Universidad Politécnica de Valencia
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Study of the association of premixed and diffusive combustion processes on the combustion and pollutant emissions in a mid-size Diesel engine

Tesis Doctoral

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Valencia, February the 5^{th} 2016

DOCTORAL THESIS

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Resumen. El objetivo principal de la Tesis es el análisis y la comprensión de la asociación de dos conceptos de combustión diferentes (combustión en premezcla y por difusión) en las emisiones contaminantes y las prestaciones en un motor Diesel de cilindrada media. La evaluación se realiza en condiciones de media y alta carga, en la cuales la implementación de una combustión premezclada es generalmente complicada.

La asociación de los dos modos de combustión es difícil de conseguir en un motor Diesel convencional, especialmente la preparación de la carga premezclada con inyección piloto adelantada. Por esa razón, el estudio se divide en dos partes principales: primero se revisa la bibliografía acerca del tema, centrando la atención en dos estrategias que permiten evitar los principales problemas evocados en la literatura, determinando su potencial para la reducción de las emisiones contaminantes. En un segundo lugar, se realiza un estudio más profundo de los procesos de combustión y de formación de contaminantes, centrándose únicamente en las estrategias de combustión parcialmente premezclada que sí tienen el potencial para reducir las emisiones contaminantes.

En la segunda parte del estudio, se aborda la asociación de combustiones premezclada y por difusión junto con la variación de parámetros de calibración convencionales como la concentración de oxígeno en la admisión (por medio de recirculación de los gases de escape), la presión de sobrealimentación y el inicio de la inyección principal, en diferentes condiciones de régimen y de carga del motor. El análisis cruzado de los resultados se realiza con el afán de entender las razones claves de los procesos que permiten la reducción de las emisiones contaminantes con esta estrategia.

Como etapa final de esta tesis, se confrontan las estrategias de combustión parcialmente premezclada estudiadas con los problemas a los cuales podrían llevar si realmente se consideraría su implementación y un motor de serie (dilución de aceite, ruido...) para finalmente concluir sobre su potencial tecnológico.

Resum. L'objectiu principal de la tesi és l'anàlisi i la comprensió de l'associació de dos conceptes de combustió diferents (combustió en premescla i per difusió) en les emissions contaminants i les prestacions en un motor Dièsel de cilindrada mitjana. L'avaluació es realitza en condicions de mitja i alta càrrega, en las quals la implementació d'una combustió premesclada és generalment complicada.

L'associació dels dos modes de combustió és difícil d'aconseguir en un motor Dièsel convencional, especialment la preparació de la càrrega premesclada amb injecció pilot avançada. Per eixa raó, l'estudi es divideix en dos parts principals: primer es revisa la bibliografia sobre el tema, centrant l'atenció en dos estratègies que permeten evitar els principals problemes evocats en la literatura, determinant el seu potencial per a la reducció de les emissions contaminants. En un segon lloc, es realitza un estudi més profund dels processos de combustió i de formació de contaminants, centrant-se únicament en les estratègies de combustió parcialment premesclada que sí que tenen el potencial per a reduir les emissions contaminants.

En la segona part de l'estudi, s'aborda l'associació de combustions premesclada i per difusió junt amb la variació de paràmetres de calibratge convencionals com la concentració d'oxigen en l'admissió (per mitjà de recirculació dels gasos d'escapament), la pressió de sobrealimentació i l'inici de la injecció principal, en diferents condicions de règim i de càrrega del motor. L'anàlisi creuat dels resultats es realitza amb l'afany d'entendre les raons claus dels processos que permeten la reducció de les emissions contaminants amb esta estratègia.

Com a etapa final d'esta tesi, es confronten les estratègies de combustió parcialment premesclada estudiades amb els problemes als quals podrien portar si realment es consideraria la seua implementació en un motor de sèrie (dilució d'oli, soroll...) per a finalment concloure sobre el seu potencial tecnològic.

Abstract. The main objective of this thesis is the analysis and comprehension of the association of two different combustion concepts (premixed and diffusive combustion) on the pollutant emissions and engine performance in a mid-size Diesel engine. The evaluation is performed at mid and high load conditions, where the implementation of premixed combustion is generally challenging.

The association of the two combustion modes is hard to attain in a conventional Diesel engine, especially the preparation of the premixed charge with early pilot injection. Therefore, the approach followed during the study has been divided in two main steps: first, the bibliography on the subject is reviewed and two strategies avoiding the main issues mentioned in the literature are grossly evaluated in order to estimate their potential for emission reduction. Second, a deeper study of the combustion processes and emissions formation is performed, focusing only on the partially premixed combustion strategies that actually have the potential for emissions reduction.

Along the second part of the study, the association of premixed and diffusive combustion is evaluated together with variation of conventional calibration parameters such as the intake oxygen concentration (via exhaust gas recirculation), the boost pressure and the start of the main injection timing, at different engine speed and load conditions. A cross analysis of the results obtained is performed in order to understand the key reasons that permit the reduction of the pollutant emissions with this strategy.

In a final part of the thesis, the partially premixed combustion strategies studied are confronted with the challenges they might face when really considered for their introduction in a production engine (oil-dilution, noise...) to finally conclude on their technological potential.

Aknowlegements.

I would like to thank the heads of the research department, Francisco Payri and José-María Desantes for giving me the opportunity to prepare my thesis in the CMT Motores Térmicos. In the same way, I would like to thank my thesis director Jesús Benajes for his guidance in my work during those years.

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Finally I'd like to thank my parents and my brother for their constant support in the whole course of my life and to extend those thanks to Mariany, with whom I shared my life for most of the time spent for the realization of this thesis. She has been the most supportive person during that period and I'll probably never be able to tell her how important she has been to me along those years.

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Acronyms

ACT	Apparent combustion time
CAD	Crank angle degree
CO	Carbon Oxide
CO_2	Carbon Dioxide
CS	Compression stroke
EGR	Exhaust gas recirculation
EoC	End of combustion
EoI	End of injection
EPI	Early pilot injection
EVC	Exhaust valve closing angle
EVO	Exhaust valve opening angle
FMI	Fraction of fuel mass injected
FMB	Fraction of fuel mass burned
HC	Hydrocarbons
HCCI	Homogeneous combustion compression ignition
LP-TDC	Low-pressure top dead center
NO_x	Nitrous Oxides (NO and NO_2)
IVC	Intake valve closing angle
IVO	Intake valve opening angle
LOL	Lift-off length
NVO	Negative valve overlap
PCCI	Premixed combustion compression ignition
RoHR	Rate of heat release
SoC	Start of combustion

SoI	Start of injection
VVA	Variable valve actuation

Chapter 1

Introduction

Contents

1.1 Introduction

Due to its high thermal efficiency and operating economy compared with the spark-ignition engine concept, the direct-injection Diesel engine is the most widely applied power plant in transportation applications, especially for medium- and heavy-duty vehicles.

However, important drawbacks arise with the conventional mixing controlled Diesel combustion, converting it in an important pollutant source for both nitrous oxides (NO_x) and soot emissions and, in a lower extend, HC and CO emissions. All those pollutants are dangerous for human health and also for the environment, leading many world governments to impose increasingly stringent exhaust emissions limits. Despite its invention being more than a century old[1], it is only in the late 1980s in the USA and in the beginning of the 1990s in Europe that the first emissions legislation for Diesel engine were created. From this date on however, the emission limits have been many time revised and largely strengthened. Following table shows the evolution of the emissions limits in Europe.

Table 1.1. *EU Emission Standards for Heavy-Duty Diesel Engines: Transient Testing. Source: Dieselnet.com.*

Stage	Date	Test	CO	HC	NO_x	PM	PN
			g/kWh				1/kWh
Euro III	1999.10 for EEV[a]	ETC	3	0.40	2.0	0.02	-
	2000.10		5.45	0.78	5.0	0.16[b]	-
Euro IV	2005.10		4.0	0.55	3.5	0.03	-
Euro V	2008.10		4.0	0.55	2.0	0.03	-
Euro VI	2013.01	WHTC	4.0	0.16[c]	0.5	0.01	$6.0 \cdot 10\,\mathrm{exp}\,11$

[a] In 1999, the EU adopted Directive 1999/96/EC, which introduced Euro III standards (2000), as well as Euro IV/V standards (2005/2008). This rule also set voluntary, stricter emission limits for extra low emission vehicles, known as "enhanced environmentally friendly vehicles" or EEVs.
[b] PM = $0.21 g/kWh$ for engines $\leq 0.75 dm^3$ swept volume per cylinder and a rated power speed $\geq 3000 rpm$
[c] THC

[1] The concept of Diesel combustion engine was invented by Rudolf Diesel in 1898

Table 1.2. *EU Emission Standards for Heavy-Duty Diesel Engines: Steady-State Testing. Source: Dieselnet.com.*

Stage	Date	Test	CO	HC	NO_x	PM	PN
			g/kWh				1/kWh
Euro I	1992, $\leq 85kW$	ECE R-49	4.5	1.1	8.0	0.612	-
	1992, $\geq 85kW$		4.5	1.1	8.0	0.36	-
Euro II	1996.10		4.0	1.1	7.0	0.25	-
	1998.10		4.0	1.1	7.0	0.15	-
Euro III	1999.10 for EEV[a]	ESC and ELR	1.5	0.25	2.0	0.02	-
	2000.10		2.1	0.66	5.0	0.10[b]	-
Euro IV	2005.10		1.5	0.46	3.5	0.02	-
Euro V	2008.10		1.5	0.46	2.0	0.02	-
Euro VI	2013.01	WHSC	1.5	0.13	0.4	0.01	$8.0 \cdot 10^{11}$

[a]In 1999, the EU adopted Directive 1999/96/EC, which introduced Euro III standards (2000), as well as Euro IV/V standards (2005/2008). This rule also set voluntary, stricter emission limits for extra low emission vehicles, known as "enhanced environmentally friendly vehicles" or EEVs.
[b]PM = $0.13g/kWh$ for engines $\leq 0.75dm^3$ swept volume per cylinder and a rated power speed $\geq 3000rpm$

The emission limits, together with the increasing pressure to reduce fuel consumption and CO_2 emissions for both economical and ecological reasons, moved forward the development of the Diesel engine. The major evolutions that were applied to the Diesel engine concept in order to reduce its pollutant emissions, not considering the important improvements brought on the mechanical side, can be separated in three main classes:

- The air-exchange process: processes by which the fresh air (or mixture of fresh air and recirculated exhaust gases) is introduced into the cylinders. This aspect of the engine was mainly marked by the introduction and evolution of the turbocharging technologies, the introduction of intercooler and that of the Exhaust Gas Recirculation (EGR) in its different variant (uncooled, cooled and low-pressure).
- The fuel injection process: process by which the fuel is introduced in the combustion chamber. The major evolution of this aspect in the latest

years was the introduction and generalization of the Direct Injection (DI) that, together with the continuous development of the injector nozzle and needle actuation technology, permitted to largely improve the mixing processes and the thermal efficiency of the Diesel engine compared to the former combustion pre-chambers technology. Later, with the generalization of the Common-Rail technology, the flexibility of the injection pattern also improved, authorizing different multiple injection strategies to be introduced.

- The exhaust gas after-treatment: process by which the pollutants emitted during the combustion process are reduced after their evacuation from the combustion chamber. This aspect was first marked by the introduction of Diesel Oxidation Catalyst (DOC) that permits the reduction of HC and CO emissions. For soot emissions, the introduction the Diesel Particulate Filter (DPF) that permits to retain and accumulate most of the particulate-matter (PM) emitted by the engine before oxidizing it outside the combustion chamber either by Continuous Regeneration Technology (CRT) effect or by thermal filter regeneration marked the major evolution. Finally, several technologies that aim at converting the NOx emissions were introduced. In the case of heavy-duty engine, mainly the Selective Catalyst Reduction (SCR) technology is used, while the light-duty applications also use Lean NOx Trap (LNT) and Passive NOx Absorber (PNA)

Efficiency of after-treatment is very high but the cost of their use during the life-time of the vehicle is not negligible and is generally proportional to the engine-out emissions level. In case of non-CRT DPF, the accumulated PM in the DPF must be oxidized. The oxidation process[2] is obtained at the cost of a large increase of the fuel consumption and its occurrence has to be limited as much as possible. In the case of NO_x, the LNT must also be emptied when it gets overloaded leading as well to a large fuel consumption increase during the so called "rich-events" required for this operation, while the SCR technology requires the injection of ammonia (*ad-blue*) in the exhaust, which also leads to extra cost.

As a consequence, the need of engine-out emissions limitation remains strong. The continuous improvement of air-exchange and injection processes

[2]This process, also called DPF regeneration process, is achieved via a large deterioration of the combustion process (combustion delay, boost reduction, richness increase) and late-post injection that do not burn inside the combustion chamber in order to create a strong exothermic process in the DOC which permits to bring the temperature of the DPF is over 600°C

permits a progressive reduction of the engine-out emissions, however the analysis of the causes of the formation of the two major pollutants of the Diesel engine (NO_x and soot) led the researchers to uncover a new combustion concept permitting to reduce largely the formation of those two species, the Homogeneous Combustion Compression Ignition (HCCI) that was first presented already in 1983 [1] while its introduction in a Diesel engine was thoroughly detailed later [2].

In the last years, many researches have been performed around this new combustion concept [3] and have shown its capability to reduce both soot and NO_xemissions even keeping some degree of inhomogeneity in the air/fuel mixture [4]. However HCCI combustion process is limited to low load engine operation and its introduction in serial engine is therefore compromised because, on one hand, an increase in local equivalence ratios results in a fast transition to knocking combustion and, on another hand, ignition timing control is hard to attain [5].

1.2 Objectives and methodology

Considering the above, it appears that the premixed combustion concept using Diesel fuel has a large potential for emission reduction but cannot be easily introduced in a commercial engine due to several strong limitations, especially at mid and high loads. Nowadays however, advanced injection devices permit new strategies based on mixing the conventional diffusion controlled combustion and the highly premixed combustion concepts by dividing injection in several events. This association could permit to extend the range of application of the premixed combustion concept while limiting/erasing the issues related to control of the combustion.

The main objective of the investigation exposed in this thesis is to determine and analyze the potential of this hybrid combustion concept based on splitting the injection and also the combustion processes into two almost independent events.

In order to successfully reach the main objective, several partial objectives have been defined:

- Understand the potential and the challenges of the association of premixed and diffusive combustion schemes in a Diesel engine through a review of the literature

- Define, implement, and evaluate advanced injection schemes that permit the introduction of a split injection strategy avoiding the main limitations identified in the literature review

- For effective injection strategies, understand the key processes that lead to the pollutant emission reduction

- Analyze the impact of the main engine calibration parameters such as the injection timing, the variation of boost pressure and the variation of the oxygen concentration via EGR on the implementation of a partially premixed combustion

- Analyze the impact of the engine speed and load on the implementation of a partially premixed combustion

- Offer an overview of the impact on Noise Vibration and Harshness (NVH) and oil dilution of the introduction of the association of split injection and combustion concept that may possibly impede its introduction in a serial application

In order to attain the defined objectives, a certain methodology was applied during the investigation. This methodology can be described together with the structure of the document that offers a view of the different steps that were followed:

- **Literature review:** The literature review covers first the characterization of the both the conventional diffusion controlled and the premixed Diesel combustion processes from the injection process to the emission formation. In a second part, the former studies based on the association of these two processes in a Diesel engine are reviewed.

- **Experimental and Simulation Tools:** The chapter describes the different tools that were selected in order to conduct the investigation.

- **Evaluation of innovative strategies to attain a partially premixed combustion process:** Based on the literature review and considering the tools selected to perform the study, a first step of evaluation of the implementation new injection-combustion strategy is performed in this chapter. The objective is then to define the strategies with best potential for further investigations.

- **Extended parametric study of the selected advanced strategies:** Once the strategies with interesting potential are selected, an extended study is performed including variation of several calibration parameters and engine speed and load. The objective of this study is the understanding of the processes that lead to emissions reduction and the discussion of the potential of the strategy for an implementation in a serial engine.

Finally, a last chapter will regroup the different conclusion of the study and discuss tehem considering the current state of engine development. Directions for further work are then proposed in order to continue with the investigations initiated with the present thesis.

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Chapter 2

Diesel Combustion Strategies and Pollutant Emissions

Contents

2.1 Introduction

This chapter presents a summary of the most relevant aspects related to direct-injection compression-ignition (Diesel) combustion. The different processes involved in different combustion strategies applied in direct injection compression-ignition engines are described. Aside from this description, the formation/destruction processes of pollutant emissions is also addressed.

The literature review presented does not pretend to expose all the features concerning every Diesel combustion strategy that has been studied by researchers around the world. The objective here is to give to the reader the background necessary to understand the motivations that has led to carry out the work presented in this thesis.

The sequence of the literature review is the following: first some considerations on fuel injection and air/fuel mixture preparation are addressed. Then the principal processes of the conventional Diesel process i.e. the diffusive combustion process are depicted. In the third section a description of diverse alternative combustion modes that aim to solve the main issues of the conventional combustion is given. Finally injection/combustion strategies that associates two modes of combustion (conventional and alternative) are detailed. Based on the considerations described in the three previous sections, in the last section of this chapter, the justification of the thesis is exposed.

2.2 Fuel Injection and Air/Fuel Mixture Preparation

2.2.1 Processes Induced by the Fuel Injection into the Combustion Chamber

In direct injection Diesel engines, liquid fuel is injected at high pressure directly inside the combustion chamber through an injector nozzle that includes various orifices. As the injection occurs, fuel starts to mix with the air[1] present in the combustion chamber following various steps. The further description will explain what occurs for one spray, however all the sprays follow the same processes.

[1]When describing the air/fuel mixture formation, air refers to the gas present inside the cylinder before the fuel injection. The composition of this gas might not be that of the air since residual burned gas from previous cycle or recirculated gas from the exhaust might be present in the mixture. However the word *air* will be used as a simplification

The first step, right at the orifice exit is known as the primary break-up. It corresponds to the atomization of the liquid vein of the fuel injected inside the combustion chamber in a collection of small droplets due to the turbulence of the flow, the instability of the flow, and the interaction of the jet with the ambient air. Due to the high injection velocity (induced by the high injection pressure) the droplets formed during the primary break-up process continue to interact with the ambient air and are further broken into smaller droplets, this phase is known as the secondary break-up. The opposite phenomenon also can occur, and two droplets that collide can be re-associated into a bigger one (this process is known as coalescence). However the predominant trend during the secondary break-up process is to a large reduction of the average size of the (still) liquid fuel droplets.

Further events will depend on the in-cylinder gas conditions. If the temperature is below that of the boiling point of the components of the fuel, the droplets will simply continue to penetrate inside the combustion chamber until they reach a wall, however this is unlikely to happen. Indeed, even if the in-cylinder conditions can vary largely depending on the engine load or the instant of the fuel injection, this last is chosen so that the temperature is high enough for the fuel to evaporate.

As a result of the primary and secondary break-up processes, the fuel is atomized in large number of very small droplets so the contact surface between the ambient air and liquid fuel is very large. This favors a very fast heat transfer process between the two fluids leading to the vaporization of the droplets. As the fuel gets vaporized, a gaseous mixture of fuel and air is formed. If the ratio between oxygen and fuel included in this mixture is close enough to the stoichiometry and if the temperature of the mixture is high enough, the mixture ignites.

2.2.2 Macroscopic Description of a Diesel Spray

Describing the air/fuel mixing process from a macroscopic point of view helps the understanding of the most important parameters influencing the morphology of the spray.

The main aspects of a fuel spray are the following:

- *Spray penetration.* It is defined as the distance from the nozzle orifice to the tip of the spray inside the combustion chamber. The penetration evolves with the time elapsed from the SoI and depends basically on the spray momentum at the nozzle exit $\dot{M}$, the opening angle of the spray

θ and the density of the ambient air ρ_a as it is detailed in the work from Desantes et al. [1] who proposed the following equation 2.1 for its calculation.

$$S(t) \propto \dot{M}^{0.25} \cdot \rho_a^{-0.25} \cdot tan^{-0.5}\left(\frac{\theta}{2}\right) \cdot t^{0.5} \tag{2.1}$$

- *Spray opening angle.* The spray opening angle is defined as the angle of the cone formed by the spray inside the combustion chamber, whose origin is the nozzle orifice. The opening angle of the spray is induced by the primary break-up process. Various jet break-up regimes exist, depending on the injection conditions. In direct injection Diesel systems, a complete atomization regime is always found, meaning that the break-up occurs right at the nozzle exit and that the opening angle of the jet depends only on the characteristic of the nozzle (mainly the conicity and diameter of the orifice geometry) and the ratio between ambient air density and fuel density. Various correlations have been proposed for the calculation of the tangent of the angle θ. Following equation 2.2 was proposed by Naber and Siebers in their extensive study [2], other researcher proposed very similar equation for the description of the spray opening angle as it can be found in references [3, 4].

$$tan\left(\frac{\theta}{2}\right) \propto \left(\frac{\rho_a}{\rho_f}\right)^{0.19} \tag{2.2}$$

- *Air entrainment.* As the spray penetrates inside the combustion chamber it forms a cone in which air is included. The entrainment of air inside the spray leads to a reduction in local fuel concentration, while local air concentration increases. The fuel concentration in the spray axis decreases with the distance from the nozzle exit (where only fuel is present) up to the tip of the spray where only ambient air is present. The same happens when considering the radial direction. The concentration of fuel is maximal in the jet axis and decreases toward the jet border. How fast is the decay in fuel concentration Y_f in the spray axis with respect to the distance from the nozzle depends basically on two parameters: the spray opening angle θ and the nozzle orifice diameter ϕ as shown in equation 2.3 that was first developed by Spalding [5]

$$Y_f \propto \frac{\phi \cdot (\rho_f/\rho_a)}{tan(\theta/2) \cdot x} \tag{2.3}$$

The evolution of the spray velocity u along the spray axis follows a similar development and is proportional to the fuel velocity at the nozzle exit u_0 as described by Ricou and Spalding [6]:

$$u \propto \frac{u_o \cdot \phi \cdot (\rho_f/\rho_a)^{0.5}}{tan(\theta/2) \cdot x} \tag{2.4}$$

- *Liquid length.* The atomization process is so fast and the size of the droplets formed is so small that the dynamic equilibrium between the two phases (fuel droplets and ambient air) is reached instantaneously from a thermal and velocity point of view. Being the heat and momentum transfer almost instantaneous, the mixture temperature and velocity at a point of the spray depends on the mass concentration of (cool and fast) fuel and (hot and quiescent) air, i.e. the air entrainment process. When enough hot air is entrained in the spray so that the temperature of the mixture reaches that of the fuel vaporization, fuel droplets evaporate. Equation 2.3 shows how fuel concentration decreases with the distance from the nozzle exit. Then considering the initial fuel and air temperatures, it is possible to calculate the length at which liquid fuel will turn into vapor i.e. the *maximum liquid length* as it was exposed in the study by Desantes et al. [7] following a similar rationale than Siebers [8]. Before the fuel spray reaches that limit, the liquid length coincides with the spray penetration (see equation 2.1) as it was remarked in the literature review by Siebers [9].

$$LL_{max} \propto \frac{\phi_o}{tan(\theta/2)} \cdot \rho_f^{0.5} \cdot \rho_g^{-0.645} \cdot T_g^{-1.73} \tag{2.5}$$

  It is interesting to note that the maximum liquid length does not depend on the injection pressure, while the spray penetration does.

- *Characteristic mixing time.* It corresponds to the time required for a fuel portion injected that follows the spray axis to reach the point where enough oxygen has been entrained into the spray so that the stoichiometric conditions are attained. This time basically depends on how fast the fuel packet travels along the spray axis and the length at which the stoichiometric conditions are found. Starting from equations 2.3 and 2.4, Fenollosa [10] establishes the relation 2.6 that, considering the fuel properties constant, gives the solution for the characteristic mixing time.

$$t_{mix} \propto \frac{\phi}{\rho_g^{0.5} \cdot u \cdot Y_{O2,a}^2} \tag{2.6}$$

Equation 2.6 is valid for inert conditions, i.e. when there is no heat released at the stoichiometric location. In reactive conditions, the high local temperature in that zone has a negative effect on the mixing process [11]. García [12] details that this is due to a reduction in local density and proposes a correction of the previous equation with a factor $(\rho_{mix}/\rho_{comb})^{0.5}$, where ρ_{mix} is the local (in stoichiometric conditions) density of the mixture fuel/gas in inert conditions and ρ_{comb} the local density of the mixture in reactive conditions. Considering a constant pressure in the combustion chamber this factor of correction can be also approximated as the square root of the ratio between the flame temperature (T_{flame}) and the unburned gas temperature (T_{ub}). The concept of the characteristic mixing time in reactive conditions was studied by Arrègle et al. [13], the authors named that parameter the Apparent Combustion Time (ACT)[2].

$$ACT \propto \frac{\phi}{\rho_a^{0.5} \cdot u_o \cdot (T_{flame}/T_{ub})^{0.5}} \tag{2.7}$$

2.3 Conventional Diffusive Combustion Process

The diffusive combustion process is the most widely applied combustion strategy in direct injection Diesel engines. In this strategy, fuel is injected close to the high pressure TDC and follows the processes detailed in section 2.2.1. The high temperature and density conditions found around the TDC leads to rapid ignition of the fuel, so that combustion process begins before the injection finishes. Important part of the fuel is burned while injection and combustion processes are overlapped, however when injection finishes, part of fuel remains unburned and gets oxidized slowly during the expansion stroke.

Diffusive combustion typically shows low levels of HC and CO emissions but leads to high formation of both NO_x and soot. NO_x are mostly formed because of oxygen availability at the periphery of the very high temperature flame while soot appear inside the fuel spray due to both high temperature and

[2]In their reasoning the authors assumed that the flame temperature is mainly function of the oxygen concentration and included the effect of heat release on the mixing process by modifying the coefficient on the oxygen concentration term instead of including the term $(T_{flame}/T_{ub})^{0.5}$

lack of oxygen. Despite the high soot formation process, diffusive combustion can show low level of soot emissions thank to strong oxidation process at the periphery of the diffusive flame.

2.3.1 Temporal and Spatial Evolution of the Conventional Diffusive Combustion

The diffusive combustion process can be temporally separated in four phases that are represented in the following figure 2.1 were injection and heat release rates are displayed together with accumulated heat release.

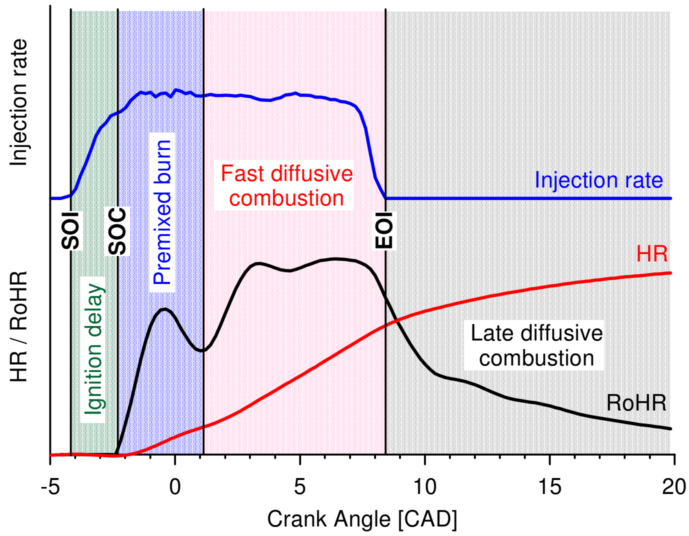

Figure 2.1. *Temporal evolution of the diffusive combustion process.*

- *Ignition delay.* It corresponds to the time elapsed from the SOI until the SOC. This time is required for the first fuel packets injected to mix enough with air to reach the local temperature and equivalence ratio conditions necessary to enable the oxidation process. The duration of the ignition delay varies in function of the oxygen concentration, the thermodynamic conditions of the air (pressure and temperature), and the chemical characteristic of the fuel.

- *Premixed burn.* The first peak observed in the RoHR corresponds to the fast oxidation of all the fuel that has been mixed enough with air during the ignition delay but has not started to burn since the thermo-chemical conditions required for auto-ignition were not attained. Dec [14] observed that in conventional conditions, the majority of the fuel getting oxidized during the premixed burn is found in locally rich zones with equivalence ratios between 2 and 4.

- *Fast diffusive combustion.* After the premixed burn, the combustion attains a quasi-steady state during which the ACT parameter defined previously in section 2.2.2 remains approximately constant. Dec [14] studied extensively the diffusive combustion process in a constant volume chamber through optical techniques and built the conceptual model of the flame structure during the fast diffusive combustion stage shown in figure 2.2. Note that the combustion process during this stage occurs in two zones: a rich (incomplete) premixed burn just downstream the lift-off that has been shown to account for approximately 10 to 15% of the heat released and a stoichiometric combustion at the periphery of the flame where the other 85 to 90% of the chemical energy of the fuel is released.

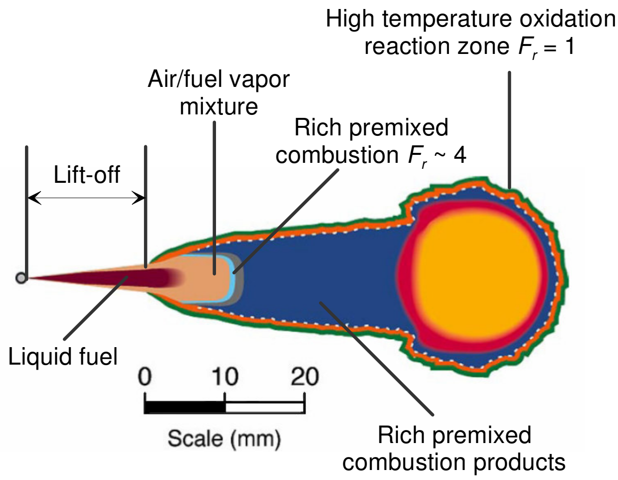

Figure 2.2. *Conceptual model of the flame structure during the fast diffusive combustion stage. Source: Dec [14].*

- *Late diffusive burn.* When the injection process finishes, the fuel and the partially burned products that are present inside the established flame remain unoxidized. Dec and Kelly-Zion [15] detailed that, as the injection process ends, the reactive jet enters in a transient stage during which its structure changes to be completely surrounded by the high temperature stoichiometric oxidation zone (the lift-off disappears). Eventually the flame structure evolves in one or two pockets still surrounded by the reactive zone and loses completely the structure of the quasi-stationary jet. The oxidation reaction ends when either all the reactants have been oxidized or when the temperature of the reaction zone drops too much (due to expansion process or exhaust valve opening) to sustain the reaction.

2.3.2 Formation and Destruction of Pollutant Emissions in Diffusive Combustion

2.3.2.1 Nitrous Oxides Formation

Emissions of nitrous oxides (NO_x) in Diesel engines are composed in a large majority of NO but also, to a much lower extend, of NO_2. The percentage of NO_2/NO varies depending on the engine and the operation point between 10 and 30%. NO and NO_2 are usually grouped together as NO_x for their measurement since emissions regulation is currently set for the sum of the two species.

Three mechanisms lead to the formation of NO_x; *Thermal*, *Prompt* and *Fuel*. It is generally accepted that in combustion of near stoichiometric mixture the mechanism *Thermal*[3] is the most relevant. The rate of NO_x formation via this mechanism shows an exponential dependence on temperature, while oxygen and nitrogen concentrations play a less important role [18].

Dec and Canaan [19] studied extensively via optical techniques the formation of NO_x during the diffusive combustion process. They stipulated that during the premixed combustion phase the equivalence ratios of the reacting zones are too high for NO_x to be formed. The authors observed that most of the NO_x is formed during the fast diffusive combustion phase, just outside the flame, where the temperature is maximal and there is both presence of O_2 and N_2. The formation of NO_x then continues during the late diffusive stage since the high temperature reactive zone still exists, however the rate of formation decreases rapidly because of the temperature drop due to the expansion process.

[3]This mechanism is often referred as the extended Zeldovich's since Zeldovich [16] established the firsts two reactions of the mechanism while Lavoie et al. [17] complemented it with a third reaction later on.

Plee et al. [20, 21] studied the influence of the engine operating conditions on the emissions of NO_x and found that they can be well correlated with the peak of adiabatic stoichiometric flame temperature alone despite important changes in load, speed or intake oxygen concentration. In his experimental work, Molina [22] observed a similar behavior. In order to ascertain the dependence of NO_x emissions on flame temperature whatever the oxygen concentration, in the same work Molina went further and used an equilibrium model to compare the NO_x concentration in different cases where the flame temperature was maintained constant and the oxygen concentration was varied (this implies that the temperature of the reactants was adjusted). The author repeated the simulation for two flame temperature levels and showed that if the NO_x concentrations were largely affected by the flame temperature (an increase of 4.3% in flame temperature resulted in an increase of 28% in NO_x concentration), the dependence of NO_x formation on YO_2 was negligible (a diminution by 21.6% of the oxygen concentration resulted in a reduction by 0.6% in NO_x concentration). This highlights how the NO_x formation process is highly dependent on the temperature and much less on the concentrations of oxygen and nitrogen despite the fact that their presence is required for the formation process to occur.

