Dilemma Diesel" or trade-off between PM and NOx. This particular term ‘’trade-off’’ showed that at the edge of the spray flame, fuel lean zones produce abundant NOx, and fuel rich zones inside the spray flame produce abundant soot element of PM [4].

The formation of NOx and particles depends mainly on local temperatures generated in the combustion chamber. NOx formed in regions of high temperature combustion and formation becomes greater in regions close to the stoichiometric. In order to reduce NOx combustion should develop local temperatures close to 2000 - 2200 K. On the other hand, the formation of particles takes place at a temperature between 1000 K and 2800 K, is precisely in the temperatures at which NOx is reduced.

An example in which the trade-off evidenced between the particles and NOx is when advancing the injection is varied. By injecting fuel in a delayed for the purpose of shifting the phase of principal combustion expansion career advancement, a significant reduction in the maximum temperature, which helps to reduce NOx emissions is developed. However, low temperatures make fewer particles are oxidized and low thermal performance is taken. Moreover, by using early injections, effective engine performance is increased and particulate levels drop, but the levels of NOx suffer an increase. Another case of trade-off is when trying to reduce NOx levels by reducing the temperature of combustion, generating an increase in the levels of particles due to the low oxidation [9].

The development of passive actions, is not sufficient, and also requires the use of active actions to control the formation of pollutants into the cylinder, which requires new combustion systems.

To achieve a simultaneous reduction of NOx and PM, new combustion strategies seek to develop a good mixture of air and fuel into the cylinder before ignition occurs, to avoid annoying CO ratio which leads to the formation of PM; and to reduce the temperature of combustion to minimize NOx.

Last years, have emerged solutions focused on combustion systems of low temperature controlled by mixing and premixed combustion.

First one, improves air surroundings to achieve an air-fuel ratio closed to 2 avoiding soot-formation. It is necessary high supercharging pressure, low intake air temperatures and high rates of EGR. With this type it is achieved high NOx reductions but it is penalized indicated efficiency.

Otherwise, with premixed combustion it is sought to increase delay time. This is achieved acting on the injection point and with large quantities of EGR. The ideal model of this type of combustion is homogenous charge, HCCI.

2.3. Low temperature combustion

2.3.1. Introduction

The term low temperature combustion (LTC) covers a number of advanced combustion strategies, including homogeneous charge compression ignition (HCCI) or premixed charge compression ignition (PCCI). LTC combustion can produce very low emissions of NOx and PM, but often results in increased CO and HC. The performance and emissions of engines using LTC strategies depend on the fuel properties, in our case diesel [4].

LTC is the main concept of advanced diesel combustion. It englobes the terms Homogeneous Charge Compression Ignition (HCCI) and Premixed Charge Compression Ignition (PCCI)
In which the combustion phasing is decoupled from the injection timing and the kinetics of the chemical reactions governs the combustion, is known as HCCI mode. In this category, air and fuel are thoroughly premixed in such a way that at the combustion start, the mixture is nearly homogenous and characterized by a lower equivalence ratio.

When combustion phasing is closely coupled to the fuel injection event, this is known as PCCI mode. In this case, pre-mixing occurs between the fuel injection and the combustion start, but there is significantly regions where equivalence ratio is greater at the start of combustion

2.3.2. New combustion systems

In order to simultaneously reduce NOx and particulate levels have been proposed new combustion concepts. These concepts are based on the development of combustion processes with local air-fuel ratio under 2 and a lower temperature combustion to 2200 K. With this purpose, in recent years, have emerged solutions focused on the attainment systems of low combustion temperature controlled by mixing and combustion in premixed phase.

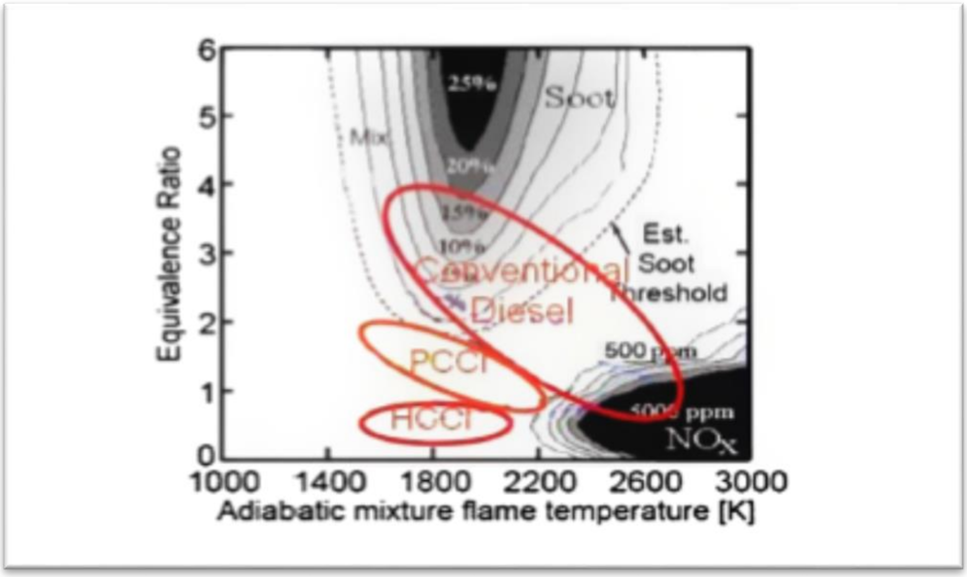

Figure 3. Plot of local equivalence ratio vs. flame temperature with different combustion mechanisms.

In Figure 3 is showed a plot of local equivalence ratio vs flame temperature, with the combustion strategies explained previously. It can be seen how NOx forms in the lean mixture zone where the highest flame temperature above 2200 K, by contrast where temperature is above 1800 K soot formation takes place. Conventional diesel combustion takes place in the overlap of NOx and soot formation whereas LTC techniques avoid this problem reducing simultaneously NOx and soot [9].

The attainment of these strategies is subjected to the application of EGR, injection pressure, injection timing and compression ratio or changes in the fuel composition with blends. It will be discussed later.

2.3.3. Homogenous charge compression ignition

This strategy is a combination between a homogenous charge spark ignition and a charge compression ignition engine, using the premixed charge like SI (spark ignition) engine but with the autoignition of a CI (compression ignition) engine [4].

On this combustion technique, fuel is injected before the combustion event which allows a homogeneous air-fuel mixture. With this, combustion initiates simultaneously at different sites of the combustion chamber. To form a homogeneous mixture, combustion is initiated simultaneously in many parts of the combustion chamber and occurs without propagation of a flame front, as it is showed on Figure 4. This prevents local temperature increase significantly, which reduces the formation of NOx. Furthermore, the absence of a phase of combustion by diffusion and regions rich air-fuel ratio prevents the formation of particles.

It is proved that on diesel fuel, HCCI combustion shows two-stage heat release [8]. First of the two, is low temperature kinetic reactions and the second stage is main heat release engine.
The autoignition on HCCI combustion is controlled by low temperature chemistry and the main heat release is dominated by CO oxidation. As we just signalized, the main advantage of the HCCI combustion over conventional combustion mode is the reduction of NOx and soot in the exhaust.

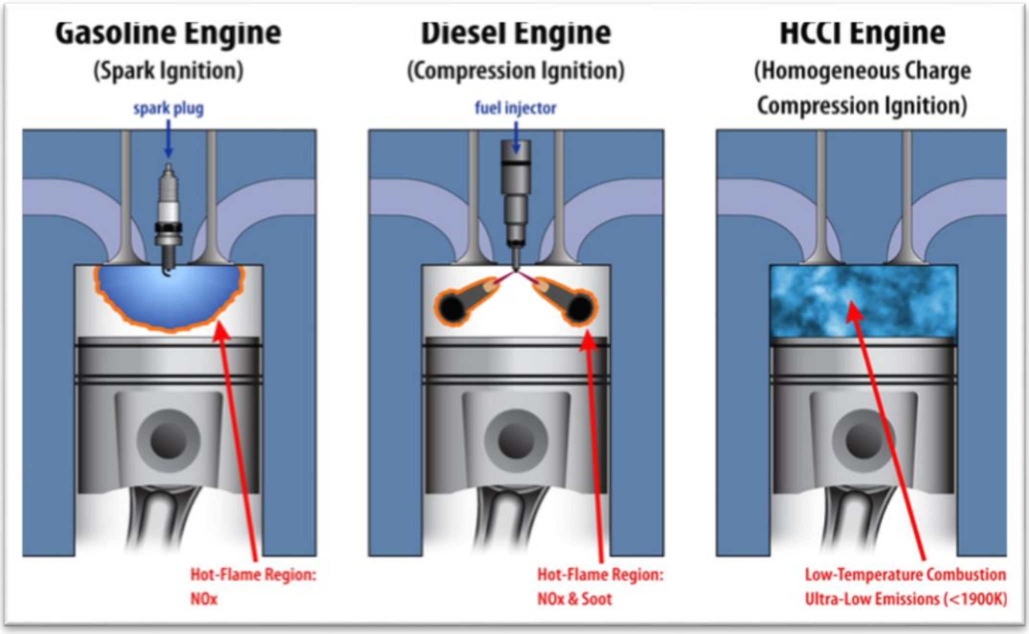

Figure 4. Comparison of HCCI combustion front of flame and conventional gasoline and diesel combustion.

Experimentally the HCCI combustion has been achieved reductions of up to 95% in NOx levels, but despite this great potential [15], this solution has some problems that need still to be resolved: the difficult control of the ignition and combustion of a wide range of operations (speeds and loads). In conventional diesel combustion engines and more particularly in homogeneous or premixed combustion, combustion at the start of an important temporal variation pressure signal occurs, causing the term ''Diesel Knock''. Diesel knock is the clanking, rattling sound emitted from a running diesel engine. This noise is caused by the compression of air in the cylinders and the ignition of the fuel as it is injected into the cylinder. By injecting raw fuel into extremely hot compressed air, the fuel ignites as the piston is still traveling up in the cylinder, causing a detonation and subsequent rattling sound to be heard.

Though the concept gives higher indicated thermal efficiency, inability to control the combustion phasing has led the researchers to try different combustion control strategies for example: port fuel injection, early direct injection , multiple fuel injection, compound combustion technology, narrow angle injection, late direct injection , variable inlet temperature, variable valve timing, internal or external EGR, etc.

Furthermore, use of alternative fuels and fuel blends [15] according to compression ratios and operating conditions have much potential to control the combustion phasing. Actually, fuels with different autoignition points can be blended at varying ratios to control the ignition point at various load–speed regions.

The combination of several strategies helps in achieving better effects on the combustion mechanism.

2.3.4. Premixed charge compression ignition

2.3.4.1. Background

Premixed charge compression ignition evolved from the HCCI combustion mode improving the control over the start of combustion, is an intermediate solution between HCCI and conventional combustion. In-cylinder homogeneity causes rapid combustion by simultaneous ignition throughout the cylinder space and produces great combustion noise in the HCCI mode.

It is also very tough to control the combustion phases in HCCI mode. PCCI process is introduced to solve these problems, although it is not so fully homogeneous. It achieves desired ignition delay through enhanced charge motion, reduced compression ratio, higher injection pressure and extensive use of EGR.

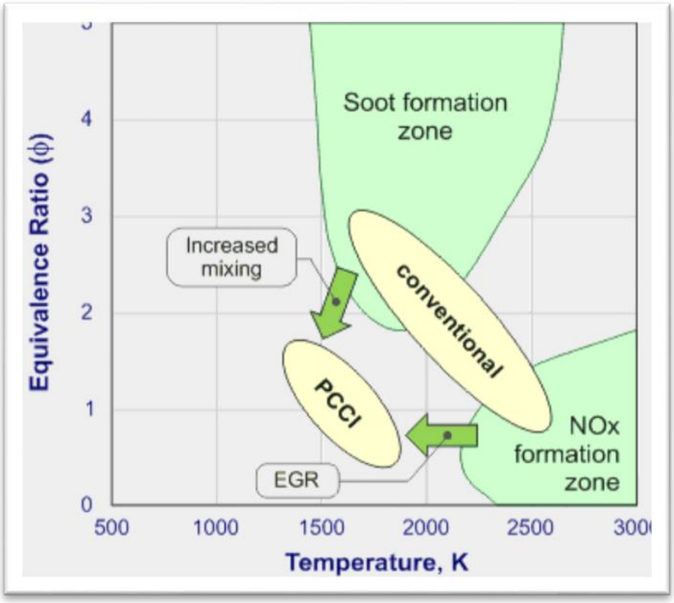

Figure 5. PCCI strategy in a plot with temperature vs. equivalence ratio and NOx-soot formation zone

In the PCCI combustion process, fuel can be injected into the combustion chamber in three ways, they are, advanced direct injection, port fuel injection and late direct injection [9].
Advanced direct injection and port fuel injection suffer from fuel spray impingement on the cylinder walls and incomplete fuel evaporation. Consequently HC and CO emissions increased. However, narrow spray angle injectors and EGR reduce the wall impingement. Late direct injection avoids the fuel-wall impingement and gives a way to switch the combustion style to the conventional at higher loads.

Researchers have tried to increase the high load limits and reduce the emissions of PCCI by applying additives and tuning fuel properties, variable valve timing, multiple injections, and fuel–air mixing enhancement. Early injected fuel stratifies in the cylinder with the air and as the compression stroke reaches near the TDC (top dead center) it creates HCCI like condition.

When the late direct injection occurs, the fuel-rich area of the late injection burns before the fuel lean homogeneous mixture. This variable fuel–air mixture prevents the entire charge from igniting instantaneously which gives a better control over the combustion phase and rate.

Moreover adoption of higher EGR permits longer ignition delay. It permits better premixing of air–fuel, results in less fuel-rich pockets followed by a low temperature combustion, which simultaneously reduces NOx and soot level.

Moreover, in contrast to conventional diesel combustion, the PCCI combustion requires high rates of EGR to lower the temperature of combustion and increase the delay time [7], which helps to reduce NOx levels. Additionally, the EGR is also used as a basic method for controlling ignition timing and rate of release of heat, causing the combustion duration is prolonged and the start of combustion is delayed.

Although the important emission reduction in NOx and PM, those new combustion concepts have to solve some challenges for their implementation:

- Especially in early injection, one of the main problems is to control the combustion phase. Combustion start depends of the fuel properties, mixture homogeneity, rate of EGR, compression rate, intake temperature…
- Operation range: It can be achieved a homogenous combustion through with low air-fuel ratios or large amounts of EGR. But, when we achieve stoichiometric values of air-fuel ratio, combustion stability decreases, the rate of heat release (RHOR) is increased.

- With a premixed strategy is observed high levels of noise especially at a high load where occurs significant increase in the temporal variation of the in-cylinder pressure.
- The low temperature combustion techniques is employed on producing low NOx emission but sometimes hinders the full development of the combustion, increasing UHC and CO emissions.

2.3.4.2. Premise for simultaneous NOx and soot reduction

As it has been talked, NOx and soot formation in diesel engines are complex. Several authors have provided in-depth insight into the physics and chemistry of NO and soot formation inside an engine, which is necessary if one wishes to create a PCI combustion strategy.

It is known that lowering temperature will increase soot. This would be correct if combustion temperatures were above the temperature at which maximum net soot release occurs. Since net soot release is the difference between soot formation and soot oxidation, both having dependencies on temperature, it is possible for soot formation to decrease at a faster rate than soot oxidation [16] as temperature decreases. This is what occurs in PCI combustion.

It will be considered the work of Khan and others, who developed an empirical Arrhenius expression correlating engine exhaust soot to temperature and equivalence ratio. The expression [10], given as Equation 1, is partly a function of equivalence ratio, /, and unburned fuel temperature, T.

$$\frac{\mathrm{d}S}{\mathrm{d}t} = ZF\phi^{n}Pe^{\left(-\frac{E}{RT}\right)}$$

Equation 1

The rate of formation of soot mass concentration, S, is also a function of the rate coefficient, Z, the soot volume fraction, F, and the unburned fuel pressure, P. The constants n and E are given as 3 and 167,472 J/mol, respectively. Several normalized curves calculated from Equation 1 at various equivalence ratios are shown in Figure 6. Notice that as combustion temperatures fall below 1500 K, the soot formation curves all collapse on top of each other, irrespective of the equivalence ratios [17]. While Figure 6 only shows part of the net soot release equation, soot formation, others have reported that soot oxidation does not have a dependency on equivalence ratio. Figure 6 confirms the independence that soot formation has on equivalence ratio at low combustion temperatures. Based on the theoretical insights, it becomes clear that low-temperature combustion can result in simultaneous reductions in NOx and soot.

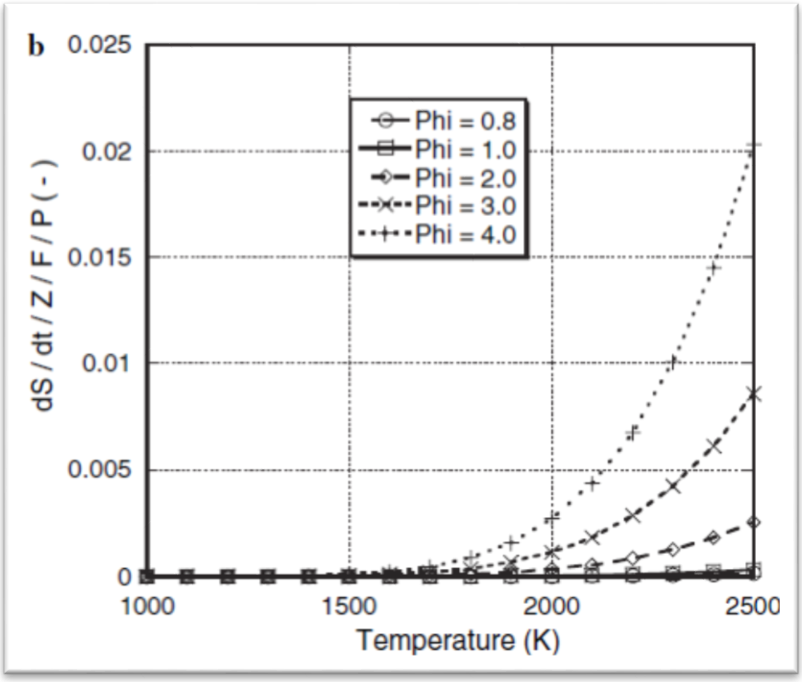

Figure 6. Normalize curve of soot formation rate versus temperature and prediction of soot formation versus temperature and equivalence ratio.