2.3.2.2 Soot Formation

Soot processes in conventional diffusive combustion has been extensively studied but remain until now not completely understood due to their complexity. Five processes are commonly identified in the soot formation from the injected fuel: pyrolysis, nucleation, coalescence, surface growth and agglomeration as shown in figure 2.3.

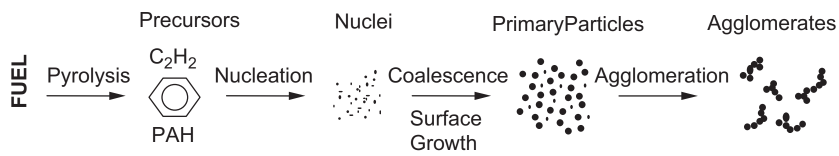

Figure 2.3. *Processes involved in soot formation. Source: Tree and Svensson [23].*

Pyrolysis is a generally endothermic reaction that undergo all fuels in presence of high temperature and lack of oxidant. Through this reaction, the molecular structure of the fuel is altered and produces species such as unsaturated hydrocarbons, polyacetylenes, polycyclic aromatic hydrocarbons (PAH) and acetylene that are the precursors of the first solid particles that are formed during the nucleation process. The particle nuclei formed during nucleation only represent a small part of the total soot mass, but they have a significant influence on the mass added later since they provide sites for surface growth. Surface growth is the process during which soot particles adsorb gas-phase hydrocarbons and see their mass increasing largely while agglomeration is the process by which particles combine to form large groups of primary particles.

Summarizing, the mass of soot formed is the result of: first, the formation of precursors due to the coincident presence of high temperature and lack of oxygen that lead to the formation of nuclei and, second, the growing of those nuclei, promoted by long residence time and high temperatures.

The premixed burn takes place in zones with relatively rich equivalence ratios. Higgins et al. [24] state that soot starts to form when the local temperature raises sufficiently, instants around the peak of heat release of premixed burn. Dec et al. [25] performed visualization of soot during the premixed burn and observed that the small particles of soot are formed during this process throughout the entire zone where the reaction takes place.

At the end of the premixed burn, the soot formed is distributed in the leading portion of the spray, with a higher concentration toward the front and a lower concentration upstream. As the quasi steady diffusive flame (figure 2.2) is established, Dec [14] states that soot is distributed inside the volume of the flame with low to high concentration from the location just downstream the lift-off until the tip of the flame.

Going more into the details, Flynn et al. [26] speculate that during the fast diffusive burn hydrocarbon fragments (soot precursors) are formed as the fuel passes through the rich premixed burn zone (see figure 2.2), the high temperature and oxygen less ambiance found inside the spray plume after that point is ideal for those elemental building blocks to turn into nuclei and growing soot particles. Kosaka et al. [27] have found more recently the same results and stated that the soot precursors are formed through the central fuel rich region of the spray surrounded by OH (i.e. by the high temperature oxidation reaction). The young soot formed then grows by surface growth and coagulation during convection to the spray head.

During the late combustion process, the lift-off disappears and no more soot precursors are formed. On the other hand, particle growth process continues and it is shown by Kosaka [27] that the maximum concentration of soot is found just after the EOI.

2.3.2.3 Soot Oxidation

Soot oxidation is the conversion of carbon into CO or CO_2. The oxidation process can take place at any time of the soot formation if oxidants are available. The most active specie for soot oxidation depends on the conditions of the mixture. It is stated by Bartok and Sarofim [28] and also by Heywood [18] that the most important specie in fuel rich and stoichiometric conditions is OH radical, while in fuel lean conditions soot is oxidized by both OH and O_2.

In conventional diffusive combustion, the great majority of the fuel burns either in locally rich or stoichiometric conditions. OH attack on soot for its oxidation is hence thought to be the predominant mechanism. Bergman and Golovitchev [29] mentioned that the formation of OH requires both high temperature and presence of O_2, and, according to the work presented by Fujimoto et al. [30] the rate of formation of OH increases exponentially with the temperature.

In a study performed by Dec and Coy [31] the authors observed with optical techniques the evolution of OH concentration during the premixed burn and the early stages of the fast diffusive combustion. Their results depicted that OH radicals are only present in very small concentration during the premixed burn (due to the lack of oxygen during this locally rich combustion). This implies that soot oxidation during the premixed burn is weak, if existent.

The authors later explained that as the combustion enters the fast diffusive phase, a thin layer with very high concentration of OH radicals surrounds the flame, just outside the location where the mixture is stoichiometric i.e. where the main oxidation reaction takes place. Similar observation was performed by Kosaka et al. [32]. This result is in agreement with previous consideration on OH formation since at the location of the stoichiometric reaction zone the temperature is maximum and, on the outside of the flame, oxygen is available.

In the reaction speculation for a free jet presented by Flynn et al. [26], the authors assumed that the thin layer of high OH concentration around the flame permits the total oxidation of the soot formed inside the jet plume since during the fast diffusive combustion, no soot signal is detected on the outside of the flame[4].

Dec and Tree [34] performed visualizations of a diffusive flame confined in an engine combustion chamber, with the flame impinging the piston wall.

[4] The observation that all the soot is distributed inside the flame in diffusive combustion was not a new result. It was observed for instance in works performed by Dec [14, 33] and also by Kosaka et al. [32]

They reached the conclusion that even in those conditions, no soot is able to cross the OH layer. However they noted that some soot is deposited onto the piston wall, on the location of the flame/wall impingement. In a similar work, Tree and Dec [35] explained that the amount of soot deposited on the cylinder wall varies largely depending on the engine condition and also that the soot deposited in previous cycles is oxidized with the new spray impingement. They finally compared the net soot deposition (the soot deposited minus the soot oxidized) with the exhaust soot emissions measurements and found that the two values did not correlate meaning that the soot wall-deposition is not the source of the soot found in the exhaust.

Flynn et al. [26] argued then that the soot found in the exhaust is due to the quenching of the oxidation reaction in the last stages of the combustion process. Dec and Kelly-Zion [15] performed a specific study on the evolution of soot and OH during the late diffusive stage. They found that at the beginning of this stage, OH surrounds the soot cloud still impeding that any soot goes out of the flame. As the soot burnout continues, the main cloud splits into smaller ones still surrounded by OH radicals, however the distribution of OH becomes broader in the burned gases. Authors showed that for early injection timing and high oxygen concentration, all the soot is oxidized before the OH disappears. However, if the injection timing is retarded or if the oxygen concentration is lowered, the soot can survive the combustion process in two cases: 1-the exhaust valve opens before the soot burnout is completed or 2-the temperature of the stoichiometric reaction zone becomes too low so that OH concentration around the sooty product becomes weak and some soot is able to cross the oxidation reaction layer.

Authors concluded comparing the changes in the late soot burnout with the evolution of the soot emissions results in the exhaust and found a strong correlation suggesting that incomplete soot burnout is the major contributor to the soot emissions in conventional diffusive combustion.

2.3.2.4 The Conventional Diffusive Combustion Dilemma

Previous sections 2.3.2.1 and 2.3.2.3 highlighted the importance of the flame temperature in the formation of NO_x and the oxidation of soot in conventional Diesel combustion. High flame temperature condition leading to high concentration of OH, permits a strong soot oxidation process and results in low soot emissions but at the same time, it promotes a strong NO_x formation process. Any strategy based on reducing the flame temperature in order to weaken NO_x formation process affects the efficiency of the soot oxidation appearing, thus, the so called conventional diffusive combustion dilemma: a trade-off between NO_x and soot emissions.

Bergman and Golovitchev [29] produced *Equivalence ratio - Temperature* maps of chemical species concentration during the combustion process, and pointed-out explicitly (see figure 2.4 reprinted from their work) that the dilemma finds its origin in the concordance of the conditions required for NO_x and OH formation.

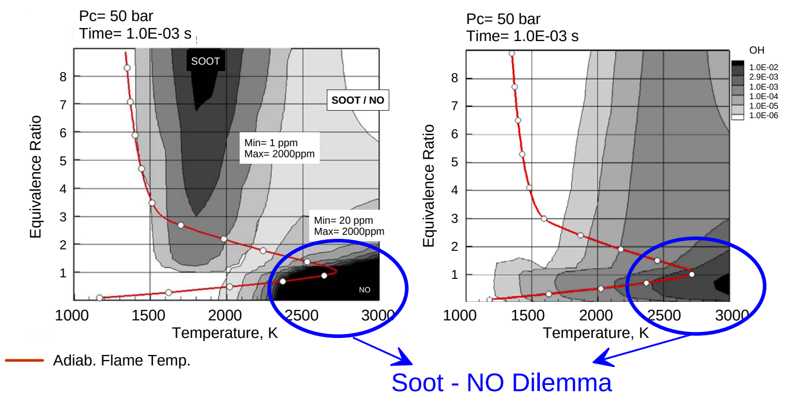

Figure 2.4. *Coincidence of the Equivalence ratio - Temperature regions for the formation of OH and NO leading to the conventional diffusive combustion dilemma. Source: Bergman and Golovitchev [29].*

Eismark et al. [36] compared the evolution of the NO_x formation and soot oxidation rates with temperature and found that NO_x formation rates drop much faster than rates of soot oxidation as the temperature decreases suggesting that there is a temperature "window" where still strong soot oxidation and yet weak NO_x formation conditions are attained. Authors later explained that this trend permits efficient soot oxidation process during the late-diffusive stage while NO_x formation becomes insignificant and stated that a fast late-diffusive stage in diffusive combustion is a key to reduce soot emissions without increasing NO_x emissions.

2.4 Premixed and Low Temperature Combustion Strategies

Considering the fact that conditions for strong soot oxidation lead to high NO_x formation, avoiding the soot formation appears to be a solution for solving the conventional Diesel combustion dilemma. The no formation of soot would make unnecessary its oxidation and would allow reducing the flame temperature to reduce NO_x emissions without drawbacks in soot emissions.

As explained in previous section 2.3.2.2, soot precursors formation is mainly due to lack of oxygen and high temperature conditions in the zone of the premixed burn of the diffusive flame while the conversion of the precursors into growing soot particles is promoted by high temperature and long residence time. It appears so that there are two possibilities to avoid the formation of soot: reducing the temperature below the threshold of formation and/or avoiding that the combustion reaction takes place with locally rich equivalence ratios.

2.4.1 HCCI Strategy

2.4.1.1 Main Features of the Process

HCCI (Homogeneous Charge Compression Ignition) strategy is the most extreme one to solve the NO_x-Soot dilemma. In this strategy, a homogeneous charge has to be formed before it auto-ignites due to compression temperature. The injection and combustion processes are thus totally separated. The homogenization of the charge before its ignition permits that no rich zone is found during the combustion, impeding thus the soot formation process to take place while the distributed lean equivalence ratios result in low flame temperature inhibiting strong NO_x formation.

In order to obtain a homogeneous mixture, a very large ignition delay is required and fuel needs thus to be injected very early in the cycle. Two options are commonly used for the preparation of the homogeneous mixture: port fuel fumigation and early direct injection.

The resulting combustion process shows generally two distinctive phases (see figure 2.5): a first relatively weak peak of heat release corresponding to low temperature reactions (also called cool and blue flames) followed by a higher peak corresponding to a higher temperature reaction.

The cool and blue flame regime is activated when the bulk gas temperature reaches approximately 700-750K, according to a work by Zheng [37] and accounts for 7 to 10% of the total HR so its intensity depends on the amount of premixed fuel. The low temperature reactions consist in the thermal cracking of the fuel molecule. As it is widely described by Kosaka [38] this correspond to the removal of H atoms from *alkane* to form various radicals. This process being exothermic (the RoHR rises), the charge is further heated up leading to

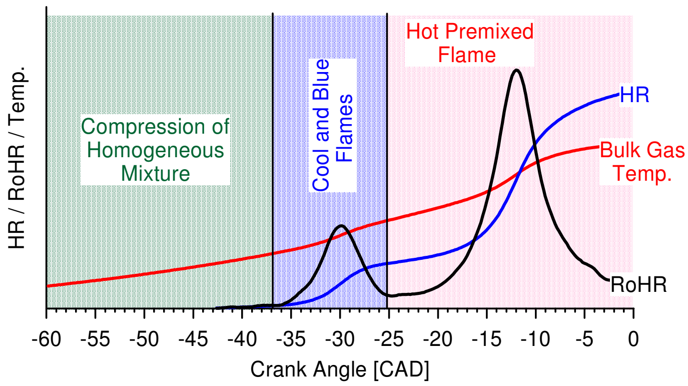

Figure 2.5. *Temporal evolution of HCCI combustion process.*

chain propagation reactions that reverse the rate of radical formation. Hence, although radicals keep forming, the RoHR eventually decreases.

During the cool and blue flame stage, the bulk gas temperature keeps increasing due to the heat released and the piston compression. When the appropriate temperature is reached (around $920K$ [37]), various chemical reactions[5] are enabled leading to the combustion process of the premixed fuel i.e. the hot premixed stage of HCCI combustion. Once the combustion process is activated, the rate of the reaction depends on the temperature evolution and thus indirectly on the equivalence ratio and the EGR rate [40].

In some cases, a third stage in the rate of heat released appears in HCCI combustion. It corresponds to the conversion of CO formed during the initial part of hot premixed stage into CO_2. Sjöberg and Dec [41] studied the influence the temperature on completeness of HCCI combustion process and concluded that the conversion of CO into CO_2 was promoted by the presence of OH-radicals and was hence strongly temperature dependent. It starts when the bulk gas temperature of $1200K$ is attained and is complete if temperature reaches $1500K$.

2.4.1.2 Drawbacks of HCCI Strategy

Despite its great potential for low NO_x and soot emissions combustion and relatively high efficiency, HCCI combustion with Diesel fuel suffers from still unresolved issues.

[5]Details of the chain of reactions are presented in the work of Aceves et al. [39]

First of all, the complexity of the mixture preparation that is inherent to the use of Diesel fuel. Epping et al. [42] stipulated in their work that Diesel fuel is particularly non-suitable for HCCI operation since it is difficult to evaporate so its injection needs to be performed in a high temperature ambiance. On the other hand, the necessity of long ignition delay to get a good homogenization of the charge implies that the injection occurs early in the cycle. These two aspects led to the installation of intake heater system upstream of the injector [43] in case of port fuel injection thus reducing the volumetric efficiency. For cases where early direct injection is used to form the homogeneous mixture, the low temperature found during the compression process leads usually to poor fuel evaporation and subsequent piston/cylinder wall wetting [44, 45] that can lead to locally rich mixture during combustion [46]. This inhibits the breakthrough of HCCI combustion and causes oil dilution and hardware wear [47, 48].

Combustion phasing and control is another unresolved challenge of HCCI strategy. The HCCI combustion process being controlled by chemical kinetics, it is highly affected by the ambient temperature whose evolution depends, among others, on compression ratio, intake temperature, EGR rate and injected fuel amount. Aroosrisopon et al. [49] observed that the application of HCCI combustion for a conventional Diesel engine with relatively high CR was limited to a small region in low engine load range. Out of this region, if load is increased, HCCI causes a fast transition to knock combustion, while if the engine CR is reduced and high amounts of EGR are used to extend the application of HCCI strategy to slightly higher load as it was performed by Ryan and Callahan [43], then at idle and very low load conditions, misfire is often found. Since the extension of HCCI combustion to whole engine operation range cannot be reached, in some cases a halfway solution was used, with the engine running in conventional combustion mode in cold start conditions and at mid to high load and operating in HCCI combustion in a restricted range [42]. The control over transient is also a challenge that can only be overcome through expensive solutions such as fast thermal management, variable valve timing and variable compression ratio as it was shown in different studies [50–52].

Regarding pollutant emissions, if HCCI combustion is very efficient to avoid NO_x and soot formation, it is the source of very high HC and CO emissions compared to conventional combustion. Dec et al. studied extensively the origin of the unburned species and concluded that they mainly appear under very light load conditions due to the quenching of the oxidation reaction. In those conditions, bulk-gas quenching occurs because of the low combustion temperatures produced by the very diluted homogeneous mixtures. Aceves et al. [53] also studied the cause of unburned or partially burn HC via multi-zone models and stipulated that wall cooling effect also plays an important role in the completeness of the combustion reactions i.e. the parts of the homogeneous mixture that are found close to the walls often display partial reaction or even do not react due to the locally lower temperature. The high HC-CO levels found with HCCI combustion imply the utilization of exhaust emission control devices and although the catalyst of those species is well understood, the cooler exhaust temperature of HCCI engines would likely lead to a decrease in the average efficiency of the commonly used devices [42].

2.4.2 PCCI Strategy

Following Flowers et al. [54], *The phrase "Premixed Charge Compression Ignition" or "PCCI" can be used to describe this class of combustion processes in which combustion occurs similarly to HCCI engines as a non-mixing controlled, chemical kinetic dominated, auto ignition process, but the fuel, air, and residual gas mixture need not be homogeneous.* The evolution of the combustion process in PCCI combustion shows thus the same phases as the HCCI combustion that were described in previous section.

However, the mixtures inhomogeneities found in PCCI combustion affect the ignition and emissions formation process. PCCI conditions can be separated in two classes regarding mixture inhomogeneities: conditions where local equivalence ratios are kept below the stoichiometry and conditions in which local equivalence ratios goes from lean to rich mixtures. PCCI strategies can also be separated regarding the injection timing; from extremely early/relatively early injection PCCI strategies to late injection PCCI strategies.

2.4.2.1 Main Features of the Different PCCI Strategies

Flowers et al. [54] studied the influence of the charge non-uniformity in lean local equivalence ratio cases and concluded that compared to completely homogeneous mixture, stratified mixture resulted in a slight advance in combustion timing and a slightly more rapid combustion process. However, opposite trends were found by Yu et al. [55] that found that the ignition process was smoothened as the charge was stratified.

Regarding pollutant emissions it appears in the study by Flowers [54] that HC and CO are not strongly affected by the charge non-uniformity however it was clearly identified that NO_x emissions increase with increased stratification of the charge. This is a consequence of sharp augmentation in local temperature during the second stage of the combustion that is found as the local equivalence ratio increases.

Indeed, if no EGR is used, Flynn et al. [56] have shown that local equivalence ratio as low as 0.5 is required to keep flame temperature around $2000-2100K$ where NO_x formation is largely limited. This implies that in this strategy, low NO_x emissions can be attained only in relatively light load conditions and with very early fuel injection timing that provide sufficient mixing time to create lean enough mixture.

With EGR dilution however, it is possible to reduce NO_x emissions even if fuel-lean conditions are not attained before combustion. In this case, the premixing of the charge becomes a mean to limit soot formation, which is avoided with much less premixing compared to NO_x formation. According to different studies [18, 57, 58], the equivalence ratio threshold value below which soot formation is inhibited is approximately 2. This feature facilitates largely the introduction of PCCI strategy since it allows injecting fuel much later in the cycle compared to an HCCI case, typically 20 to 40 CAD before the top dead center.

Musculus [59] studied extensively the evolution of the PCCI combustion process with low intake oxygen concentration (12.7% in volume) simulating high amount of EGR and relatively close to TDC injection (SoI timing set 22^{o} bTDC). Applying several optical technique, he was able to build up the conceptual model for early PCCI combustion shown in figure 2.7 corresponding to a typical early PCCI injection/combustion process whose injection rate and RoHR traces are shown in figure 2.6.

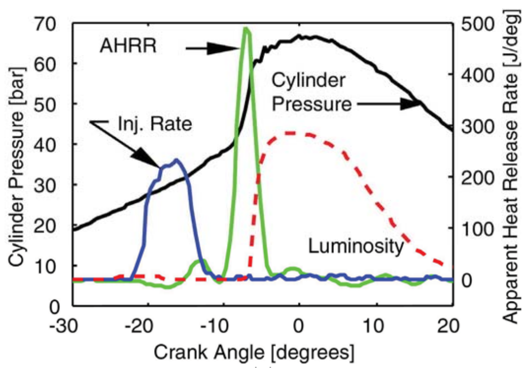

Figure 2.6. *Temporal Evolution of a Typical early PCCI injection combustion process. Source: Musculus [59].*

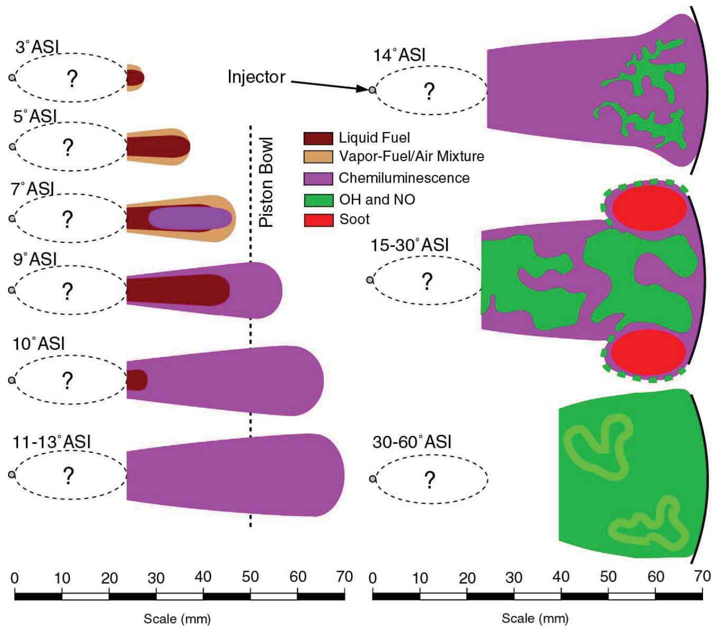

Figure 2.7. *Conceptual model for PCCI combustion in heavy-duty engine with low oxygen concentration and early injection. Source: Musculus [59].*

The author explains that with this strategy, because of the still early injection timing and consequent low temperature and density in the combustion chamber, extended liquid length (approximately twice as long as the liquid length for conventional injection/combustion) found before ignition can potentially wet in-cylinder surfaces. However, the cool and blue flame stage (observed through chemiluminescence) that is activated upstream the tip of the jet, overlaps fuel injection and likely contributes to fuel vaporization, limiting the wall wetting phenomena. Later, the second stage of the combustion is activated (chemiluminescence signal becomes stronger) and OH appears when the peak of HR is attained, marking the zones where the equivalence ratio is close to the stoichiometry. It is highlighted that, contrarily to conventional diffusive combustion where OH is located in a thin sheet around the sooty products, in PCCI conditions OH is broadly distributed thorough the fuel jet. Also, the soot that is formed[6] in the regions where equivalence ratio is above 2 shows a different location compared to

[6]It is important to note that in his study the author stipulates that *"the SOI was purposely adjusted to the threshold level for soot formation"* i.e. if the SoI would have been further advanced, no soot would have been formed.

conventional diffusive combustion; instead of being formed upstream in the jet, at the lift-off length, it appears located mainly at the tip of the jet and in the head vortex.

Musculus also performed measurements in a light-duty and was able to establish a similar conceptual model for this case [60], that is presented in the following figure 2.8.

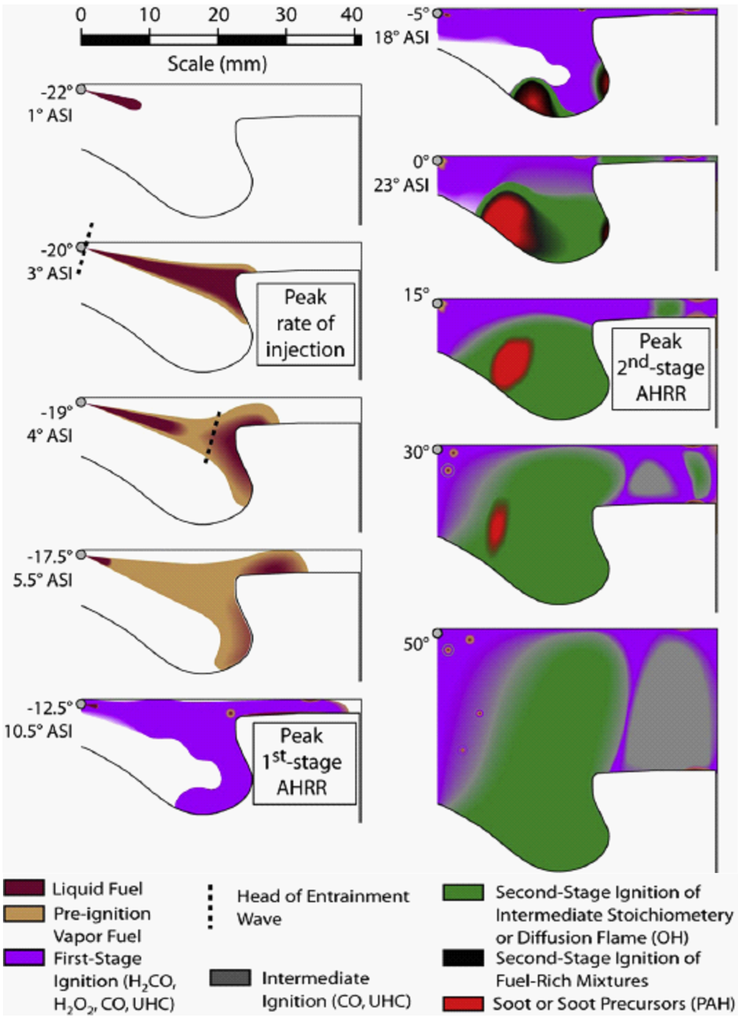

Figure 2.8. *Conceptual model for PCCI combustion in light-duty engine with low oxygen concentration and early injection. Source: Musculus [60].*

The evolution of soot emissions over a wide range of SoI timing in similar PCCI conditions was studied by Benajes et al. [61, 62]. It was shown that for advanced injection timings (SoI set before 27^o bTDC) important soot emissions were found, probably due to important piston wall wetting and subsequent pool fire as it was also found in a study by Kiplimo et al. [63]. Low soot emissions were attained for SoI timing around 24^o bTDC. From that point, if SoI was delayed, soot emissions increased due to stronger formation and weaker oxidation process. It was argued that this increase in soot formation when delaying SoI is due to a reduced mixing time between EoI and SoC that leads to higher mixture mass under rich enough conditions (equivalence ratio > 2) to form soot during the high temperature combustion process. Indeed, as shown by Musculus et al. [64], local maximum equivalence ratio reduces very fast after the EoI, and a slight change in time elapsed between EoI and SoC will results in important differences in mixture mass with equivalence ratio > 2 during the combustion process.

A maximum in soot emissions was observed for SoI timing around 9^o bTDC. For further delayed SoI, soot emissions begin to reduce again, as ignition delay increases and combustion temperature decreases. This so called "late PCCI" or "MK" (Modulated Kinetic) strategy was widely studied by Kimura et al. [65, 66]. Like in early PCCI strategy with locally rich mixture during combustion, MK combustion strategies uses high rate of EGR to reduce NO_x and large premixing time to avoid soot formation.

2.4.2.2 Limitation of the Different PCCI Strategies

The limitations of lean mixture PCCI combustion are basically the same as those of HCCI strategy (combustion control, HC and CO emissions, oil-dilution...). PCCI strategy with locally rich mixture together with high rate of EGR, that has also a strong potential for low NO_x and soot emissions, does not suffer those issues. Compared to HCCI combustion, this strategy offers a much better control of the SoC since it is related to SoI timing [61]. Furthermore, under PCCI combustion it is possible to attain relatively high engine efficiency, with much lower HC and CO emissions compared to HCCI strategy because it limits over-lean mixture (premixing time is much lower) and shows higher combustion temperature.

However, even at low-load conditions, due to fast and early combustion process, early PCCI strategy shows high level of noise for high engine efficiency operation points [67, 68]. Late PCCI strategy seems to be able to limit this issue [66] but the use of such strategy at higher load (with settings that permit to maintain low NO_x and soot emissions) shows dramatic drop in fuel efficiency [69]. In the case of early PCCI, Benajes et al. [70] showed that low NO_x and soot emissions together with high engine efficiency can be attained at mid-load, but in this case combustion noise becomes the limiting factor.

2.4.3 Non-Sooting Diffusive Combustion Strategies

Contrarily to HCCI or PCCI strategies, non-sooting diffusive combustion strategies are not based on decoupling injection and combustion processes to avoid soot formation. In those strategies, the combustion is mixing controlled, as it is the case of conventional Diesel combustion, however the formation of soot in the flame is avoided either by drastically reducing the combustion temperature and/or favoring high oxygen entrainment in the lift-off so that the (rich) premixed burn just downstream the lift-off is not rich enough to form soot.

2.4.3.1 Non-Sooting Diffusive Combustion through Extreme Combustion Temperature Reduction

Akihama et al. [58] performed engine tests at low-load and showed that the realization of a "smoke-less rich" combustion was possible by using large amount of cooled EGR. The authors performed a detailed analysis of the evolution of combustion process using this strategy and stated that the mixture formation is similar compared to conventional Diesel combustion but with largely reduced temperatures. They hence concluded that *"the smoke suppression is realized by the combustion taking place at temperatures below that needed to form soot. In such lower temperature region, the soot formation itself can be suppressed because the reactions forming soot particles from PAH do not progress even if the rich combustion occurs"*.

Pickett and Siebers [71] reach a similar conclusion when studying a non-sooting diffusive flame in a constant volume vessel, using a nozzle hole orifice diameter representative of a heavy-duty applications and varying the oxygen volume concentration from 21 to 5%. They observed that soot formation was avoided for oxygen concentration below 8% despite rich equivalence ratio at lift-off (equivalence ratio at lift-off was higher than 5) and concluded that the temperature of this mixture was too low for soot inception.

In both previously cited works, it is mentioned that for the cases where soot formation was avoided due to extremely low combustion temperature, NO_x formation was also largely reduced since flame temperature was around $2000K$

thank to low oxygen concentration. This strategy hence yields the possibility for low NO_x and soot combustion and avoids the issues of the decoupling of injection and combustion processes.

Benajes et al. [72] successfully implemented this strategy in a light duty engine at mid-load ($IMEP \approx 9bar$); however they showed that the use of very high boost pressure ($\approx 3bar$) together with high EGR rate ($\approx 70\%$) were necessary at such engine load to reach low enough intake oxygen concentration maintaining lean overall equivalence ratio. Using a similar boost pressure level, Wakisaka et al. [73] intended to reach extremely low temperature combustion in a small bore engine at higher load ($IMEP \approx 16bar$) and showed that if low NO_x emissions were attained, soot inception was not avoided and engine out soot emissions were found to be unacceptably high.

Aside from the limitation regarding boost pressure, in his work, Amorim [74] found that as the non-sooting diffusive combustion regime was attained with a conventional injection system reducing intake oxygen concentration from 12 down to 9%, the combustion process was slowed down and HC/CO emissions increased largely (approximately 6 fold increase) while IMEP reduces from 9 to $8bar$. Modifying the injection system (reducing nozzle hole diameter and increasing injection pressure) and using moderate intake temperature, he nevertheless found that non-sooting diffusive combustion was attained more easily, i.e. a reduction of intake oxygen concentration down to 10% was enough to produce a soot free combustion, limiting but not eliminating the previously mentioned drawbacks in term of HC/CO emissions and engine efficiency.

The nozzle hole diameter reduction, injection pressure increase and in-cylinder gas temperature reduction are three measures that enhance oxygen inclusion in the lift-off [57] i.e. that reduces the equivalence ratio of the premixed burn reaction that occurs just downstream the lift-off during the fast diffusive stage of the combustion. This strategy, that has also the potential of reducing the soot formation in diffusive combustion is detailed in the following section.

2.4.3.2 Non-Sooting Diffusive Combustion through Extreme Oxygen Entrainment in the Lift-Off

Siebers and Higgins [57] experimentally demonstrated that soot formation tends to decrease as more oxygen is entrained in the lift-off and found that equivalence ratio value of 2 at lift-off is a threshold below which soot formation is inhibited. Going more into the details, Idicheria and Pickett [75] explained that, when equivalent ratio at lift-off is reduced, the premixed burn zone of the diffusive spray (defined in section 2.3.2.2) and the zone where PAH (main soot precursors) are formed start to split. In this case, PAH are formed further downstream and in a lower amount leading to a reduction in soot formation. If the equivalence ratio at the lift-off is further reduced, the formation of PAH is avoided, inhibiting by the way completely the formation of soot.

The parameters that affect the equivalence ratio at lift-off were extensively studied by Siebers et al. [57, 76, 77] and Pickett et al. [78]. Novella [79] and also Amorim [74] performed extended reviews of the previously cited works and summarized that the main parameters affecting the equivalence ratio at lift-off using a conventional Diesel fuel are the following:

- *In-cylinder gas temperature.* A reduction in ambient gas temperature increases the lift-off length with no effect on gas entrainment hence resulting in reduced equivalence ratio at lift-off.