Additionally, for combustion temperatures lower than 1500 K, the potential exists for PCI combustion to operate with rich AF ratios without forming soot.

2.3.4.3. Implementing the PCI strategy

To demonstrate PCI combustion conditions, experiments were conducted using a light-duty diesel engine [10] run through several EGR rate and injection timing conditions, as summarized in Table 1.

Table 1
Operating conditions for Lean PCI combustion development

	Lean PCI
Speed (RPM)	1500
BMEP (bar)	3.5–3.9
Fuel injection demand quantity (mm^3/stroke)	11.0–11.6
Measured fuel flow (g/s)	0.59–0.62
Rail pressure (bar)	1000
Injection timing (° BTDC)	9–18
EGR (%)	41.1–45.7
Measured exhaust AF ratio (–)	19.1–16.2

The impact of injection timing on ignition delay near PCI combustion conditions is shown in Figure 7a. Notice that EGR rate has a stronger influence on ignition delay than does injection timing, thus necessitating its use to create PCI combustion. Figure 7b shows the variation in combustion duration with respect to injection timing and EGR rate. For conventional combustion, ignition delay has an inverse relationship to combustion duration as a longer ignition delay creates more premixed burn which burns faster than the ensuing diffusion burn. For PCI combustion however, the combustion duration depends only on the premixed burn rate rather than a tradeoff between the levels of premixed and diffusion burn.

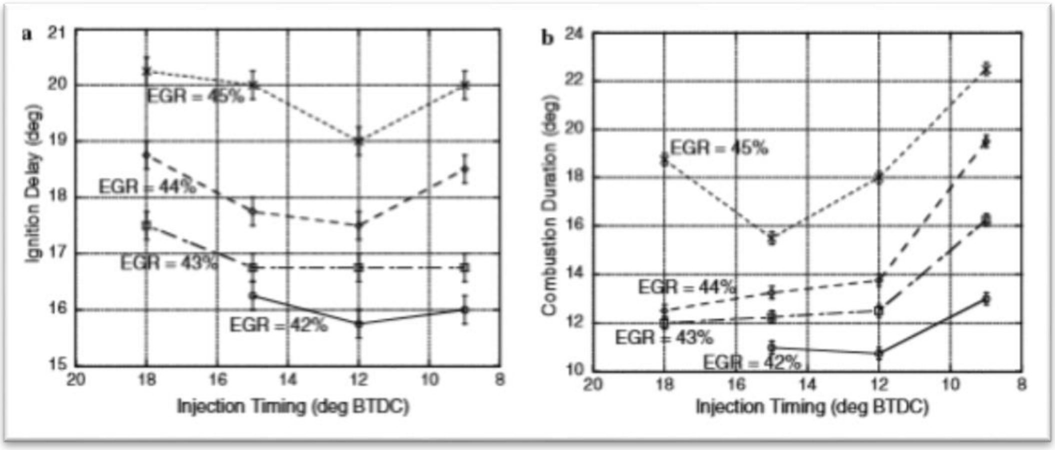

Figure 7. Ignition delay and combustion duration versus injection timing under PCCI combustion mode.

The implication of PCI combustion behavior is that an increase in combustion duration indicates a decrease in burn rate, as shown in Figure 8. An alteration to injection timing results in dramatic decreases in the burn rate for a given EGR rate [10]. Similarly, an increase in EGR rate also results in a dramatic decrease in burn rate regardless of the injection timing. As shown later, the low combustion burn rates result in thermodynamic gas temperatures below 1500 K, which as identified earlier is the temperature at which PCI combustion is realized. Both soot formation and soot oxidation mechanisms are direct functions of combustion temperature and net soot release is maximized when the two mechanisms equally compete with each other [14]. A decrease in temperature along an equivalence ratio constant from the location of maximum net soot release results in a more dramatic reduction of soot formation than soot oxidation, causing net soot release to decrease.

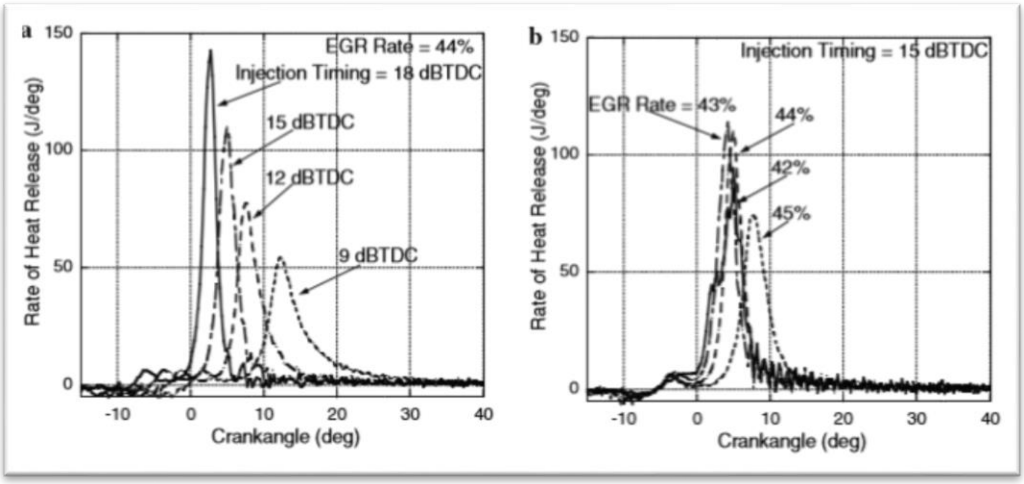

Figure 8. Rate of heat release versus engine crank angle for an EGR rate of 44% and variable injection timings, an injection timing of 15_ BTDC and variable EGR rates.

However, if combustion temperature is higher than the location of maximum net soot release, then a decrease in combustion temperature will result in soot formation decreasing more slowly than soot oxidation, causing net soot release to increase.

At this point, the stage has been set to explore how the behavior of soot and NOx is governed by local equivalence ratios and combustion temperatures. Figure 9a illustrates the peak bulk gas temperature, calculated from measured in-cylinder pressure using the ideal gas law, versus injection timing and EGR rate. Notice for all conditions, the bulk gas temperature is below 1500 K, where rates of soot formation are theoretically low[17]. While bulk gas temperatures do not precisely indicate the more critical local combustion temperatures, they can suggest general trends.

The implication of low-temperature combustion is a defeat of the soot-NOx tradeoff. By considering the influence of local equivalence ratio and temperature on soot and NO formation, one can rationalize the defeat of the soot-NOx tradeoff observed in Figure 9b. When comparing emissions for an EGR rate of 42% to 43%, NOx is lower while soot is higher. The lower NOx suggests a lower combustion temperature. Ignition delay for an EGR rate of 43% is longer than that for 42%; the longer ignition delay suggests more time for mixing, thus lower local equivalence ratios. However as EGR rate increases from 42% to 43%, AF ratio decreases from 18.75 to 18.25, ultimately raising local equivalence ratios. The net result is to push local combustion conditions to higher soot formation. As EGR rate increases to 44%, NOx and soot both decrease. The lower NOx again corresponds to the lower combustion temperature with increased EGR.

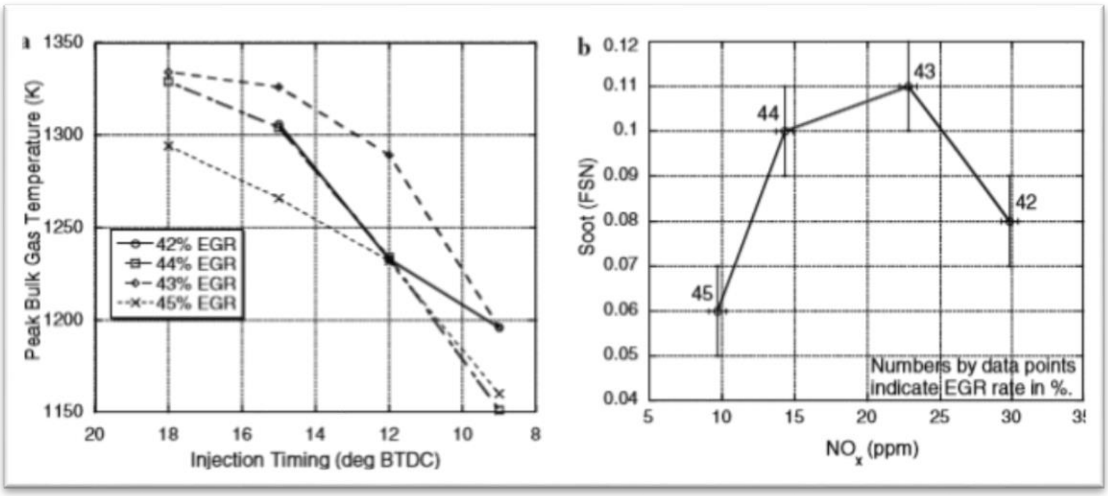

Figure 9. Injection timing vs. peak bulb gas temperature, and soot-NOx emissions for several EGR rates.

The decrease in soot suggests that combustion temperatures are below those where maximum net soot release occurs. As EGR rate continues to increase to 45%, combustion temperature continues to decrease, causing simultaneous reductions in NOx and soot irrespective of what the local equivalence ratios may be. Defeating the soot-NOx tradeoff does not always come easily with an EGR sweep, as close examination of Fig. 4 will reveal for injection timings advanced of 12_ BTDC. In spite of this, emission levels for all the cases shown

Defeating the soot-NOx tradeoff for an all premixed combustion does indicate that local combustion conditions are below the temperature at which maximum net soot release occurs. Based on examination of Figure 11, combustion temperatures that are below this maximum correspond to the PCI regime where both soot and NO formation are generally insensitive to local equivalence ratios.

2.3.4.4. Smokeless rich PCI

Since it were designed, Diesel engines have always been designed to run lean; Rudolf Diesel even stipulated as such in his lecture given to the general meeting of the Society at Cassell in 1897. However, lean conditions traditionally work unfavorably with exhaust after treatment catalyst systems; lean exhaust generally has a low-temperature and high-oxidant and low-reductant concentrations, which likely will prevent the introduction of a ‘‘diesel three-way catalyst.’’ Notwithstanding, several diesel after treatment systems including the lean NOx trap (LNT) are receiving significant research attention [6]. The process is showed in Figure 10. The LNT works by oxidizing and storing NO until its trapping efficiency decreases to a point at which regeneration is required. Regeneration of the trap requires exposing the device to exhaust constituents that favorably desorb the stored molecules and reduce them to molecular nitrogen.

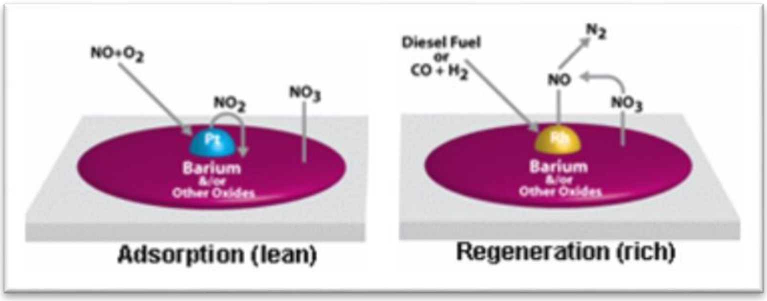

Figure 10. Adsorption and regeneration process under Lean NOx Trap system.

One strategy to regenerate a LNT is to create rich exhaust conditions where oxygen concentrations are low [6] and carbon monoxide concentrations. From the discussion describing lean PCI combustion, it was shown that soot has very little dependence on local equivalence ratios once combustion temperatures are lowered below the temperature at which maximum net soot releases occurs. A significant increase in local equivalence ratio results in a small increase in soot, while a small decrease in local combustion temperature results in a large decrease in soot.
Such behavior, i.e. insensitivity of soot emission to equivalence ratio, is demonstrated in Figure 11, which shows exhaust soot concentration versus AF ratio.

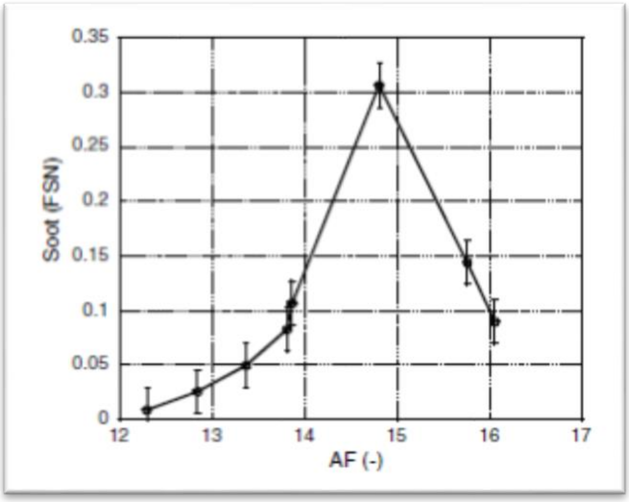

Figure 11. Soot concentration vs. air-fuel ratio

The AF ratio sweep was experimentally performed by increasing EGR and fuel rates and adjusting injection timing to ensure decreasing combustion temperatures with decreasing AF ratio. Notice that initially as AF ratio decreases, soot increases. This would correspond to a temperature decrease while temperatures are still near the point of maximum net soot release [10]. A further decrease in global AF ratio via increased EGR and fuelling rates decreases local combustion temperatures and places local conditions within the PCI combustion regime, causing soot to decrease as well. Naturally, a decreasing global AF ratio [16]—particularly as combustion falls below the stoichiometric AF ratio (approximately 14.7)—indicates an increasing local equivalence ratio. In spite of this, soot continues to fall, which confirms not only that combustion temperatures are falling, but also that within the PCI combustion regime net soot release has very little sensitivity to local equivalence ratio. As a result, an aggressive rich PCI combustion strategy yields exhaust concentrations of soot = 0.03 FSN, NOx = 3 ppm, and CO greater than 5%.

The tremendous success PCI combustion has in simultaneously reducing NOx and soot comes with drawbacks such as increased levels of HC, CO, and fuel consumption [8] . The issues of increased emissions of hydrocarbons and carbon monoxide with lean PCI combustion are to a certain extent resolved with the use of a diesel oxidation catalyst. However, catalytic activity with rich PCI combustion dies nearly instantly, requiring further investigation into appropriate catalytic materials for PCI combustion. The issue of increased fuel consumption, reported to around 5% over lean conventional combustion at similar engine load conditions, possibly could be combated with improved combustion chamber design and EGR flow methods.

3. Chapter III: emission analysis under PCCI combustion

3.1. NOx emission analysis

NO (nitric oxide) and NO2 (nitrogen dioxide) are generally grouped under the NOx emission. But among the nitrogen oxides, NO is the predominant oxide produced inside the engine cylinder [8]. Oxidation of the atmospheric nitrogen (molecular) is the main source of NO and this is called thermal NOx. Strong triple bond of nitrogen molecules breaks down by high combustion temperature and disassociated atomic state of nitrogen takes part in series of reactions with oxygen which results in thermal NOx. This mechanism is also known as Zeldovich mechanism. NO_x formed through high temperature oxidation of the diatomic nitrogen found in combustion air. The formation rate is primarily a function of temperature and the residence time of nitrogen at that temperature. At high temperatures, usually above 1600 °C (2900 °F), molecular nitrogen (N_2) and oxygen (O_2) in the combustion air disassociate into their atomic states and participate in a series of reactions. The three principal reactions (the extended Zeldovich mechanism)[6] producing thermal NO_x are:

$N_2 + O \rightarrow NO + N$

$N + O_2 \rightarrow NO + O$

$N + OH \rightarrow NO + H$

All three reactions are reversible. Zeldovich was the first to suggest the importance of the first two reactions. The last reaction of atomic nitrogen with the hydroxyl radical, •HO, was added by Lavoie, Heywood and Keck to the mechanism and makes a significant contribution to the formation of thermal NO_x.

However if the nitrogen content of the fuel is higher, then the nitrogen containing compounds get oxidized and become a potential source of NOx, which is also called fuel NOx. Formation of fuel NOx is quite complex because numerous intermediate species are there. Several hundred reversible reactions take place and still the true rate constant values are unknown. Another process of NOx formation is prompt mechanism. By this mechanism, the amount of NOx is quite lower than fuel and thermal NOx. Mainly, free radicals formed in the flame front of the hydrocarbon flame generate this rapid production of NOx. Formation of NOx generally depends on oxygen concentration, in-cylinder temperature, air surplus coefficient and residence time.

NOx forms both in the flame front as well as in the post flame gases. In engines, flame reaction zone remains extremely thin, as the combustion pressure is very high. In addition, residence time is short within this zone. On the other hand, the burned gases, which are produced early in the combustion process, are compressed to a higher temperature than they reached just after the combustion. That is why NO formation on the post flame gases usually dominates over the flame-front-produced NO.

3.2. NOx emission under LTC mode for diesel PCCI

In LTC modes, the combustion temperature is reduced by premixed or leaner mixture with moderated use of EGR, consequently NOx emission reduces. EGR hinders the O2 flow rate into the engine and results in reduced local flame temperature, which helps to reduce thermal NOx.

Again EGR extends the ignition delay which indicates delayed start of combustion. It results in lower pressure and temperature rise during the combustion. The effect of late injection strategy on NOx emission is just like as ignition delay.

Many researchers have attained LTC modes like PCCI, HCCI or RCCI, optimizing various parameters such as fuel reactivity (Cetane number, CN of fuel), injection timing and pressure, dilution of charge by EGR, controlling the operating load [9]. Effects of these parameters for attaining the LTC modes are discussed below concerning the literature review PCI (premixed compression ignition) which showed reduced NOx. As the EGR rate increased and injection time retarded, emission of NOx decreased. It has been reported application of immense EGR gave them about 94% decrement of NOx while they were trying to get a universal determination of LTC mode attainment criteria.

Retardation of injection timing with EGR gave them even better results of emission. Kiplimo reported lower NOx even in early injection (20_ BTDC) during PCCI combustion strategy. With EGR, they got about 75% decrement of NOx emission and negligible difference of IMEP. They worked out an optimum spray-targeting zone where they got simultaneous reduction of CO, HC and soot but could not manage to reduce the NOx without EGR. Without EGR, lower injection-pressure (80 MPa) with late injection (2–15_ BTDC) gave reduced amount of NOx while for higher injection pressure (140 MPa) advanced injection (20–40_ BTDC) resulted in reduced NOx[18]. This lower NOx for higher injection pressure at advanced injection timing can be attributed to the achievement of PCCI regime. PCCI regime ensured lower in-cylinder temperature and longer premixing time, which resulted in lower NOx. On the contrary, Alriksson and Denbratt reported higher NOx when they tried advanced injection timing to keep the BSFC (brake specific fuel consumption) low.

However, Kook investigated the effect of dilution and injection timing very precisely on low temperature combustion emission. In a fixed SOI (start of injection) they observed that NOx emission decreased as the dilution increased. NOx emission was actually correlated with the adiabatic flame temperature. Again, in a fixed level of dilution, retardation of injection timing gave lower NOx. They commented that earlier injection timing assisted by high level of dilution, generated greater adiabatic flame temperature than the late injection even with less amount of dilution. From NOx formation point of view, we can infer that, as late injection LTC mode generates lower temperature than early injection LTC mode, the former one is better in this regard.

Laguitton observed the effect of compression ratio on NOx emission under a wide range of PCCI like combustion styles. Lowering the compression ratio gave them lowered NOx. This effect was more pronounced at higher loads. They also observed that at higher loads, combustion style proceeded to premixed-charge from the combination of premixed and diffusion type combustion as the injection timing was swept from very early to retarded. Hence, NOx emission was converged as the diffusion combustion suppressed. They also concluded that below a certain combustion temperature, as in fully premixed charge combustion, NOx emission was dominated by the air fuel ratio and the local oxygen concentration rather than in-cylinder pressure and temperature.

Summary:

- Lower NOx depends on higher ignition delay and lower combustion rate which result in lower in-cylinder temperature and pressure rise rate.
- NOx emission is controlled by air-fuel ratio and local oxygen concentration more than in-cylinder temperature.
- In order to improve NOx emission is better late injection premixed on LTC mode than early injection premixed.

3.3. PM emission analysis

Perhaps the most characteristic of diesel emissions, is responsible for the black smoke traditionally associated with diesel powered vehicles. Particulate matter is mainly consisted of elemental carbon, adsorbed hydrocarbons and inorganic compounds. To be precise, particulate matter is a highly complex mixture of fine particles and liquid droplets including soot, ash, hydrocarbon soluble organic fraction and water. A schematic representation of the main types of particles and its association is showed in Figure 12 [6].

PM varies in size, shape, number, surface area, solubility, chemical composition and origin. Size distribution of the PM has three modes consisting coarse particles, fine particles, and ultrafine particles. These particles exist in various shape and densities in the air thus aerodynamic diameter is used to define the size of the particle. Soot particle size can be as small as 1–2 nm at initial state. Collision of rings causes coagulation and clustering together similar to a chain, making the soot grow to agglomerates with size ranging 100–1000 nm. Soot content in the exhaust gas is indicated by the smoke opacity; hence, this parameter can be correlated with fuels tendency to form PM during combustion. Incomplete combustion of fuel hydrocarbons produces most of the particulate matter with little contribution of lubricating oil. It sources from the rich combustion zones where the equivalence ratio is higher than 1 [12]. This is the reason for the highest particulate concentrations in the core region of each fuel spray in direct injection diesel engines.

Generally, soot formation takes place at higher than 1800 K [6] temperature in diesel combustion environment. Net soot release is commonly defined as the difference between formation and oxidation of soot [19]. Formation and oxidation of soot are strongly coupled with the combustion temperature just like the NOx formation. So, conventionally soot and NOx formation have got an inverse relation known as soot–NOx tradeoff, as we talked previously.

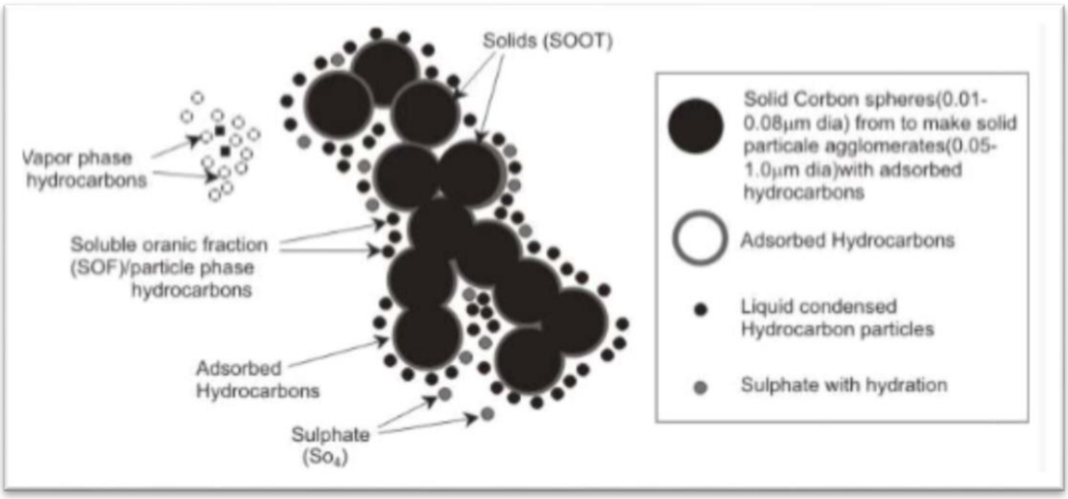

Figure 12. Particulate matter formation.

3.4. PM emission under LTC mode for diesel-PCCI

In LTC, simultaneous reduction of soot and NOx are achieved by reducing the combustion temperature lower than soot formation level. Once LTC is attained, soot formation loses its strong dependence even on equivalence ratio. In LTC mode soot formation occurs primarily downstream in the head of the jet. This is in contrast to the upstream soot-producing core, in conventional diesel jets [18]. This shift is due to the charge dilution employed in LTC and mixing between the end of injection and second-stage ignition.

Many researchers have reported successful reduction of soot when they attained LTC. Actually more complex relationship exists in soot reduction than that of NOx. Soot oxidation process is more sensitive to temperature than the soot formation process. Therefore, when application of EGR reduces the combustion temperature, oxidation rate falls dramatically and emission of soot increases. Further reduction of temperature by higher level of EGR or retarding the SOI, below the soot formation level gives very low amount of soot. For example, PCI combustion condition attained by late injection have been reported to produce very low amount of PM at about 48% EGR. Alriksson and Denbratt also reported that they needed almost 50% EGR even on 25% loading when the peak combustion temperature was comparatively low. They observed an increment of soot up to 50% EGR and then it suddenly decreased, which supports the soot oxidation and formation relationship discussed previously.

It is reported that PCI or PCCI achieved by advanced injection and extensive use of EGR also produced lower soot emission. Jacobs and Assanis experimented with an air–fuel ratio which is lower than stoichiometric (14.7) and the injection timing was 25_ BTDC, confirming the low peak combustion temperature in the combustion. They reported tremendous low soot (0.03 FSN) emission. Though, lower global air-fuel ratio indicated rich mixture, soot decreased. Therefore, within the PCI combustion regime net soot release has very little sensitivity to local equivalence ratio.

But opposite results are also been reported by Benajes and others. They studied the PM emission with advanced injection timing within the premixed LTC regime. Advancing the injection timing empowered them to control ignition delay and local equivalence ratio of the injected fuel. Higher value of the ignition delay and maximum local equivalence ratio for advanced injection timing were supposed to give less PM formation. Surprisingly they observed increased PM with injection advancement. Most drastic increment of PM (185%) happened when they advanced the injection from _30_ ATDC to _33_ ATDC [12]. Though here the end of injection occurred before the start of combustion as in premixed LTC mode, PM formed because of fuel deposition on the surface of piston bowl. As the injection was advanced, the fuel spray trajectory crossed the piston bowl surface at a higher point. It caused increased liquid fuel spray and combustion chamber surface interaction, hence more deposition. This was the reason behind increased PM formation, though it was a premixed LTC mode.
Regarding soot/PM emission for LTC conditions, can be noted the following summary:

- Soot emission is independent from local equivalence ratio, once soot formation temperature is overcome.

- With the increase of EGR rate, PM and soot is increased with the advancement or retardation of injection timing to develop LTC strategy, until the combustion temperature is above the formation temperature of soot.

- On one hand, through very late injection in premixed mode could increase soot emission due to fuel rich zones.

- On the other hand, it can be created from premixed load HCCI mode through extremely advanced injection fuel deposition but may increase PM and soot emission.

3.5. Unburned hydrocarbons (UHC) and CO emissions analysis.

The formation of HC emission in conventional diesel combustion is due to cooling at low temperature oxidation reactions, excess richness or poverty defect in the mix, incomplete evaporation of fuel and clamping of overspray [6].

Among these methods of formation, include: wealth or poverty of the mixture. For rich mixture, there is a good mix with enough air and do not get enough time to oxidation, which increases the amount of Co and UHC. To lean mixture, flame cannot spread on combustion and partial oxidation causes the formation of CO and UHC.

In LTC mode, deeper fuel mixture before combustion, to reduce local air-fuel rich region. With this temperature combustion is reduced thereby reducing NOx and soot but produce products of incomplete combustion [18]. A significant amount of fuel is stored in the crevices at the time of compression and exhaust combustion. Since, the burned gas temperature is not that much high in original processes to consume the fuel, When They are back into the cylinder During expansion, CO and UHC formation Becomes unavoidable.

Han and others have experimented with increasing EGR. The ignition delay during mixture produced unburned fuel by the poverty of the mixture. Before TDC to inject the lower peak temperature oxidation process difficult. Indeed, with each crank angle, the cylinder content is getting cooler for expansion, whereby the cooling mechanism is greater, responsible for HC and CO emissions [19].

Other experiments by Opat and others discovered others responsible for emissions of HC and CO. Among them, the fuel film on the piston bowl generated by the spray shock on premixed combustion. By injecting the spray path meets the surface of the piston cup at a higher point, so stools occur. This can be avoided with changes in injection angle
The dual injection strategy of fuel to achieve reduced levels of HC and CO consists of two injections at 50 early and late BTDC to 20 BTDC. Again increased load becomes greater thus improving the oxidation and complete fuel combustion peak temperature.

By advancing or retarding the combustion timing, UHC emissions occur along the squish area produced by the wealth or poverty of the mixture.

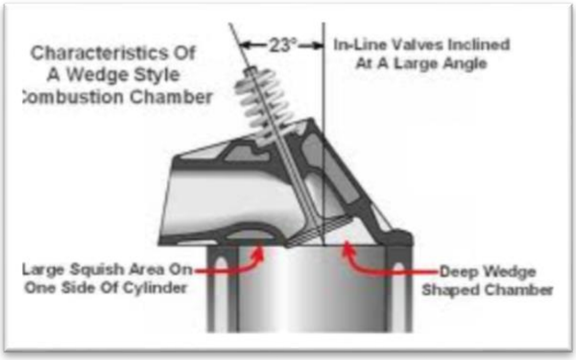

Figure 13. The squish effect on the piston chamber

"Squish" is an effect in internal combustion engines which creates sudden turbulence of the fuel/air mixture as the piston approaches top dead center (TDC).In an engine designed to use the squish effect, at top dead centre (TDC) the piston crown comes very close, (typically less than 1mm), to the cylinder head. The gases are suddenly "squished" out within the combustion chamber, creating turbulence which promotes thorough fuel/air mixing, which is beneficial for efficient combustion. This beneficial effect is showed in Figure 13.

A low load, UHC is produced on the centerline due to the poverty of the main mix. Concerning CO, is observed in general distributed in the central line in the displacement volume of the mixture. Using advanced injection timing can increase the displaced volume of CO mixture.

By any increase in the dilution slows oxidation mixture around the cylinder. Musculus and others tested the UHC and CO increase when the end of injection (EOI) was shorter than the ignition delay. Analyses revealed within the fuel vapor as the maximum concentration of HC came near the nozzle, due to incomplete combustion.

We can summarize the main striking points concerning UHC and CO emissions:

- The main reason for HC and CO formation are the reduction in combustion temperature and oxygen concentration.
- On one hand, over-mixing the charge is produced when is employed the ignition delay that carries to increase HC and CO emissions.
- On the other hand, the advanced injection on PCCI causes spray impingement on the piston bowl which also carries to increase HC and CO emissions.
- Researches have proved that HC emission depends on ignition delay while CO emission depends on equivalence ratio.

4. Chapter IV: Development of the PCCI strategy: effects of performance parameters.

4.1. Introduction

On this chapter is going to be discussed the effects of several parameters on the performance of a premixed charge compression ignition (PCCI) diesel engine in order to characterize the effect of these parameters on the combustion performance and exhaust emissions. Specially, the effects of injection parameters, spray impingement and exhaust gas recirculation (EGR).
Due to stricter emission regulations, diesel engines are more needed for powering light duty and heavy commercial vehicles, but conventional diesel combustion suffers high increases of particulate matter (PM) and nitrogen oxide (NOx) emissions. For that reason, new combustion strategies were developed widely known on this project previously.

As we talked, homogenous charge compression ignition (HCCI) eliminates locally rich air-fuel mixtures and reduces the combustion temperature achieving promising reductions on PM and NOx emissions. But HCCI faces many challenges like a lack of combustion phase control, high pressure rates, and limitations in homogenous mixtures.

Several combustion concepts were developed, one of them is premixed charge compression ignition (PCCI). In the PCCI combustion fuel is introduced into the combustion chamber through port fuel injection, early direct injection, or late direct injection. First two cause high levels of HC and CO due to fuel vapourisation. Late direct injection avoids that problem and provides good phasing control.

With the PCCI combustion strategy ignition delay is achieved through a lower compression ratio, enhanced charge motion, higher injection pressures, and relatively large amounts of cooled external EGR. HCCI, is not fully homogeneous, but it uses injection timing and EGR to increase the controllability of combustion phasing and the rate of combustion. In order to simultaneously achieve low soot and NOx under moderately early injection timings, there is a need to optimize the injection pressure, EGR rate, compression ratio, and injection timing.
This chapter develops the effects of spray impingement, injection pressure, injection timing, and EGR on the formation of emissions in a PCCI strategy.

4.2. Experimental setup review

On this part, it is studied and reviewed the investigation on an engine 4-strokes, single-cylinder, water cooled [12].
The simulated EGR based on N_2 gas dilution was calculated in the formula:

$$\text{EGR rate} = \frac{N_2}{\text{Air} + N_2}$$

They used a four-stroke, single-cylinder, direct-injected, supercharged diesel engine with the following specifications and operating conditions, showed in Table 2:

Engine specification and operating conditions.	
Engine type	4-stroke, single-cylinder, water cooled
Bore × Stroke	96 × 108 mm
Swept volume	781.7 cm^3
Compression ratio	13
Combustion system	PCCI, direct injection
Combustion chamber	Derby hat
Engine speed	1000 rpm
Intake pressure	101 kPa
Injection system	Common rail system
Fuel injection pressure	80 MPa, 140 MPa
Fuel injection quantity mf	12.2 mg/cycle
Straight-hole injector	Φ 0.1 mm × 4holes
Nozzle (VCO type)	
Included cone angle	140°
Injection timing sweep	2–40°BTDC
EGR rate	0–40%
Intake temperature	40 °C
Coolant temperature	80 °C
Lube oil temperature	80 °C

Table 2. Engine specification and operating conditions.

Also a schematic diagram of the installation is showed in Figure 14 , which is equipped with:

- Common rail injection system which develops an injection pressure of 180 MPa
- Photo interrupters detecting top dead centre (TDC) signals and every half-degree crank angle (CA)
- A controller controlling the injection timing and its duration.
- A valve-covered orifice (VCO) injector with four symmetrically holes placed in the nozzle.
- A piezoelectric pressure transducer to measure in-cylinder pressure
- A NOx-CO analyzer to capture exhaust emissions, HC analyzer and smoke meter.

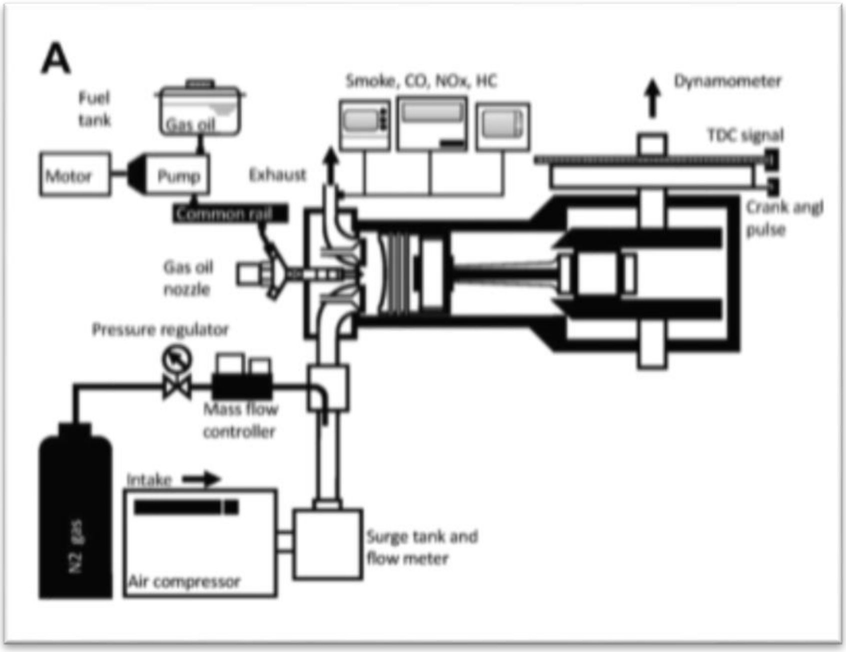

Figure 14. Engine experimental design

4.1. Effect of injection pressure and injection timing

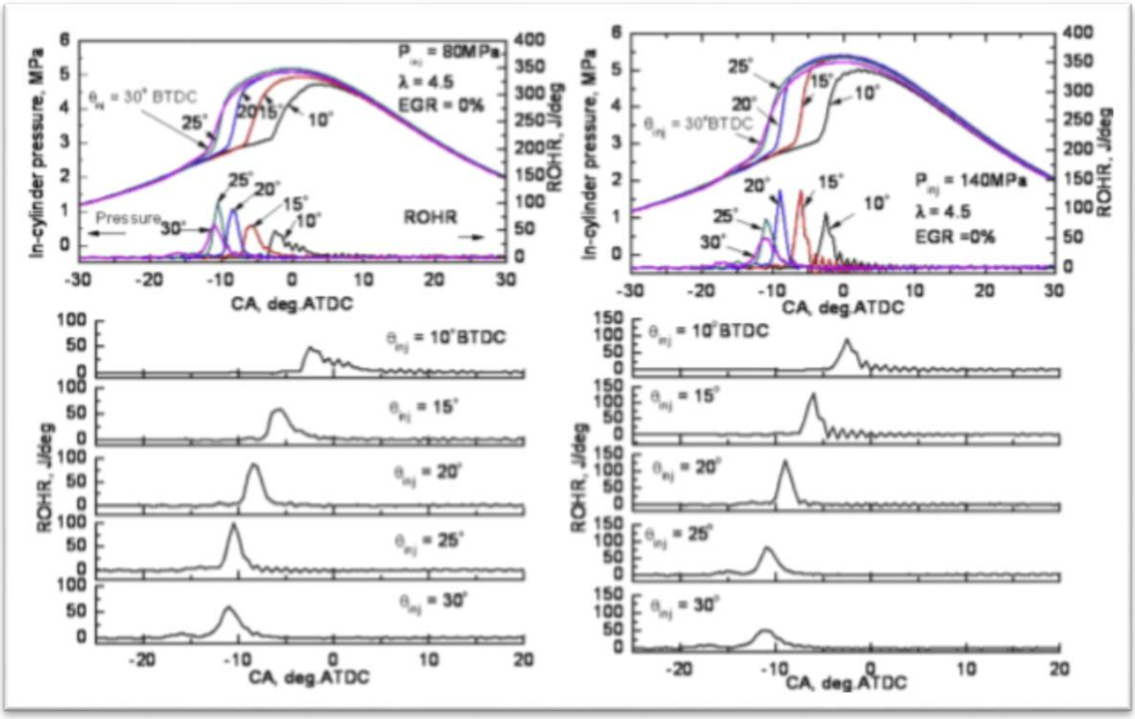

Figure 15. In-cylinder pressure and ROHR.