- *In-cylinder gas density.* An increase in ambient gas density leads to an increase in gas entrainment and a decrease in lift-off length. The net effect is a slight increase in the equivalence ratio at lift-off.

- *In-cylinder gas oxygen concentration.* A reduction in ambient gas oxygen concentration results in an increase in lift-off length with no effect on gas entrainment; the amount of gas entrained in the lift-off hence increases. However, this increase in *gas* entrained in the lift-off is compensated by the reduction of the gas oxygen concentration so that the amount of *oxygen* entrained in the lift-off is maintained approximately constant i.e. the equivalence ratio at lift-off is not affected.

- *Injection pressure.* An increase in injection pressure increases injection velocity and lift-off length with no effect on gas entrainment, leading to a reduction in the equivalence ratio at lift-off.

- *Nozzle hole diameter.* A reduction in nozzle hole diameter reduces the lift-off length but increases largely the ratio of the amount of gas entrained with respect the fuel mass injected. The net effect is a reduction of the equivalence ratio in lift-off.

Basing their experiments on the previous considerations, Pickett and Siebers [71] showed how mixing controlled soot-less combustion with pure air ($\approx$ 21% oxygen concentration per volume) and conventional Diesel fuel in a constant volume vessel, under many different ambient gas conditions, could be achieved using various nozzles for a constant injection pressure of

1380bar. The following figure 2.9 reprinted from their work shows for each nozzle tested where the limit between sooting and non-sooting conditions is found in function of the ambient gas density and temperature.

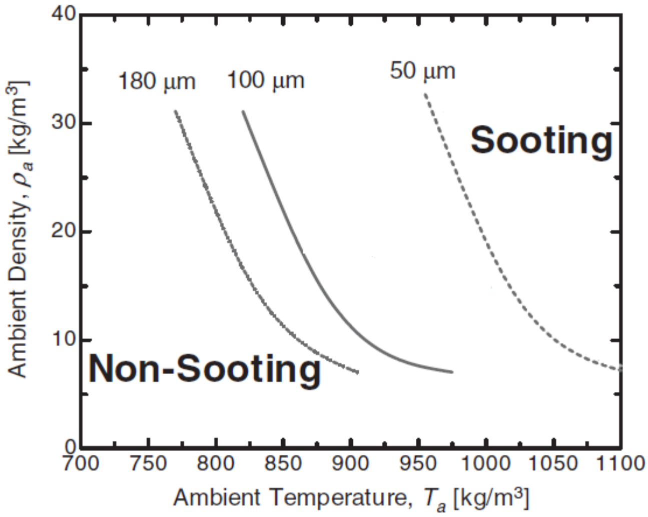

Figure 2.9. *Limit between sooting (to the right of each curve) and non-sooting (to the left of each curve) conditions as function of ambient gas density and temperature for single jets in constant volume chamber. Source: Pickett and Siebers [71].*

Their results show that when using a conventional Diesel fuel either the use of micro-hole nozzle (conventional nozzle hole diameter is 180 μm in heavy-duty Diesel engines) or a large reduction of the ambient gas temperature (conventional ambient gas temperature in Diesel combustion condition is approximately 1000K) is necessary to reach non-sooting regime.

Musculus [80] showed, comparing the results in term of lift-off length obtained in a constant volume vessel and within a optical engine combustion chamber, that in realistic conditions lift-off were shorter and that important cycle-to-cycle variation yielded considerable uncertainty in most of the reported trends. He also pointed out that the lift-off length was further reduced as the inter-jet spacing was reduced i.e. as the number of nozzle holes was increased.

No study was found in which non-sooting diffusive combustion (i.e. a diffusive combustion in which soot formation is avoided) was attained in an engine and in a high oxygen concentration environment. Amorim [74] studied extensively the low-temperature diffusive controlled combustion in a small-bore engine with high rate of EGR but was not able to produce conditions in which equivalence ratio at lift-off was lower than the threshold limit of 2. He stipulated however that, when maintaining nozzle hole number, smaller hole greatly helped attaining the non-sooting regime but that a reduction in hole diameter accompanied by an increase in nozzle hole number to maintain the total hydraulic flow of the nozzle lead to a switch from non-sooting to sooting conditions, which was attributed to an increase in jet-to-jet interactions as reported by Musculus [80]. Amorim also explored the possibility of reducing the ambient temperature during combustion by lowering intake temperature but was not able to attain ambient temperature below $900K$.

2.5 Association of Conventional and Advanced Combustion Strategies

Limited load range application has been described as an important drawback of the introduction of premixed combustion strategy. This observation inspired the idea of extending the load range of HCCI/PCCI combustion strategies by adding a second injection event that produces a diffusive combustion. In case of port-fuel fumigation injection for homogeneous charge preparation, this requires the use of two injection systems. However in the case of direct-injection premixed combustion, the flexibility of current injection systems permits multiple injection event during the combustion cycle. A single injection system would hence permit to associate early injection that produces premixed combustion (early PCCI type) with an additional injection close to TDC generating a diffusive combustion process and/or late injection that produces MK like combustion. In this section, studies involving those types of strategy are reviewed.

2.5.1 Early Pilot Injection Strategy

Already in 1998, Hashizume et al. [81] presented the combustion concept MULDIC (MULtiple stage DIesel Combustion) in which partially premixed combustion was attained with the use of a complicated injection system where two side injectors prepared a premixed charge injecting early in the compression stroke and without wetting the cylinder walls, while a third conventional injector was used to inject close to the TDC producing a diffusive combustion. Their results showed that, compared to a conventional combustion, this strategy permitted a simultaneous reduction in NO_x and soot emissions with a slight cost in term of fuel consumption. The authors detailed that the first combustion event was lean so that low NO_x and soot were produced. The second injection was performed in hot, low oxygen ambiance so that soot formation was important; however it was observed that soot oxidation process was strong enough to produce low engine out emissions.

In their study, Okude et al. [82] pointed out the necessity to switch from ”PCI” (totally premixed combustion) to ”split-PCI” (premixed combustion associated with diffusive combustion) strategy at mid and high load to avoid knock and unacceptable noise level due to fast heat release process, similar conclusion were found later by Fang et al. [83]. Figure 2.10, reprinted form their work, illustrates how the combustion process is smoothed with the introduction of this strategy.

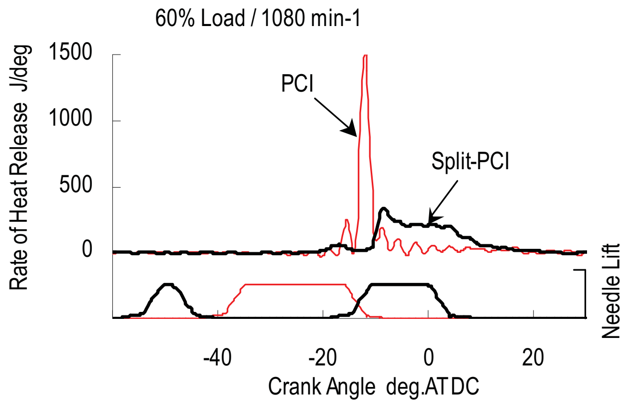

Figure 2.10. *Combustion process smoothing by switching from totally premixed to partially premixed combustion strategy. Source: Okude et al. [82].*

The authors demonstrated that this strategy could be used at full load with an important reduction in NO_x and soot emissions compared to conventional diffusive combustion. The implementation of the strategy resulted in an increase in fuel consumption. The authors also pointed out that, even if compared to a totally premixed strategy the combustion process was smoothened, the partially premixed strategy resulted in higher pressure gradients than a conventional diffusive combustion. They also found that due to early first injection event (around 60° bTDC), fuel sprays miss the piston bowl and impinge onto the cylinder liner. Authors state that the subsequent adhesion of liquid fuel to the cylinder line was an important issue of the ”split PCI” strategy.

Benajes et al. [48] studied experimentally and by means of CFD calculations the introduction of early pilot injection (EPI) strategy in a heavy-duty engine at mid-load and provided a good description of the injection pattern and the resulting combustion evolution. It is clearly observable in figure 2.11 that the first injection leads to a typical HCCI/PCCI combustion process with two peak of heat released while the second injection near TDC produces a conventional diffusive combustion.

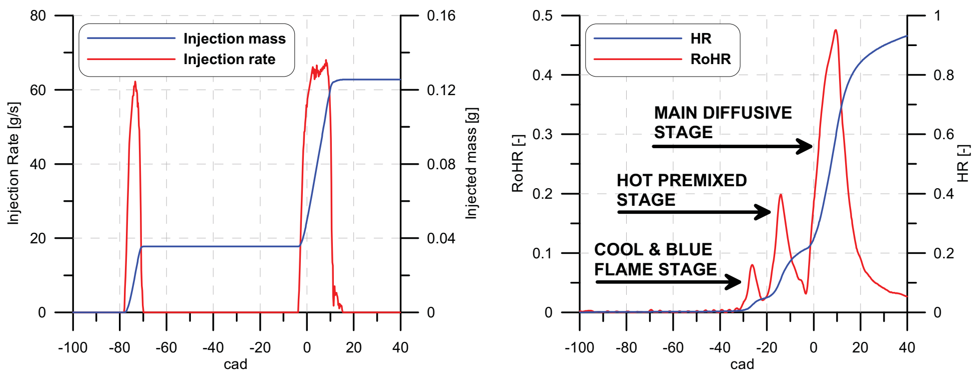

Figure 2.11. *Injection pattern and combustion evolution when operating with EPI strategy. Source: Benajes et al. [48].*

The authors also explained that this partially premixed combustion strategy was efficient to reduce both NO_x and soot emissions compared to a conventional strategy. They however also noted that an important part of the fuel introduced during the EPI event was adhering on cylinder wall and concluded that *"spray impingement and associated fuel losses and oil dilution are [...] the most important drawbacks of the strategy"*.

This latter issue of spray impingement on cylinder wall was also observed by Dronniou et al. [47]. The picture displayed in figure 2.12 shows the damage of the cylinder liner resulting from the impingement of the fuel sprays during the early injection event. The authors stated that switching early injection event from 60º to 30º bTDC keeps the sprays inside the piston bowl and avoids the impingement. However, this measure reduces drastically the premixing time for the premixed combustion and limits the breakthrough of the strategy in terms of soot emissions reduction.

In another work by Payri et al. [84], the potential of increasing swirl motion to avoid spray impingement during early pilot injection was analyzed. The authors found that liquid fuel deposition onto cylinder liner was reduced with increased swirl ratio but could not be avoided.

Figure 2.12. *Cylinder liner wear due to spray impingement during early pilot injection. Source: Dronniou et al. [47].*

2.5.2 Post Injection Strategy

The use of post-injection i.e. the injection of a small amount of fuel after the main injection event, is widely used in Diesel engines because it permits an interesting reduction in soot emissions with no drawback in term of fuel consumption and NO_x emissions. Detailed studies performed at the University of Wisconsin [85–87] and at the CMT Motores Térmicos [88–90] found that the soot reduction is due to reduced soot emissions from the main injection while the post-injection itself, if the amount injected is low enough, does not produce additional soot. Other studies also point-out that the reduction of soot emissions with the introduction of post injection is also due to the fact that the resulting post combustion permits to oxidize part of the soot formed during the previous injection [87, 91, 92].

In his extended study García [90] details that in order not to form additional soot, the post injection pulse needs to be short enough to avoid the establishment of a diffusive flame i.e. the post injected fuel needs to burn dominantly under premixed conditions. He also found that for larger post injection pulses, it was still possible to avoid soot formation if the fuel was injected late enough in the cycle, when the in-cylinder temperature is low enough to produce a non-sooting combustion even if part of the latter occurs in diffusive conditions.

The limitations of the use of post-injection strategy results from the previous considerations. First, considering a short pulse for a premixed post-injection, the reduction of the mass injected in the main injection pulse (that permits the reduction of the soot emissions) is limited to the extent of the fuel mass that can be injected during the post-injection maintaining the transitive character of the post-injection combustion process. It is mentioned in the study by Desantes et al. [88] that in this type of strategy, NO_x emissions slightly increase while the fuel consumption decreases to some extent. On the other hand, considering larger post-injections that need to be injected late in the cycle to avoid additional soot formation, important increase in fuel consumption due to largely decreased cycle efficiency is the limiting factor.

2.5.3 Reduction in the fuel Mass Injected during the Main Injection: Key Parameter of the Association of Conventional and Advanced Combustion Processes

The mechanisms of the soot emission reduction with the introduction of post-injection are well understood. They consist basically in the reduction of the soot emissions resulting from the main injection process while the post-injected fuel does not produces additional soot since the post-injected fuel burns under a non-sooting combustion regime.

The reduction of the fuel injected during the main injection event appears to be a mean of overcoming the conventional diffusive combustion dilemma described in section 2.3.2.4 while the addition of a second combustion that does not produces additional NO_x and soot is a mean to allow recovering the total fuel mass injected without loosing the benefit of the main injection reduction.

When associating a HCCI/PCCI strategy with diffusive combustion through an EPI strategy, the mechanism of soot reduction is not as clearly identified as it is the case for post-injection strategies. Indeed there is a major difference between the two strategies considering that in the case of EPI strategy, the premixed combustion occurs before the diffusive combustion, hence the in-cylinder conditions during the main combustion are largely affected by the premixed combustion. However, as depicted in section 2.5.1, similar results are obtained: soot emissions can be reduced for an equivalent NO_x emissions level i.e. the NO_x-soot dilemma can be overcome.

Furthermore, while strategies involving short post-injections are limited to small main injection fuel amount reduction, early pilot injection can account for an important part of the total injected fuel, the limit being that of a pure premixed combustion process i.e. around 25% of the full engine load.

Important challenges remain for the introduction of an early pilot injection strategy. As they were described in section 2.5.1, they rely principally in the first injection event that leads to hardware wear and unacceptable oil-dilution, while the slight increase in fuel consumption is also a concern.

2.6 Conclusions of the Literature Study and Thesis Justification

This last section of the chapter concludes the literature review and, based on the main points raised, the justification of the work performed in this doctoral thesis is presented.

A detailed description of the processes involved in the conventional diffusive combustion process was given, especially regarding the formation and destruction of the main pollutant emissions resulting from the process i.e. NO_x and soot. As a conclusion it was shown that if the conventional combustion process is efficient and can be applied in the whole load range of the engine, it suffers an intrinsic issue that is the NO_x-soot dilemma.

Overcoming the NO_x-soot dilemma led to the introduction of advanced combustion strategies such as low temperature and/or premixed combustion. Those advanced strategies are efficiently leading to low NO_x-soot combustion however they suffer usually from issues regarding the mixture preparation and above all, all of them are limited to the low load range of the engine operation range.

The association of premixed and conventional strategies allows to extend the load range application of advanced combustion strategies. The resulting combustion process appears to produce less soot emissions for a given NO_x emission level, overcoming hence the NO_x-soot dilemma at higher loads.

Among those strategies, the post-injection strategy was widely studied and the mechanisms of the soot emissions reduction were clearly identified. On the other hand, the EPI strategy that also shows important improvement in terms of pollutant emissions compared to a conventional strategy is not as well understood. Also, EPI strategy suffers from an important issue of hardware wear and oil dilution due to liquid fuel spray impingement onto cylinder liner during EPI event.

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Chapter 3

Experimental and Modeling Tools

Contents

3.1 Introduction

This chapter gives a description of the methodology, the material and the hypothesis with which the results presented in this thesis were obtained.

Given the experimental character of the study, an important part of this chapter focuses on the description of the research engine and the test cell in which it was installed. Together with the description of the experimental tools, the accuracy of the different measurement devices is given, so that the validity of the results produced can be evaluated.

The second part of this chapter details the post-processing and simulation tools used to define and interpret the experiments.

3.2 Single Cylinder Engine

The aim of the study is to introduce partially premixed strategies and analyze the combustion process and the pollutant emissions in a heavy-duty Diesel engine. Given the focus of the analysis on the combustion process, the use of a single cylinder engine is preferable as it permits a better control and a higher flexibility regarding air management.

In order to be able to introduce innovative strategies, high flexibility of the engine is necessary. This is evidently the case for the injection system that is able to perform numerous injection events in one cycle, but the flexibility of the engine also resides in the presence of a totally variable hydraulic valve actuation (HVA) system. Although it was not a prerequisite for the study, the HVA system revealed itself very useful for the introduction of an innovative injection strategy that is described in chapter 4.

3.2.1 Engine Characteristics

The engine used in this research is a single-cylinder, four-stroke, direct-injection Diesel research engine representative of medium-large truck engines.

The engine base is a unit built by AVL, whose geometry has been adjusted to a VOLVO TRUCKS production engine, precisely the MD11US2007 model. The basic unit is composed of the engine block with its crankcase and a counter rotating mass system to ensure the balance of the running engine.

In order to reproduce the conditions found in the equivalent multi-cylinder engine, connecting rod and cylinder head are that of the production engine while the crankshaft has the same geometry but is adapted to dimensions of the basic unit hence with a single crank pin. The basic specifications of the engine are included in table 3.1.

The original piston and injector nozzle, whose specifications are also given in table 3.1, were used in parts of the study. However in some cases they were replaced with adapted design piston and/or nozzle for the introduction of advanced injection/combustion strategy. The main specifications of the adapted hardware and the justification of their introduction is given in the result chapters 4 and 5.

Table 3.1. *Engine Main Characteristics.*

Engine	
Type	4 stroke Direct Injection Diesel Engine
Basic Unit	AVL model 5300
In-cylinder max pressure	250 bar
Production Engine	VOLVO model MD11US07
Bore x Stroke	123 mm x 152 mm
Connecting Rod Length	225 mm
Displacement	1806 cm^3
Intake Valves Diameter	39.5 mm
Exhaust Valves Diameter	38.5 mm
Mean Swirl Number	0.3
Original Piston	
Bowl Diameter	83.6 mm
Bowl Depth	19.98 mm
Bowl Volume	95.65 cm^3
Compression Ratio	16.3 : 1
Original Nozzle	
Number of Hole	6
Hole Diameter	0.214 mm
Spray Included Angle	140^o
Hydraulic Flow	745 $cm^3/30s/100\ bar$

3.2.2 Variable Valve Actuation System

The most peculiar aspect the research engine is the presence of a fully variable cam-less valve actuation system in place of the conventional mechanical camshaft. In this system, all the valves are independently actuated by different hydraulic pistons (one per valve), which are controlled by a specific electronic control unit. The system flexibility includes variable timing, variable duration, and variable lift for each intake and exhaust valve and, if desired, two intake and/or exhaust events per cycle can be carried out.

The system was developed by *Sturman Industries* under the name HVA 4A [1, 2]. The following figure 3.1 shows a scheme of the HVA system. It can be observed that the system requires three oil pressure levels for the valve actuation: a variable (high) pressure circuit ($\approx$ 100 to 200 *bar*), a low pressure circuit ($\approx$ 35 *bar*) and an atmospheric pressure circuit. The control of the position of a proportional valve located between those three circuits permits the modulation of the force applied on the hydraulic actuator to open the engine valve, while the counter force is insured by a return spring.

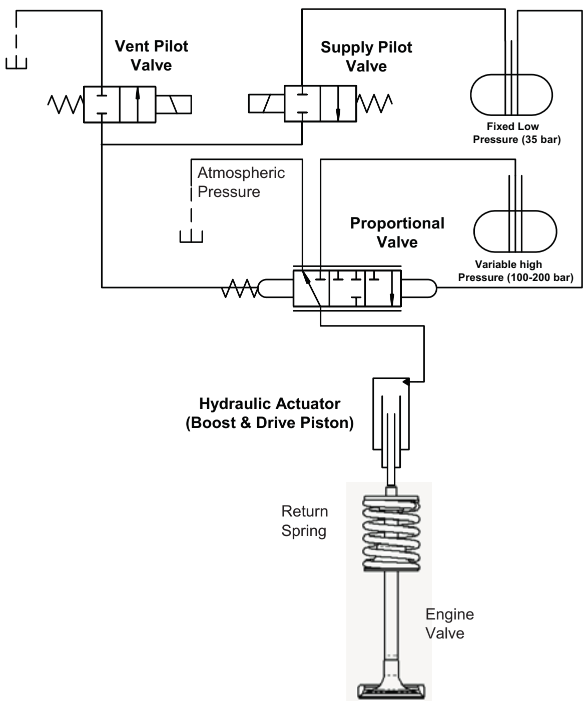

Figure 3.1. *Scheme of the oil circuit of the HVA system.*

This system permits the valve actuation over a large range of engine speed. The advantages of the system rely on its adaptability to variations in oil viscosity and its ability to control the engine valve seating velocity to reduce mechanical stress so that the life duration of the valves is increased.

The system allows high flexibility regarding opening and closing angle for each of the four valves as it is illustrated in figure 3.2. Opening velocity, maximum lift and seating velocity can also be modified independently. Furthermore, the systems allow two opening events per cycle and two stages lift for the intake valves to permit and control valve overlap. Additional information about this VVA system and its application on HD Diesel engines can be found in the work of Lombard et al. [3] and also in the work of Novella [4].

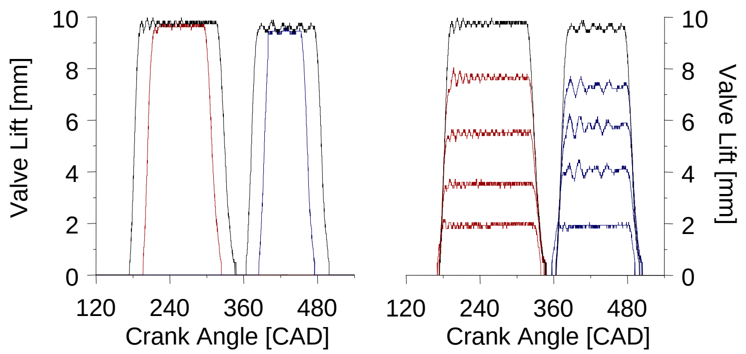

Figure 3.2. *Flexibility of the HVA system regarding maximum valve lift, opening and closing angles.*

Following the study by Lombard et al. [3], the accuracy attained by the HVA system over the different parameters is the following. Opening angle is controlled with a accuracy of ± 2 CAD while the closing angle accuracy is ± 3 CAD, the repeatability is 3 CAD in both cases. Regarding maximum lift, the accuracy is ± 0.5 mm and the repeatability 0.8 mm when the valve lift is higher than 3.5 mm (always the case in the present study).

Along the study, the oil pressure for the HVA system was provided via external electrical pumps. The energy consumption of the system is not taken into account when estimating the engine efficiency because, due to the prototypical character of the HVA system, the electrical pumps are over dimensioned.

3.2.3 Injection System

The engine is equipped with a Bosch injection APCRS (Amplifier Piston Common Rail System), model CRSN4.2 [5] that is able to perform up to five injections per cycle. The main characteristic of this hardware is its capability to amplify the common-rail fuel pressure for one of the injections (main injection) by means of a hydraulic piston directly installed inside the injector. Following figure 3.3 shows a scheme of the injection system.

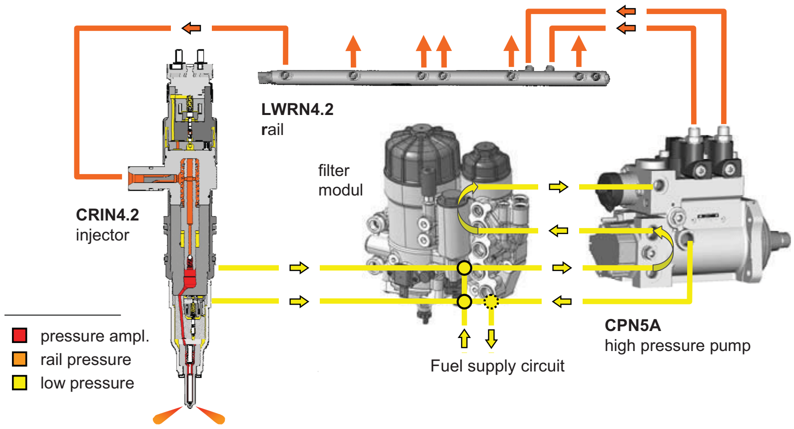

Figure 3.3. *Scheme of the APCR injection system installed in the engine. Source: Albrecht et al. [5].*

The user can adjust, via the engine electronic control unit, the rail pressure up to a maximum value of 1000 *bar*. Then, the activation of the amplifier piston allows a multiplication of the actual injection pressure with a ratio of 2.2 i.e. the maximal pressure for the main injection event is 2200 *bar*.

The system control is ensured by two solenoids that are present in the core of the injector. One is used to control the needle lift while the other acts on the amplifier piston. As previously mentioned the activation of the amplifier piston is possible only once in a cycle. Adjusting the relative timing of its activation with respect to the actual action on the needle lift allows a shaping of the injection rate and permits to produce square, boot or ramp injection rate type for the main injection.

3.3 Test Cell Description

The engine is installed in a fully instrumented test cell, with all the auxiliary facilities required for its operation and control. The test cell is equipped with state of the art devices that permit the measurement of pollutant emissions, fuel consumption, engine effective power and torque. Also, a series of sensors are installed for the control and the diagnostic of the combustion. During engine tests, the data measured are recorded following two approaches: a fast acquisition is available for the measurement of the dynamic parameters along the combustion cycle (mainly the in-cylinder pressure) while a slow acquisition is performed for average parameters such as pollutant emissions, air mass flow etc. Following figure 3.4 shows a scheme of the test cell.

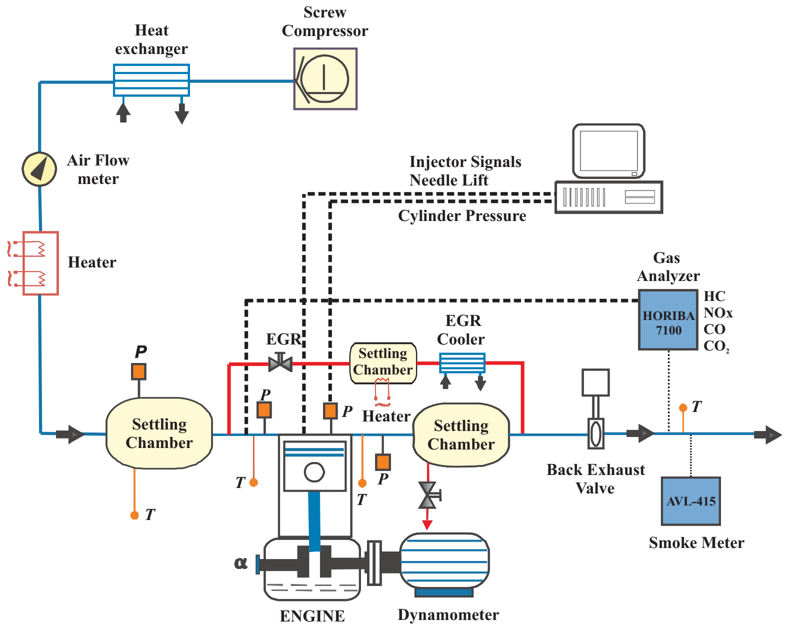

Figure 3.4. *Scheme of the engine test cell.*

3.3.1 Engine Air Management

The capabilities of the test cell regarding boost pressure and EGR rate are beyond that of a real engine, therefore permitting to explore the introduction of a partially premixed charge over a wide range of conditions.

3.3.1.1 Intake and Exhaust Circuits and Air Flow Measurement

To achieve stable intake air conditions, a screw compressor[1] supplies the required boost pressure before passing through an air dryer. An intake settling chamber was installed in the test cell in order to avoid that the pressure waves generated by the engine reach the air flow meter system. Its size (500 L) was defined in a previous study by Molina [6] according to two different methods (SAE J244 [7] and Kastner [8]). The exhaust back-pressure produced by the turbine in the real engine was replicated by means of a valve placed in the exhaust system, controlling the pressure in the exhaust settling chamber.

A roots type air flow meter is installed upstream the settling chamber provides both volumetric and mass flows[2]. The model used, Elster RVG G100, offers a accuracy of $\pm 0.1\%$ over the measured value.

Boost pressure is adjusted within the intake settling chamber through a PID control, with an accuracy of ± 0.001 bar in the low to mid pressure range (< 4 bar) and an accuracy of ± 0.005 bar in the high pressure range (> 4 bar). The control of the exhaust pressure is by the operator, maintaining an accuracy from ± 0.010 bar to ± 0.025 bar depending on the pressure range. The temperature of the fresh air (before mixing with EGR) was adjusted inside the intake settling chamber controlling an electrical heater via PID, with an accuracy of $\pm 1^oC$ while intake temperature is controlled in the intake manifold after mixing with EGR as described hereafter.

3.3.1.2 EGR Circuits

Two types of EGR circuits were used along the study. In a first instance a relatively simple circuit that is showed in the scheme presented in figure 3.4 was used. In this case, hot exhaust gas are cooled down enough to permit attaining the desired intake temperature, while the fine adjustment of the intake temperature was performed by re-heating EGR gases. The intake temperature is therefore controlled via a PID acting on the electrical heater installed inside the EGR settling chamber. The accuracy attained for the intake temperature is $\pm 1^oC$. The limitation of this system is principally the lack of control of the hydrometry of the recirculated gases as an uncontrolled

[1] Along the study, two different screw compressors were used depending on the required boost pressure. A low to mid pressure range (1 to 4 bar) compressor and a high pressure (3.5 to 8 bar) compressor. Whatever the compressor used, the intake circuit was similar.

[2] The volumetric mass flow is directly interpolated from the rotating speed of flow meter lobes while the mass flow is obtained by calculating the density in the device from temperature and pressure measuring.

amount of the water produced by the combustion condenses in the circuit due to the cooling.

In a second part of the study, a different EGR circuit that is presented in figure 3.5 was used.

In this case, the exhaust gases pass through a commercial DOC/DPF in order to clean them and complete the oxidation process of the remaining HC and CO present in the exhaust[3]. The gases are then cooled down to a controlled temperature (usually 30^oC) so that the amount of water that is able to condensate can be calculated. This amount is then condensed and accumulated in a water condenser that is purged regularly. Right after the water condenser, the gases are largely heated up to avoid further condensation in the rest of the EGR circuit. The reheating is adjusted so that the temperature before the EGR settling chamber is maintained at a higher value (usually 40^oC) than that before the water condenser.

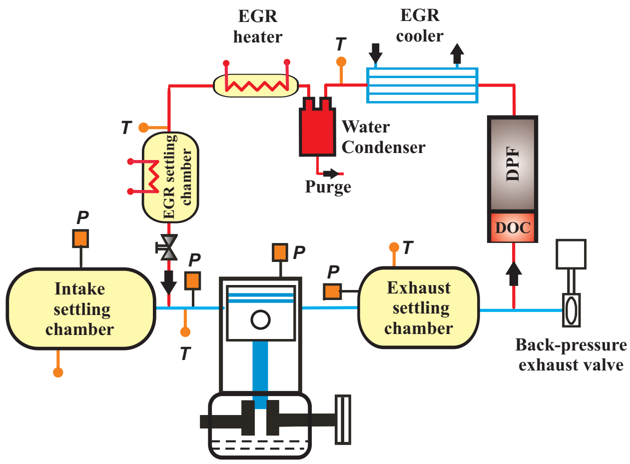

Figure 3.5. *Scheme of the evolved version of the EGR circuit.*

As it is the case for the more basic circuit, the fine adjustment of the intake temperature is performed by heating further the exhaust gases in the EGR settling chamber.

In both cases, the EGR rate is measured by the gas analyzer as described in section 3.3.4.1. To ensure the required external EGR mass flow rate, the

[3]This way the composition of the gases recirculated could be calculated from the A/F ratio considering a 100% efficiency of the combustion process, therefore permitting the definition of an advance engine test plan as it is detailed in chapter 5.

exhaust pressure was maintained above the intake pressure. The exact external EGR rate was controlled with an accuracy of ±0.2% by means of a valve between the EGR settling chamber and the intake pipe.

3.3.1.3 Blow-by Measurement

Despite the fact that large bore engines typically do not produce large amount of blow-by loses through the piston rings, the measurement of this gas leakage is useful to estimate with accuracy the instantaneous trapped mass inside the combustion chamber during the compression-expansion strokes and to produce a better diagnostic of the combustion.

During the tests, the measurement of the blow-by is performed by the commercial device AVL 442 in terms of volume flow rate, by using the pressure difference in the flow passing through a calibrated orifice. The mass flow rate is calculated considering the ambient pressure and temperature. The accuracy of the equipment is ±1.5% upon the measured value.

3.3.2 Engine Auxiliary Systems

3.3.2.1 Fuel Supply Circuit and Flow Measurement

A specific circuit supplies fuel to the injection system from an external fuel tank as shown in the scheme of figure 3.6.

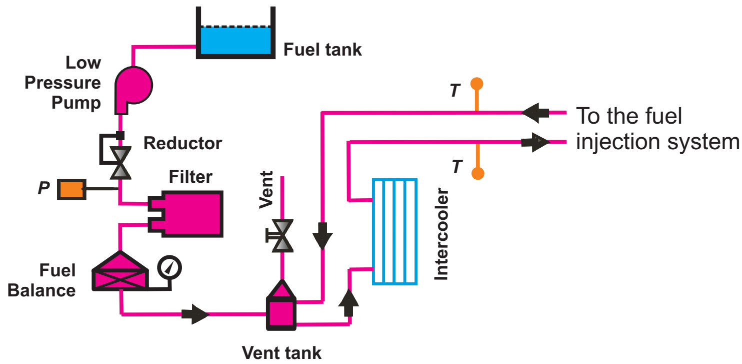

Figure 3.6. *Scheme of the fuel supply circuit.*

An AVL fuel balance model 733S is installed in the supply circuit to measure the engine fuel consumption with an accuracy of ±0.2% over the measured value. This accuracy can be largely affected by the fuel density variation if the fuel temperature changes rapidly in the circuit as it was observed by de Rudder [9] and Novella [4]. For this reason, the temperature in the fuel supply circuit is constantly controlled: the water temperature in the inter-cooler (see figure 3.6) is adjusted to maintain constant the fuel temperature before it enters the injection system at a value of 40^oC $\pm 0.5^oC$. The temperature of the fuel in the returning circuit is controlled by the operator so that engine measurements are initiated only once the temperature in the returning circuit is stable[4].

3.3.2.2 Lubrication and Cooling Systems

Due to the specificity of single cylinder engines, conventional lubrication and cooling systems in which the oil and water pumps are actioned by the engine cannot be used. Specific circuits are therefore installed with electrical water and oil pumps. Also, due to the lower heat rejected by a single cylinder compared to a full size engine, electric heaters are installed in the water and oil circuits in addition to the conventional heat exchanger systems in order to reach nominal fluid temperature relatively fast during the engine warm up process.

During the tests, the temperature was set to the nominal value of the full size engine under conventional operation i.e. 90^oC for oil and 80^oC for water. The adjustment of the fluids temperature was performed upon the engine out temperature measurements via a PID control with an error of $\pm 1^oC$. The oil circuit pressure is constantly monitored for security reasons.

3.3.3 Torque and Speed Measurement and Regulation Systems

The engine crankshaft is coupled via a drive shaft to a dynamometer that permits maintaining the engine at the desired operation point. The main characteristics of the dynamometer used in the study are given in the following table 3.2.

In this study, the engine performance is evaluated in steady conditions i.e. the engine speed and load are kept constant. The torque regulation

[4]The temperature at which the fuel stabilized in the returning circuit is different depending on the engine mode. The temperature is considered stable when the variation is less than 0.1^oC in two minutes. This permits, according to Novella [4] an error lower than 0.02 kg/h corresponding to an error of 5% in the worst case for the present study.

Table 3.2. *Main characteristics of the dynamometer.*

Type	4 poles dynamometric brake
Manufacturer	Wittur Electric Drives GmbH
Model	DSB3-16.3-B0I.41-M1KN
Nominal power	65 kW
Nominal torque	414 Nm
Maximal torque	1150 Nm
Nominal speed	1500 rpm
Maximal speed	6500 rpm

is performed via a PID control, the engine speed being monitored with an encoder installed directly inside the dynamo-meter, the action of control is performed upon a frequency dimmer that permits modifying the braking torque. The accuracy attained by the system regarding engine speed regulation is ± 1 rpm.

The torque measurement, that is not necessary for the regulation, is performed by a strain gage torque-meter installed directly on the drive shaft. The nominal torque of the measurement system is 1000 Nm while its accuracy is 0.2% of the nominal value i.e. ± 1 Nm.

If necessary, the dynamometer can be used as an electrical motor for spinning the engine when no fuel is injected. This functionality is used for the starting process.

3.3.4 Pollutant Emissions Measurement

3.3.4.1 Gas Analyzer

The experimental installation was equipped with a gas analyzer HORIBA MEXA 7100 DEGR. The analyzer is used to measure main gaseous pollutant emissions: nitrous oxides (NO and NO_2 lumped in NO_xemissions), total unburned hydrocarbons THC and carbon monoxide CO. Concentration of oxygen O_2 and carbon dioxide CO_2 in the exhaust are also measured. Following table 3.3 shows the measuring technique used for each one of the chemical species. The accuracy of the analyzer is theoretically $\pm 2\%$ over the measured value however Molina [6] found experimentally an accuracy of $\pm 4\%$

for this equipment using a similar engine. Considering the complexity of the measurement, this accuracy is considered sufficient for the present study.

Table 3.3. *Measuring techniques of the gas analyzer.*

NO and NO_2	Chemiluminescence Analyzer (CLA)
CO and CO_2	Non-Dispersive Infra-Red Detector (NDIR)
THC	Flame Ionization Analyzer (FIA)
O_2	Para-Magnetic Analyzer (PMA)

The main sampling probe of the analyzer is installed in the exhaust pipe, downstream the back-pressure valve since an inlet pressure maintained below 1.2 *bar* is required for a proper functioning. From this point, the temperature of the gas is maintained above 192^oC to insure that unburned HC do not condensate before the measuring. All the chemical species except HC are measured in dry conditions, a correction is later applied to take into account the humidity in the exhaust gases, following European directive 2005/55/EC [10] currently effective.

Another probe is located in the engine intake manifold (see figure 3.4), as close as possible to the cylinder head. This second probe is used only to measure the CO_2 concentration in the intake, in order to determine the EGR rate following the formula 3.1.

$$EGR\,rate \;=\; \frac{[CO_2]_{Intake} - [CO_2]_{Atmosphere}}{[CO_2]_{Exhaust} - [CO_2]_{Atmosphere}} \tag{3.1}$$

The CO_2 concentration in the atmosphere is assumed as the commonly accepted value of 0.04% since the engine test cell is perfectly ventilated and the exhaust gas are efficiently extracted.

3.3.4.2 Smoke-Meter

In complement of the gas analyzer, the installation is equipped with a variable sampling smoke meter model AVL 415. With this equipment, the measurement is performed by taking a sample of the exhaust gas downstream the back-pressure valve.

The results obtained in FSN units are transformed into dry soot mass emissions by means of a correlation proposed by Christian et al. [11] that is shown in following equation 3.2.

$$Soot = \frac{1}{0.405} \cdot 4.95 \cdot FSN \cdot e^{(0.38 \cdot FSN)} \tag{3.2}$$

In this study, soot will be considered as a suitable tracer of particulate matter. Although the measurements provided do not entirely take into account the soluble organic fraction (SOF) of the total particulate matter, this measurement has been shown to give reliable trends [12].

3.3.5 Injection Rate Measurement

A specific equipment is used for the measurement of the injection rate that is useful for later spray simulation and combustion diagnostic.

The measurement is performed removing the injectors from the engine[5] and installing it in a specific test rig called injection rate discharge curve indicator.

The model used, EVI commercialized by IAV, permits the measuring of the injection rate following the Bosch method [13]. According to Plazas [14] who performed a comparative study between various injection rate measuring devices, this equipment is the most suitable for the measurement of fast changing signals as it is the case of injection rate, especially at the beginning and at the end of the injection process.

The fuel balance described in section 3.2.3 is used during the injection rate tests to determine the exact amount of fuel injected during a cycle. The ratio between the value obtained from the balance and the integral of the recorded injection rate signal is used as a correction coefficient for the injection rate signal. This means that the *raw* injection rate trace is multiplied by the correction coefficient so that the integral of the *post processed* injection rate equals the value obtained by the fuel balance.

3.3.6 Data Acquisition

Two different acquisition methodologies were used during engine tests, depending on the parameter recorded. A fast acquisition process is required for the "constantly varying" or "dynamic" variables such as the in-cylinder pressure or the valve lift. These signals constantly change along the cycle,

[5]The injection circuit and high pressure pump used for the measurement of injection rate are the same as that of the engine test meaning that the engine has to be running (motored by the dynamometer) during injection rate tests to entrain the high pressure pump. During those tests the injector in the engine is replaced by a dummy injector to avoid air leaks in the cylinder.

but are repeated similarly from one cycle to another. For this reason a high frequency acquisition procedure was used over a large number of cycles in order to compensate from cycle to cycle dispersion.

On the other hand, parameters such as engine speed, boost pressure or pollutant emissions are kept constant when the engine is running in steady conditions. For those signals a low frequency acquisition system was used.

The dynamic signals are registered with a fast measuring system *Yokogawa* model DL716. The measuring system was synchronized with the engine so that the data were recorded every 0.2 CAD i.e. 3600 points during each engine cycle.

The synchronization of the measuring system with the engine is performed using an optical angular encoder model AVL 364 coupled to the engine crankshaft providing a signal every 0.5 CAD. The frequency was increased electronically to obtain a signal every 0.2 CAD. The accuracy of the encoder is ± 0.02 CAD.

The encoder also provides a reference signal once every engine rotation in order to define the crank position with respect to the TDC. The frequency of this signal is electronically divided by two in order to obtain one reference signal in an engine cycle and therefore be able to identify the crank position with respect to the *combustion* TDC. This reference signal is used to trigger the measuring system. Once the measurement is launched, fifty cycles are recorded in order to compensate from engine dispersion. The post treatment of each signal is latter performed upon the average obtained from the fifty cycles.

The most important measurement acquired with the system is the in-cylinder pressure. The pressure sensor used along the study is a piezoelectric model 6125B from the company Kistler, which measurement range is appropriate for the engine (0 to 250 bar). The other parameters measured using the fast acquisition system are the four valves opening and the electric pulses of the needle and amplifier piston from the injection system.

All the parameters acquired with low frequency are recorded on a computer using a specific software called *Samaruc*, internally developed under Labview code at CMT Motores Térmicos. The software permits the acquisition of the information recorded via specific modules, developed by *National Instrument*, that condition the measurement of the temperature and pressure sensors, the torque-meter, the encoder and the air mass flow meter. Also the computer receives directly the information from other sources: fuel balance, pollutant emissions and blow-by measuring devices.

The acquisition frequency for those stationary parameters is set to 1 Hz while the duration of the acquisition chosen is 40 s in order to provide a representative average for each measurement.

3.3.7 Engine Test Routine

The routine adopted during the engine tests is the following:

- *Engine warm-up and auxiliary systems calibration.* During this phase, the fluids (oil, water and fuel) are brought up to their nominal temperature. The HVA system is calibrated to ensure a good control of the valve timing. Also both the gas analyzer and smoke-meter are calibrated and purged.
- *Reference test measurement.* Before every measuring session, the same engine operation point is evaluated in term of engine breathing process, performance and pollutant emissions as well as the in-cylinder pressure trace. This measurement permits to ensure test repeatability along the study.
- *Engine tests.* Engine test routine consists in bringing the engine to the specific operation point defined in the test plan. Once the operating conditions are stable, the measurement of both the fast and slow varying signals is repeated three times to control the stability of the results.

It is important to note that the load is adjusted maintaining the fuel mass injected, not the engine torque or IMEP. The present study focuses principally on the evolution of the combustion process and the resulting pollutant emissions. In order to be able to compare one strategy with another in this frame, it is important to maintain constant the injection duration, the SoIand the EoI. For this reason, the fuel mass injected is maintained constant for a given load level.

In the first part of the study, presented in chapter 4, the engine conditions regarding breathing process are defined with respect to the boost pressure and the EGR level. However in the second part, presented in chapter 5, the EGR level is a parameter that is adjusted to meet the desired oxygen concentration level in the intake. The prediction of the boost pressure and EGR levels to meet the required intake oxygen concentration is obtained through the building of A/F-EGR maps that are presented at the beginning of the chapter 5. The use of those maps implies that the fuel injected per cycle is maintained constant for a given load level. This is another reason that led to the decision of controlling the load level via the fuel mass injected per cycle instead of the engine IMEP.

3.4 Modeling Tools

3.4.1 Combustion Diagnosis

The post processing of the information collected during the tests is fundamental for the description and the understanding of the evolution of the combustion process that leads to the measured pollutant emissions and engine performance. Without a doubt, one of the most representative information describing the combustion process is the heat release law, however the estimation of the flame temperature might be an even more useful tool for the understanding of the pollutant emissions results.

3.4.1.1 In-house Code CALMEC

The calculation of the heat released (HR) and rate of heat released (RoHR) is performed resolving the first law of thermodynamics for open systems[6] and that is applied the to controlled volume represented by the combustion chamber from the intake valve closing until the exhaust valve opening. The model used for the calculation, that has been entirely developed within the CMT Motores Térmicos, is named CALMEC [15, 16].

The main assumption used by the model for the calculation of the heat release law are the following:

- The pressure inside the combustion chamber is uniform.
- The gas present inside the combustion chamber is a mixture of fresh air, gaseous fuel and combustion products.
- The thermodynamic evolution of the gas mixture inside the combustion chamber is assimilated to that of a perfect gas.

An important point for a good calculation of the heat release law is the right estimation of the trapped mass inside the combustion chamber at the beginning of the calculation i.e. at the instant of the intake valve closure.

[6]The system is considered open as there is mass exchange induced by the fuel injection (mass addition to the system) and the blow-by (mass reduction).

The total gas mass trapped in the cylinder is calculated from the fresh air and EGR flows and the mass of residuals. The estimation of the residual mass is obtained using a quasi-steady filling and emptying model described in reference [17]. At each step of the calculation the in-cylinder mass is then recalculated considering the blow-by and the fuel injection flows.

3.4.1.2 Flame Temperature Calculation

The combustion diagnostic assumes that there is only one zone inside the combustion chamber, i.e. the gas temperature estimated at each step of the calculation is representative of the average temperature inside the combustion chamber.

However the formation and destruction processes of pollutants are mainly dependent on the local conditions (equivalence ratio and temperature). For this reason, the estimation temperature in the stoichiometric reaction zone i.e. the flame temperature is very useful to understand the emissions results obtained.

The flame temperature is calculated assuming the three main hypotheses listed below, which are widely accepted:

- The combustion is considered to take place at constant pressure at each time step.
- The fuel/air mixture burns adiabatically, which implies that all the heat released during combustion is used to increase the temperature of the combustion products.
- A conventional chemical equilibrium model taking into account the dissociation phenomena is assumed.

Considering these assumptions, the absolute enthalpy of the reactants at the initial state equals the absolute enthalpy of the products at the final state. This statement is expressed in equation 3.3.

$$H_{initial}(p, T_{ub}) \;=\; H_{final}(p, T_{flame}) \tag{3.3}$$

where p represents the instantaneous in-cylinder pressure that is directly measured in the research engine, T_{flame} is the desired output variable and T_{ub} is the temperature of the unburned gases.

In this calculation scheme, T_{ub} acts as an input and it should be obtained before computing T_{flame}. It is worthy to note that the temperature of the unburned gases and the in-cylinder mean temperature are the same until the start of combustion. Once the combustion process starts, the unburned gases are assumed to undergo a sequence of adiabatic compression processes each time step according to measured in-cylinder pressure profile. This idea is reflected in equation 3.4.

$$T_{ub,i} = T_{ub,i-1} \cdot (\frac{p_i}{p_{i-1}})^{\frac{\gamma-1}{\gamma}} \tag{3.4}$$

where p_i and $T_{ub,i}$ are the respective in-cylinder pressure and unburned gases temperature at the i^{th} instant, p_{i-1} and $T_{ub,i-1}$ are the same parameters at the $(i-1)^{th}$ instant, and γ is the ratio of specific heats.

According to equation 3.4, the unburned gases temperature at a given instant $T_{ub,i}$ is calculated from that of the previous instant one time step before $T_{ub,i-1}$. This algorithm starts together with the combustion process, therefore the initial $T_{ub,0}$ and p_0 correspond with the mean in-cylinder temperature and pressure at the start of combustion directly provided by the combustion diagnosis code.

In case of EPI strategy, the first part of the combustion occurs in highly premixed, possibly non stoichiometric conditions. In this case, the calculation of the flame temperature considering a locally stoichiometric reaction is not valid.

Fort this reason, in EPI conditions the flame temperature calculation is not valid until the start of diffusive combustion. Therefore, initial $T_{ub,0}$ and p_0 for the calculation are that obtained from the CALMEC code at the instant the start of diffusive combustion despite the fact that the premixed combustion started before.

Once the computation of Tub is explained, the adiabatic flame temperature can be computed from 3.3. To compute the enthalpies at both the initial and final states of the combustion process i.e. the enthalpy of the reactants and the combustion products, the initial and final composition of the gases has to be known.

Regarding the details of the thermodynamic calculations, the enthalpy for a mixture of elements can be expressed as:

$$H_{mixture}(T) = \sum_{i=1}^{n} N_i \cdot h_i \tag{3.5}$$

$$h_i(T) \;=\; h_{i,form} + \int_{T=298}^{T} c_{p,i}(T) \cdot dT \tag{3.6}$$

where $H_{mixture}$ is the total enthalpy of the mixture, which depends on its temperature and composition, h_i is the enthalpy of each individual element of the mixture, $h_{i,form}$ is the formation enthalpy for each element (computed at standard conditions: 298.15 K and 1 bar). N_i is the moles number of each element of the mixture, $c_{p,i}$ is the heat capacity at constant pressure for each element of the mixture, available in thermodynamic databases as a polynomial expression in terms of temperature and n is the number of chemical elements in the mixture.

The unburned gases composition is known, whereas the combustion products composition depends on their final temperature, that is, the adiabatic flame temperature. Because of this link between products composition and temperature, the computation of the adiabatic flame temperature becomes an iterative process.

For attaining a suitable balance between complexity and accuracy, n-Dodecane ($C_{12}H_{26}$) has been selected as the fuel surrogate for the calculation because its physical and chemical properties are similar to those of the conventional gas-oil fuel used in the present investigation. The adiabatic flame temperature is calculated considering a conventional chemical equilibrium model with dissociation following the scheme proposed by Way [18]. Thirteen species are involved in this calculation, as listed in the following table 3.4:

Table 3.4. *List of chemical species taken into account for the calculation of the adiabatic flame temperature.*

1	2	3	4	5	6	7	8	9	10	11	12	13
$C_{12}H_{26}$	O_2	N_2	CO_2	H_2O	CO	H_2	H	N	O	OH	NO	Ar

3.4.2 1-D Spray Simulation

3.4.2.1 In-house Code DIES - Inert Spray Model

DIES (Direct Injection Engine Spray) simulation code is capable of simulating the formation of the fuel spray and its interaction with other elements in the combustion chamber. The model is fully described in the work of López [19].

The model is based on the gas jet theory, it follows the scaling laws detailed by Desantes et al. [20] and is developed under two main assumptions that are reported in another work by Desantes et al. [21] as follow:

- There is an existing analogy between a turbulent gas jet and a direct-injection diesel spray under current engine boundary conditions i.e. high injection pressure, small nozzle diameter, and high air density.
- The transient spray can be described by means of quasi-steady-state equations. This is a reasonable assumption when using a constant injection rate, as is often found in direct-injection diesel engines.

The model is separating the jet in two main zones. First the zone where the evaporation process is taking place, in which the presence of liquid and vapor fuel as well as ambient gas is found, and second the zone where the evaporation process is completed, in which only vapor fuel and ambient air are found.

As stated in the first main assumption, the model considers the analogy between the Diesel spray and the turbulent gas jet, however compared to single-phase gas jets, additional aspect (related to the fact that two phases are present in the spray) are taken into account by the model. First, the change in local properties are considered due to the phase shift of the liquid fuel due to the entrainment of warm ambient gas using state relationships that serve as the equation of state of the mixture, so that if the fuel mass fraction distribution is known at a certain point inside the spray, any thermodynamic property (density, temperature, etc.) can be known. Second, the spacial distribution of spray properties, that is obtained coupling the previously mentioned state relationships to a one dimensional spray model that is used to calculate the spacial distribution of velocity and fuel mass fraction based on the momentum and mass conservation balances inside the spray. Following figure3.7 reprinted from the work of Desantes et al. [21] offers a schematic of the coupling procedure described above.

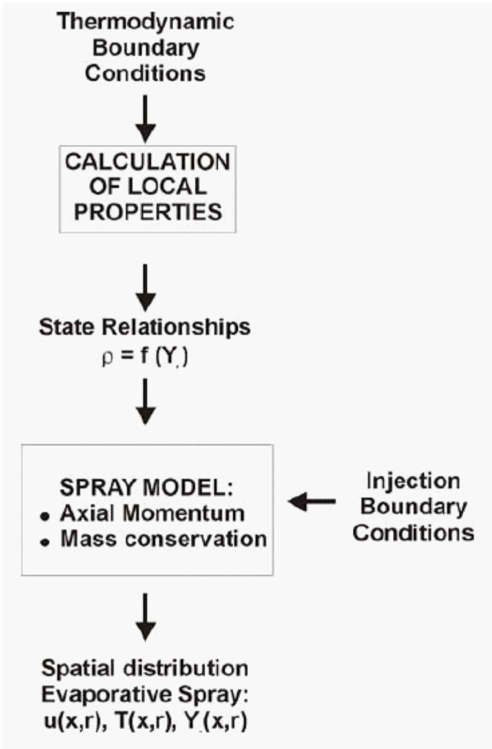

Figure 3.7. *Schematic of the coupling procedure between the spray model and the evaporation information in DIES model.*

The approach used for the simulation is Lagrangian type. This means that each portion of fuel injected, defined in terms of fuel mass and axial position, is tracked along the spray axis and their state (air included, temperature...) is recalculated at each step of the calculation. The definition of the portion of fuel is performed via a discretization of the injection rate.

The in-cylinder conditions are revised at each step of the calculation to take into account the piston motion. Although the model is one dimensional, the radial profile of the spray is estimated from the axial conditions. The model offers a 3-D visualization of the spray evolution into the combustion chamber.

The limitation of the model relies principally in its second important hypothesis that impedes a good predictions of the mixing evolution of the spray in the transient conditions found after the end of injection. For this reason, the evolution of the local equivalence ratio after the end of injection (fundamental parameter in case of premixed combustion as depicted in section 2.4.2), will be estimated using another one dimensional spray model that is described hereafter.

Along the study, this model has been used to estimate the liquid spray penetration and eventual liquid deposition onto the piston.

3.4.2.2 In-house Code DICOM - Reacting Spray Model

The DICOM (DIesel COMbustion) model permits the simulation of free liquid fuel spray under non-evaporative, evaporative and reactive conditions. The model is thoroughly described in the works of Pastor et al. [22] and Desantes et al. [23].

The main hypothesis of the model is, similarly as in the case of the DIES model, that the behavior of the liquid fuel spray can be assimilated as that of a gas jet. The other main hypothesis under which the model has been established are:

- The spray is considered as perfectly symmetrical. As a consequence, the swirl motion inside the cylinder cannot be taken into account.
- The flow is considered turbulent, it can therefore be assumed that the radial profiles for the different variable follow a Gaussian shape.
- The flow is locally homogeneous. There is an local equilibrium between the different species for the temperature and the velocity.
- The ambient pressure is constant in the complete calculation domain, including the spray.
- the local density is calculated assuming an ideal mixture.

Following figure3.8, reprinted from the work of Pastor et al. [22] shows the conceptual view of the model and of its main inputs and outputs:

The model uses an Eulerian approach to resolve the state relationships and the conservation equations. Compared to the approach used by the DIES model, this permits a much better prediction of the mixing evolution in transient conditions such as that found during the late spray mixing, after the end of injection. The model provides the results of the local velocity, temperature and fuel mass fractions as function of time, axial and radial position.

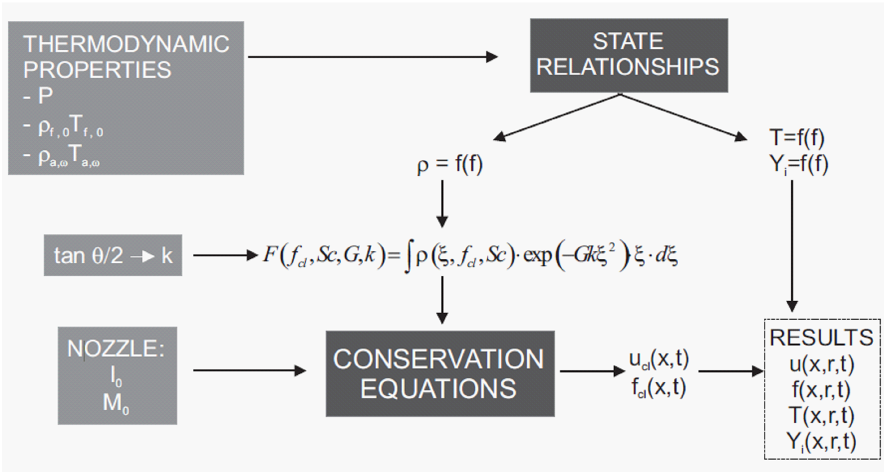

Figure 3.8. *Schematic description of the DICOM model.*

In this study, this model has been used to estimate the maximum local equivalence ratio at the beginning of the premixed combustion. The model is not intended to simulate the spray in a limited volume as it is the case in the combustion chamber, however since spray to wall interaction does not affect strongly the mixing process [24], this limitation should not greatly affect the results used.

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Chapter 4

Evaluation of Two Innovative EPI Strategies

Contents

4.1 Introduction

The literature review presented in chapter 2 showed that for mid to high load operation, the introduction of partially premixed combustion by means of Early Pilot Injection (EPI) strategy is an effective way to reduce major pollutant emissions (i.e. NO_x and soot) with low penalty on fuel consumption.

However, it was also pointed out that a major issue related to the liquid fuel impinging with the cylinder liner during the first injection event exists with the introduction of EPI strategy. A simple solution to this problem is the retarding of the pilot injection so that fuel jets aim toward the piston bowl during this event. Results found in literature show that in this case, the lack of mixing time for the premixed combustion limits the breakthrough of the EPI strategy in term of pollutant emissions.

In this chapter, an evaluation of two different EPI strategies that avoid the fuel impingement onto the cylinder liner, are implemented in the research engine described in chapter 3.

The evaluation of the EPI strategies is conducted at mid-load, low speed as this operation point is central in the engine map. The EGR circuit used for this evaluation is the basic circuit were hydrometry is not controlled as detailed in previous chapter 3.

In the first strategy, the nozzle geometry is modified so that the sprays are directed toward the piston bowl despite the early timing of the pilot injection. In this case, the geometry of the piston is adapted to limit spray/piston interaction during the main injection. In the second strategy, the hardware is not modified but the injection strategy is. In this case, the pilot injection is performed around the low-pressure TDC (LP-TDC), between the exhaust and the intake stroke.

In both cases, first the description of the strategy is given. A presentation and analysis of the results in term of engine performance and pollutant emissions is then proposed. The evaluation of the EPI strategies is performed comparing the results with a conventional single injection strategy.

The purpose of this chapter is to evaluate the potential of each strategy in term of main pollutant emissions (NO_x and soot) and fuel consumption. This means that the detailed understanding of all the processes and mechanisms will not be performed, although the major insights that explain the results are mentioned. A deeper analysis will be made in the following chapter6 with the strategies that have demonstrated the best potential.

4.2 EPI Strategies with Narrow Angle Nozzle

In this section, a narrow angle nozzle configuration is investigated with two different adapted piston bowl geometries. Parametric tests are performed with the aim of appreciating advantages and limitations of the strategy. In each case, results are compared with equivalent conventional single injection strategy. The analysis is focused on the combustion process evolution, NO_x-soot emissions and engine performance.

4.2.1 Hardware Selection

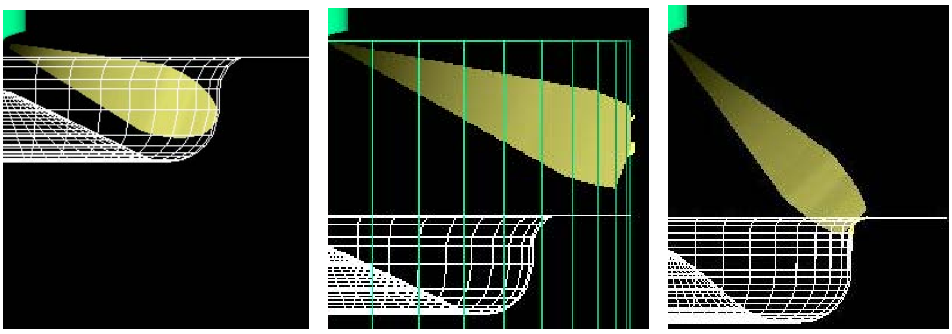
The nozzle is designed with a reduced spray included angle that permits the injection of fuel early in the compression stroke, directing fuel sprays toward the piston bowl. The original piston was not maintained for this study as the shape of the bowl is so that fuel spray during main injection around TDC would directly impinge onto the piston protrusion.

4.2.1.1 Narrow-Angle Nozzle Selection

The conventional nozzle, whose full specifications were given in chapter 3, has a 140° included spray angle for a good adaptation with the original piston bowl design as it is shown in figure 4.1(a). However, as it is also shown in figure 4.1(b), it is not suitable for an EPI injection strategy injecting at 60° $bTDC$.

The spray included angle of the modified nozzle has been calculated so that its value is the maximal that permits the injection at 60° $bTDC$ with the spray targeting the piston bowl as shown in 4.1(c). The included angle was hence reduced from 140° down to 90°. The number and the geometry of the nozzle holes are the same as for the original nozzle i.e. 6 holes with a diameter of 0.214 mm.

Note that in the representation of the spray path in previous figure 4.1, original nozzle spray evolution is shown together with original bowl while modified nozzle spray evolution is shown with modified bowl shape. The selection of the shape of the adapted bowl is described in next section.



(a) Original configuration, injection around TDC
(b) Original configuration, injection at 60°bTDC
(c) Modified configuration, injection at 60°bTDC

Figure 4.1. *Spray-bowl interaction for original and modified configuration. Illustrations were obtained from simulations with DIES program (see section 3.4.2.1).*

4.2.1.2 Adapted Piston

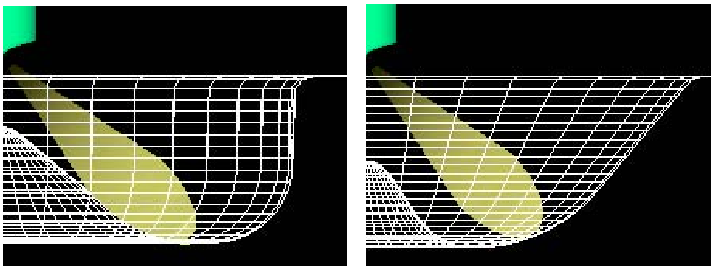
Piston bowl design was adapted to the narrow angle nozzle in order to limit the interaction between spray and piston during the main injection i.e. when the piston is close to the TDC.

The protrusion of the original piston has an angle of 128.7°, wider than that of the included angle of the narrow-angle nozzle, meaning that in case of close to TDC injection conditions, the fuel sprays are directed toward the protrusion. Strong interaction between spray and protrusion wall would limit the inclusion of air inside the fuel spray, affecting hence the combustion process.

Considering this aspect for the adapted pistons, the protrusion angle was reduced. In the first case, it was reduced down to 100° meaning that the fuel jet would follow the protrusion wall. In a second approach, the protrusion angle was further reduced, also its height was limited in order to maximize air inclusion into the fuel jets. The evolution of the fuel jet into the combustion chamber for the two modified piston bowl designs is shown in figure 4.2.

It can be observed in this representation that the two piston bowls are relatively different from each other. The first design shown in figure 4.2(a) is more conventional with vertical wall on the outside of the bowl while the other design (figure 4.2(b)) shows an open bowl shape with inclined wall.

The volume of the two adapted bowls was increased compared to the original design in an objective of reducing the compression ratio. Indeed, as the adapted design was set for the introduction of EPI strategy, compression



(a) Vertical wall bowl, injection around TDC (b) Inclined wall bowl, injection around TDC

Figure 4.2. *Spray-bowl interaction for narrow angle nozzle and two adapted bowl designs. Illustrations were obtained from simulation with DIES program (see section 3.4.2.1).*

ratio reduction is desirable to delay premixed combustion, therefore providing longer ignition delay and more homogeneous mixture and also bringing the premixed combustion process as close as possible to the TDC to enhance cycle efficiency.

Main characteristics of the adapted pistons are given in the following table 4.1.

Table 4.1. *Main Characteristics of the Adapted Bowl Designs.*

	Vertical Wall Bowl Piston	Inclined Wall Bowl Piston
Bowl Diameter	80.5 mm	92 mm
Bowl Depth	23 mm	24.5 mm
Bowl Volume	102.4 cm^3	103.65 cm^3
Compression Ratio	15.3 : 1	15.2 : 1

4.2.2 Hardware Influence in Conventional Combustion

A first analysis has been carried out injecting the fuel in one shot around TDC (i.e. conventional injection strategy). The objective is to investigate the possibility of operating the engine in conventional diffusion-controlled combustion with narrow-angle nozzle configurations. However, as discussed in the previous section, narrow-angle nozzle is not supposed to provide good results in conventional diffusion-controlled combustion conditions since the jet path across the combustion chamber is not optimized, which might compromise the correct in-cylinder air utilization.