In Figure 15, it is showed the in-cylinder pressure represented with the injection timing in the crank angle (CA) degree. Below is showed the rate of heat release (ROHR).
If it is compared both injection pressure, it is showed how higher injection pressure carries to a higher in-cylinder pressure. In fact, to P_{inj}=140 MPa it is seen a peak in-cylinder pressure about 5'5 MPa.

In so far as the ROHR, for the earlier injection timing is showed two-stage heat release, the first one is associated to the low temperature reactions, the second one is related with the typical heat release of reactions at high temperature. But for the most retarded injection, it is showed only one heat release peak that means premixed combustion dominate the combustion [7]. Also, the highest peak was associated to the P_{inj}=140 MPa. Moreover, for the higher injection pressure P_{inj}=140 MPa, also was led to shorter combustion time and more rapid.

Summarizing, high injection pressure led to shorter, higher heat release peak, while moderately advanced injection timing led to the same higher heat release peak.

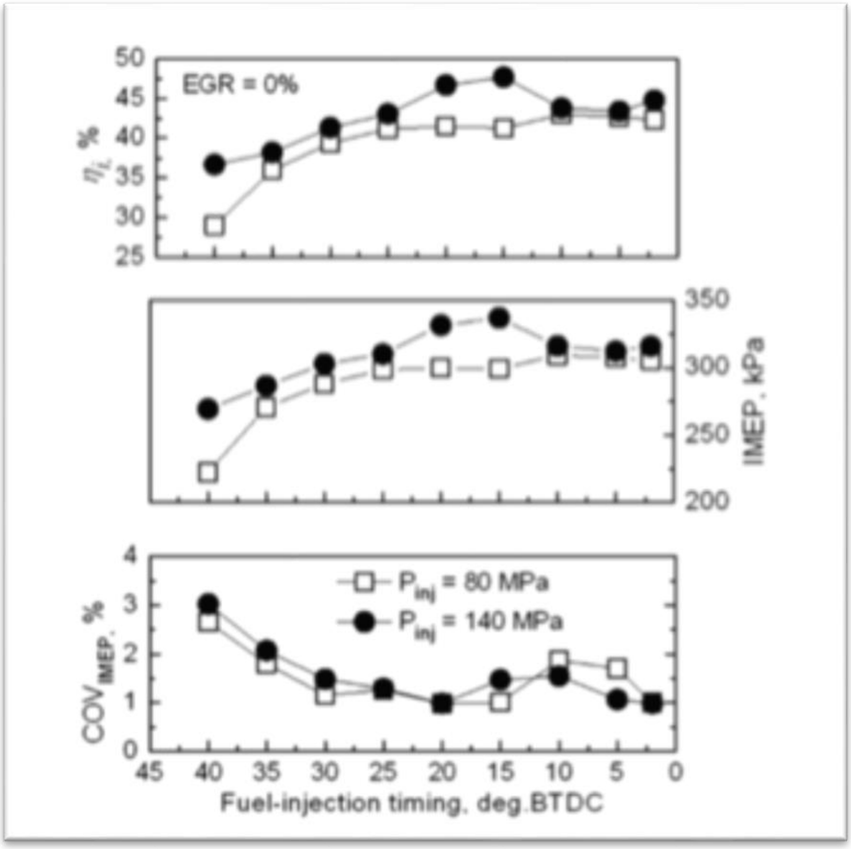

Figure 16. Effect of injection pressure on indicated thermal efficiency, IMEP, and COV of the IMEP

This time in Figure 16, it is being discussed the effect of injection pressure compared with a new introduced parameter: indicated thermal efficiency,

$$\eta_i = \frac{\text{Indicated work per cycle}}{\text{Input energy}} = \frac{\text{IMEP}x V_d}{m_f Q_{LHV}}$$

where is defined as the indicated mean effective pressure per displacement volume divided by the mass of fuel employed per the low heating value of the fuel.

On white is represented the lower injection pressure an on black the higher. It is showed how the indicated efficiency is stable with a phasing near the TDC. Specifying in values between 20º and 15º BTDC both indicated mean effective pressure and indicated thermal efficiency were achieved, due to avoid fuel impingement on the piston bowl.

Broadly, IMEP and indicated thermal efficiency were superiors with the higher injection pressure P_{inj}=140 MPa.

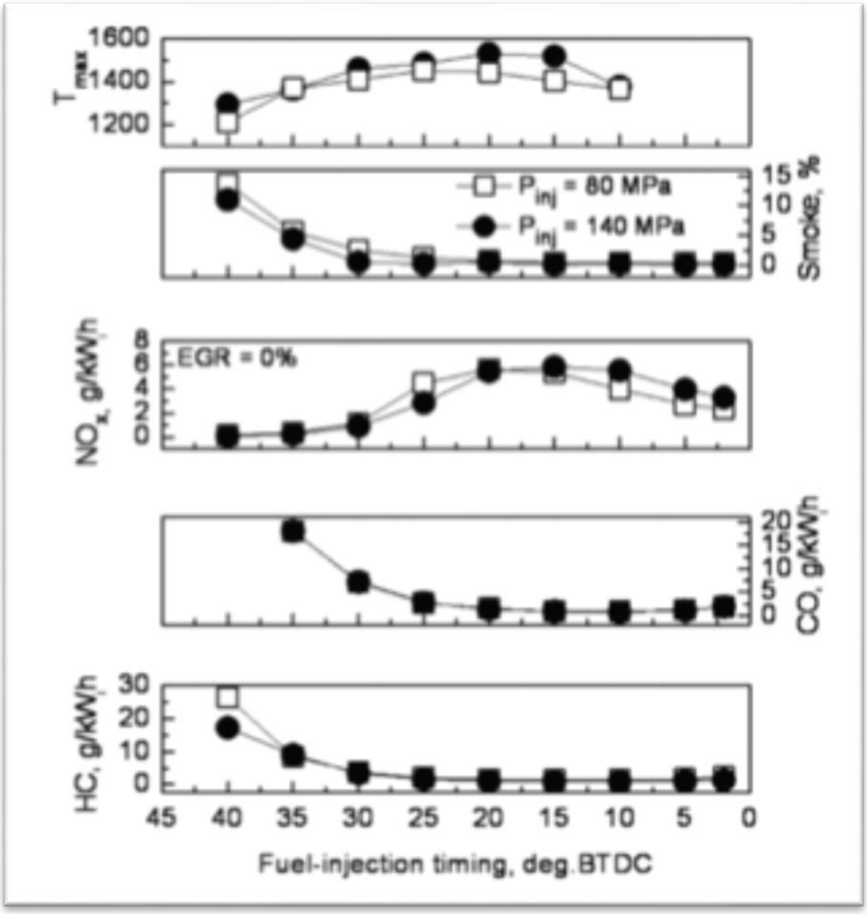

Figure 17. Effect of injection pressure on maximum temperature and specific emissions

In Figure 17, it is showed the relationship between the injection pressure and the exhaust emissions. It is showed how for the higher injection pressure led to the maximum in-cylinder temperature, except for the step 35º BTDC. Whereas, for all cases smoke emission were also lower for the higher injection pressure, due to a better atomization by the premixing combustion. It is also possible to check the NOx-soot trade off in the 40º-30º range. Moreover, from 30º to earlier injections resulted in higher smoke emissions, as well as HC emission.

Whereas, NOx emission decreased in that point. This is due to the PCCI combustion, the in-cylinder temperature gets reduce and it carries to lower NOx emission. As we already have commented previously in the PCCI combustion the in-cylinder temperature is lower and the time while premixing the charge is longer which led to a soft combustion.

For both injection pressures the CO emissions are equals but get increased with early injections. It is showed how with early injection timing, from 25º BTDC, CO emission get increased drastically due to the fuel impingement on the piston bowl surface.
Except for very early injection timing where P_{inj}=80 MPa. Were lower, HC emission are similar for both injection pressures.

4.2. Effect of soot formation and oxidation

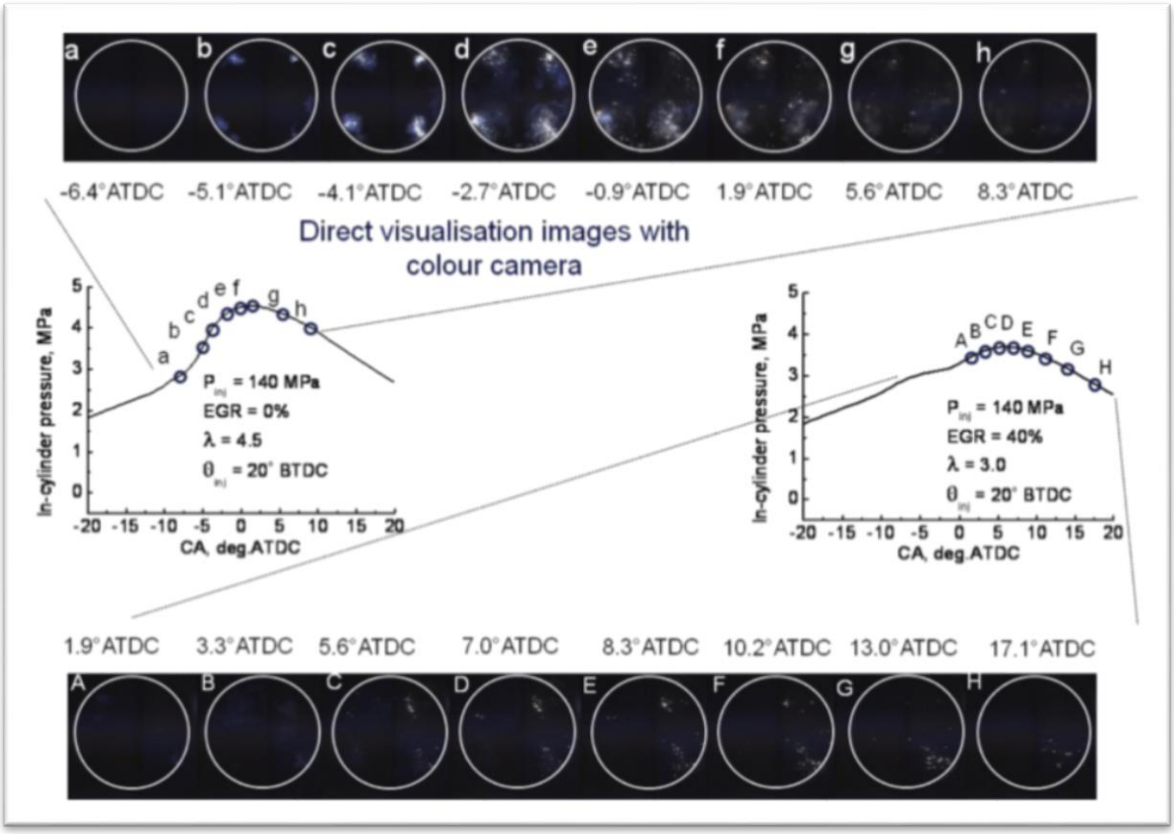

Figure 18. Visualization of combustion images from the optical instrumentation

The images displayed by the optical instrumentation [12], we will make an analysis of the formation of smoke and oxidation. The images showed in Figure 18, are the bottom-view images. It is showed the time series of the combustion images for $P_{inj=}$140 MPa and injection timing of 20º BTDC for the cases with EGR OF 40% and without EGR.

For the case without EGR, there was a short premixing time. Early combustion occurred at 6.4 after top dead centre (ATDC) with only a short premixing time. The early flame locations were located below of the spray and located near the piston bowl corner. A blue flame was observed during early flame development, quickly filled the bowl region uniformly due to the higher injection velocity. From image c to d, was observed some more luminous flame were observed between the spray directions. Due to the short premixing time, the regions with high luminous flames indicated some no homogeneities in the air fuel mixture in the piston bowl.

For the case with an EGR rate of 40%, first signals of flame were observed lately, at approx. -0.5 ATDC. Due to the long ignition delay, air-fuel mixture had a longer premixing time, the fuel that struck the piston bowl wall is a well mixing air-fuel, burning with low luminosity, and it is clearly showed how all the images are very homogenous and the lack of highly flame regions, preventing the sources of soot. With EGR, is showed how at injection time of 20º BTDC, simultaneously achieve low engine soot and NOx emissions.

Hence enhancing the mixing of the air and promoting oxidation so that the engine soot emissions at the exhaust pipe were a minimum.

For a deeply knowledge of soot formation and oxidation, it will be introduced as a parameter of study the spatially integrated flame luminosity (SIFL). It is known that SIFL is directly related with soot formation.

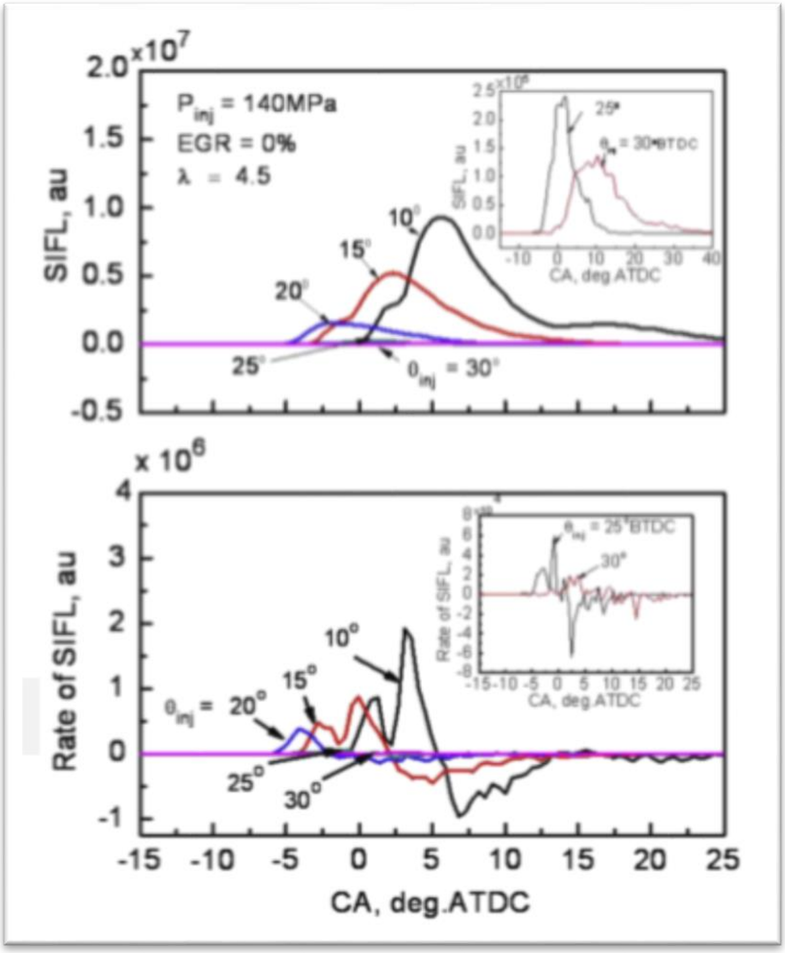

Figure 19. SIFL and rate of SIFL.

As it showed in Figure 19, for P_{inj}=40MPa and without EGR, late injection timing led to a high SIFL. Flame luminosity is dependent of the soot concentration and local temperature and under similar local temperatures, higher flame luminosity led to higher soot formation.

For θ= 10º BTDC, without EGR, temperature led to higher oxidation rates that is traduced into lees engine soot at the exhaust pipe. With the advance of the injection timing, in-cylinder temperature decreased and with it the SIFL, leading into low oxidation rate and more soot emissions, which can be showed with the derivate of SFL, the rate of SIFL which provides the soot formation and oxidation information. Cases without EGR showed higher positive peaks compared with EGR cases, which indicate faster combustion and soot formation processes. Negative peaks indicate higher oxidation rate, which will be traduced into higher oxidation rate, as it is showed this fact is also present for the θ= 10º BTDC.

4.3. Effect of EGR

Ultimately, it will be discussed the effect of EGR to improve the performance and emissions [3] . If it is remembered, EGR was used to improve the NOx emissions by lowering t the adiabatic flame temperature and reducing NOx emissions through charge dilution. On the figure below, it is represented the in-cylinder pressure and the rate of heat release (ROHR) with the crank angle for P_{inj}=140 MPa, for the cases with and without EGR[12].