4.2.2.1 Tests Definition

The test conditions set for the evaluation of the hardware in conventional combustion strategy is mid-load and low speed as it is also the case for the evaluation of the introduction of EPI strategy with adapted hardware described in this section.

Main data of the operating conditions are displayed in following table 4.2. As explained in chapter 3 (section 3.3.7), the fuel mass injected was maintained constant for all those tests, given that the operation point is always the same. The injection timing was slightly varied around the TDC in order to offer an more ample view of the results shown in term of fuel consumption and pollutant emissions.

Breathing conditions were set to the nominal value of the equivalent full size engine running in the same operation conditions. Note that the equivalent multi-cylinder engine was equipped with a two stage boost system that is described for instance in reference [1].

Table 4.2. *Main Settings for Conventional Injection Strategy.*

Load (%)	Speed (rpm)	Fuel Mass (mg/cycle)	SoI (°aTDC)	IP (bar)	P_{int} (bar)	P_{exh} (bar)	T_{int} (°C)	EGR (%)
50	1200	117	-4/+0/+4	1950	2.3	2.6	50	33

4.2.2.2 Combustion Process Analysis

In following figure 4.3, the RoHR traces obtained for the three configurations are displayed in the bottom part of the graph. In the top part, the evolution of the estimated flame temperature is shown. The figure shows only the results obtained for the intermediate SoI timing (i.e. $+0^{o}$ $aTDC$) for the sake of conciseness. Together with the traces of the RoHR, the injection rate trace is shown.

Looking first at the bottom part of figure 4.3, it appears that the ignition delay (defined as the angle between SoI and SoC) increases in case of narrow angle configurations. The amount of fuel burned during the premixed combustion stage also increases but to a greater extent in case of open bowl configuration.

During the fast diffusive combustion, narrow-angle configurations lead to reduced RoHR compared to the original wide angle nozzle case. On the other hand, both narrow-angle nozzle configurations show a higher RoHR after the end of injection.

The trend observed on the top part of the graph showed a clear reduction in the flame temperature along the whole combustion for both narrow angle nozzle configurations compared to the original configuration, with only slight differences between vertical and inclined bowl cases.

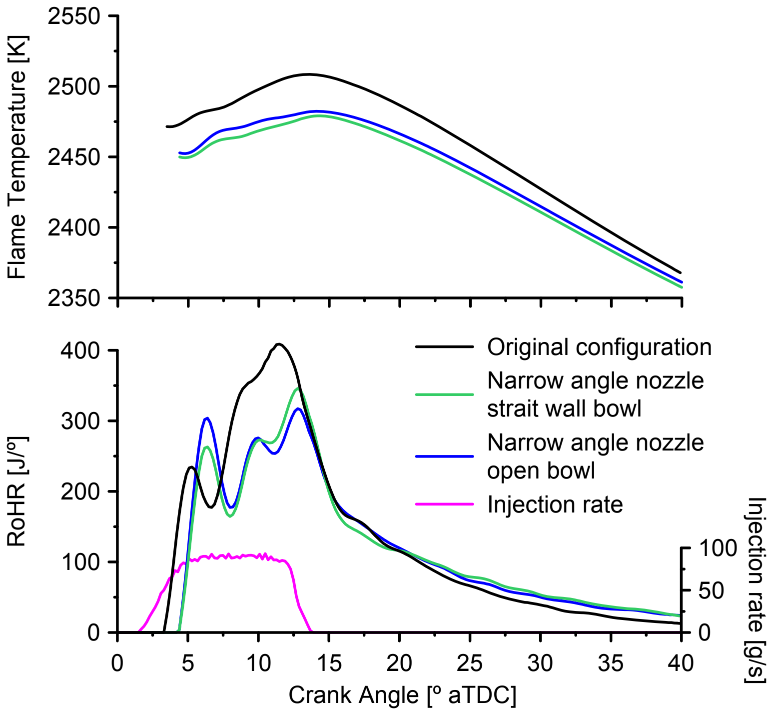


Figure 4.3. *RoHR and injection rate traces under conventional combustion for the different configurations tested.*

Ignition delay depends on in-cylinder temperature and pressure, thus the increment observed with the narrow angle nozzle configurations is a direct consequence of the lower compression ratio of those cases. The higher amount of energy released during the premixed phase of the combustion in the case of the open bowl configuration compared to the strait wall bowl configuration despite the same ignition delay might be the consequence of the major quantity of fuel mixed with air during ignition delay for the open bowl configuration. This means that, during the initial stages of the injection process, the important interaction between the spray and the piston protrusion observed in the straight wall configuration worsens the air/fuel mixing process.

Mixing rate during fast diffusive combustion is directly related to both gas density and oxygen concentration [2, 3]. Intake oxygen concentration was kept constant in all cases by maintaining EGR rate, but gas density was decreased in narrow angle nozzle configuration due to the compression ratio reduction. Additionally, the important spray-piston wall interactions during the diffusion-controlled combustion result in an air/fuel mixing process deterioration and poor utilization of the oxygen present in the cylinder.

The fact that RoHR is higher for narrow angle nozzle configurations during the late diffusive stage means that more quantity of fuel is burned after the end of injection event (during the expansion stroke) when in-cylinder gas temperature and density rapidly decrease. The higher mass of fuel burned in this poor thermodynamic environment (i.e. when soot oxidation process is slowed down) should affect soot emissions. It will, in addition, have a negative impact on the cycle efficiency.

The differences observed in terms of flame temperature are a direct consequence of the compression ratio difference between the original and the narrow-angle nozzle configurations. Indeed, the equivalent oxygen concentration in all cases leads the flame temperature differences to be a function of the differences in ambient gas temperature. Given that the intake temperature is similar in all cases, the higher compression ratio leads to a higher in-cylinder gas temperature at the end of the compression stroke i.e. right at the beginning of the combustion process.

4.2.2.3 Pollutant Emissions Results and Analysis

Results in terms of pollutant emissions and fuel consumption are displayed in figure 4.4. In this figure SoI sweep results are displayed for the three configurations tested.

Trends regarding SoI variations are the commonly known for conventional combustion process in all cases: a delay in fuel injection results in a NO_x reduction while both soot emissions and fuel consumption increase.

It can be observed that the flame temperature reduction observed when changing from the original configuration to the narrow angle nozzle configurations has a impact on NO_x emissions that are reduced considering a same SoI timing.

Soot emissions on the other hand are increased with both narrow angle nozzle configurations however with an important difference from one case to another. Soot emissions increase is much more important in case of the vertical wall than it is in the inclined wall bowl configuration.

Impact of configuration on the fuel efficiency is limited to a slight increase when comparing open bowl configuration with original configuration for a same SoI. Vertical wall bowl configuration however results in a more noticeable increase.

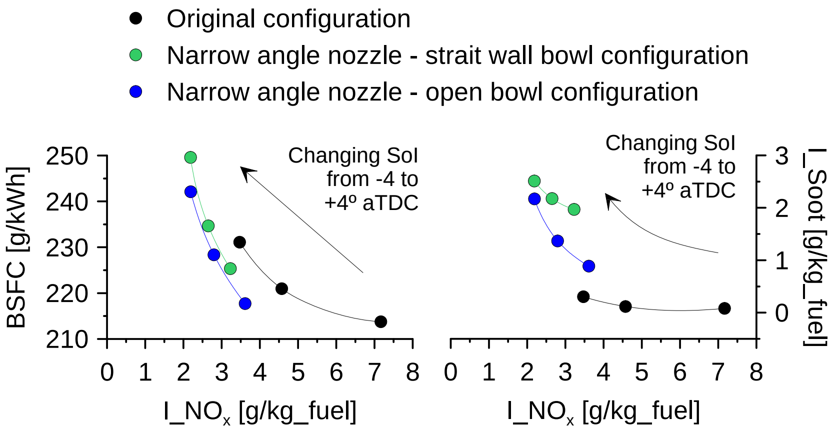


Figure 4.4. *Main pollutant emissions and fuel efficiency under conventional combustion for the different configurations tested.*

Although the trends are relatively clear for NO_x emissions (as they follow the trends observed in terms of flame temperature), the evolution of soot emissions from one configuration to another needs some more analysis.

From the literature review, it is known that in conventional combustion soot emissions are mainly the result of weak oxidation process during the late diffusive stage. This easily explains that switching from original configuration to narrow angle nozzle configurations, soot emissions are increased. However, the difference in terms of soot emissions observed between the two narrow angle configurations are not reflected in the trends of the flame temperature.

The increase is therefore more likely to be due to a largely worsened mixing process for the strait wall bowl that was observed previously in the analysis of the RoHR curve. This affected mixing process slows down slightly the combustion process that ends later in the expansion stroke, when flame temperature dropped further, weakening the soot oxidation process.

This affected mixing process and slowed down combustion is also reflected by increased fuel consumption for the vertical wall bowl compared to the open bowl configuration. The slight difference between the open bowl configuration and the original is also due to a slight slowdown but also to the compression ratio difference that affects the cycle efficiency.

4.2.3 Results with EPI Strategy

As discussed before, narrow angle nozzle configurations were not designed to improve the diffusion-controlled combustion process. The objective is to allow injecting fuel early in the compression stroke without impinging the cylinder liner walls. In this case, original wide angle configuration has not been tested together with EPI strategy due to the extreme liquid spray impingement on cylinder liner described in the literature review in chapter 2.

4.2.3.1 Tests Definition

The two adapted bowls have been tested in the same conditions at mid load and varying the fuel mass injected during the EPI following two steps: in a first step, 20% of the total fuel mass was injected during EPI and in the second step, this amount was increased up to 40%. Full details of injection repartition and SoI timings are given in following table 4.3. As commented in chapter 3, the injection system permits the amplification of injection pressure for only one event in the cycle, so injection pressure was different between EPI and main injection.

Table 4.3. *Main injection settings for EPI strategy operation.*

$m_{f-total}$ (mg/cycle)	SoI_{EPI} (°aTDC)	m_{f-EPI} (mg/cycle)	IP_{EPI} (bar)	SoI_{main} (°aTDC)	m_{f-main} (mg/cycle)	IP_{main} (bar)
117	-60	23.4 (20%)	900	-4/+0/+4	93.6 (80%)	1950
		46.8 (40%)			70.2 (60%)	

4.2.3.2 Combustion Process Analysis

Figure 4.5 shows the evolution of RoHR along the combustion process when injecting part of the fuel as an EPI.

The combustion can be divided in three different stages as described in the literature review: cool and blue flames, hot premixed combustion and main diffusive stage.

The injection rate trace is also displayed on the graphs, making clearly visible that for the EPI event, injection and combustion process are totally separated. On the other hand, the second injection produces a conventional diffusive combustion with the RoHR controlled by the injection process.

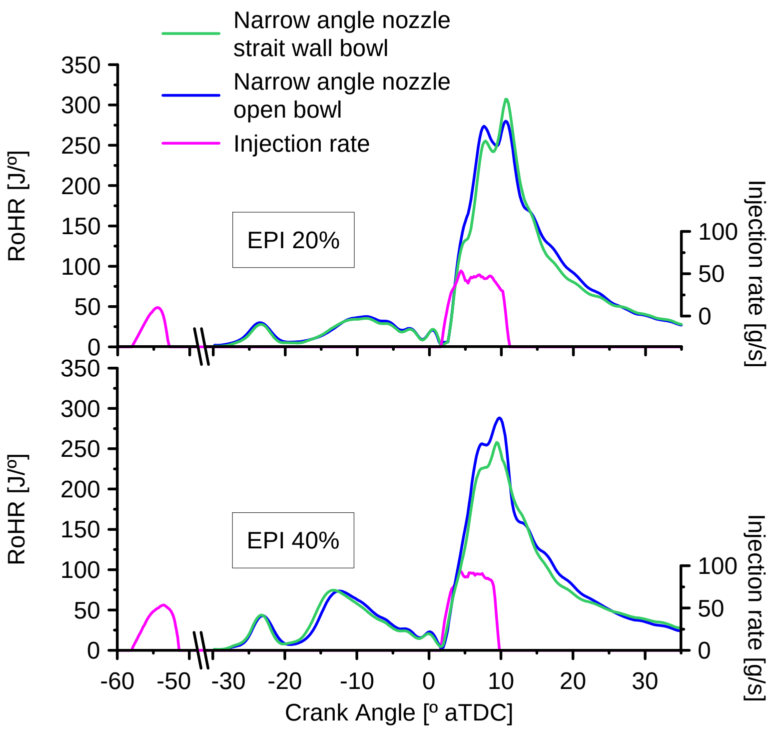


Figure 4.5. *RoHR and injection rate traces with EPI strategy for the different configurations tested.*

On the top graph, where RoHR traces for m_{f-EPI} 20% case are displayed, it can be observed that the SoC of cool and blue flame occurs at the same time for both configurations, which was expected since in-cylinder thermodynamic conditions are the same in both cases. Hot premixed combustion also shows very similar behavior. While the influence of the bowl shape on the premixed combustion is small, differences are more significant during the main diffusive combustion. Ignition delay is similar in both cases but higher mass of fuel is burnt in the first part of the diffusive combustion with open bowl configuration because of the lower piston head as detailed previously.

The bottom part of the graph presents the results for m_{f-EPI} increased up to 40%. In this case, cool flame combustion stage is still similar in both cases but hot premixed combustion is slightly enhanced for the straight bowl wall configuration. RoHR during main diffusive combustion stage is much slower with straight wall bowl geometry compared to the open bowl. Mixing process seems to be more largely affected in this case because main injection duration is reduced and flame does not have sufficient time to grow up and stays on poor mixing conditions along all the injection process.

4.2.3.3 Pollutant Emissions Results and Analysis

Highly premixed combustions have been described to produce an amount of NO_x and soot close to zero. Injecting part of the fuel in those conditions permits a reduction of main injection duration and so might lead to a reduction of soot emissions. On the other hand, premixed combustion has two main effects on in-cylinder conditions at the start of the main injection combustion (SoC_{main}). It increases gas temperature but at the same time reduces oxygen concentration. Effects on NO_x are opposite so NO_x emissions are not expected to be largely affected.

Figure 4.6 shows emissions and fuel consumption results obtained for both EPI and conventional strategies, for the two piston bowl designs tested with narrow angle nozzle. In all cases, the whole SoI_{main} sweep tested is shown.

As expected NO_x emissions are not strongly affected by the introduction of EPI strategy, however soot emission results are totally different from the expectations. In fact, aside from the slight reduction obtained when injecting the lowest m_{f-EPI} with the open bowl configuration, all other cases show a an augmentation of soot emissions. This is particularly the case for straight wall bowl configuration where a dramatic increment is observed.

Engine efficiency is not affected by the piston bowl geometry and EPI strategy does not have a great influence either. Comparing with a single injection case, BSFC is reduced for early main injection timing because part of the fuel is burnt too early in the compression stroke. But when the main injection is delayed, this trend changes owing to the shortening of the main injection and the ignition delay, which is favorable in the case of retarded SoI_{main} conditions.

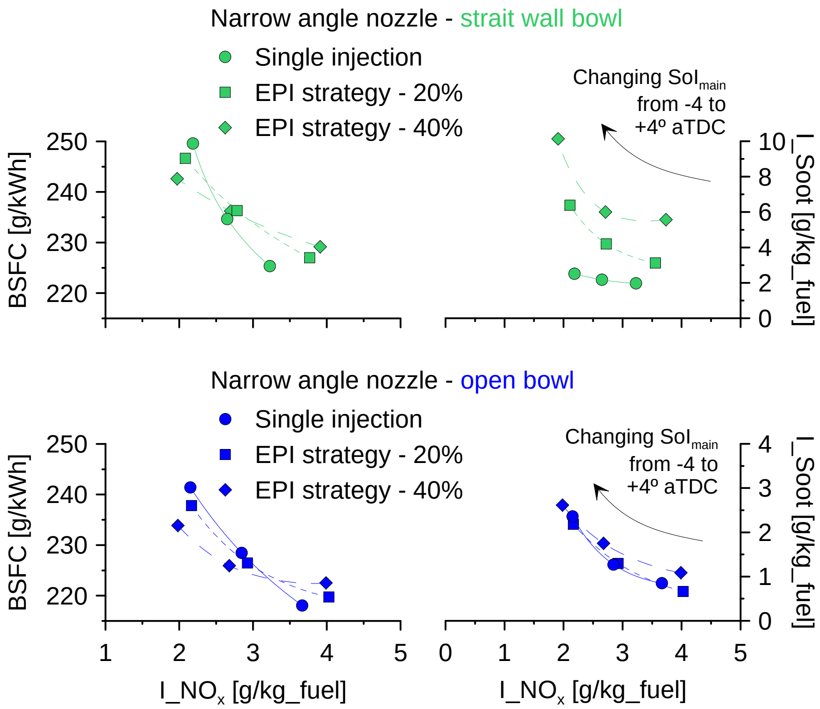
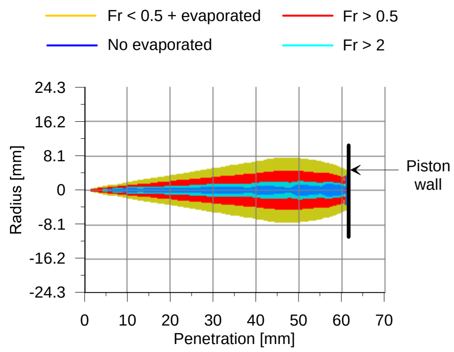


__Figure 4.6.__ Main pollutant emissions and fuel efficiency with EPI strategy for the different configurations tested.

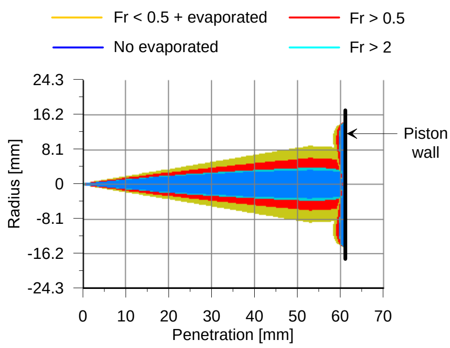
In order to understand the reasons for the unexpected soot increase with the introduction of EPI strategy, calculations were run with DIES code to understand the evolution of the spray during the EPI event.

A representation of the spray composition taken close to the end of the injection process for the two EPI cases (20% and 40%) is shown in figure 4.7. Results show that due to early injection timing ($60^{o}bTDC$) and consecutive low in-cylinder gas density and temperature environment (around 6.3 kg/m^3 and 550 K respectively), the liquid core of the spray reaches the piston wall. The importance of the impingement varies depending on the amount of fuel injected, from apparently slight deposition for the 20% EPI case, to large impingement for the 40% EPI case.

The large deposition of liquid fuel onto the piston bowl during the EPI event can explain part of the unexpected results in term of soot emissions.



(a) EPI 20% case



(b) EPI 40% case

Figure 4.7. *Fuel spray penetration and mixing process during EPI event. Illustrations were obtained from simulation with DIES program (see section 3.4.2.1).*

Indeed, liquid fuel impingement on piston leads to pool fire formation during combustion process [4] with poor air fuel mixing rate and high local equivalence ratio favorable to soot formation. The higher is the quantity of liquid fuel impinging on the piston wall, the higher is the soot formation [5].

Considering the open bowl configuration this phenomenon could explain why soot emissions are higher with higher EPI mass in case of early main injection timing but are not in the case of late main injection. Soot produced by fuel film combustion would define an off-set in soot emissions. This offset would be of lower importance as total soot emissions increases as it is the case when delaying main injection timing.

In the case of straight wall bowl, liquid fuel impingement also occurs during EPI but it does not completely explain the large increase in soot emission. Indeed soot produced by the combustion of the fuel film might be similar to the open bowl case. Moreover, soot is also increased at the lowest mf-EPI where impingement is almost negligible. Soot emission levels reached with vertical wall bowl are extremely high and another parameter has to influence those results.

The bowl shape affects the interaction between the EPI and the main combustion processes. This interaction between both combustion stages is critical, especially for soot, because in-cylinder oxygen concentration is locally reduced by pilot injection combustion. The zones with less oxygen should be avoided by the flame during the last part of the main diffusion-controlled combustion process because the lack of oxygen worsens the soot oxidation process. A non-optimized combustion interaction is believed to play an important role in the soot emission increment observed with straight wall bowl.

4.3 EPI Strategy with Injection at LP-TDC

This section describes a new injection strategy that was implemented for the preparation of the premixed charge. Here, the EPI event, usually performed during the compression stroke, has been advanced up to the low-pressure TDC (LP-TDC), between the exhaust and the intake strokes.

4.3.1 Injection Strategy Description and Hardware Selection

Figure 4.8 describes the valve motion and injection events. The presence of a negative valve overlap (NVO) during EPI ensures that no fuel will go out of the combustion chamber (valves are closed during the entire duration of the EPI). After the EPI, the intake valves begin to open. As this occurs, fresh air enters the cylinder and impedes the EPI fuel from leaving the cylinder. It is interesting to note that the mixing time is greatly increased compared to EPI during the compression stroke leading to a more homogeneous premixed combustion.

The HVA system permits full flexibility regarding the instant of the closing of the exhaust valve (EVC) and that of the opening of the intake (IVO). In a first approximation, those timings were defined in order to insure that valves are completely closed during the EPI event. This resulted in an NVO duration of 40^o, considering that EVC and IVO are defined at a valve lift of 1mm.

Since the EPI event is performed while the piston is in its TDC position, the fuel jets of the EPI remain within the piston bowl (at this instant the piston is in the same position as it is for conventional close to TDC injections). For this reason, the original hardware (nozzle and piston bowl) was maintained for the evaluation of this strategy.

4.3.2 Preliminary Results

As it was the case for the evaluation of the narrow angle nozzle configuration, the evaluation of the new injection strategy was performed at mid-load and low speed, with the same overall settings as described in table 4.2.

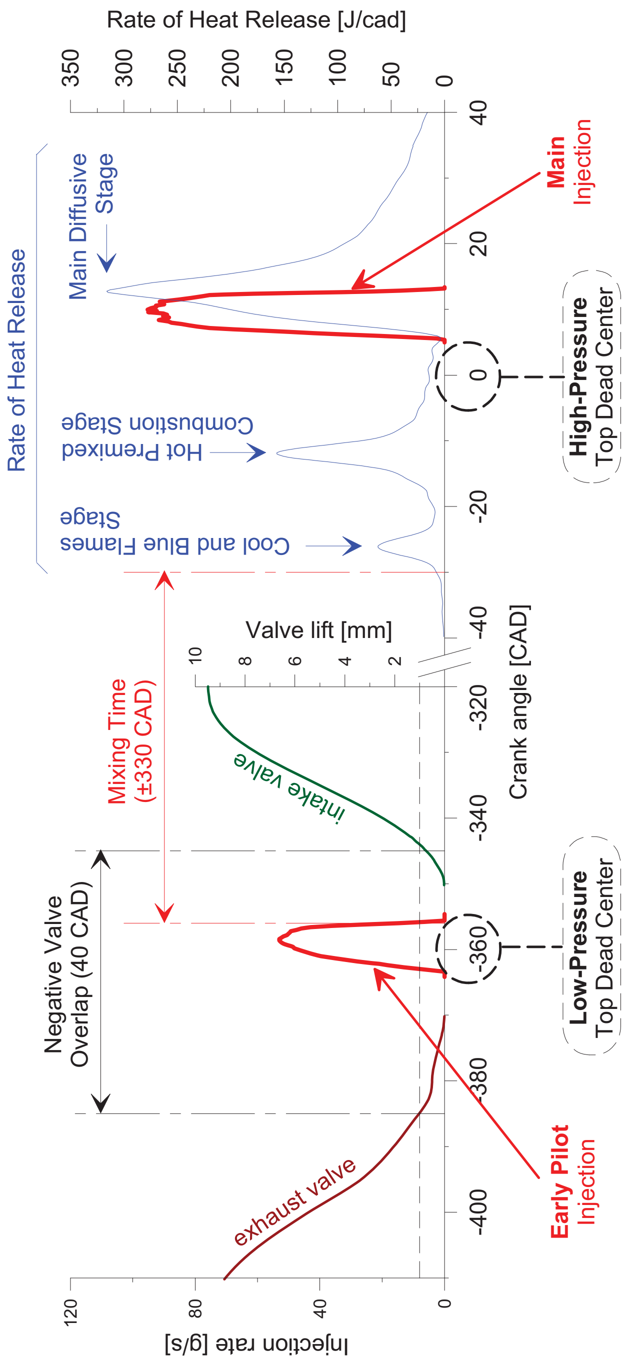


Figure 4.8. EPI strategy injecting part of the fuel at LP-TDC.

In a first approach, tests were run with 40% of the total fuel injected during EPI event at LP-TDC. The exact SoI for EPI event was set at 365° bTDC in order to center the injection with respect to the LP-TDC as shown in previous figure 4.8. As commented before, the duration of the NVO was set to 40°. Tests with conventional strategy (single injection around TDC), were also run with the duration of the NVO set to 40°.

Results in term of pollutant emissions and engine performance of both conventional strategy and advanced EPI strategy are shown hereafter in figure 4.9.

It can be observed that once again, the results obtained in term of soot emissions do not meet the expectations. Soot emissions are largely increased with the introduction of EPI strategy when considering the two most advanced SoI_{main} timings while they remain approximately the same for the most retarded SoI_{main} timing.

Results in term of NO_x are not strongly affected by the introduction of the EPI strategy while fuel consumption results show the same trend as depicted before in the evaluation of narrow angle nozzle configuration i.e. EPI strategy affects slightly negatively the fuel consumption for advanced SoI_{main} timing while results are slightly better for retarded SoI_{main} timing.

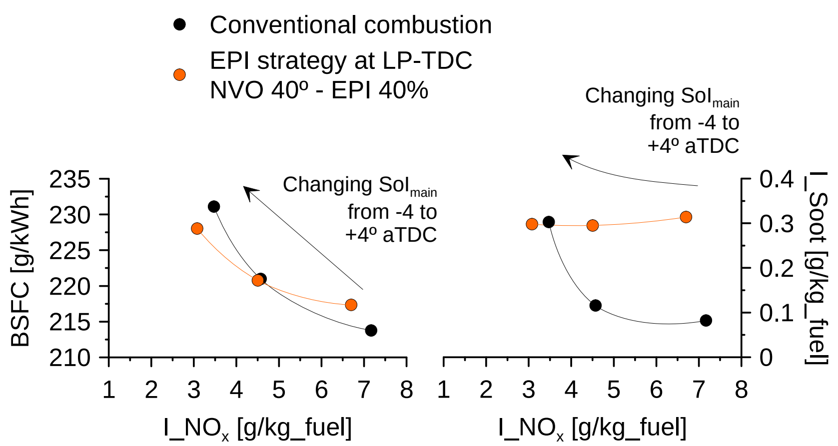


Figure 4.9. *Pollutant emissions and fuel consumption results obtained with EPI strategy injecting part of the fuel at LP-TDC.*

A surprising aspect of the soot emission trends observed with the introduction of EPI at LP-TDC strategy is that they seem to be unaffected by the variation of the SoI_{main} timing while it is widely known that this particular parameter usually have a very strong influence on the soot emissions in diffusive combustion conditions.

A possibility that would explain the trends is that soot emissions originated by the diffusive combustion are very low for all SOI_{main} timing and that most of the soot is formed by the combustion resulting from the EPI event. Highly premixed combustion should not form soot at all, however the in-cylinder conditions found at LP-TDC are of extremely low density and it is assumed that large liquid fuel impingement onto piston wall occurs during the EPI event, possibly leading to the formation of pool fire during the combustion with negative consequences on soot emissions.

4.3.3 Increased NVO Duration

The fuel evaporation process is controlled by mixing and depends highly on in-cylinder gas density and temperature [6]. At LP-TDC, in-cylinder gas conditions are high temperature, but very low density (approximately that of the exhaust system). Thus, a low fuel evaporation rate and high spray penetration are expected, leading to piston wall wetting during the EPI.

Due to impingement, part of the fuel-air mixing process is dependent on fuel film evaporation. Depending on the evaporation rate of the fuel film and the mixing time, liquid fuel could be still present on the piston head during premixed combustion. A diffusive flame from the combustion of the fuel deposited on the piston would appear, potentially producing high amount of soot because of high local A/F ratios and reduced air fuel mixing

One possibility for enhancing air fuel mixing during EPI and reduce liquid fuel impingement is to increase NVO duration. If the exhaust valve is closed earlier, a higher amount of residual gas is trapped in the cylinder during the NVO, thus increasing the gas density at LP-TDC (same volume, higher mass). As the EVC is advanced, the IVO is retarded to expand the gas trapped in the cylinder before opening the intake valve avoiding back-flows toward the intake manifold.

4.3.3.1 Tests Definition

In order to appreciate the influence of the NVO duration on the EPI at LP-TDC strategy, a set of tests have been defined in which the injection scheme was maintained (i.e. 40% of the total mass injected was introduced during EPI event at LP-TDC) and the NVO duration was increased from the original value of 40^o up to 70^o, 80^o and finally a maximum value of 90^o.

The higher mass of residuals retained in the cylinder during the NVO has an impact on the composition of the trapped mass at IVC. Indeed, since the composition of the gas trapped during NVO is the same as that of the exhaust, increasing the residual mass is equivalent to increasing the EGR rate i.e. reducing the oxygen concentration of the trapped mass at IVC. In order to avoid this effect that would largely affect the pollutant emission results, the external EGR rate was modified together with the NVO duration in order to provide a similar total EGR rate (external + internal) in all cases.

The strategy was evaluated for various SoI_{main} timing, however for the sake of clarity, only the results obtained with two SoI_{main} will be displayed and put in comparison with the conventional combustion strategy. Following table 4.4 shows the main settings used for the evaluation of this strategy.

Table 4.4. *Main settings for evaluation of EPI at LP-TDC strategy with increased NVO duration.*

Main injection settings

$m_{f-total}$ (mg/cycle)	SoI_{EPI} (°aTDC)	m_{f-EPI} (mg/cycle)	IP_{EPI} (bar)	SoI_{main} (°aTDC)	m_{f-main} (mg/cycle)	IP_{main} (bar)
117	-365	46.8 (40%)	900	+0/+4	70.2 (60%)	1950

Main valve and EGR settings

$NVO_{duration}$ (°)	EVC (°aTDC)	IVO (°aTDC)	$EGR_{internal}$ (%)	$EGR_{external}$ (%)	EGR_{total} (%)
40	335	375	8	33	41
70	320	390	12.5	28.5	
80	315	395	15.2	25.8	
90	310	400	17.3	23.7	

4.3.3.2 Influence of NVO duration on in-Cylinder Conditions at LP-TDC and during High Pressure Cycle

Figure 4.10(a) shows the evolution of the gas temperature and density at LP-TDC as a function of the NVO duration. Enlarging it from 40° up to 90°, the density is multiplied by 2.3 while the temperature increases by 300K.

The evolution of the fuel spray during the pilot injection has been simulated by means of DIES spray model. Results presented in figure 4.10(b) show that as the NVO is enlarged, liquid fuel impingement is reduced, however it appears that the liquid impingement is avoided only in the case of the largest NVO.

Longer values of NVO enhanced fuel evaporation during EPI by trapping more residual gas before intake process producing higher internal EGR. To keep similar gas composition for all tests during the high pressure cycle, total EGR rate (external + internal) has been maintained constant by reducing cooled external EGR for cases with increased NVO duration (see figure 4.11).

Intake temperature and pressure have been kept constant in all cases. Consequently, temperature at IVC is higher when enlarging NVO as shown in figure 4.11. Also, density during the high pressure cycle is slightly reduced. The effects on the premixed combustion and mixing process for main injection will be analyzed later.

4.3.3.3 Combustion Process Analysis

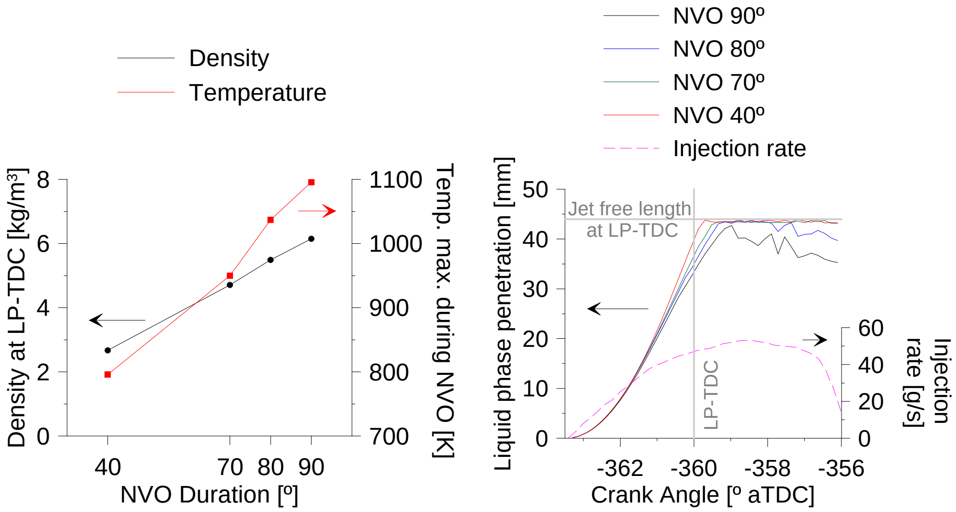
The curves showed in this section correspond to a SoI_{main} timing set at $+4^{o}aTDC$. The trends described are the same in case of other injection timing, but this delayed SoI_{main} was chosen because it permits to separate clearly the premixed combustion from the diffusive combustion.

The evolution of premixed combustion resulting from EPI is displayed in figure 4.12 for all cases. Since temperature is an important factor in premixed combustion evolution, the figure also shows the average in-cylinder temperature evolution during the premixed combustion.

The first peak in the curve of RoHR corresponds the cool and blue flame stage. It starts when the in-cylinder temperature is around $700 - 750\ K$ and does not depend on the local air to fuel ratio. With longer NVO, more residual hot gas is trapped in the cylinder and the temperature is higher during the compression process. The cool and blue flame stage is hence advanced toward the start of the compression stroke when NVO is increased.

The activation and the evolution of the hot premixed stage (found immediately after cool and blue flame stage) depends on both in cylinder conditions (mainly temperature) and local air to fuel ratio. Temperature is higher for longer NVO, and the advanced cool and blue flame stage timing increases it earlier in the compression stroke. Chemical reactions are so accelerated in case of longer NVO, the hot premixed combustion is advanced in the compression stroke and the peak of RoHR is higher.

Commonly, premixed combustions do not show a third stage but here, it is clearly visible in cases of long NVO that after the second peak, the curve of



(a) In-cylinder gas conditions at LP-TDC (b) Evolution of the spray liquid length during EPI

Figure 4.10. *Gas conditions and liquid length evolution during EPI at LP-TDC with varying NVO duration.*

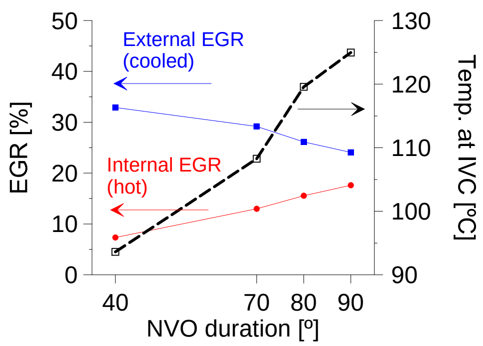


Figure 4.11. *Internal and external EGR rates and temperature at IVC when varying NVO duration.*

RoHR rises again softly. Plausibly this third stage corresponds to the oxidation of CO formed during hot premixed stage, which is activated because of the high gas temperatures. Because the third stage begins before the end of the second, it is hard to define the beginning of its evolution. Following Sjoberg and Dec [7] the start of CO oxidation occurs when gas temperature reaches 1200K. They also pointed out that 1500K is required for complete CO oxidation.

In the present study, CO oxidation process occurs earlier in case of longer NVO since 1200K is attained before. In case of NVO 40, the third combustion process is almost inexistent whereas for NVO 90 the process seems to be completed (RoHR is 0 before main injection) with a peak temperature value of 1500K.

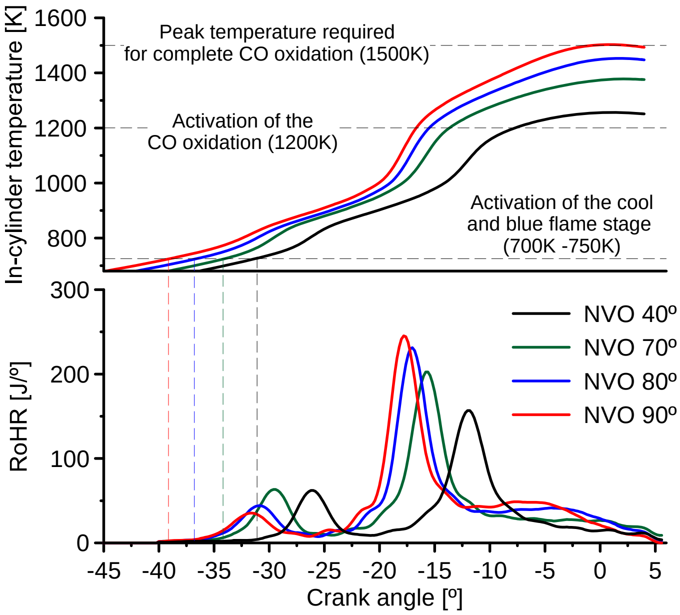


Figure 4.12. *Premixed combustion. Evolution of RoHR and in-cylinder temperature.*

Figure 4.13 shows the RoHR during the main combustion together with the evolution of the in-cylinder temperature and the fuel mass burned. It is evident how the mass burned at the beginning of the diffusive combustion, i.e. at the end of the premixed combustion, depends on the duration of the NVO. This stems from the incomplete CO oxidation described before. During the diffusive combustion, the CO oxidation process is completed. Indeed, at 100^{o} $aTDC$ (not shown in the figure) no difference is observable between the curves of the mass fuel burnt confirming that the entire fuel mass is burnt in all cases.

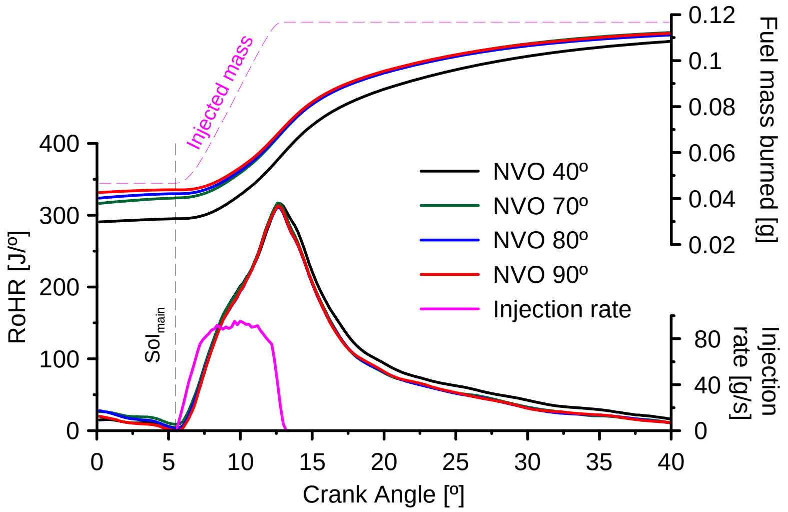


Figure 4.13. *Diffusive combustion. Evolution of RoHR and fuel mass burned.*

In all cases, the same RoHR is observed during the injection process showing that the mixing process is not affected by the slight reduction of the density mentioned in previous section when increasing NVO duration.

The RoHR of the main combustion seems unaffected by the value of NVO except in the case of the shortest NVO, where more heat is released in the late stage of the diffusive combustion, after the end of injection. The origin of this higher RoHR is not clear and could be explained by different conjectures. It could be the oxidation of the CO occurring later in this case because of the lower gas temperature during the overall combustion process. The late burning of the fuel film formed during EPI could be another explanation.

This phenomenon will certainly affect soot emissions because of a higher fuel quantity burned in the late stage when mixing intensity is reduced. Moreover, temperature is higher during the whole combustion process as NVO increases, what should lead to a stronger soot oxidation but also a higher NO_x level.

4.3.3.4 Pollutant Emission Results and Analysis

Emission results obtained with EPI strategy at LP-TDC have been compared with those of conventional Diesel combustion. The results obtained are displayed in following figure 4.14.

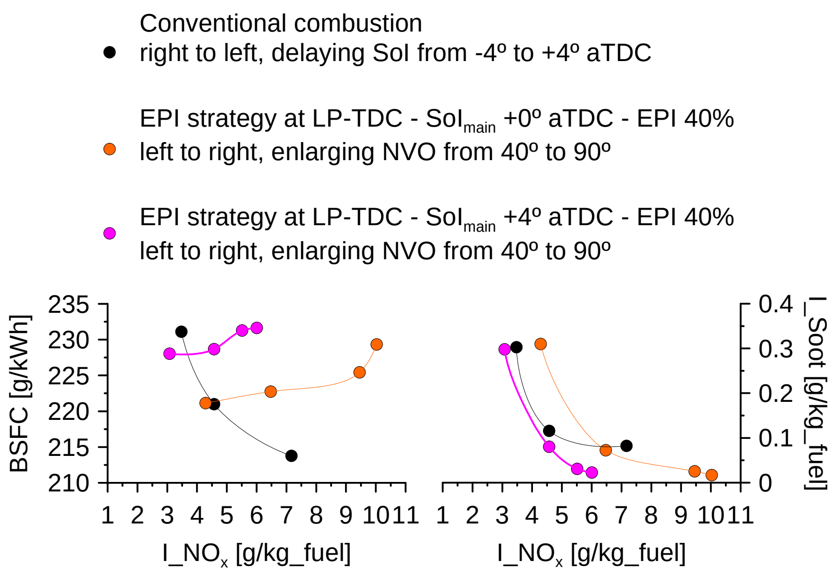


Figure 4.14. *Pollutant emissions and fuel consumption results obtained with EPI strategy injecting part of the fuel at LP-TDC and varying NVO duration.*

Increasing NVO improves the air fuel mixing process during EPI, but it increases the overall temperature during the high pressure cycle. As a result, NO_x emissions are increased as the NVO is enlarged.

Fuel consumption is also increased when increasing NVO, because the premixed combustion is advanced toward the compression stroke. The evolution is not significant when increasing NVO from 40º to 70º because the effect of the earlier premixed combustion is compensated by the reduction of the heat released during the late diffusive combustion stage. For longer NVO however, the main combustion is no longer affected while the premixed combustion is further advanced.

Soot emissions are reduced as the NVO duration is increased. The higher overall temperature during the combustion process as the NVO duration is increased could explain this trend, however it appears that the soot emission level considering a equivalent NVO duration is the same for the two

SoI_{main} tested, suggesting that soot emissions do not result from the diffusive combustion process as it was commented in previous section but possibly from pool fire induced by liquid fuel deposition onto piston wall during EPI process.

This suggests that increased NVO duration has actually the potential for limiting pool fire phenomena permitting therefore the introduction of a EPI strategy at LP-TDC avoiding this drawback. It appears that an NVO duration of 80^o is required to almost completely avoid the soot emissions is those conditions. This results is in accordance with the liquid length calculation during EPI process presented in previous section 4.3.3.2 where it was shown that liquid fuel impingement was almost avoided for an NVO duration of 80^o.

Comparing the pollutant emission results obtained for large NVO cases with those of the conventional diffusive combustion, it appears that the breakthrough described in the literature review of the introduction of EPI strategy is found i.e., soot emissions are reduced at similar NO_x levels.

4.3.3.5 Complementary Results

Previous results have shown that with the introduction of an EPI strategy at LP-TDC with sufficiently enlarged NVO, the NO_x-soot trade-off was enhanced compared to a conventional combustion.

In case of large NVO (80^o and 90^o) soot emissions are reduced to close to zero level, however NO_x emissions are relatively high even for the most delayed SoI_{main} timing due to the high temperature during the cycle induced by the important amount of hot residuals. Two conventional and relatively simple solution can be introduced to reduce NO_x emissions back to more acceptable level: a further delay of the SoI_{main} timing or an increase of the external EGR rate.

A further delay of the SoI_{main} timing would increase further the fuel consumption, that was already relatively high due to the introduction of enlarged NVO duration. It was therefore decided to evaluated emissions and engine performance results increasing the total EGR rate from the original value of 41% up to 43% and 45%. Note that even in this last case, the *external* EGR level was still lower than for the original NVO duration.

Tests were run with SoI_{main} timing set at $+4^o aTDC$. The results in terms of pollutant emissions and fuel consumption are displayed in following figure 4.15.

Results show that the breakthrough obtained with the introduction of EPI at LP-TDC is maintained with increased EGR. In terms of pollutant emissions,

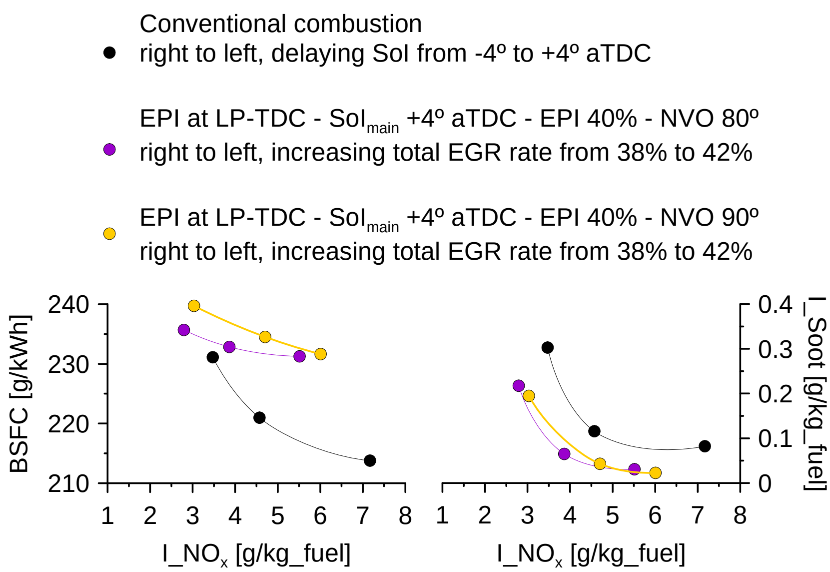


Figure 4.15. *Pollutant emissions and fuel consumption results obtained with EPI strategy injecting part of the fuel at LP-TDC, large NVO duration and increasing total EGR rate.*

it appears that both large NVO have the same potential while regarding fuel efficiency, NVO set at 80º yields a clear advantage compared to NVO set at 90º.

4.4 Conclusions

4.4.1 EPI operation with narrow angle nozzle

Two hardware configurations were tested using a narrow angle nozzle for the introduction of EPI during the compression stroke at $-60^o\ aTDC$.

It was observed that both narrow angle nozzle configurations deteriorate the diffusion-controlled combustion process, mainly due to the non-optimized fuel jet path and the spray interaction with piston walls. Both configurations worsened the NO_x-soot compromise when considering a conventional single injection strategy.

Comparing between both narrow-angle nozzle configurations, the open bowl design offers better emission levels and engine efficiency at the operation mode evaluated (low speed, medium load) than the straight wall bowl design.

EPI strategy has only little impact on NO_x emissions, but soot emissions have been higher than those expected operating with EPI strategy. In this case, 1-D simulations suggest that a liquid fuel film is formed on the piston surface, possibly leading to a pool fire combustion process with poor air/fuel mixing conditions and high levels of soot. Additionally, the non-optimized interaction between the pilot premixed and the main diffusion-controlled combustion stages also contributes to increase soot emissions.

Overall, the results obtained in terms of pollutant emissions and fuel consumption with the introduction of EPI strategy with narrow angle nozzle and adapted bowl design are largely negative. The interesting breakthrough depicted in the literature review in term of soot emissions was not found with this engine configuration. Moreover, in the case of the vertical wall bowl, soot emissions were largely increased.

Main reasons for those unexpected results are believed to be:

- The non-optimized jet path and important spray bowl interaction during the main injection (around TDC).
- The liquid fuel deposition onto the piston during the EPI event, that leads to pool fire and subsequent soot emissions formation.
- The suggested interaction between main injection and premixed combustion that lead to a diffusive combustion process in low oxygen local region and subsequent affected air fuel mixing process.

The benefit of EPI strategy with narrow angle nozzle was not found. On the contrary, the NO_x-soot trade-off was worsened with both the introduction of the modified hardware and the EPI strategy. Another study [8] reached a similar conclusion confirming that narrow angle nozzle is not efficient in operating conditions other than low-load.

4.4.2 EPI operation at LP-TDC

EPI at LP-TDC strategy has the potential of enhancing the NO_x-soot trade-off while avoiding fuel spray impingement onto the cylinder wall during EPI event.

The operation of EPI at LP-TDC requires that the NVO duration is enlarged to provide good results in terms of main pollutant emissions. Indeed, if the NVO duration is kept to its lowest value, the results in term of soot emissions show a deterioration compared to a conventional combustion operation.

It was suggested along the analysis of the results that the increase in soot emissions for short NVO duration are due to the fact that important liquid fuel impingement occurs during the EPI event due to the extremely low density conditions at LP-TDC probably leading to pool fire during premixed combustion with poor mixing process and important local equivalence ratio.

Increasing NVO duration permits to increase largely both density and temperature during the NVO so that the evaporation process during EPI event is largely enhanced, and fuel impingement largely reduced or even avoided. The increasing of the NVO duration has also an impact on the conditions during the high pressure cycle, principally regarding temperature that is increased during the whole cycle due to the higher amount of hot residuals.

In this case, soot emissions are largely reduced. The NO_x emissions on the other hand are increased because of the overall temperature increase during the combustion process. The NO_x-soot trade-off appears to be enhanced compared to a conventional combustion strategy, while the fuel consumption is slightly increased due to an advancing of premixed combustion toward the compression stroke.

Increasing total EGR amount to compensate the temperature increase due to enlarged NVO duration permits to reduce NO_x back to its original value. In this case, the breakthrough of the EPI at LP-TDC is further evidenced with an important reduction of soot emissions at similar NO_x level compared to a conventional combustion strategy.

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Chapter 5

Extended Parametric Study of EPI Strategy

Contents

5.1 Introduction

The evaluation of two EPI strategies in previous chapter 4 evidenced the potential for emissions reduction of the introduction of EPI at LP-TDC. On the other hand, the use of narrow angle nozzle was shown to negatively affect the emissions results.

In this chapter, the analysis of the EPI at LP-TDC strategy is deepened through an extended parametric study. All the results obtained are put in comparison with a conventional single injection strategy and an EPI strategy in which the pilot injection is introduced during the compression stroke, late enough to inject fuel within the piston bowl using a wide open nozzle.

The objectives of this chapter are:

- To present a detailed analysis of the combustion process and understand the mechanisms of the emissions reduction found with EPI strategies.
- To analyze the influence of three of the commonly calibrated engine parameter over the EPI strategy. Those parameters are: the main injection timing, the EGR rate, and the boost pressure.
- To evaluate the potential of the EPI strategy in different engine speed and engine load conditions.
- To present the impact of the EPI strategy on aspects that may limit its introduction in engine via the evaluation of NVH and oil-dilution.

5.1.1 Adapted Hardware for EPI Strategies Introduction

It was identified in previous chapter 4 that a challenging issue of the introduction of EPI was the possible liquid fuel impingement on the cylinder wall or on the piston bowl itself. In order to limit the spray liquid length, the nozzle selected has a largely reduced hole size: 168 μm down from 214 μm for the nozzle used in the previous study in Chapter 4. To maintain a constant fuel flow (mandatory condition so that the maximum power of the engine can still be reached), the number of holes of the nozzle was increased from 6 to 9. The rightness of the selection of this nozzle was confirmed by a study of several nozzle alternatives in low-load and early injection conditions [1], in which this small hole diameter nozzle was shown to provide better emission results in premixed combustion conditions, thanks to a limited spray liquid length and enhanced mixing behavior.

The piston selected for the study has a large bowl (87, 5 mm) and a limited compression ratio (14, 4). The large bowl permits to maximize the free length of the fuel jet during injection, hence helps limiting the impingement of liquid fuel onto its surface. The limited compression ratio leads to limited in-cylinder temperature during the compression process, therefore delaying the ignition of the premixed charge and leading to a premixed combustion process closer to the TDC.

5.1.2 Approach for the Global Study of EPI Strategy - Definition of the Test Plan

5.1.2.1 New Certification Cycles

The EPI strategy studied aims at reducing the pollutant emissions of a heavy-duty Diesel engine. For this reason it is sensible to elaborate the study in operation conditions that are representative of emission certification cycle. In case of heavy-duty, both transient and stationary cycles have been recently harmonized over the EU, USA, Japan and Australia to cover typical driving conditions in those regions.

Compared to the former European Cycles (ETC for Transient and ESC for Stationary), the World Harmonized Cycles (WHTC for Transient and WHSC for Stationary), give more importance to the low load and low speed operation range of the engine as it can be observed in following figure 5.1 (case of stationary cycles).

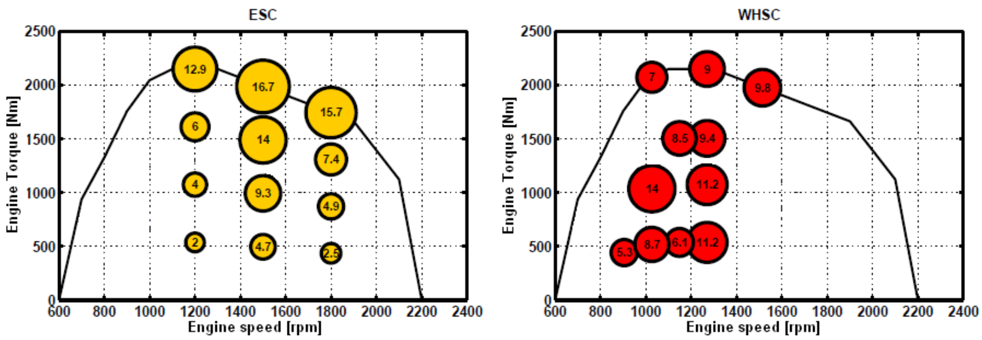


Figure 5.1. *Comparison between ESC and WHSC operation points - The numbers indicated on each operation point represents the weight of the point over the cycle, energy wise.*

5.1.2.2 Selection of the relevant Operation Modes for the parametric Study

In order to limit the measurement effort on the test-bench to a reasonable extend, the number of operation points studied is limited. As the speed dispersion of the WHSC is relatively narrow, the evaluation at one engine relatively low speed only ($\approx$ 1200 rpm) appears to be adequate. The distribution of the load on the other hand is much broader, varying from 25% to 100%. As a consequence, several load levels should be studied.

Going more in the details of the load distribution during the WHSC, mid-load appear to be the operation condition with the most importance: 43.1% of the cycle is run in mid-load conditions (i.e. between 45% and 70% of the full-load), while 31.3% is run in low load conditions (i.e. at $\approx$ 25% of the full-load) and 25.8% in full-load conditions.

Low-load conditions were extensively studied on the same engine [1, 2] in the frame of other combustion studies that are not the object of the current thesis. Those studies showed that large emission reduction in low-load operation conditions can be successfully attained through a PCCI strategy. It was also shown in [3], that the PCCI strategy cannot be implemented in mid-load conditions due to combustion roughness.

The low-load conditions being covered by the PCCI combustion strategy, mid and high load conditions are left. Considering the bigger importance of mid-load of certification cycle, the present parametric study focuses on this operation condition. Additionally, a reduced set of tests is run in high-load conditions as these conditions still bear non-negligible importance for the certification. Finally some complementary tests are also run at mid-load and higher engine speed. In spite of the fact that the mid-load and high-speed operation is not relevant for certification, it is still of interest to observe the impact over a such important parameter as the engine speed on the EPI strategy.

5.1.2.3 Injection Strategies Selected

Previous chapter 4 showed the potential of EPI strategy at LP-TDC, this strategy will hence be further evaluated in the parametric study. In order to a have a base for comparison, tests are also run with a conventional single injection strategy and also introducing an EPI during the compression stroke, late enough in order to avoid injecting fuel outside the piston bowl, even with the use of a wide-angle nozzle.

In case of mid-load conditions, the ratio of fuel injected during EPI is varied from 0% (case of conventional single injection strategy) up to 45% by steps of 15%. The limit of 45% is chosen as it is known from previous studies [3, 4] that a further increase of the premixed charge leads to a very rough combustion process, potentially dangerous for the engine.

In case of full-load conditions, the EPI ratio is varied from 0% to 20% by steps of 10%. In this case the premixed charge ratio is limited to 20% because the maximum in-cylinder pressure of the engine is already reached in those conditions.

As largely described in the previous chapter 4, the introduction of EPI at LP-TDC requires an enlarged NVO. For this study, the required density to avoid liquid fuel impingement onto the piston during EPI was calculated, and the duration of the NVO subsequently adapted considering the exhaust pressure and temperature level.

5.1.2.4 Description of the Parameters Varied in the Study and their Ranges

In the process of engine tunning, the emission calibration aims at adjusting the main combustion inputs in order to find the best compromise mainly between emission requirements for certification and fuel consumption. The present study focuses on the combustion analysis of EPI strategy and the calibration of the engine is not an objective. It is however of great interest to understand the possible cross-effects between the EPI strategy and the main inputs that would be adjusted during an engine calibration process. In a typical emission calibration process, those are usually the EGR rate (or any other similar variant: fresh air mass flow, intake oxygen concentration...), the boost pressure, the main injection timing and the injection pressure.

The injection pressure is usually a variable of adjustment between soot emissions and combustion noise, but in the case of heavy-duty, the NVH consideration are of limited importance. In order to limit the extend of the test plan, its variation is not included in the present study.

Main injection timing has an important influence on both efficiency and pollutant emissions. In typical conventional diffusive combustion process at relatively low engine speed, the injection timing is set close to the TDC. For this study, the nominal injection timing is set $-2°CA\ aTDC$. The timing is then varied in both earlier and later directions by $4°CA$, a representative range of calibration adjustment.

Boost pressure is affecting many processes in the injection-combustion processes because it influences directly the density in the cylinder. In case of mid-load operation, the nominal value of boost pressure is 2.3bar. Variations of ± 0.4 *bar* (i.e. approximately $\pm 20\%$) are applied from this value. For full-load operation the nominal value is 4.6 bar, this value is not increased in the study as the maximum in-cylinder pressure is already reached in case of nominal setting. The variation is therefore only going in the direction of boost pressure reduction. Again the value is varied by approximately 20%, down to 3.8bar. In both load conditions the exhaust pressure is varied according to the boost pressure increase so that the work losses during low-pressure loop of the cycle (exhaust and intake processes) are kept constant[1]. Intake temperature is also maintained constant to the nominal level when boost pressure is varied.

Intake oxygen concentration (Y_{O2}) variation via EGR has a very strong impact on pollutant emissions, especially on the NO_x-Soot trade-off. It is the last parameter that is varied in the study. Several studies [5, 6] showed that the EGR rate itself has no relevance on the combustion processes when the total quantity of gas entering the cylinder is varying (as it is the case when the boost pressure is varied). For this reason the EGR rate was adapted in-order to target a certain Y_{O2} in the cylinder before the combustion (i.e. considering both external and internal EGR). In the mid-load conditions, the nominal value of Y_{O2} (15.5%) is varied by $\pm 1\%$. In full-load conditions two levels of Y_{O2} are evaluated: 16% and 18%.

Table 5.1 presents the test plan of the parametric study for all the operation points evaluated. At mid-load and low engine speed, the entire number of possible input combinations is not covered by the test plan as this would have been too extended. Oxygen concentration, that has a large and complex impact on the emissions formation, is varied on all levels of premixing ratio for the nominal injection timing and boost pressure. Boost pressure is also varied for all injection setting conditions, but high and low boost levels are only combined with high and low oxygen concentration levels so that the maximum range of air to fuel ratio is covered. Injection timing variation is covered for the reference case (without any EPI) and for the 30% premixing ratio conditions, for all the oxygen concentration level considered.

The test plan for the other operation points (at higher engine speed and higher engine load) is detailed in the same table. Number of variation is largely

[1]In case of complete, multi-cylinder engine equipped with a turbocharger the boost pressure variation has an impact on the T/C efficiency and influences the pressure difference between intake and exhaust (P3 - P2). This changing differential pressure impacts the cycle efficiency as it modifies the work during the low-pressure loop of the cycle. This study focuses on combustion analysis only and therefore these aspects are not considered

Table 5.1. *Test Plan for Parametric Study.*

<table>
<tr><th rowspan="2">m_{fuel}</th><th>Engine Speed</th><th colspan="2">EPI_{ratio}</th><th rowspan="2">NVO</th><th rowspan="2">SoI$_{main}$</th><th rowspan="2">P_{int}</th><th rowspan="2">Y_{O2}</th></tr>
<tr><th></th><th>LP-TDC</th><th>-40°aTDC</th></tr>
<tr><th>(mg)</th><th>(rpm)</th><th>(%)</th><th>(%)</th><th>(°)</th><th>(°aTDC)</th><th>(bar)</th><th>(%)</th></tr>
<tr><td rowspan="9">117</td><td rowspan="9">1200</td><td rowspan="2">0</td><td rowspan="2">0</td><td rowspan="2">28</td><td>-6/-2/+2</td><td>2.3</td><td>14.5/15.5/16.5</td></tr>
<tr><td>-2</td><td>1.9/2.7</td><td>14.5/16.5</td></tr>
<tr><td rowspan="3">0</td><td>30</td><td rowspan="3">28</td><td>-6/-2/+2</td><td rowspan="2">2.3</td><td rowspan="2">14.5/15.5/16.5</td></tr>
<tr><td rowspan="2">15/30/45</td><td rowspan="2">-2</td></tr>
<tr><td>1.9/2.7</td><td>14.5/16.5</td></tr>
<tr><td>30</td><td rowspan="4">0</td><td rowspan="2">90</td><td>-6/-2/+2</td><td rowspan="2">2.3</td><td rowspan="2">14.5/15.5/16.5</td></tr>
<tr><td rowspan="3">15/30/45</td><td rowspan="3">-2</td></tr>
<tr><td>100</td><td>1.9</td><td rowspan="2">14.5/16.5</td></tr>
<tr><td>80</td><td>2.7</td></tr>
<tr><td rowspan="3">117</td><td rowspan="3">1650</td><td>0</td><td>0</td><td>28</td><td rowspan="3">-7</td><td rowspan="3">2.3</td><td rowspan="3">14.5/16.5</td></tr>
<tr><td>0</td><td>15/30/45</td><td>28</td></tr>
<tr><td>15/30/45</td><td>0</td><td>90</td></tr>
<tr><td rowspan="4">232</td><td rowspan="4">1200</td><td>0</td><td>0</td><td>28</td><td rowspan="4">-2</td><td rowspan="2">3.8/4.6</td><td rowspan="4">16/18</td></tr>
<tr><td>0</td><td>10/20</td><td>28</td></tr>
<tr><td rowspan="2">10/20</td><td rowspan="2">0</td><td>60</td><td>3.8</td></tr>
<tr><td>50</td><td>4.6</td></tr>
</table>

reduced compared to the mid-load low-speed evaluation points, in the case of high speed all EPI strategies are evaluated on two oxygen concentration levels while the boost pressure and injection timing is not varied. As for high load cases, again all EPI strategies are evaluated while all combinations of two oxygen concentration levels with two boost pressure are measured.

5.2 Influence of the EPI Ratio for EPI Strategies at LP-TDC and during Compression-Stroke

5.2.1 Introduction

The analysis of the influence of the EPI ratio over the combustion process and pollutant emissions is a major objective of this thesis.

This section is divided in two main parts:

First, the injection-combustion processes are analyzed for the different EPI strategies considered. This includes an analysis of the injection conditions at LP-TDC and during the compression-stroke, the possible inhomogeneities of the premixed charge before the beginning of the combustion process, and the analysis of the combustion process itself.

Second, the results in term of pollutant emissions and fuel consumption are presented and related to the key aspects of the combustion that permit to explain how and why the trends presented are found.

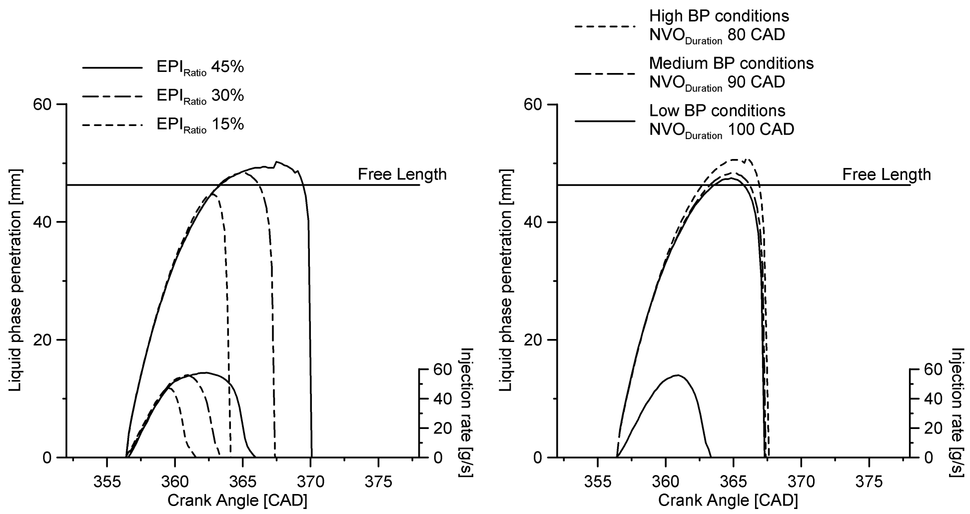
5.2.2 Conditions during EPI Event and Combustion Process Analysis

5.2.2.1 In-Cylinder Conditions during EPI at LP-TDC

As detailed in previous chapter 4, one of the key aspect of introducing an EPI at the LP-TDC is the enlargement of the NVO in order to ensure the density and temperature conditions so that the impingement of liquid fuel on the surface of the piston bowl is limited or even avoided.