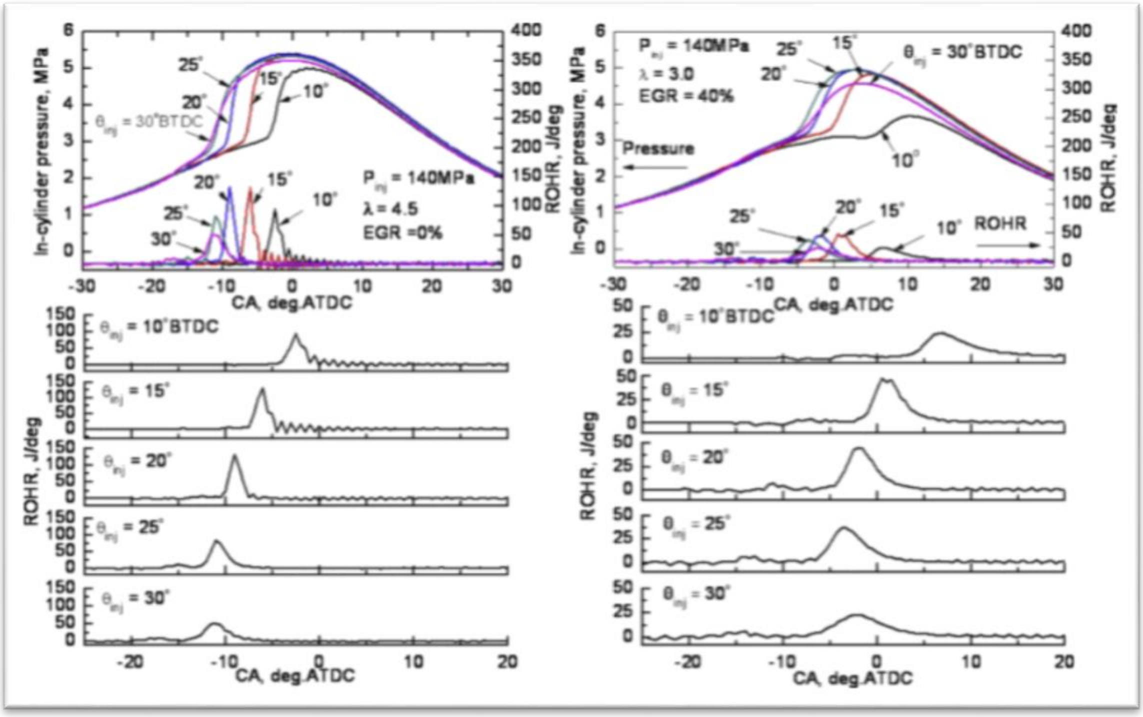

Figure 20. In-cylinder pressure and ROHR.

As it is showed in Figure 20, for EGR case, the in-cylinder pressure is lower being at maximum about 5 MPa, being in the case without EGR higher to this value. The same for the ROHR where for the case of EGR, it is showed an extremely good reduction. As it is known, with the introduction of EGR, the ignition delay is longer and the combustion phasing is retarded, as it can be seen for the EGR case where for all de cases, the in-cylinder pressure curve and ROHR curve were displaced at the right. Due to the longer ignition delay, in-cylinder temperature is lower as well as is better the air-fuel mixing which led to a slight combustion.

For the next image, it will be discussed the effect of EGR on the indicated thermal efficiency, IMEP and coefficient of variance of IMEP.

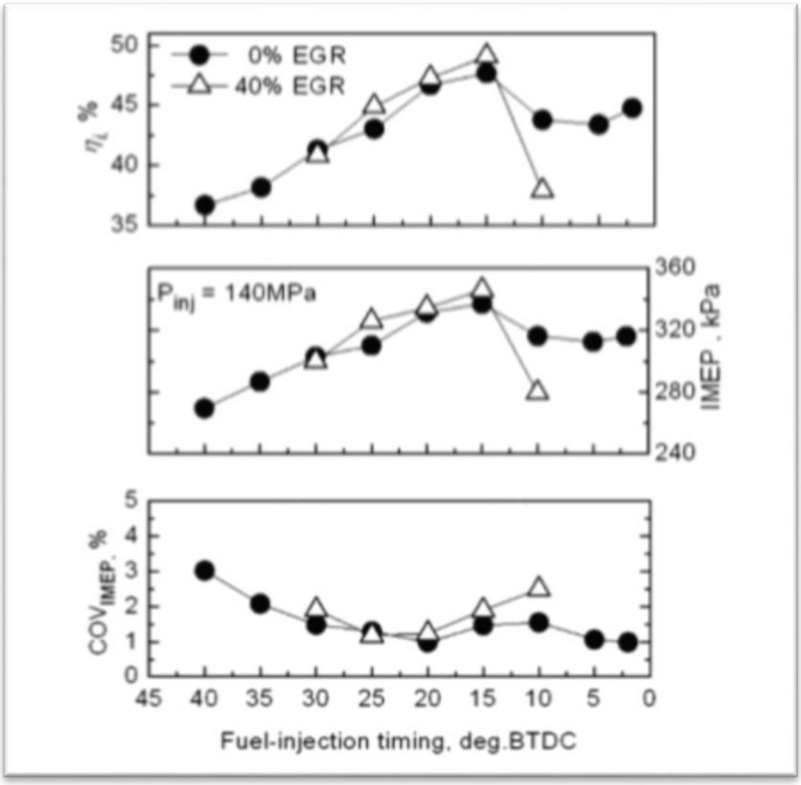

Figure 21. Effect of EGR on indicated thermal efficiency, IMEP, and COV of the IMEP

In the case of indicated thermal efficiency, with a θ= 2-10º BTDC it is showed in Figure 21 how with retarded injection timing improves the indicated thermal efficiency for the case without EGR. It is thought to be connected with the over-mixing air-fuel which carries to a too lean mixture unable to burn completely. At 15º BTDC is reached a maximum and then decreased for both cases. Right this point and forward, using 40% of EGR improves the indicated thermal efficiency. If it is remembered is the same timing in order to improve the engine performance for different injection pressures. θ= 15º BTDC is the optimum point to achieve highest indicated thermal efficiency for high injection pressure and use of EGR.

As it was commented, EGR on PCCI combustion is used to lower adiabatic flame temperature and oxygen concentration, also EGR had a strong impact in achieving a longer air-fuel premixing which carries to better performance and lower soot emissions. As it is showed below EGR has an important role on ignition delay:

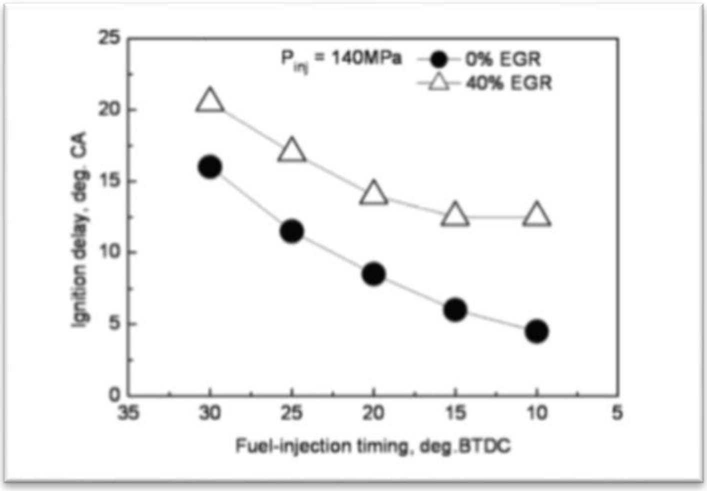

Figure 22. Effect of EGR on ignition delay

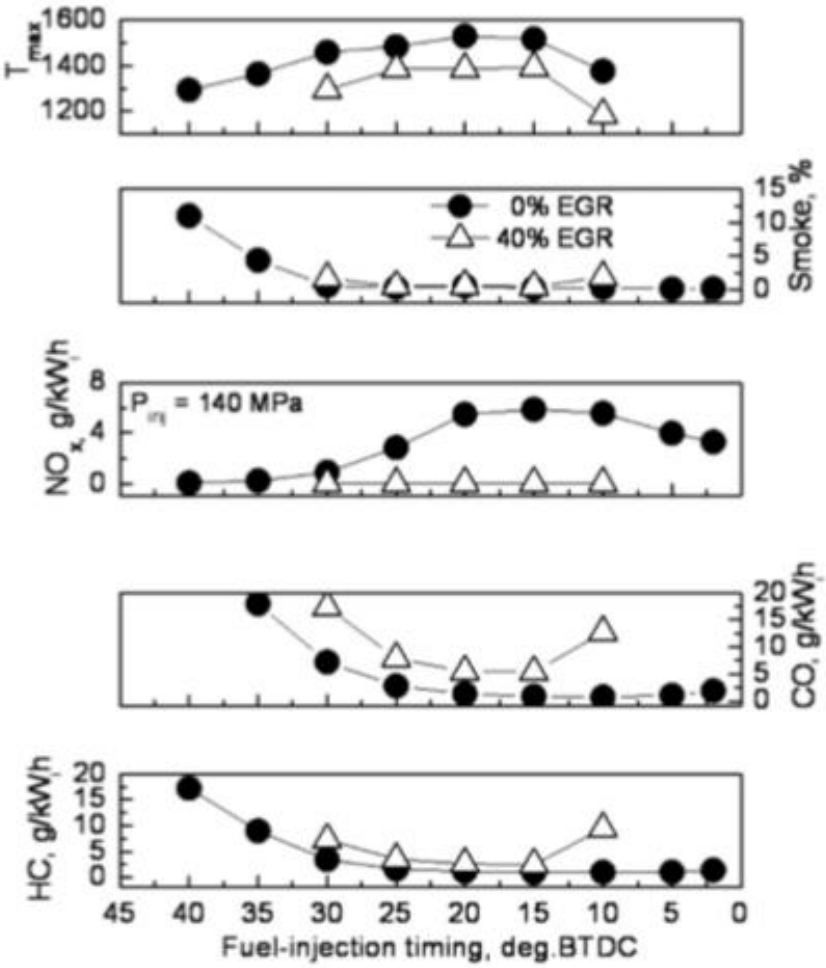

Figure 23. Effect of EGR on maximum in-cylinder temperature and specific emissions.

In Figure 23 is discussed the effect of EGR on the in-cylinder temperature and the exhaust emissions:

On first one plot, it is showed how for all the injection timing maximum temperature was always lower with the use of EGR. It can be seen clearly how while advancing injection timing the maximum in-cylinder temperature rises till the well-known θ= 15º BTDC is achieved, where maximum in-cylinder temperature decreases drastically for PCCI combustion.

For the smoke emission, in the case of use of EGR, it can be identified for θ= 10º and 30º BTDC two excessive smoke emissions. It is attributed to the reduction of in-cylinder temperature and the uncompleted soot oxidation formed by impingement on the piston surface. PCCI combustion is not meant to completely eliminate smoke emissions but rather reduce them to acceptable levels. Many researchers have noted that it is possible to obtain a simultaneous reduction in NOx and soot by incorporating large amounts of EGR, but this comes at the expense of high fuel consumption and increased CO and HC emissions. For conventional diesel combustion without EGR, NOx emissions drastically increased as the injection timing was advanced up to a maximum at 15 BTDC, after which they decreased drastically in the PCCI combustion regime. This could be attributed to the high in-cylinder temperature, which promotes NOx emissions, and a short duration for fuel air mixing prior to combustion. EGR effectively reduced NOx emissions. This was attributed to the reduced oxygen concentration and the decrease in flame temperature in the combustible mixture. A moderately early injection of 30 BTDC and above led to low NOx emissions without EGR due to the reduced in-cylinder temperature but also led to an increase in smoke emission.

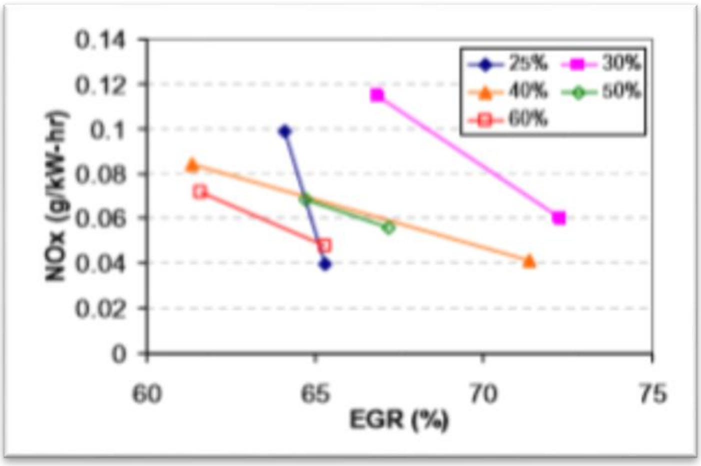

Figure 24. NOx emissions vs. EGR rate at each percent.

Expanding the information about the sensitivity of EGR on NOx emissions, came to corroborate what was expected. NOx maintained the trend established that its decrease with increasing EGR. They expand the information about the sensitivity of EGR on NOx emissions, came to corroborate expected. NOx maintained the trend established its decrease with increasing EGR. This conclusions are showed in Figure 24.

With the advance of injection timing, CO emissions, as it is showed on the CO plot, were increased. The responsible of this CO emission growth was the use of EGR which reduces in-cylinder temperature and reduce the oxygen formation for a complete combustion. In Figure 25 is showed the CO emissions with the high and low levels of EGR and can be cross-referenced with PM emissions, which shows that the equivalence ratio indeed increased. CO emissions are at least partially influenced by the availability of oxygen, increasing the equivalence ratio through the use of EGR would be expected to result in an increase in CO emissions.

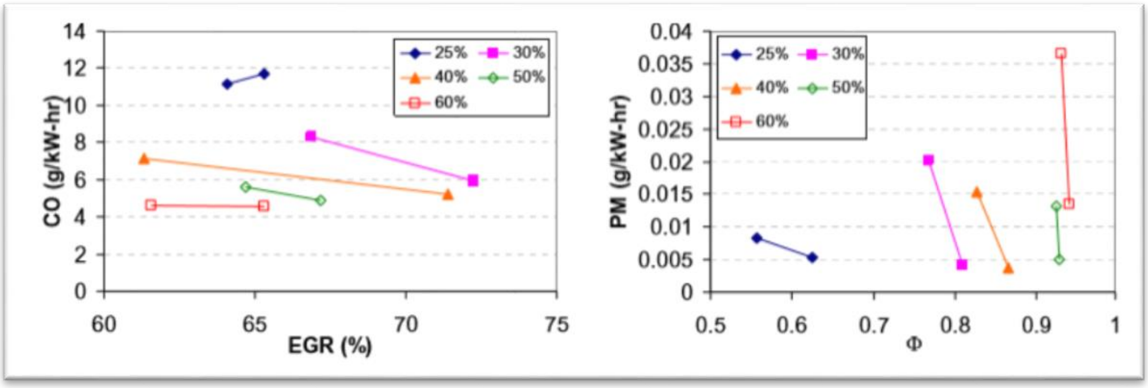

Figure 25. CO emissions vs. several EGR rate and PM vs. equivalence ratio.

The common pattern for advanced injection timing on HC emissions was always high due to the same reason than CO emission: limited amount of oxygen. It is especially significant the HC increase for θ= 10º BTDC which is attributed to the excessive lean mixture to burn due to over-mixing. Some researchers suggested air mass supercharging while using high amounts of EGR

4.4. Effect of the equivalence ratio

For last, it will be discussed the effect of equivalence ratio, given that this one is such an important parameter on the development of a PCCI combustion.
This time, it is studied the investigation of Hardy and Reitz [11]
In the table, it can be seen the experimental running parameters.

Φ	Boost Pressure (kPa)	Back Pressure (kPa)	EGR (%)	Injection SOI (CA-ATDC)	Injection Duration (μs)	Intake T (C)	Exhaust T (C)	η_F (%)	BSFC (g/kW-hr)	NOx (g/kW-hr)	HC (g/kW-hr)	PM (g/kW-hr)	CO (g/kW-hr)
0.54	199.3	210.3	64.53	-48.25	1600	33	203	27.7	332	0.4165	0.656	0.0101	10.59
0.71	213.7	231.7	73.86	-48.25	1900	34	193	25.2	269.8	0.1226	0.6209	0.0084	12.09
0.76	203.4	220.6	70.62	-59.25	2000	34	194	25.6	287.7	0.0839	0.952	0.0054	11.14
0.80	210.3	235.1	69.65	-60.25	2200	38	223	29.1	262.7	0.043	1.3934	0.0078	8.48
0.85	203.4	221.3	70.47	-59.25	2200	38	247	31.5	274.8	0.0396	1.7767	0.0047	5.96
0.92	209.6	231.7	66.17	-60.25	2500	41	262	31.3	244.5	0.0673	1.9229	0.0064	5.24
0.93	209.6	231.0	67.19	-60.25	2600	43	289	33.6	252.3	0.0557	2.6358	0.005	4.97
0.94	210.3	239.2	65.31	-60.25	2900	46	311	33.6	284.9	0.0476	9.2434	0.0135	4.90

Table 2. Equivalence ratio investigation experimental parameters.

With the increase of equivalence ratio at high EGR, it is showed the increase of intake temperature and exhaust gases. By other side, it is proved how the brake specific fuel consumption lowers with the commented increase. With regard to contaminant emissions, NOx and CO emissions were found decreasing with the higher equivalence ratio at high EGR.

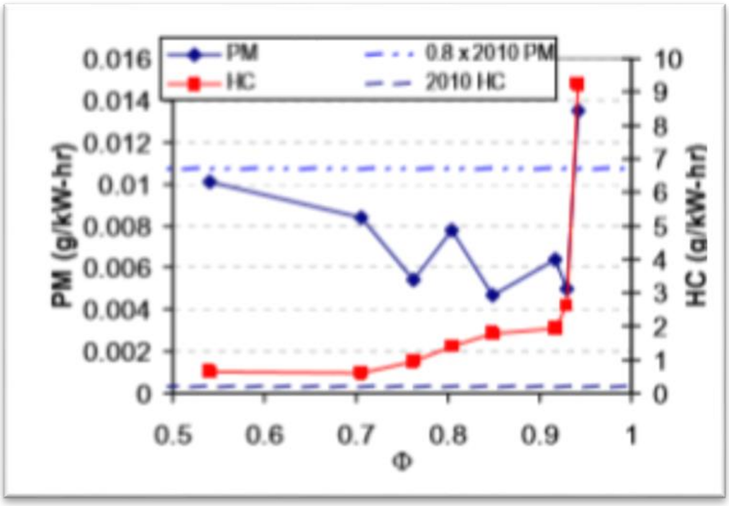

Figure 26. HC and PM emissions versus equivalence ratio.

As can be seen in Figure 26, a multi parameter plot where is represented the 2010 PM and HC emissions mandate. At equivalence ratios up to approximately 0.9, PM emissions levels well below 80% of 2010 levels were attained. This is a very promising result, since high-EGR, high-load PCCI will require high equivalence ratios in order to achieve acceptable BSFC levels. By contrast, as the equivalence ratio increased beyond approximately 0.8, the HC emissions levels became too high to meet the 2010 emissions mandates. Better targeting and fuel delivery could also reduce HC emissions by increasing mixing and decreasing impingement

4.5. Conclusions

On this chapter has been analyzed the effects of injection parameters, spray impingement and EGR on the combustion characteristics and emissions of a PCCI diesel engine. To achieve a global vision, it has been achieved the following conclusions:

- It has been reported how with the higher injection pressure of P_{inj}=140 MPa both indicated thermal efficiency and IMEP were better, specifying in values between 20º and 15º BTDC, than lower injection pressure of P_{inj}=80 MPa. Also the HC emissions and smoke were lower and CO emissions remained constant. For late injection timing, NOx emissions were higher for the higher injection pressure, by contrast, for early injection timing occurred the opposite.
- Low smoke emission was achieved with late and moderately early injection timings, due to the interaction between spray and piston bowl geometry. Higher flame luminosity led to higher soot formation. It was found that low luminosity flame was attributed to fuel spray impingement on the surface of the piston bowl. At θ= 20º BTDC was the optimum point for simultaneously low soot and NOx emissions.
- By the use of EGR was found that for θ= 15º BTDC both reduction in NOx and soot this was attributed to the reduced oxygen concentration and the decrease in flame temperature in the combustible mixture. Also the indicated thermal efficiency and the IMEP were higher with the use of EGR in the PCCI combustion, at the expense of HC and CO emissions.
- EGR rates must be closely metered to appropriately phase the start of combustion and conserve appropriate equivalence ratios with that PCCI combustion is capable of meeting 2010 NOx and PM emissions, however, the HC emissions must be improved.

5. Chapter V: Experimental instrumentation review on the tuning of a FTP industrial engine

5.1. Introduction

On this chapter is discussed an experimental review of the test bench which is carried out at Politecnico di Torino in collaboration inside Erasmus+ program. A review of the main experimental apparatus which are necessary on the tuning of a diesel engine in order to the implementation of a PCCI strategy.

It is a dynamic bench, where an electric brake (dynamometer) is installed. The brake is able to work in two different operating conditions: it can work as a typical brake able to reproduce a resistant torque, opposite in direction to that delivered by the engine, and as an electric motor that is able to drive the internal combustion engine. The advantage deriving from the presence of the brake consists in the capability of performing experimental tests under dynamic conditions.

5.2. The engine

The tested engine is a FPT Industrial F1C diesel engine, prepared for the test by the FPT Industrial R&D office in Arbon (Switzerland), Euro 6 ready. It is an in-line 4 cylinders 3 liters diesel engine, with a rated power of 180 [HP] and maximum torque of 350 [Nm] at 1400 - 2800 [rpm]. In Figure 27 the engine is shown.

Figure 27. FTP Industrial F1C Euro6 diesel engine

The main technical characteristics of the engine are resumed in the following table.

F1C Technical data	
Layout	4 cylinders in-line
Bore	95.8 [mm]
Stroke	104 [mm]
Valves per cylinder	4
Intake mode	VGT
Fuel injection	High-pressure CR system

Table 3. Technical data

5.3. Engine instrumentation

The engine is fitted with its own ECU that monitors the main functional parameters, but other quantities have to be monitored in order to correctly calibrate the engine to make it running on PCCI mode. For this reason the engine has been provided of other sensors that provide additional information in terms of pressures, temperatures, blow-by gases flow rate, engine oil flow rate, exhaust gases flow rate and emissions. The added sensing elements are described in the following sections.

5.3.1. Pressure sensors

5.3.1.1. Intake manifold pressure sensor

The sensor is of the piezoresistive type and it is a high temperature pressure sensor from Kistler. The sensor in question is 4007C type (Figure 28), a miniature type suitable where the size available for mounting is limited, able to work in continuous high temperature operation up to 200◦C.

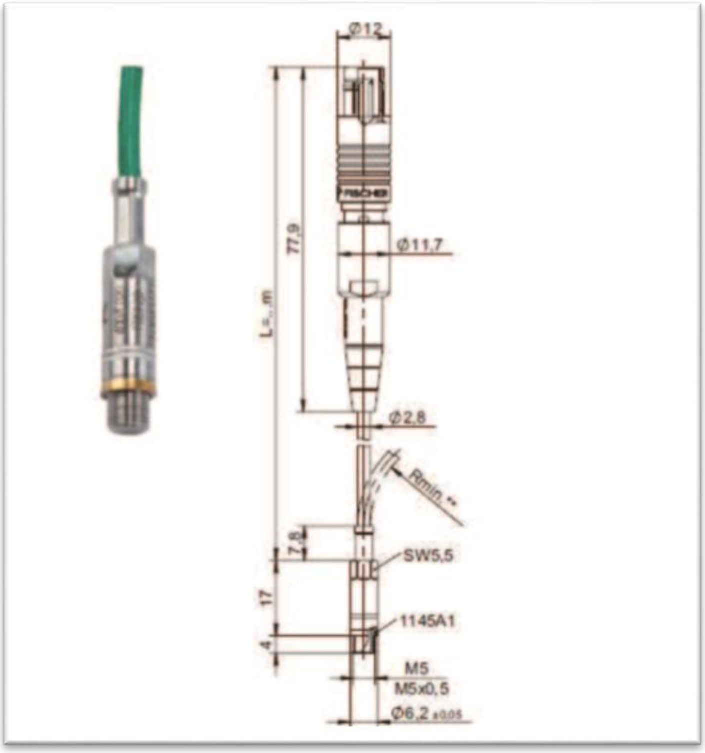

Figure 28. Kistler 4007…DS pressure sensor for intake manifold.

It utilizes a fully active four arm Wheatstone bridge to generate an electrical signal which is proportional to the applied pressure. The resistors making up the Wheatstone bridge are implanted into a micro-machined silicon diaphragm which is formed using Silicon on Insulator (SOI) technology. This technology allows to minimize the hysteresis and the repeatability errors. The small size coupled to the very fast dynamic response allows for high quality pressure measurements to be made in locations where other sensors cannot fit.

The sensor can be easily installed directly into a simple threated measuring port; it is essential to comply with the clamping torque specified in the previous table

5.3.1.2. Exhaust manifold pressure sensor

The high temperature gas pressure measurement within the exhaust manifold is made through a water cooled absolute pressure sensor from Kistler of type 4049B...DS (Figure 29). It is a piezoresistive sensor provided with water cooling and capable of continuous high temperature operations. It allows the application within high temperature environments like that of the exhaust manifold, without the need of additional water cooled adapters. The measuring element is situated behind a thin steel isolation diaphragm and an oil fill providing excellent media compatibility. The core element is placed within a cooling jacket whereby the internal temperature can be suitably managed. This configuration allows the sensor to be applied within hot environments and since the constant water cooling thermal effects are minimized. The Figure 5.4: Pressure sensor mounting and particular of the bore for direct mounting presence of an integrated amplifier also allows a further monitoring digital compensation.

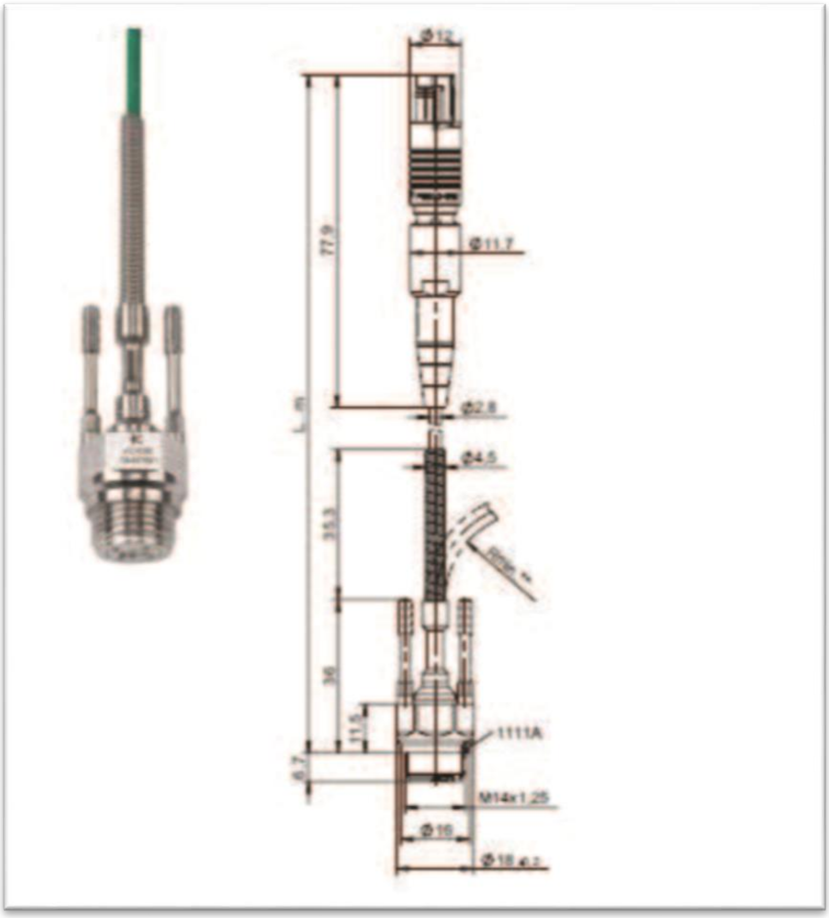

Figure 29. Exhaust manifold pressure sensor Kistler 4049B…DS

The sensor can be installed directly within the measuring port and it is recommended to install it with the integrated screen that acts as heat protector. The seat hole must be machined according to the sensor bore and the tightening torque must be equal to that shown in the previous table.

5.3.1.3. In-Cylinder pressure sensor

The last pressure sensor is mounted within the combustion chamber. It is a Kistler 6058A sensor type with glow plug adapter (Figure 30). The adaptability to the glow plug avoids the machining of an additional hole within the cylinder head. It shows good temperature stability of the sensitivity, low thermal shock errors and high sensitivity. Within a diesel engine could be difficult to find space enough for the sensor installation and so the reason of a glow plug adapter.

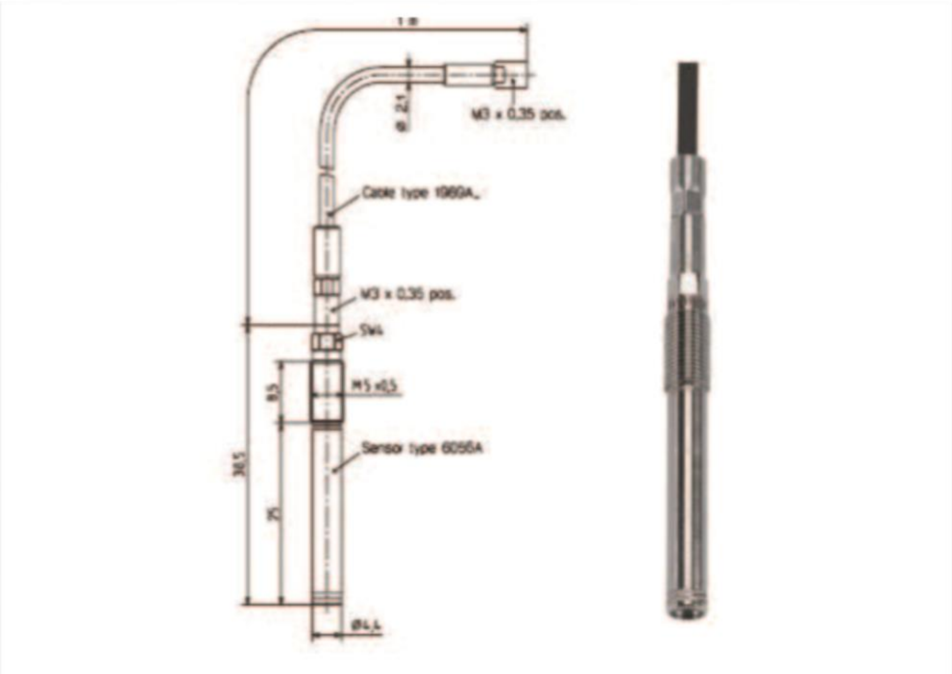

Figure 5.7: In-cylinder pressure sensor Kistler 6058A with glow plug adapter

Figure 30. In-cylinder pressure sensor Kistler 6058ª with glow plug adapter

The adapter is installed into the existing bore of the series glow plug. The cylinder pressure sensor remains in contact with the combustion gases via short, star-shaped bores in the glow plug tip. The PiezoStar, a new piezoelectric crystal from Kistler, allows high thermal stability. The connector allows standard length for pressure sensors to be installed in varying length of glow plug adapters.

Glow plugs are used primarily to optimize engine starting with minimized emissions. However, provided they are not being used in extremely cold climates, engines can always be started without glow plugs. An adapter fitted with a sensor can therefore be mounted in place of the original glow plug. The gap between adapter and mounting bore has a significant influence on the measuring performance and so the precise diameter of the mounting bore is required. Another problem could be pipe oscillations that can arise in diesel engines since the high pressure gradients and that can disturb the measuring signal. These superimposed high-frequency oscillations occur from the start of combustion and can lead to errors in determining the maximum cylinder pressure, while the mean effective pressure results not affected. Pipe oscillations sensitivity can be reduced by mounting the sensor deep into the glow plug adapter.

5.3.2. Temperature sensors

Temperature values are detected by a sensing element called thermocouple. The thermocouple is a sensor that consists of two wire legs made from different metals. The wires legs are welded together at one end, creating a junction (Figure 31). This junction is where the temperature is measured, while the other side junction is at 0◦C or at a predefined reference temperature. When, at the junction, a change in temperature is experienced, a voltage is created. The voltage value, function of the temperature difference between the measuring junction and the reference junction, is then read on tables where for each voltage value a temperature value exists.

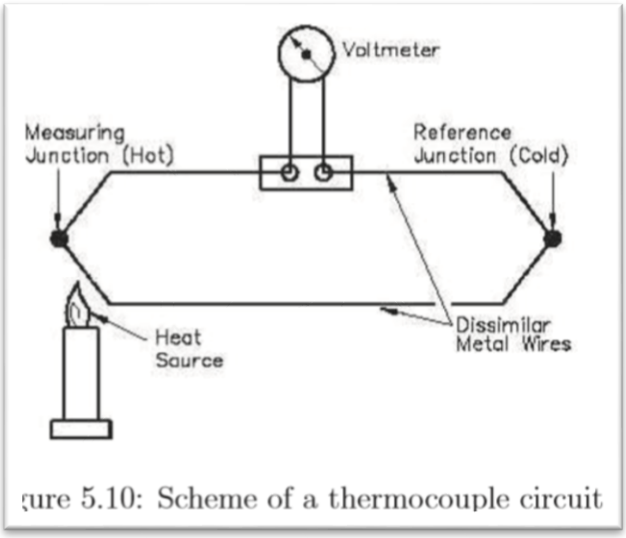

Figure 31. Scheme of a thermocouple circuit

Nickel alloys circuit thermocouples are the most used since they are inexpensive and a great variety of probes is available in their temperature range of −200◦C/1350◦C. A typical sensor is the type K thermocouple (Figure 32) that can be used up to 1260◦C in a non-oxidizing environment without aging, or in marginally oxidizing environment like carbon monoxide. Thermocouples are mounted and fixed through adjustable compression joints.

Figure 32. Kistler type K thermocuple.

5.4. Sensor positioning on the F1C engine

Test bed runs of the F1C engine from FPT Industrial require the engine to be instrumented with a certain number of sensors. Temperature, pressure and other sensors types are placed in suitable positions both at the engine inlet and outlet in order to analyze the engine steady-state and dynamic behavior both in conventional diesel combustion mode and PCCI mode. F1C engine, arriving from the FPT Industrial R&D department of Arbon (Switzerland), was already run on a test bed for emissions and performances tests before its placement on the market. Since the experimental activity to be carried out at Politecnico di Torino is quite different and complex from an usual diesel engine calibration, specific sensors have to be mounted in fundamental areas of the engine. For this reason, F1C was equipped only with few sensors thus leaving a certain freedom to the research group of the ICE advanced laboratory of Politecnico di Torino.

5.4.1. F1C existing sensors

5.4.1.1. Exhaust manifold thermocouples

The F1C engine is provided of a cast iron exhaust manifold directly connected to the cylinder head without the presence of cylinders runners. FPT Industrial provided the engine with four type K thermocouples, each of which is placed in direction of the cylinder axis plane (Figure 33).