The valve timing settings were hence adapted based on the experience gained during pre-study described in chapter 4 and refined after calculation so that, for different exhaust pressure cases, the right conditions are obtained.

Figure 5.2(a) shows the results in terms of liquid length calculation together with the free length for different EPI quantities, in the case of mid-load low speed operation points and nominal boost and exhaust pressure level, while figure 5.2(b) displays the same evaluation for the three different boost pressure levels tested and the middle fuel quantity. It can be observed that the liquid impingement is limited in all cases, whatever the conditions of boost pressure and injected quantity.



(a) 15%, 30% and 45% EPI_{ratio}, nominal Boost Pressure Level (90 CAD $NVO_{duration}$)

(b) Different Boost-Pressure and $NVO_{duration}$ Levels for the Case of 30% EPI_{ratio}

Figure 5.2. *Evolution of the Spray Liquid Length for different in-Cylinder and Injection Conditions at LP-TDC. Calculations performed using DICOM Program (see section 3.4.2.2).*

5.2.2.2 Mixture Homogeneity in Case of EPI during Compression-Stroke

In the case of EPI during the compression-stroke, the possibility of liquid fuel deposition onto the piston head is a possible critical aspect, as it is for injection at LP-TDC. However in this case, the mixture homogeneity at the start of the premixed combustion needs also to be taken in consideration, as it can also affect the emission results.

Indeed, the concept of the EPI relies on the fact that the premixed combustion is lean enough to avoid both soot (local equivalence ratio < 2) and NO_x (local equivalence ratio < 1) formation. In the case of injection during the compression stroke, the ignition delay is limited, and the required homogeneity conditions may not be reached.

Calculations show that the maximum local equivalent ratio at the instant of the start of the premixed combustion never reaches the value of two in any of the conditions tested, meaning that the formation of soot during the premixed combustion should not occur. The calculations however predict that in several

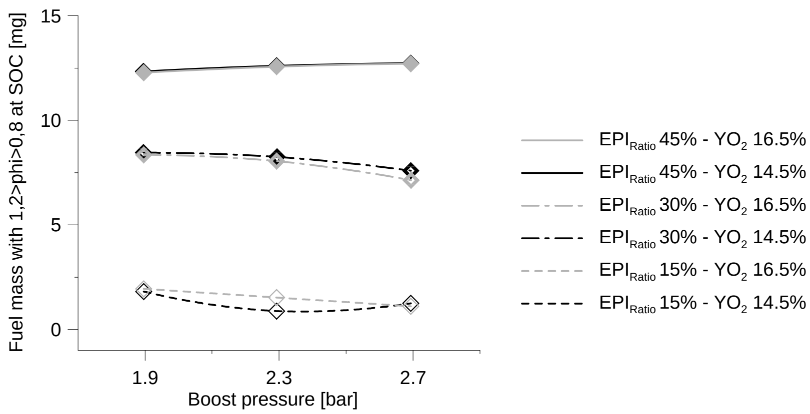
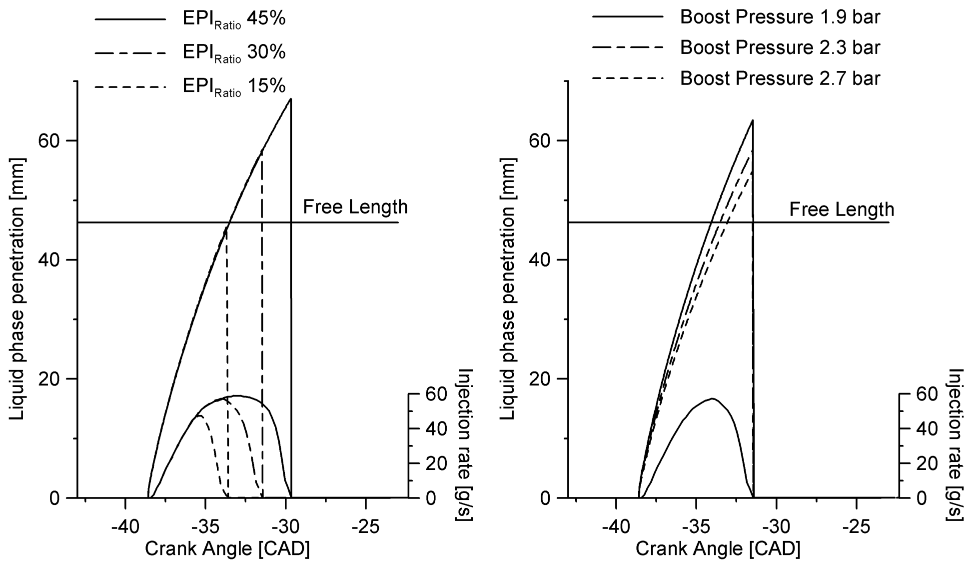


Figure 5.3. *Mass of Fuel found in Conditions of Equivalence Ratio around* 1 *at the Beginning of the Premixed Combustion Process, for different in-Cylinder conditions. Calculations performed using DICOM Program (see section 3.4.2.2).*

conditions part of the fuel is not premixed enough to avoid local equivalent ratios around one at the beginning of the premixed combustion as it is shown in figure 5.3. As a consequence, it is likely that the formation of NO_x during the premixed combustion is not avoided in all cases of EPI during the compression stroke. Indeed, if in the case of lowest EPI ratio, the amount of fuel found in those conditions to form NO_x may be neglected, it is not the case for the two other EPI ratios, where the quantity of fuel in the range of equivalence ratio between 0.8 and 1.2 is respectively around 8 mg and 12 mg. In all conditions, it appears that the boost pressure level is not affecting the amount of fuel found in conditions to produce NO_x during the premixed combustion.

Additionally to this issue, following figure 5.4 shows that the deposition of fuel on the piston during the EPI event is far from avoided when considering the EPI during the compression-stroke for the EPI ratios above 15%



(a) 15%, 30% and 45% EPI_{ratio}, nominal BP level (90 $CAD\ NVO_{duration}$)

(b) Different BP levels for the case of 30% EPI_{ratio}

Figure 5.4. *Evolution of the Spray Liquid Length for different in-Cylinder and Injection Conditions during the Compression-Stroke. Calculations performed using DICOM Program (see Section 3.4.2.2).*

5.2.2.3 Combustion Process Analysis

This section details the analysis of the combustion process when varying the EPI ratio for both EPI at LP-TDC and during the compression-stroke.

For each cases, figures 5.5 and 5.6 show the evolution of the RoHR and the in-cylinder temperature along the combustion process. Additionally the corresponding curves for the case of single injection are also displayed for comparison.

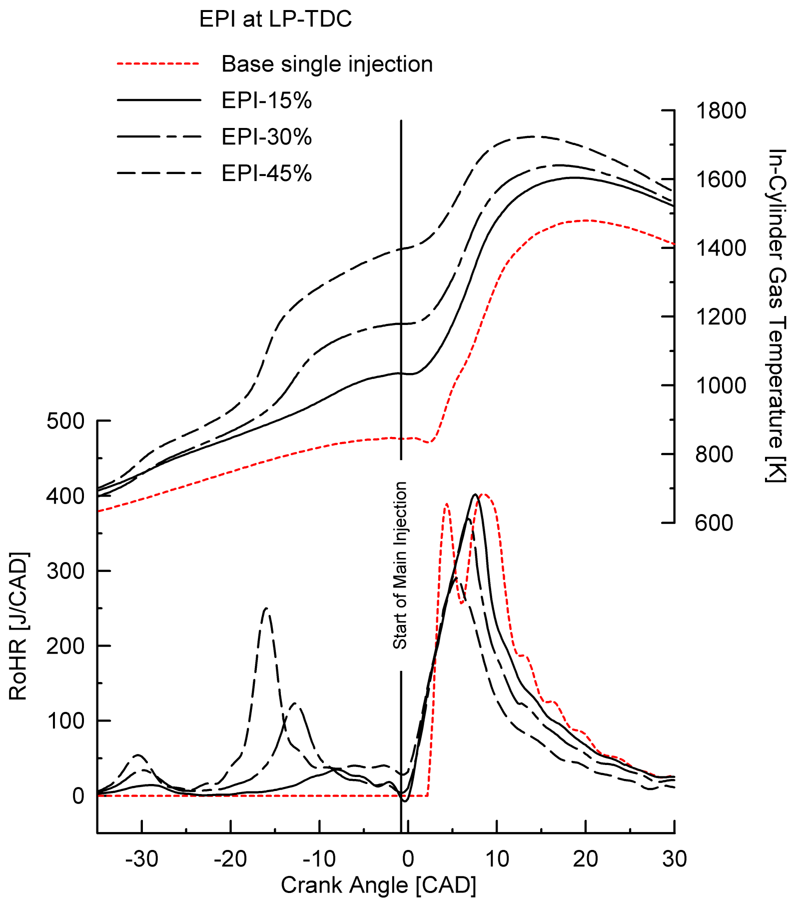


Figure 5.5. *RoHR and in-Cylinder Temperature Evolution with various EPI Ratios Conditions, Case of EPI at LP-TDC.*

In all EPI cases, the premixed combustion starts close to the end of the compression stroke, and the RoHR curve shows the two peaks generally observed in HCCI combustion. Increasing the EPI ratio enhances the cool and blue flame stage since reaction rates are proportional to the fuel concentration, as many authors have observed [7, 8]. The cool flame stage is then closely followed by the hot premixed combustion stage. The activation and the evolution of this stage are affected by both in-cylinder temperature and air-fuel ratio. In the case of high EPI ratio, the enhanced cool and blue flame stage increases sensibly the in-cylinder temperature. This effect, together with the higher overall richness, results in an early activation of the hot

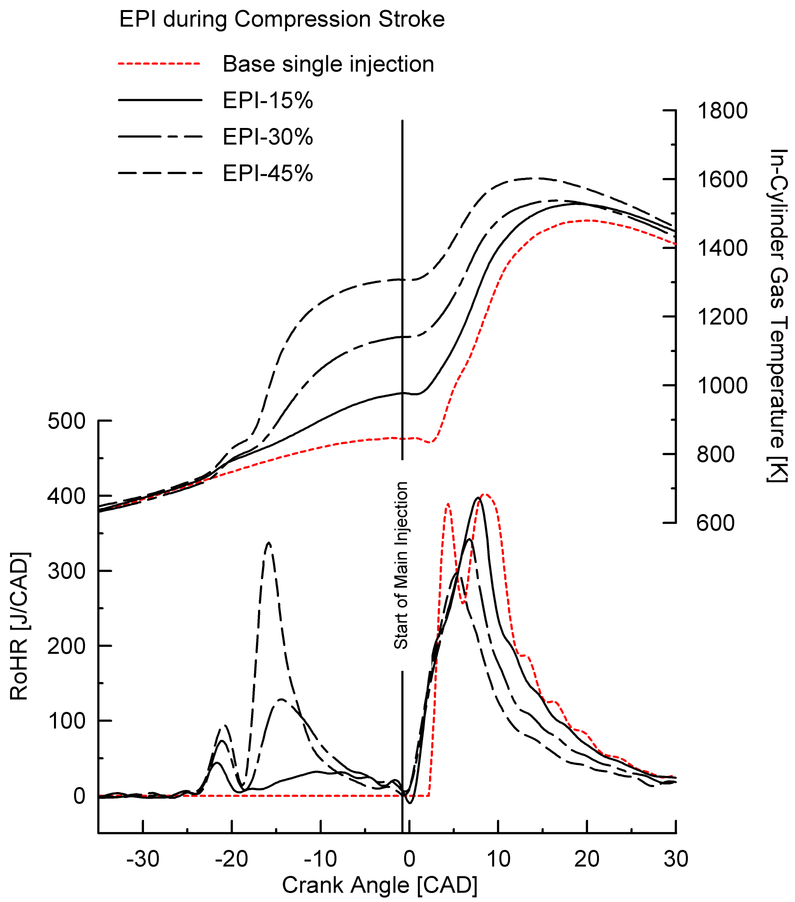


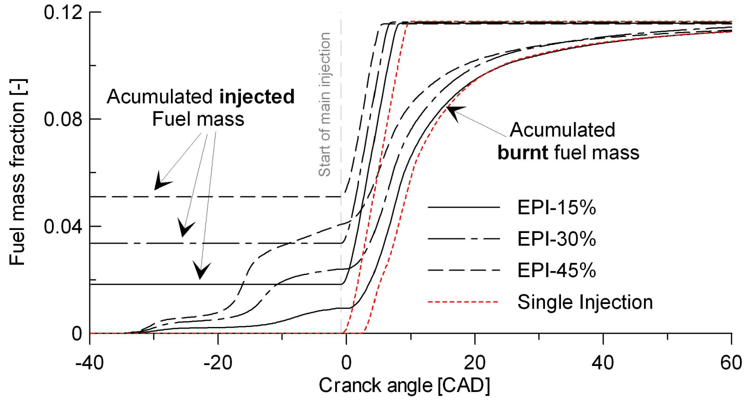
Figure 5.6. *RoHR and in-Cylinder Temperature Evolution with various EPI Ratios Conditions, Case of EPI during Compression-Stroke.*

premixed combustion stage and also in a faster heat release, as it is confirmed by the higher second peak observed in the RoHR. In the case of highest EPI ratio at LP-TDC a third premixed combustion phase is also detected. As detailed in previous chapter 4, this third stage, corresponding to CO oxidation, is activated when the in-cylinder temperature rises over $1200K$. After the premixed combustion the main injection is introduced, leading to a conventional diffusive combustion process that, in all EPI strategies, is characterized by a very short ignition delay and the absence of premixed phase due to the high in-cylinder temperature conditions. This is not the case for the single injection strategy that displays two peak of RoHR after a non negligible ignition delay as it is normally found.

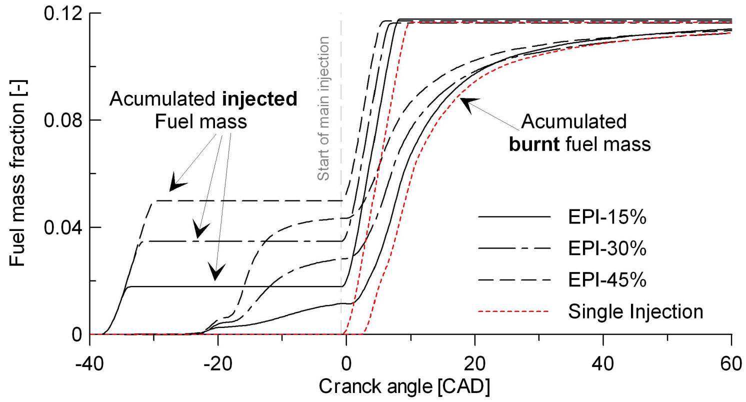
The comparison between the EPI strategies shows that the cool and blue flame stage is activated earlier in case of the EPI at LP-TDC strategy due to the enlarged NVO leading to higher temperature during the compression stroke. Despite the earlier first combustion stage, the main premixed combustion RoHR curve is much smoother in the case of LP-TDC strategy because of the more homogeneous charge (especially in case of the highest EPI ratios). As a result the RoHR peak for this stage is occurring approximately at the same time for the two EPI strategies. The third premixed stage is only noticeable in case of EPI at LP-TDC (for the highest EPI ratio) while the in-cylinder temperature level suggest that the CO oxidation should also take place for the highest EPI ratio in the EPI during compression-stroke strategy. It is likely that in this latest case, the CO oxidation is taking place during the main premixed stage and therefore not directly visible.

The incompleteness of the fuel oxidation process during the premixed combustion stage for all cases is highlighted in following figure 5.7 where the traces of accumulated injected and burnt fuel are shown.

It is clearly visible that the fuel mass burnt at the end of the premixed stage is much lower than the mass injected. This difference could be the indication of unburnt HC or non-oxidized CO, this second possibility being more plausible according to the emissions results described in next section. Toward the end of the main combustion process, around $60°CA\ aTDC$, mass burnt and injected mass are practically equal, meaning that great part of the HC or CO that is still present after the premixed combustion is in fact oxidized during the main combustion stage.



(a) Case of EPI at LP-TDC



(b) Case of EPI during Compression-Stroke

Figure 5.7. *Fraction of Fuel Mass injected and Fraction of Fuel Mass burnt for various EPI Ratios.*

5.2.3 Emissions and Engine Efficiency

Overall pollutant emission results for all cases previously discussed in the combustion analysis section are presented in following figure 5.8.

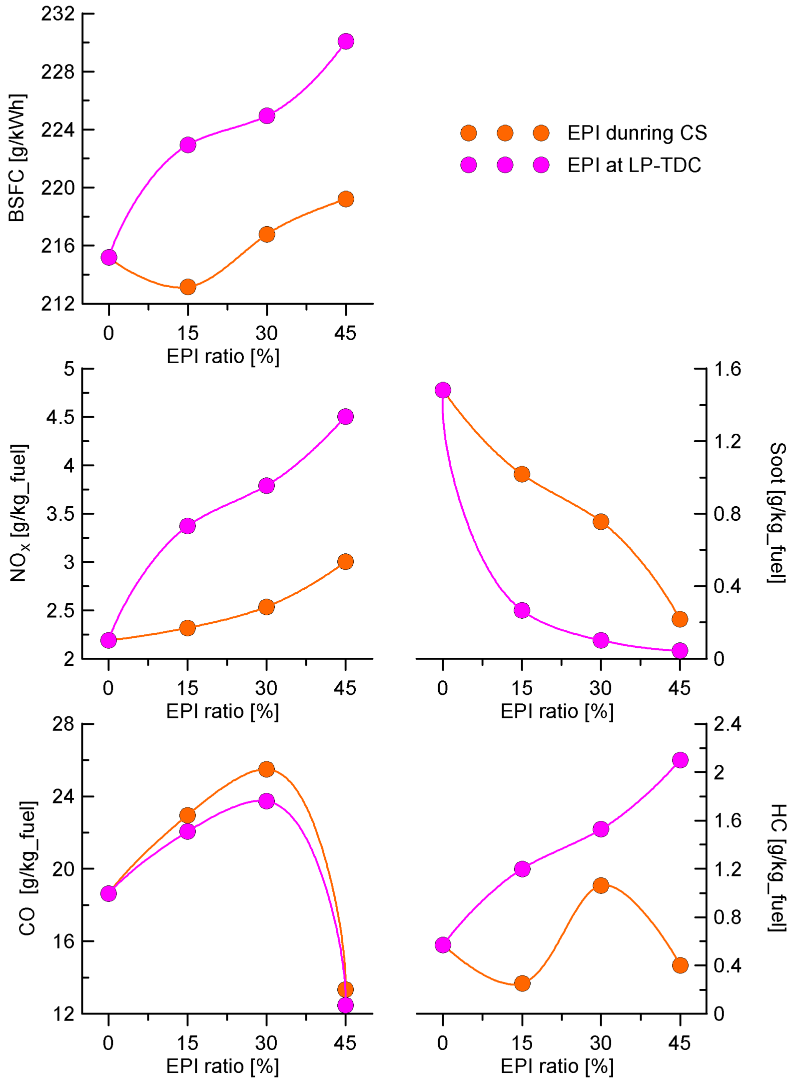


Figure 5.8. *Pollutant Emissions and Specific Fuel Consumption varying EPI Ratio for the two EPI Strategies.*

Emissions resulting from the lean premixed combustion (HCCI combustion) produced before the main injection are mainly of high HC and CO, but close to zero NO_x and soot as largely described in chapter 2. It is then considered that for the EPI at LP-TDC strategy the trends for NO_x and soot emissions should mainly be explained taking into account the diffusion controlled combustion process. In case of EPI during compression-stroke, calculations of mixture heterogeneity at premixed SOC suggest that NO_x could also be formed during premixed combustion, however soot may only be formed during diffusion controlled combustion. For both cases, HC and CO emission results will be explained by taking into account both combustion processes.

5.2.3.1 NO_x Emissions

Figure 5.8 shows that increasing the EPI ratio produces higher NO_x emissions and that this increase is more important in the case of EPI at LP-TDC strategy. These results are in agreement with the maximum flame temperature found during the diffusive combustion process displayed in figure 5.9, those trends being themselves explained by the fact that the higher heat released during the premixed combustion when increasing EPI ratio leads to a higher in-cylinder temperature at the beginning of the diffusive combustion process while, in the case of EPI at LP-TDC strategy, the enlarged NVO also leads to a higher in-cylinder temperature even before the combustion process.

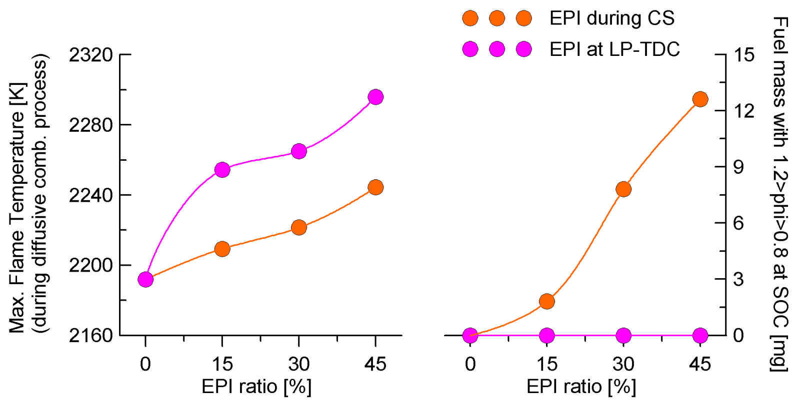


Figure 5.9. *Maximum Flame Temperature and Mass of Fuel found in Conditions of Equivalence Ratio around One at the Beginning of the Premixed Combustion Process.*

On the left part of the figure 5.9 the mass of fuel in conditions of equivalence ratio around one at the beginning at the premixed combustion process (i.e. that may lead to the formation of NO_x during the premixed combustion) are shown. This mass is zero in case of LP-TDC thanks to the very long mixing time in this case. On the other hand, a non negligible amount of fuel could lead to NO_x formation in the case of EPI during compression-stroke, the amount increasing with the EPI ratio. This drawback of the EPI during compression-stroke also follows the trends of NO_x emission with increased EPI ratio, but when comparing to EPI at LP-TDC, it appears to be less impacting than the drawback of increase NVO and subsequent higher in-cylinder temperature in this case.

5.2.3.2 Soot emissions

Figure 5.8 shows a continuous decrease in soot emissions with increased EPI ratio, with a larger reduction in case of EPI at LP-TDC than for the other EPI strategy. Soot emissions are mainly affected by the combustion process after the end of the main injection (as detailed in chapter 2). Increasing the EPI ratio, the main injection duration is shortened and the late diffusive combustion stage starts earlier in the cycle, closer to the TDC, when gas density and temperature are higher.

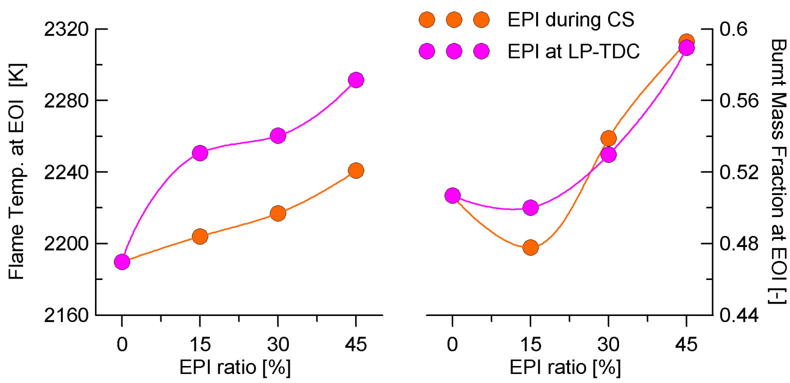


Figure 5.10. *Flame Temperature and Fraction of Fuel Mass Burnt at the Instant of the* EoI.

As shown in figure 5.10, increasing the EPI ratio leads to higher flame temperature at the EoI and, for EPI ratios over 15% it also leads to a reduction of the fuel mass burnt after the EoI. The higher flame temperature during this stage enhanced largely the soot oxidation, explaining the results in term of soot emissions. The large EPI ratio permits to burn more fuel before the late oxidation stage, thanks to a large fuel quantity burnt during the premixed combustion, it is however not the case for the lowest EPI ratio that shows an opposite trend when comparing with the single injection case.

A possible explanation for this is that, on the one hand, the amount of fuel injected during the EPI burns pretty incompletely, only 50% to 70% of the fuel injected is burnt in that phase according to figure 5.7, and, on the other hand, that the single injection case benefits from a relatively large ignition delay before the main combustion that helps getting a fast mixing of fuel with in-cylinder gas at the beginning of the main injection. However the largely increased flame temperature even in these lower EPI ratio cases overcomes this negative aspect and finally leads to lower soot emissions.

Comparing the two EPI strategies, it appears again that the enlarged NVO for the EPI at LP-TDC leads to higher flame temperature therefore explaining why the soot reduction is more important in that case compared to EPI during compression-stroke.

5.2.3.3 HC emissions

HC emissions in conventional diffusive combustion are typically very low, and this is confirmed by the results shown in figure 5.8 for the single injection case. For the cases where an EPI is used, two different trends appear, depending on the EPI strategy used.

In case of EPI at LP-TDC, a steady increase of the HC emissions while increasing the EPI ratio is measured. Because of the extreme mixing time in that case, over leaning phenomena occurs and, on top of this, fuel will reach cold zones close to the cylinder walls and in the crevices. In these cold zones, combustion reactions are slowed down or even completely inhibited, so the fuel present in these zones is not able to burn. As the EPI ratio increases, there is more quantity of fuel that can potentially reach these critical zones, therefore giving explanation to the increase of HC emissions with increased EPI ratio. Despite the noticeable increase of HC emissions in high EPI ratios, it is important to note that the overall level remains acceptable, and clearly much lower than what can be found in pure HCCI combustion strategy.

In case of EPI during compression-stroke, the trend shown when increasing the EPI ratio is inconsistent, but stays in all cases at a very low level. With this strategy, the premixing time is limited and the fuel has no time to reach the cold zones of the combustion chamber before combustion occurs, therefore limiting the HC emissions to a level comparable to that of a normal injection strategy.

5.2.3.4 CO Emissions

The evolution of CO emissions as the EPI ratio increases is similar for both strategies. First, an increase of CO emissions is observed when increasing EPI ratio from 0% to 30% before a large drop to close to zero CO emission is found for the 45% EPI cases.

This trend can be linked to the already mentioned CO oxidation process that occurs in case of premixed combustion when the temperature overcomes 1200 K and that gets close to completeness for temperatures over 1500 K. The maximum temperature reached during the premixed combustion for the different EPI strategies is shown on following figure 5.11 together with the premixed combustion efficiency (i.e. the ratio between the amount of fuel of the EPI and the amount of fuel burnt in the premixed combustion).

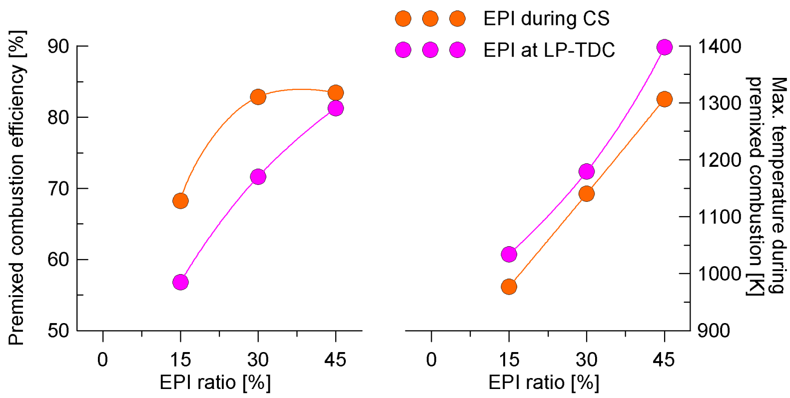


Figure 5.11. *Premixed Combustion Efficiency (Right) and maximum Temperature reached during the Premixed Combustion Process (Left).*

One can observe that the efficiency of the premixed combustion increases with the EPI ratio, and that it is higher in case of EPI during compression-stroke strategy (less over-mixing phenomena in this case, as also noticed in the analysis of the HC emissions). The temperature reached during premixed combustion remains below the threshold of CO oxidation activation (1200 K) for the 15% and 30% EPI ratio cases, where the large increase of CO emissions is noticed. In the 45% EPI ratio case, the temperature during premixed combustion rises well above the 1200 K mark, suggesting that the oxidation of CO during the premixed combustion as a great influence over the final CO emissions after the main combustion.

5.2.3.5 Fuel Consumption

Figure 5.8 shows that the EPI strategy at LP-TDC has a negative impact on fuel consumption, and that this negative impact gets more and more important as the EPI ratio increases. The EPI during compression-stroke strategy on the other hand shows that for low EPI ratio a reduction of the fuel consumption can be found. For higher EPI ratios however, this strategy also shows an increase in fuel consumption compared to a conventional single injection strategy.

In following figure, the center of combustion for both premixed and diffusive processes is shown in order to help understanding the fuel consumption results previously mentioned.

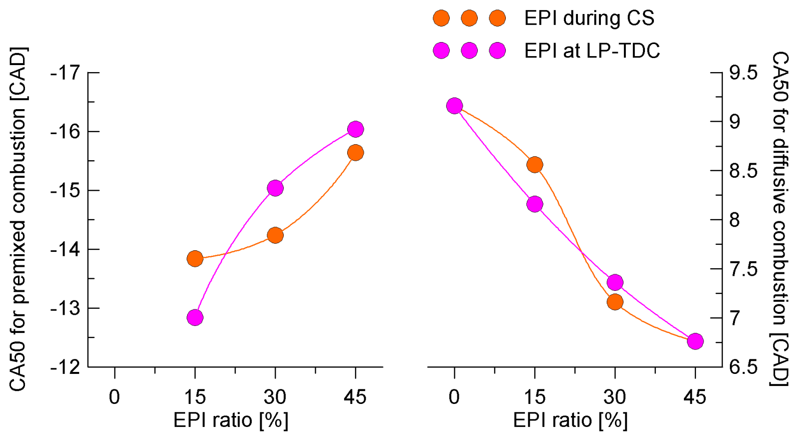


Figure 5.12. *Angle at which* 50% *of the Total Fuel burnt for each Combustion Process has already released its energy. Premixed Combustion on the Left and Diffusive Combustion on the Right.*

Figure 5.12 shows that the center of the premixed combustion is found very early in the cycle and that it gets more and more away from the TDC as the EPI ratio increases. This is due to the fact that premixed combustion reactions are enhanced with increased EPI ratio, and even though the start of the premixed combustion remains unchanged, the rest of the combustion occurs at a faster rate, as it was described in the combustion analysis. Consequently, the compression work is increased and the cycle efficiency is decreased. However, as it can be seen on the right part of figure 5.12, the exact opposite phenomenon occurs with respect to the main combustion. Increasing

EPI ratio, the mass injected during main injection is reduced, and the gravity center of the main combustion is moved closer to the TDC.

The balance of the two phenomena appears to be negative for most of the EPI cases compared to a single injection, except in the case of lowest EPI ratio. In this case, the amount of fuel burnt during the compression stroke is limited (and so is the negative impact on the compression work) while the shift over the main combustion still noticeable.

The comparison between the two EPI strategies shows that the trends and the actual values of center of combustion for both premixed and diffusive events are very similar. In spite of this similarities, the fuel consumption results show a very important drawback in the EPI at LP-TDC compared to the EPI during compression-stroke. The combustion efficiency itself being in all cases relatively high (low HC and CO emissions), the only explanation for this fuel consumption increase is the negative impact of the enlarged NVO that leads to a non-negligible re-compression work at the LP-TDC.

5.3 Influence of commonly varied engine parameters and cross-effects with EPI strategy

Previous section depicted the major trends observed in term of pollutant emissions and fuel economy when varying the quantity of fuel injected during the EPI. However, in order to define the potential of such strategy, its impact needs to be evaluated when varying other engine calibration parameters as it would be the case in a calibration optimization process.

5.3.1 Influence of Start of Injection Timing

The timing of the SoI variation was limited to the 30% EPI ratio cases for the different EPI strategies as well as the single injection case. It was varied by $4°CA$ in both advancing and delaying directions from the reference injection timing of $-2^oCA\ aTDC$.

Following figure 5.13 shows the NO_x-BSFC and NO_x-Soot trade-off when varying the SoI for three cases: single injection, and 30% EPI ratio for the EPI at LP-TDC and EPI during compression-stroke.

The main trend observed as the SoI$_{main}$ is delayed from $-6^oCA\ aTDC$ to $+2^oCA\ aTDC$ is the same for all the strategies depicted i.e. the well known decrease of NO_x and increase of soot and fuel consumption.