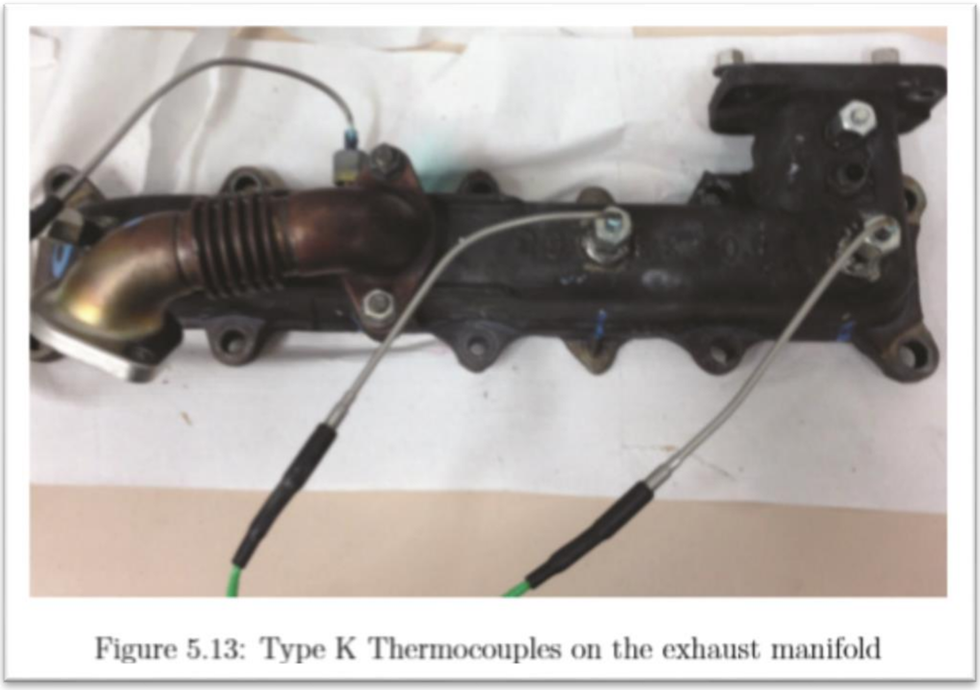

Figure 33. Type K thermocouples on the exhaust manifold.

The engine exhaust pipe is provided with a lambda sensor for emissions control.

5.4.1.2. Turbocharger speed sensor

The engine shows a turbocharger group of the VGT type with an electronic control of blade angle performed by the ECU. An additional sensor, the Pico-Turn, is fitted within the turbine in order to control its rotational speed (Figure 34).

Figure 34. Turbine rotational speed sensor

5.4.2. New sensors mounting on F1C engine

5.4.2.1. F1C exhaust side

Other sampling points have been already machined onto the engine and new sensors mountings have to be prepared for experimental proposals. By taking into account the F1C exhaust side available sampling points are present onto the exhaust manifold and before and after the turbocharger group. As it is possible to see in next figure (Figure 35), the exhaust manifold shows a sampling point for gas pressure before turbocharger and a sampling point for gas temperature before turbocharger.

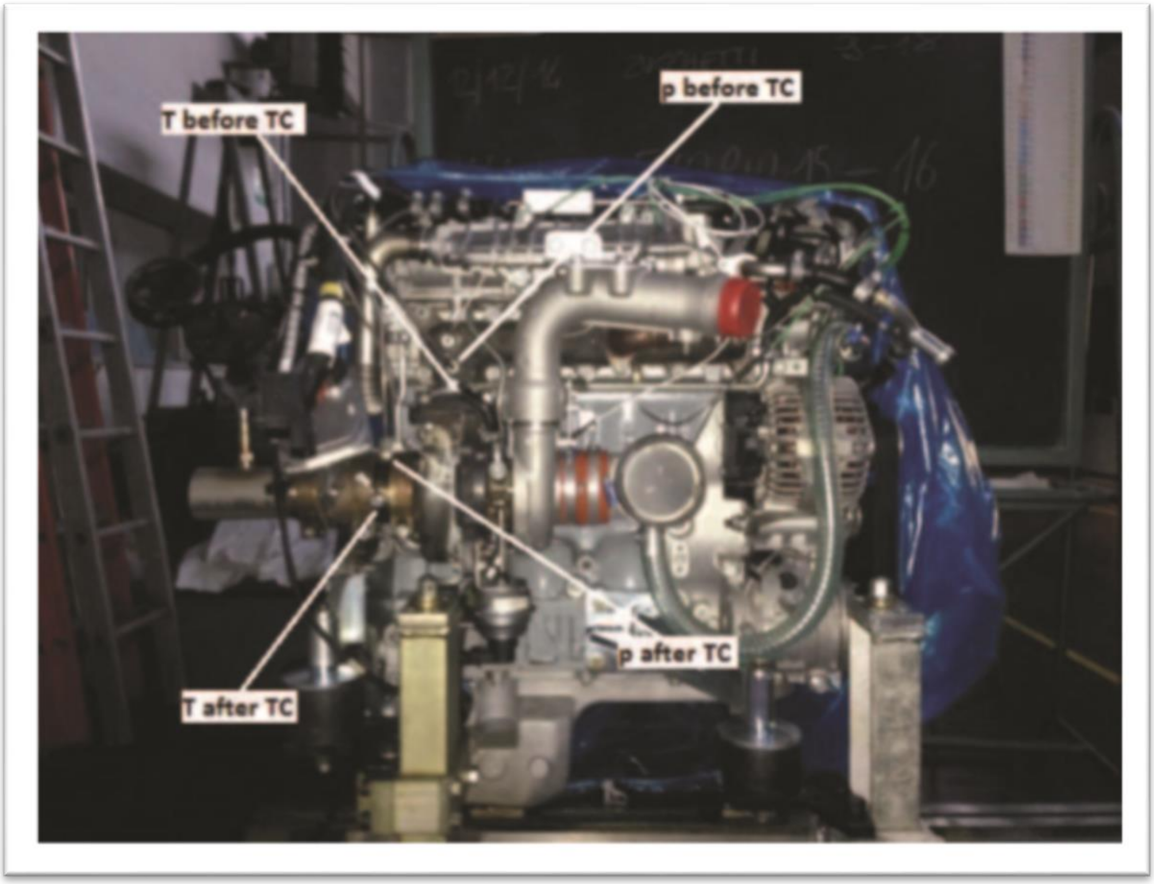

Figure 35. Sampling points on F1C exhaust side

The pressure sensor is a relative pressure sensor with a measuring range 0 - 5 [bar]; the temperature sensor is a type K thermocouple. Sampling points for similar measurements after the turbocharger group are also available. In addition to these available points, some other sampling points have to be machined onto the exhaust manifold. As now shown, the manifold will be provided with a high frequency pressure sensor; the mounting of the sensor will be machined in the direction of the axis of cylinder 1. The choice of this positioning is due to space constraints on the intake side for the mounting of the high frequency intake pressure sensor and due to the need of having the two measuring points (intake-exhaust) on the same cylinder

5.4.2.2. F1C intake side

While taking into account the intake side of F1C engine some other sensors types have to be installed. As shown in Figure 36 the intake manifold is not provided of temperature sensors: they will be installed in direction of each cylinder axis and with a 30◦ inclination with respect to the horizontal direction since existing space constraints. The intake manifold temperature sensors are type T thermocouples. Moreover, in the left part of the figure are also available two sampling points: for seek of simplicity one of them is used for the positioning of the high frequency pressure sensor.

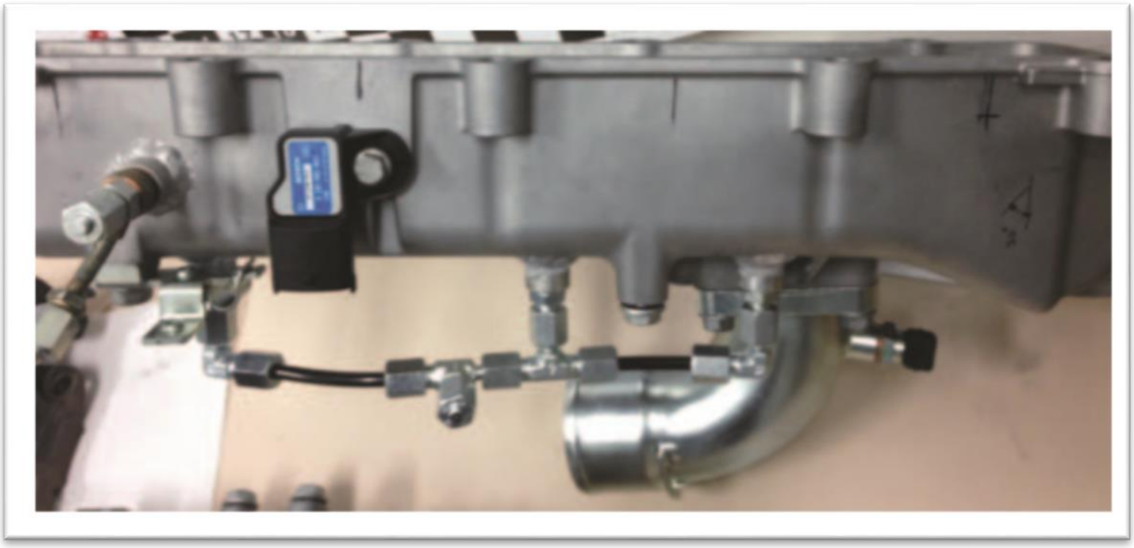

Figure 36. Sampling points for measurements on the intake manifold

FPT industrial delivered the engine with an already created Kempt chamber that in previous tests was used to sample CO2 and emissions concentrations on the intake side by using the central sampling point. Politecnico's configuration is slightly different since the emissions sampling point will be the most external on the left while the central point will be used for static pressure measurements. On the intake side, below the oil filter, two additional sampling points are present; these points will be exploited for the monitoring of the oil temperature before the filter and of the oil pressure within the gallery (Figure 37).

Figure 37. Sampling points for oil temperature and pressure upstream the filter

Cylinder pressure measurements, as already discussed, will be mounted through glow plug adapters within the cylinder head. The adapter will replace the F1C glow plugs by allowing the mounting of the pressure sensor.

5.4.2.3. F1C SENSORS GENERAL SCHEME

The sensors positioning described up to now are those directly installed on the engine. Then other sensors, called test bed sensors, are installed within the dynamic test cell and directly connected to the AVL Puma used for tests. These sensors belong to some cell systems like the intercooler and the cooling system more precisely described in the following sections. For every sensor has been chosen a norm name, i.e the ID for the Puma Open system. A general scheme of sensors installation with their relative norm names and a descriptive table are now provided (Figure 38).

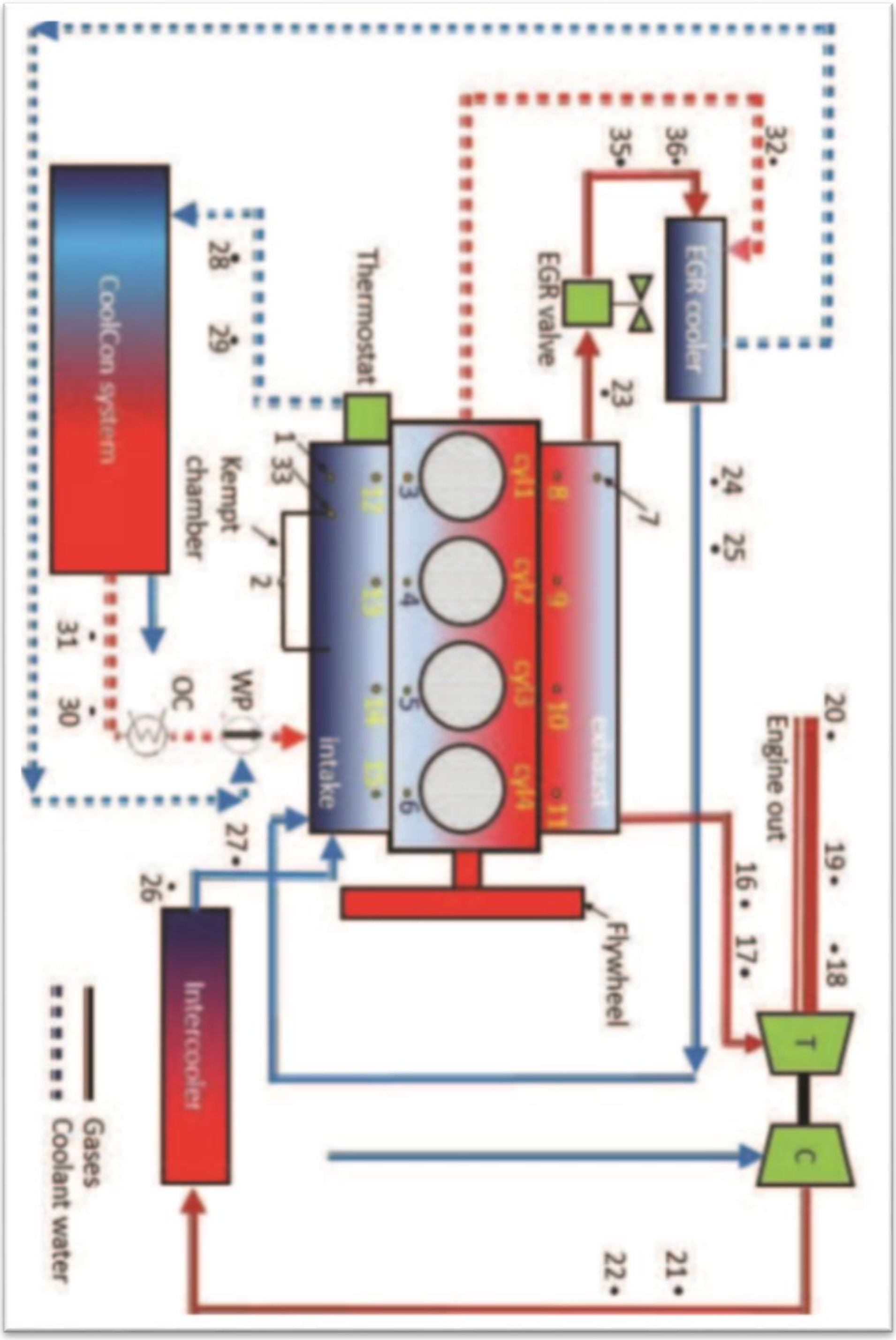

Figure 38. F1C and dynamic test cell sensors installation general scheme

F1C installation in the dynamic test cell			
Position	Normname	S-type	Description
1	PINT	Kistler	Intake manifold HF pressure
2	P-MAP	0-4 [bar] G	Intake manifold LF pressure
3	PCYL1	Kistler	Cylinder 1 pressure
4	PCYL2	Kistler	Cylinder 2 pressure
5	PCYL3	Kistler	Cylinder 3 pressure
6	PCYL4	Kistler	Cylinder 4 pressure
7	PEXH	Kistler	Exhaust manifold HF pressure
8	T-ExRun1	TC-K	Cylinder 1 exhaust temperature
9	T-ExRun2	TC-K	Cylinder 2 exhaust temperature
10	T-ExRun3	TC-K	Cylinder 3 exhaust temperature
11	T-ExRun4	TC-K	Cylinder 4 exhaust temperature
12	T-Run1	TC-T	Cylinder 1 intake temperature
13	T-Run2	TC-T	Cylinder 2 intake temperature
14	T-Run3	TC-T	Cylinder 3 intake temperature
15	T-Run4	TC-T	Cylinder 4 intake temperature
16	p-ExbTC	0-5 [bar] G	Exhaust pressure before turbo
17	T-ExbTC	TC-K	Exhaust temperature before turbo
18	p-ExaTC	0-1 [bar] G	Exhaust pressure after turbo
19	T-ExaTC	TC-K	Exhaust temperature after turbo
20	Emissions		Sampling point for emissions
21	p-abIC	0-5 [bar] A	Air pressure before intercooler
22	T-abIC	PT100	Air temperature before intercooler
23	T-ExhSy1	TC-K	Exhaust T upstream EGR valve
24	T-ExhSy2	TC-K	Exhaust T downstream EGR valve
25	p-ExhSy2	0-2.5 [bar] G	Exhaust p downstream EGR valve
26	p-aaIC	0-5 [bar] A	Air pressure after intercooler
27	T-AaIC	PT100	Air temperature after intercooler
28	p-CWC-in	0-5 [bar] A	Coolant pressure engine-in
29	T-CWC-in	PT100	Coolant temperature engine-in

Position	Normname	S-type	Description
30	p-CWC-out	0-5 [bar] A	Coolant pressure engine-out
31	T-CWC-out	PT100	Coolant T engine-out
32	T-CWEgrI	TC-T	Coolant T EGR cooler-in
33	Intake emissions		Intake manifold sampling point
34	T-CWEgrO	TC-T	Coolant T EGR cooler-out
35	T-CWEgrV	TC-T	Coolant T before EGR valve
36	T-EGRaV	TC-K	Gas T after EGR valve

5.4.2.4. Encoder mounting

Another sensor to be mounted on the FPT Industrial F1C engine is the encoder: this sensor is a transducer of the engine crankshaft angular position and of the crankshaft angular speed and it is needed for the synchronization of the engine ECU. The encoder has to be fitted on a special support external to the engine block and connected to a suitable electronic wiring. The F1C encoder support provided by FPT Industrial did not fit with the encoder already available. A sketch of the F1C support for encoder is shown in Figure 39. On the support shaft a seat for a feather key has been machined. The feather key allows a perfect centering of the new flange for the encoder.

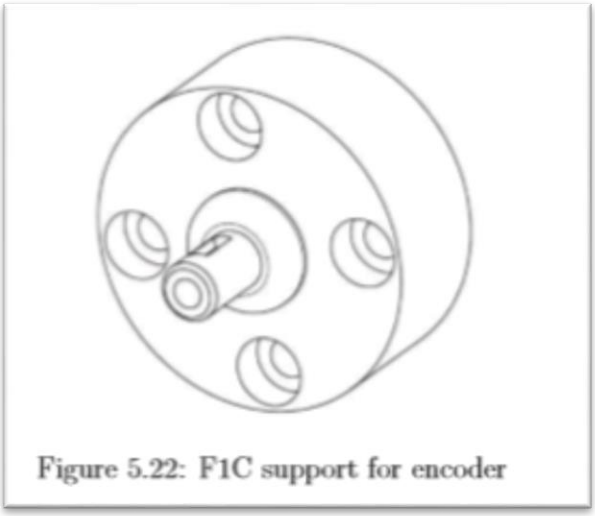

Figure 39. F1C support for encoder.

The newly designed flange for the encoder shows on one side the feather key seat that will fit on the support shaft. Moreover an additional screw will tight within the support shaft thus allowing perfect centering and stability. On the other side the encoder will be mounted; it will fit on a 4 mm thickness seat and it will be kept in the desired position through three screws fitting within the M5x1.25 holes.

5.5. Intake air flow meter

The amount of inducted air is measured by means of the flow meter. It provides an information to the ECU that delivers a signal to the electronic control unit of the injection system. The most used air flow meter, positioned within the intake manifold of the engine, is a hot wire-based sensor (Figure 40). It is useful to adjust some governor parameters and to help the ECU to select the more appropriate engine map. The electric air flow meter (hot-wire) directly sends information to the ECU (while in ancient mechanical MAF the ECU required some calculation process) through an electric circuit that stores and analyzes the received data

Figure 40. Intake manifold air flow meter.

from the air film. The operating principle is based on a membrane within the channel where the air film is flowing. By means of a resistor, the membrane is heated and it is kept at a constant temperature of 120◦C. Since the flowing air within the channel has a cooling effect on the membrane, this last absorbs current in order to maintain constant the temperature of 120◦C. The absorbed current, measured through a Wheatstone bridge, is directly proportional to the air is flowing within the manifold. Since it is a mass meter, the sensor is not subjected to deviations from changes in temperature and pressure.

5.6. Blow-by meter

With the term blow-by it is usual to indicate the flow of combustion gases through piston rings, crevice regions and cylinders, up to the engine crankcase. Accepted values are around 0,4% − 0,6% of the total amount of engine gases. This value tends to increase with the ageing and the wear of the engine, and as it increases the compression pressure proportionally decreases, with a direct deterioration of engine performances. Moreover gas into the crankcase can dilute the oil thus affecting and reducing the component life. The crankcase vent is connected to the intake system (after the compressor) of the engine for emissions reasons, so the blow-by gases are reinducted within the cylinder where they burn. Since, within the crankcase, gases are in contact with the lubricant oil, some oil separators are placed in the intake system and they are connected to the engine oil sump where oil residuals are newly sent. Oil separators are not able to provide any information about the flow rate of blow-by, so a dedicated measuring instrument is used at the test bed. The AVL blow-by meter (Figure 41) working principle is based on the pressure differential concept. The amount of leakage of

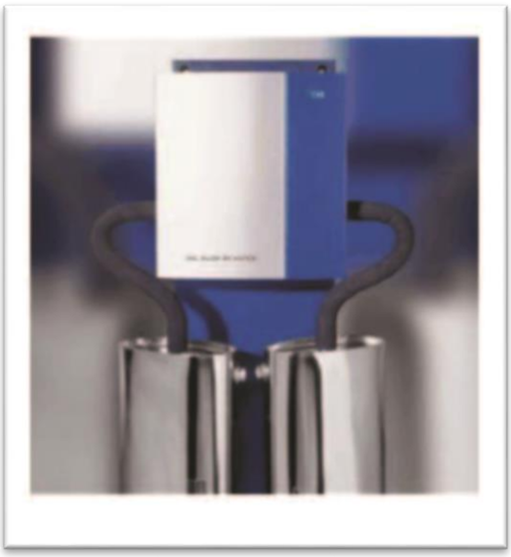

Figure 41. AVL Blow-by meter

internal combustion engine is measured with a device based on the principle of orifice measurement, through an orefice measuring pipe and evaluation electronics. The leakage gases pass from the crankacase through a filter where the lubricant oil is separated. The flow now enters a first damper bottle upstream of the meter that, together to a second one, has the aim of damping pressure oscillations thus allowing a more precise pressure measurement. Blow-by gases pass through an orefice measuring pipe that creates a differential pressure (Venturi effect) between the sections upstream and downstream the orefice. By measuring the pressure differential and by applying the ideal gas low, the blow-by flow rate can be obtained.

5.7. Fuel consumption measuring system

The test bed is provided of a AVL KMA4000 fuel metering system that allows a continuous measurement of the fuel. Fuel is aspirated from a tank through a pump that sends it to a volume flow rate meter (PLU 121) through a heat exchanger, a filter and a density meter (L-dens). Downstream the flow rate meter, a second pump sends the fuel to the engine fuel injection system through a pressure valve and a temperature regulator. The fuel measurement is made by means of the density and flow rate meter, with a mass correction since the temperature difference that exists between the PLU 121 and the L-dens devices.

5.8. Intercooler

The intercooler is a cooling system present in turbocharged engines. Turbocharging allows to increase the output power from an engine by inducting compressed air at the intake system. The turbocharger group is moved by exhaust gases from the engine that are compressed before being reintroduced in next cycle. Since gas compression leads to an increase in temperature, the compressed charge must be cooled and increased in density. The test cell is provided of its own compressed charge cooling system as that shown in Figure 42. The coolant is the water fed directly from the water piping system of Politecnico di Torino: the cooling effect on the compressed charge is regulated by means of a lamination valve before the intercooler.

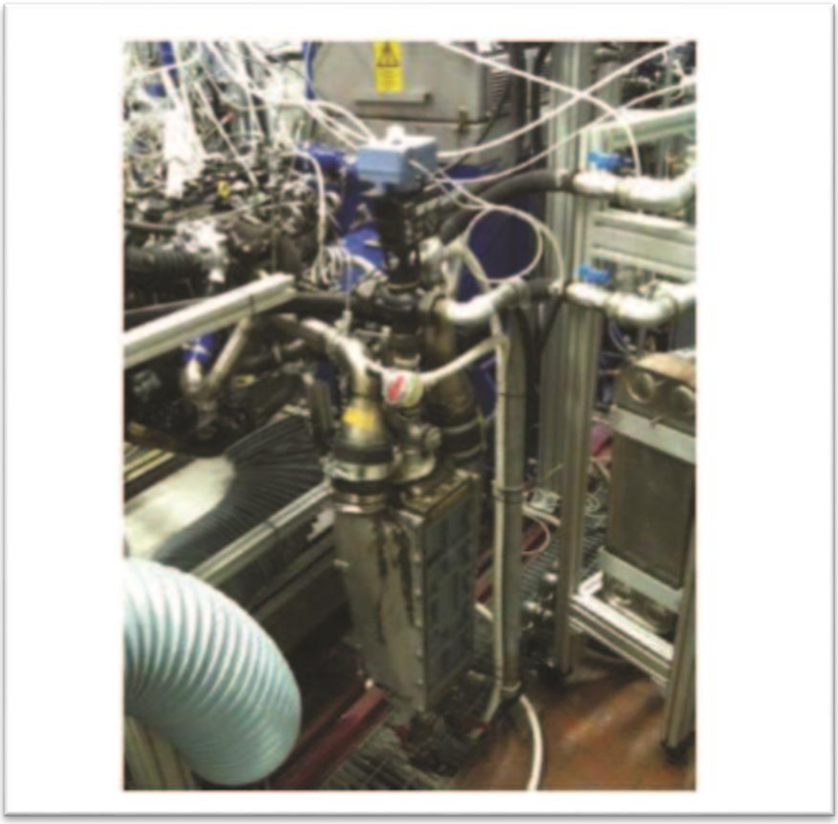

Figure 42. Compressed charge cooling system.

5.9. Engine CoolCon System

The engine to be tested is fitted on the test bed without its own heat exchanger. The engine is, for this reason, connected to a cooling system that is mainly composed by a heater, a heat exchanger, a circulation pump and a thermostat (Figure 43). While on a vehicle it is usually found an air-liquid radiator, the CoolCon system shows a conditioning system made by a liquidliquid heat exchanger, where the coolant fluid is the water directly arriving from the water piping. The coolant is coming out from the engine first enters the heater where its temperature is increased up to a well defined threshold; now the coolant enters the heat exchanger where an electrovalve regulates the coolant water flow rate in order to bring the engine coolant temperature at the desired value before it is reintroduced within the engine. The system is also provided with a level sensor and a degasser to prevent from air bubbles formation in the piping. A second piping is directly connected with the engine intake downstream the engine thermostat: through this piping the engine cooling system is fed with

Figure 43. Test bed engine cooling system

cool water in order to fasten the cooling down operation after the engine has been run on the bench. This solution allows a sensible reduction in time between two consecutive tests.

5.10. Gaseous emissions analyzer

The test engine is not provided with after-treatment systems, so an external system is used to evaluate engine-out emissions. The concentration of each pollutant species is evaluated through different gas analyzers: they compose a measurement system made by a base unit that sample's a certain volume of exhaust gases from the engine (Fig. 5.31). If regulatory tests are conducted, gas dilution in a CVS system is mandatory, while for R&D and testing operations usually raw emissions are measured without any dilution process. Once exhaust gases are sampled it is of crucial importance that no chemical reactions occur in the sampling train. It is fundamental that inert materials (teflon and stainless steel) must be used for the piping where exhausts flow. The measuring system is also provided of a conditioning system making the sampled exhausts volume to meet the analyzer specifications. The main requirement is to remove any particulate matter from the sample volume, so particulate filters have to be installed along the piping; moreover some analyzers are dry instruments, requiring the condensation and removal of water vapour. Since, as said before, the engine is fitted on the bed without after-treatment systems a throttle valve (with a well defined throttle angle) is positioned on the exhaust piping in order to reproduce the same pressure drop generated by these devices in a typical on-vehicle installation. In the dynamic cell the AVL AMA 600 is the measuring instrument used to monitor engine-out emissions: it is composed of gas analyzers that essentially measure the concentrations of CO, HC, NOx, O2 and CO2. The emissions probes are placed onto the exhaust line downstream the turbine and upstream the throttling valve; another probe is placed at the intake manifold in order to monitor the real-time EGR rate.

5.10.1. HC measurement

The FID is able to measure all the different composing species of HC thus being able of provide a cumulative value of this type of emissions. In the FID a flame generated from a constant flow of synthetic air and a gas mixture of hydrogen and helium burns inside the measuring cell. The flame burns within an electrical field between a cathode and an anode and it is blended with a constant sample gas flow. The operating principle states that the introduction of a gas sample containing hydrocarbon into a hydrogen flame will crack and ionize the hydrocarbon ions with the flame; these ions will transport a weak current between anode and cathode. The current represents the measuring signal. During the measurement of HC the temperature of exhaust gases, up to the FID, should be maintained at 190◦C since some HC compound could condense below this temperature, thus leaving a portion of HC impossible to be measured.

5.10.2. NO and NOx measurement

Usually the total amount of nitrogen oxides (NO + NO2) is measured. Since the CLD is able to measure only NO, every NO2 molecule is converted into NO before CLD, thus obtaining the NOx total amount. Measurement is based on the light from the chemiluminescent reaction of NO and O3. When a sample gas is blended with O3 in a reaction chamber a chemical reaction converts NO and O3 into NO2. Approximately the 10% of these reactions produce NO2 in an energetically excited state (NO∗ 2); the molecules return at their base state after a short time period. The excess in energy, released as photons, is the measuring signal since it is directly proportional to the NO concentration within the reaction chamber.

$$I \propto \frac{[NO][O_3]}{M}$$

Since NO2 reacts with water, any condensation should be avoided at least up to the NO2/NO converter. For this reason the entire sample gas volume is heated before the measurement. Since the CLD analyzer produces a linear response with NO concentration, a two-points calibration (zero and full scale) is periodically required.

5.10.3. CO and CO_2 measurement

These types of emissions are measured with a non-dispersive infra-red analyzer. The NDIR is composed of an emitter with a broad infra-red spectrum that sends the signal to a bipartite measuring cell. In one column (sample) of the cell is present the gas whose concentration has to be known, while the second column (reference) is filled with a non-absorbing gas (Figure 44). The infrared radiation is passed through the sample and the reference column to the detector, composed of two cells separated by a flexible metal diaphragm. The two cells are filled of the same gas to be measured: when one of the cell receives the infra-red radiation, its inner pressure increases since the absorbed energy will increase the gas temperature. However, when a sample gas is flowing within the sample column some infra-red energy is absorbed by the gas with lower energy arriving to the detector. The cell lower pressure produces an oscillation of the metal diaphragm that acts as a plate capacitor. To make the diaphragm to oscillate, in order to create a detector output signal, a copper blade driven by a synchronous motor periodically interrupts the sample and the reference energy beams in the range of 5 and

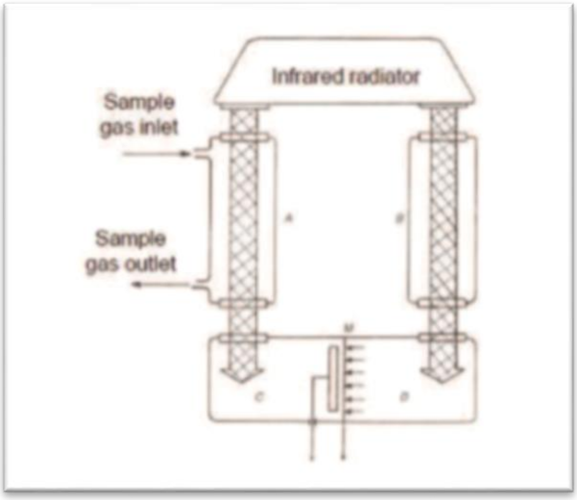

Figure 44. NDIR operating scheme

10 Hz. The diaphragm oscillation, function of the gas concentration, is converted from a variable capacitance to an AC signal. The signal is then amplified and rectified, thus obtaining a DC output signal. The signal amplitude increases as the amount of emissions to be measured increase.

5.10.4. O_2 measurement

A paramagnetic detector measures the oxygen concentration. The PMD measures the oxygen partial pressure of a gas sample by measuring its magnetic susceptibility (oxygen molecules become a temporary magnet when placed within a magnetic field). Magnetic properties of the sample volume are due to the oxygen it contains. The sample gas flows within a magnetic field and the oxygen molecules are attracted to the center of the field, where a quartz sphere is present. The instrument is symmetrical, with two magnetic fields and two quartz spheres connected by an arm. The device is mounted on a rotating axis and the oxygen molecules attempt to move the spheres from the magnetic field center. The higher the oxygen concentration the stronger the impulse on the sphere. An optical device senses the rotation of the axis and provides the voltage to control the magnetic field and to keep its rotating axis in a fixed position. This voltage is also the output voltage of the circuit and it is directly proportional to the oxygen concentration.

5.11. Particulate matter emissions

Particulate matter (PM) is a typical emission from diesel engines: PM properties and composition are function of the fuel type, engine type, engine operating conditions and after-treatment systems. Some challenges in PM measurement face with the volatile fraction of PM that is increasing with new low-emissions diesel engines. In old engines the volatile fraction condensed on the solid particulate fraction thus leading to an increasing particle diameter but no new particles were generated. Advanced combustion techniques and the mandatory use of the DPF (diesel Particulate Filter) reduce the solid fraction on which the volatile could condense. Consequently nucleation of new particles could be induced. The main parameters to quantify PM emissions are:

• Dry measurement: the volatile fraction is eliminated from the sample before entering the instrument, but the overall PM amount is not taken into account.

• Wet measurement: with this method the undiluted sample volume is cooled before entering the instrument. In this way all the material that can condense is enforced to nucleate and condense.

Critical parameters regulation imposes to take care are particle mass, particle number, particle size, particle surface area and particle composition.

• PM mass: it has been the most common parameter used by regulation in order to define emission limits from diesel engines. Today all emission standards from new diesel engines are expressed in terms of Total PM (TPM).

• PM number: PN concentration is an alternative parameter which is dominated by small nuclei mode particles. Problems can occur with the volatile fraction, since the occurring nucleation can strongly influence the PN.

• Black carbon: another parameter could be the BC, that is the solid, carbonaceous fraction of PM. However, up to now, no BC-based emission limits exist.

The measuring modes can be classified into two macro areas: the collecting one and the in-situ one.

• Collecting technique: particles are first deposited on a sampling filter and then analyzed. The gravimetric analysis is based on this principle. However the sample volumes can undergo some changes in properties with respect to their ones of the aerosol phase.

• In-situ technique: the analysis is performed directly in the aerosol phase while the gas volume is flowing through the device. Since negligible changes in sample properties are faced, this technique allows also the measurement of particle size and PN. Another advantage is the availability of continuous or quasi-continuous measurements with respect to the collecting technique.

6. Chapter VI. General conclusions, challenges and future.

- On this project it is discussed the PCCI technology as a promising system in NOx and soot emissions in forthcoming Euro VI diesel engines. It is also presented the possibility of a real application in a test bench for tuning and attaint this combustion technology.

- Three main results from the pioneering investigation of PCCI combustion have been established: first, PCCI combustion demonstrates a strong potential to improve the thermal efficiency of diesel-fuelled engines and substantially reduce NOx and soot emissions of diesel-fuelled engines; second, high amounts of EGR and injection timing strategy, injection pressure, spray impingement and equivalence ratio has a dominating role in PCCI combustion; third, difficulties associated with the successful operation of PCCI engines need to be overcome including: high HC and CO emissions and extending the operation range at high loads

- Examination of the soot and NO formation regimes on a diesel combustion equivalence ratio/temperature plot suggests that low temperature combustion has the potential to result in ultra-low soot and NO formation. Furthermore, theoretical analysis of reaction rates suggests that soot formation is insensitive to equivalence ratio at temperatures below 1500 K.

- Simultaneous reductions in exhaust NOx and soot concentrations have been observed under lean PCI combustion. NOx emissions can be avoided due to the high EGR rates and thus low combustion temperature. Furthermore, the EGR rate influences the path not only through changes in the flame temperature, but also in ignition delay and the amount of ambient fluid that must be mixed with the fuel to attain a given equivalence ratio. In addition, the injection strategies (including injection pressure, timing and multiple injections) influence the temperature during the ignition delay period, the peak flame temperature reached, and the premixing improvement. Experimental data revealing increased ignition delays and combustion durations, along with decreased rates of heat release, confirm that PCCI combustion has been attained. The resulting ultra-low concentrations of NOx and soot are the result of low-temperature combustion, a finding supported by estimated peak bulk gas temperatures that are below 1500 K. Unfortunately, increasing mixing time is seen a penalty in CO and HC emissions.

- The insensitivity of soot formation on equivalence ratio when operating within the low-temperature combustion regime has been experimentally demonstrated via measured exhaust soot concentrations at rich conditions. The possibility of rich PCI combustion creates opportunities to use diesel aftertreatment devices such as an LNT, which can be regenerated with high concentrations of CO.

- Therefore, the control and optimize of EGR rate, injection strategies and high boost are the key issue to the PCCI low temperature combustion. The PCCI has more benefits, such as high efficiency over broad load range, simple control of ignition timing, reduced pressure rise rates, high load capability. So, this strategy will be more promising in the future. Also, the lack of a dependable ignition control method detracts from fuel efficiency.

- In a longer term, engine hardware and fuels can be adapted to overcome intrinsic PCCI challenges at higher loads and higher compression ratio. A less reactive fuel is required to delay auto-ignition and phase combustion correctly. A number of low reactivity fuel blends as biodiesel blend and n-heptane [13] have shown to produce very low smoke levels.

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