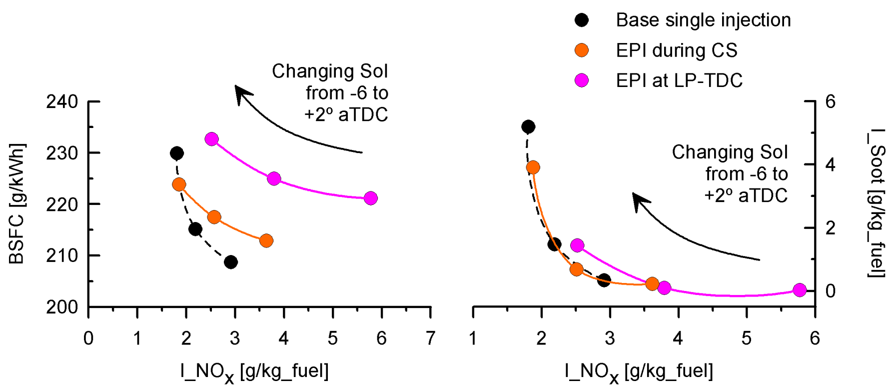


Figure 5.13. *NO_x, Soot Emissions and Fuel Consumption Results varying* SoI_{main} *in Case of Single Injection and EPI Strategies.*

One can observe that the EPI strategy with injection during compression-stroke and the conventional single injection strategy show similar NO_x-Soot compromise although the trade-off is shifted between two strategies (toward slightly higher NO_x emissions in case of the EPI strategy). Regarding the fuel efficiency, although the trends and the overall level are similar between both strategies, the impact of the variation of the SoI_{main} is stronger in the case of the conventional injection strategy than it is for the EPI during compression-stroke. The reason of this lower sensitivity in case of EPI strategy is that the injection delay affects only the diffusive part of the combustion, that in that case accounts only for 70% of the total fuel injected.

As for the EPI at LP-TDC, the global trend is a large switch toward the higher NO_x emission due to a higher combustion temperature. It is interesting to detail that in that case, the decrease in NO_x obtained when delaying the SoI_{main} by $4°CA$ corresponds approximately to the increase found with the introduction of the strategy compared to a single injection case or a EPI during compression-stroke strategy. It is then visible that the combination of introducing this strategy and then delaying the injection timing results in a worsening of the NO_x-Soot compromise.

The fuel consumption trends are similar to what is observed for the other EPI strategy (here as well the SoI_{main} affects only the diffusive part of combustion, i.e. 70% of the fuel injected) but a global increase in fuel consumption is observed. As exposed in previous section 5.2.3.5, this is mainly a consequence of the additional pumping looses found during the re-compression of the gases at LP-TDC.

Although the general trends found with the variation of the SoI_{main}, and the shift between the different injection strategy were expected, the fact that the combination of the introduction of EPI at LP-TDC and then delay of the injection timing results in a worsening of the NO_x-Soot trade-off was not. Indeed it was shown in previous chapter 4 and also in a similar study [9] that the combination of the introduction of an EPI strategy together with the delay of the SoI_{main} can lead to an enhancement of the NO_x-Soot trade-off.

In the present case indeed, the NO_x-Soot compromise is slightly deteriorated compared to the best result found in chapter4 with optimized EPI strategy at LP-TDC, higher EGR rate and different hardware (but enhanced compared to the single injection case and former hardware) and that was displayed in figure 4.15. On the other hand, fuel consumption level is enhanced compared to the former engine configuration with EPI strategy. The new configuration tested here (lower compression ratio and adapted injector nozzle) makes however those results now attainable not only implementing the EPI strategy, but also with the conventional single injection strategy.

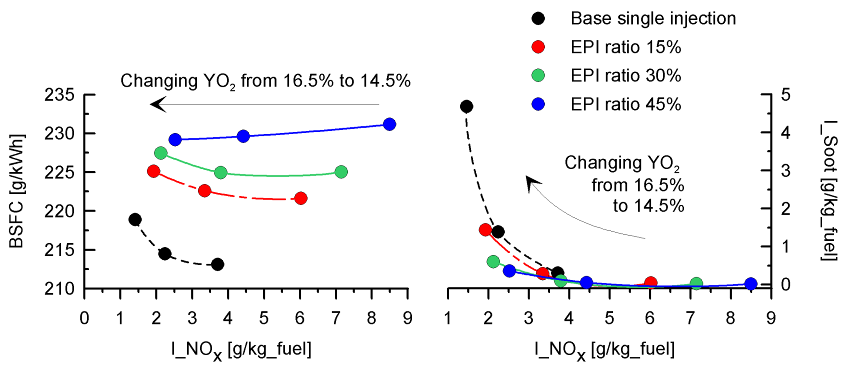
5.3.2 Influence of Oxygen Concentration (Y_{O2})

The Y_{O2} is known to have a large influence on the flame temperature and therefore affects strongly NO_x and soot emissions. It is varied from the base setting (15.5%) by 1% in both increasing and decreasing directions for all the EPI ratios tested along the study.

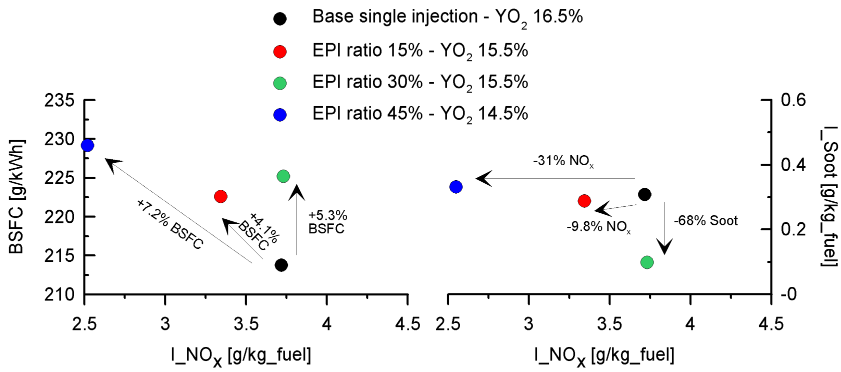
In following figure 5.14, the emission and fuel consumption results are presented for EPI at LP-TDC for ratios from 0% (i.e. conventional single injection strategy) till 45%, the maximum tested.

It globally appears that the combination of increasing the EPI ratio and later decreasing the oxygen concentration brings a better compromise in term of NO_x and soot emissions. The fuel consumption however is still negatively affected by the introduction of the EPI strategy.

Going more into the details, the comparison between the single injection case in high Y_{O2} conditions with the 15% EPI ratio case and middle Y_{O2} conditions shows a similar soot level while the EPI strategy shows 9.8% less NO_x. A further increase of the EPI ratio to 45% combined with a reduction of the Y_{O2} to the minimum tested leads to a further diminution of the NO_x emission (-31% compared to single injection case) while the soot level remains the same. The intermediate EPI ratio case (30%) offers a similar compromise as that found in EPI ratio 45% conditions but with a trade-off slightly shifted in the direction of higher soot a lower NO_x emissions.



(a) Complete Result Overview

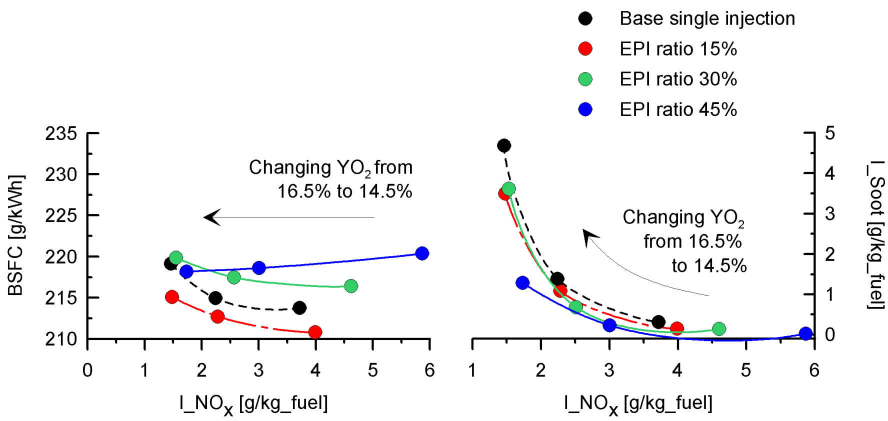


(b) Focus on Specific relevant Cases

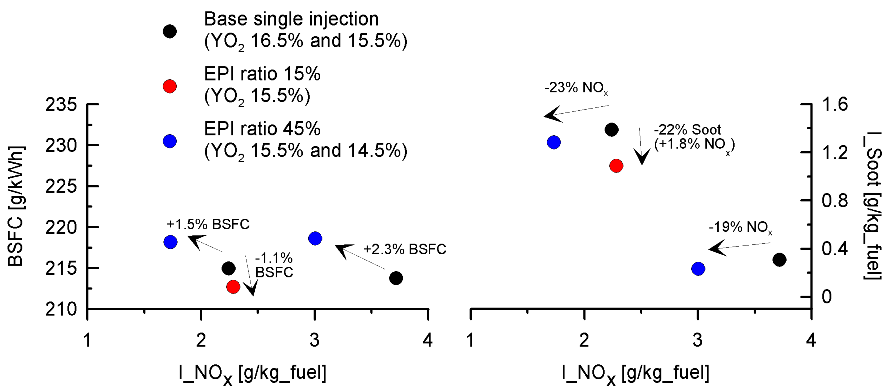
Figure 5.14. *NOx, Soot Emissions and Fuel Consumption Results varying Y_{O2} in Case of Single-Injection and EPI at LP-TDC Strategies.*

Regarding fuel economy, the introduction of the EPI at LP-TDC results in a clear drawback as mentioned in previous section 5.2.3.5. The impact of decreasing the Y_{O2} is negative in case of conventional single injection (because the lower oxygen concentration leads to a slower combustion process) but this trend is progressively inverted as the EPI ratio increases. The reason for this inversion is that a lower Y_{O2} condition leads to a less reactive mixture, therefore delaying the premixed combustion and bringing it closer to the TDC. Despite the inversion of the trend, even in the lowest Y_{O2} case, the introduction of EPI at LP-TDC still results in an increase of the fuel consumption.

The improvements of the NO_x-Soot trade-off are similarly found in the case of the EPI strategy during compression-stroke as shown in figure 5.15. It is especially noticeable when comparing the results obtained with the single injection cases in mid and high Y_{O2} conditions to the results with 45% EPI ratio and low to mid Y_{O2} conditions. Between those points one can observe that the soot level is slightly lower in the EPI strategy while the NO_x emissions are strongly reduced, by 23% and 19% respectively.



(a) Complete Result Overview



(b) Focus on Specific relevant Cases

Figure 5.15. *NOx, Soot Emissions and Fuel Consumption Results varying Y_{O2} in Case of Single-Injection and EPI during Compression-Stroke Strategies.*

Fuel consumption is higher in case of EPI during compression-stroke compared to the single injection case but, while the fuel consumption increase is steady from low to high EPI ratio when considering the high Y_{O2} cases, the trend is no longer found when considering the lowest Y_{O2} case. As it was the case in the EPI strategy at LP-TDC, here again the decrease of Y_{O2} appears to be beneficial for the highest EPI ratio case, thanks to a delay and a slow down of the premixed combustion process.

It is interesting to note that the combination of EPI strategy (either at LP-TDC or during compression-stroke) brings a similar NO_x-Soot compromise compared to the best results presented in previous chapter that were obtained with the former hardware configuration (see section 4.15), but with a largely reduced fuel consumption (around 2.8% to 3.5%) when comparing at similar NO_x and EPI ratio level.

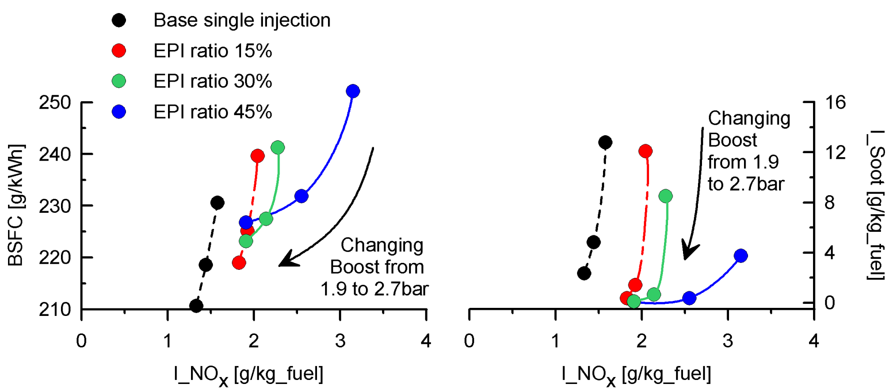
5.3.3 Influence of Boost Pressure

5.3.3.1 Pollutant Emission Trends varying Boost Pressure for the different Injection Strategies

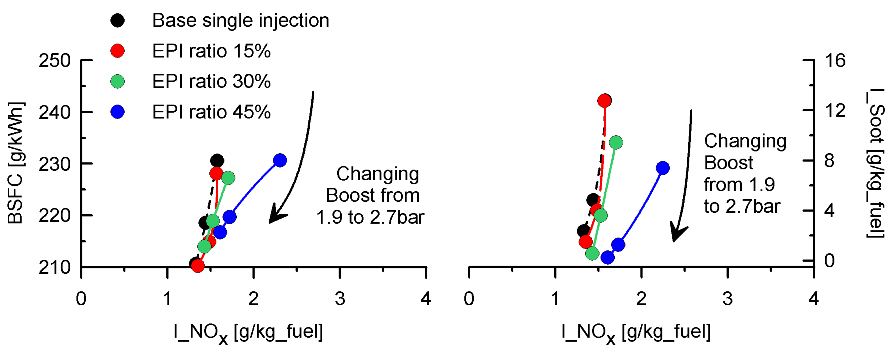
The boost pressure was varied by ±20% around the base value of 2.3 *bar* in the maximum and minimum oxygen concentration cases. Figure 5.16(a) shows the evolution of the NO_x-Soot and NO_x-fuel consumption compromises when varying boost in the lowest oxygen concentration conditions (trends are similar in the high Y_{O2} conditions) for the single injection and for the EPI strategies.

The increase of boost leads to a very important reduction of soot emissions while fuel consumption is moderately decreased for all EPI ratio conditions. The NO_x emissions are overall also slightly reduced for all cases, but it appear that the reduction is much more noticeable in the case of high EPI ratio than it is for low EPI ratio or for single injection case. The latter remark is valid for both EPI strategies however the case of EPI at LP-TDC shows a more important decrease of NO_x emissions as the boost pressure increases.

The fact that the trends observed when increasing boost pressure do not show a compromise between soot and NO_x emissions but a reduction of both emissions at the same time, makes the evaluation of the combination of a boost modification versus the introduction of EPI strategy, as it was made for the cases of main injection timing or oxygen concentration, irrelevant. It is however very interesting to understand how the increase of boost pressure permits to overcome the NO_x-Soot dilemma and to which extend the modification of the boost pressure is affected in combination of the EPI strategy introduction. Such evaluation is proposed in the following part of this subsection while the next subsection will detail a cross-analysis of the boost pressure, oxygen concentration and introduction of EPI strategy (with different EPI ratios).



(a) Case of EPI at LP-TDC



(b) Case of EPI during compression stroke

Figure 5.16. *NOx, Soot Emissions and Fuel Consumption Results varying Boost Pressure in case of different EPI Ratios and EPI Strategies.*

5.3.3.2 Analysis of the Capability of the Boost Pressure Increase to improve the NO_x-Soot Trade-Off

The main trend observed when increasing boost level is not related to NO_x emissions but to the reduction of soot emissions. To understand why the increase of boost leads to this behavior, the following figure 5.17 shows the trends of the oxygen concentration at the beginning and at the end of the combustion process, the amount of fuel left to be burned when the injection ends and the angle of the End of the Combustion (EoC), i.e. angle at which 95% of the energy contained in the injected fuel has been released by the combustion process. The trends are shown for the EPI during compression-stroke cases only for the sake of clarity, trends are similar in the other EPI strategy case.

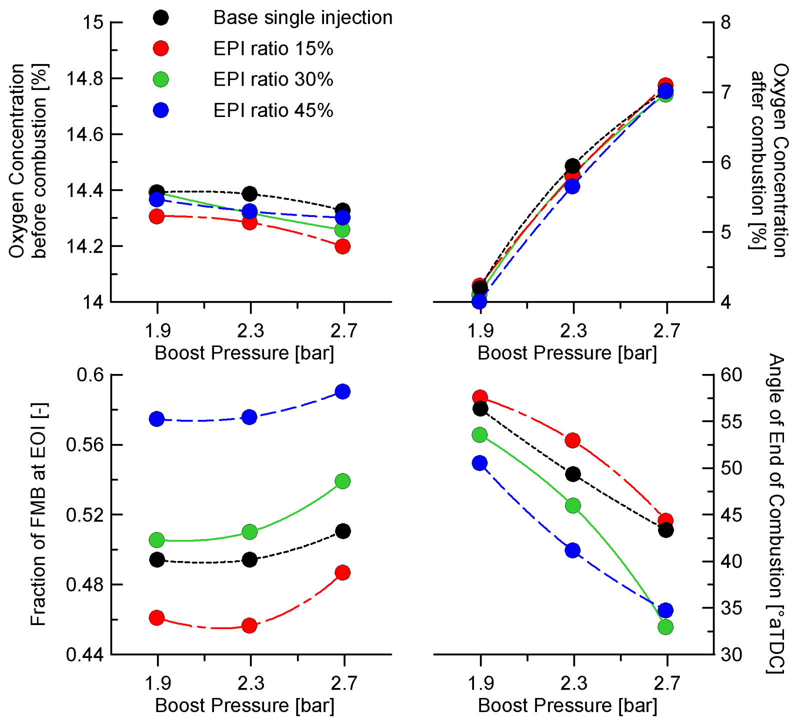


Figure 5.17. *Trends for several Key Combustion Parameters when varying Boost Pressure in Case of different EPI Ratio (EPI during Compression-Stroke).*

It is very clear that the impact of increasing boost pressure keeping a similar Y_{O2} at IVC has a strong impact on the Y_{O2} in the combustion chamber at the EoC. During the combustion, Y_{O2} diminishes as fuel is burned, and since the fuel mass injected has been maintained constant for all cases the mass of oxygen that is consumed during combustion is the same in all cases (considering that combustion efficiency is the same). When increasing the boost pressure, the in-cylinder trapped mass increases and a similar reduction in oxygen mass results in a smaller reduction in oxygen concentration.

The other trend observable in the figure 5.17 is related to the mixing process. One can see that, as the boost pressure increases, the amount of fuel burned at the EoI tends to increase and, in a clearer way, the combustion ends earlier. These two trends highlight the fact that the mixing process is enhanced as the boost pressure increases.

This is indeed consistent with the study performed by Fenollosa [10] already mentioned in the literature review and that demonstrates that the Apparent Combustion Time (ACT) required for a fuel packet injected in a steady state to reach the stoichiometric surface can be described as in the following expressions 5.1.

$$ACT \propto \frac{\phi}{\sqrt{\rho_a} \cdot u_o \cdot \sqrt{T_{Flame}/T_{ub}}} \tag{5.1}$$

The mixing process between fuel and oxygen during diffusive combustion depends on different parameters that can be separated in two classes:

- The injection parameters: injection velocity and nozzle hole diameter.
- The in-cylinder gas parameters: density, oxygen concentration, flame and unburned gas temperatures.

In the case considered here, for each injection strategy the injection parameters are maintained constant so the equation can be re-written as follow:

$$ACT \propto K_2 \cdot [{Y_{O2}}^2 \cdot \sqrt{\rho_a} \cdot \sqrt{T_{Flame}/T_{ub}}]^{-1} \tag{5.2}$$

Where

$$K_2 = \frac{\phi}{u_o} \tag{5.3}$$

The mixing process can hence be appreciated qualitatively through the mixing capacity of the in-cylinder gas (inverse of the characteristic mixing time) from the final equation given below, in arbitrary units:

$$Gas\,Mixing\,Capacity \propto \frac{1}{ACT} \tag{5.4}$$

This can be re-written as:

$$Gas\,Mixing\,Capacity \propto {Y_{O2}}^2 \cdot \sqrt{\rho_a} \cdot \sqrt{T_{ub}/T_{Flame}} \tag{5.5}$$

This parameter has been calculated from the experimental data for the different cases presented previously. Following figure 5.18 shows its evolution along the combustion process for the three boost pressure levels evaluated in three cases: that of conventional single-injection and those of maximal EPI ratio for EPI during compression-stroke and EPI at LP-TDC. Together with the gas mixing capacity, the evolution of the flame temperature from the beginning of the diffusive combustion process till the EoC is displayed.

The trends of the gas mixing capacity confirm that the increase of boost pressure has a very noticeable positive impact on the mixing process for both conventional injection and EPI strategies. This is of course due to the fact that the in-cylinder density increases as the boost pressure is augmented.

The figure 5.18 also depicts that the mixing process is better in the case of the single injection strategy than it is for the EPI strategies at the beginning of the diffusive combustion, while the levels are similar toward the end of the combustion. That is mainly due to the differences found in term of Y_{O2} in the combustion chamber when the main injection starts. In the case of the single injection, the Y_{O2} at the beginning of the diffusive combustion process is that found at the IVC while in the case of the EPI strategies, part of the fuel injected during the EPI has already burned and consumed some of the oxygen that was present in the combustion chamber at IVC. Toward the EoC however, the Y_{O2} tends to the same value in all cases (as the total amount of fuel burned in both conventional injection and EPI strategy is the same), therefore the mixing capacity of the gas is comparable. In order to get a better comparison on the influence of the injection strategy on the mixing process, in following figure 5.19 the gas mixing capacity is displayed as function of the fraction of fuel mass burnt.

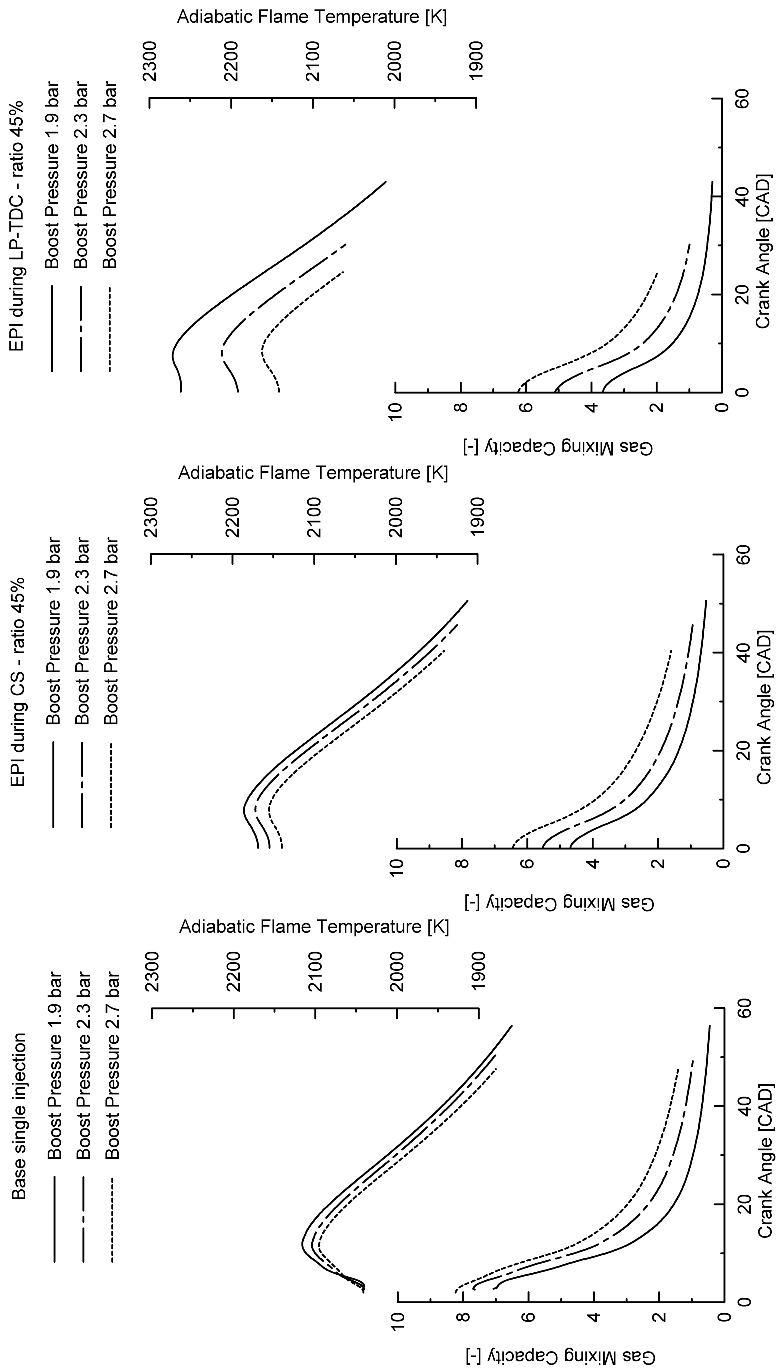


Figure 5.18. *Gas Mixing Capacity and Adiabatic Flame Temperature Evolution during Combustion Process when varying Boost Pressure for Three Injection Strategies.*

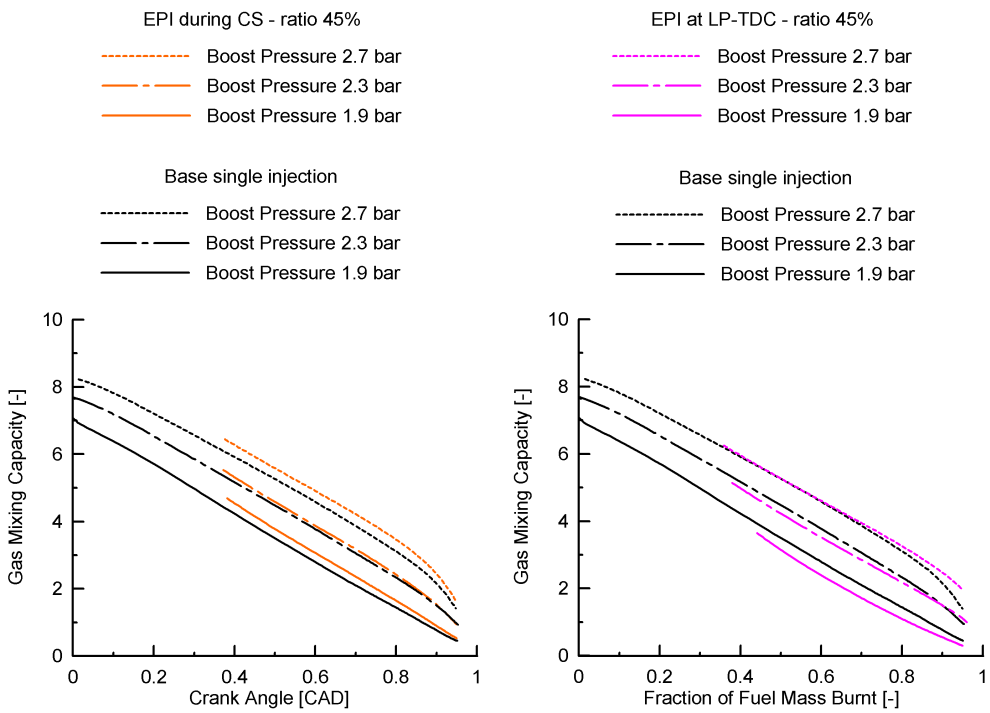


Figure 5.19. *Gas Mixing Capacity for different Boost Pressure Levels and Injection Strategies as Function of the Fraction of Fuel Mass Burnt.*

It appears hence more clearly that the different strategies offer a similar gas mixing capacity as the combustion develops. Going more into the details however, several remarks can be made:

The mixing capacity for the case of EPI during compression-stroke is slightly higher than that of conventional combustion for all the boost-pressure levels considered. This is due to the fact that the diffusive combustion starts at the same angle, i.e. at approximately the same in-cylinder density conditions. But at that point, in the case of EPI, already almost 40% of fuel has been burned. Therefore at iso-fraction of fuel mass burnt, the density is higher in the case of the EPI strategy.

In low boost pressure conditions, the mixing capacity of EPI at LP-TDC strategy is lower than the conventional injection case. Here again, the density at iso-fraction of fuel mass burnt is higher in the case of the EPI strategy than it is for the conventional injection case, but the total amount of gas trapped in the combustion chamber at IVC being lower in case of EPI at LP-TDC because of the larger NVO duration, the concentration of oxygen as the combustion takes place is lower in that case despite an equivalent value before combustion process. The balance of these two opposite trends appear to go in the direction of weaker mixing process for EPI at LP-TDC compared to conventional injection. However, the increase of boost pressure has more positive influence on the mixing capacity in case of EPI strategy at LP-TDC than it has in the case of conventional single injection because the drawback in term of Y_{O2} during combustion process gets smaller (as the NVO duration is reduced in case of higher boost pressure). Finally, for the highest boost pressure condition, the mixing capacity between both strategies is equivalent.

It is finally interesting to remark that, even if the gas mixing capacity is higher for the EPI strategy during compression-stroke than it is in the case of EPI at LP-TDC, the combustion duration is shorter in the latter case. This phenomenon depicts that the diffusive flame is somehow more disturbed by the premixed combustion in the case of the EPI during compression-stroke than it is for the EPI at LP-TDC. This statement appears plausible considering the fact that the diffusive flame is injected in the zone where the premixed combustion just took place in the case of the EPI during compression-stroke, while the premixed combustion is homogeneously distributed in the case of EPI at LP-TDC. Those local conditions are not considered in the calculation of the gas mixing capacity, showing the limit of this concept in case of complex overlapping combustion phenomena.

Regarding the flame temperature, the increase of the boost pressure appears to decrease it in the three cases, but to a different extent. In the case of the conventional single injection, the diminution of the flame temperature as the boost pressure is increased is not found at the beginning of the combustion process. As the combustion process takes place however, the traces of the adiabatic flame temperature start to diverge from one boost level to another. Indeed, as the temperature of the compressed gas at the SoC depends only on the compression ratio, the gas temperature and the gas composition at the IVC, that are roughly the same for all the boost pressure considered, it is logical to find similar temperature are the beginning of the combustion process. However, as the combustion process takes place, the pressure rise in the combustion chamber is relatively less important as the boost pressure is increased (a similar amount of heat is released in a higher amount of gas trapped in the cylinder). The relatively lower pressure rise means that the temperature of the unburned gas rises less in the case of the higher boost pressure, this being an input of the computation of the adiabatic flame temperature (as described in the previous chapter 3 section 3.4.1.2), the same trend is observed on that parameter.

In the case of EPI during compression-stroke, the differences in term of adiabatic flame temperature are more noticeable than for conventional injection strategy. In that case, the differences are found from the beginning of the diffusive combustion process. Indeed, as in the case of the conventional injection case, the in-cylinder gas temperature is similar in all cases during the compression process, however when the premixed combustion starts, the relative increase of in-cylinder gas pressure is much more elevated in the case of low boost pressure as the amount of gas trapped in the cylinder is lower while the heat released by the premixed combustion process is comparable. Therefore the temperature of the unburned gas is higher from the beginning of the diffusive combustion process as it is reflected in the trends of the adiabatic flame temperature. This trend is more marked as the EPI ratio is increased, explaining why the reduction NO_x as the boost pressure increases is much more noticeable in the case of high EPI ratio than it is for low EPI ratio or for single injection case as it was mentioned in the analysis of the figure 5.16.

Finally, the case of EPI at LP-TDC shows even bigger differences than that of EPI during compression-stroke in terms of flame temperature between the different boost pressure evaluated. In that case indeed, the in-cylinder gas temperature before the premixed combustion process are already affected by the boost pressure level as the duration of the NVO used in order to ensure that the injection at the LP-TDC is performed with limited wall impingement is being reduced as the boost pressure increases (and vice-versa) as described previously in section 5.2.2.1. For instance, the NVO duration required in the low-boost case is $100°CA$ while this number goes down to $80°CA$ in case of high-boost case. As a consequence, the internal EGR rate drops from 16.7% down to 14.1% between those two cases (oxygen concentration at IVC was kept equal by increasing the external (cooled) EGR rate consequently). The temperature of the gas inside the combustion chamber during the compression stroke is consequently decreased as the boost pressure increased. This trend adds to that found during the premixed combustion, previously described for the case of EPI during compression-stroke, and finally leads to a very large difference in terms of flame temperature explaining the higher NO_x sensitivity in cases of EPI during NVO than for the other strategies observed in figure 5.16.

It is interesting to note how the mixing process affects the trends in terms adiabatic flame temperature toward the EoC. The improvement of the mixing process when increasing the boost pressure results in a reduction of the combustion duration, hence in a earlier EoC as it was highlighted in figure 5.17. As a consequence, the adiabatic flame temperature - that tends to decrease toward the end of the combustion process due to the expansion process - decreases less in case of higher boost pressure simply because the combustion ends earlier. The figure 5.18 shows hence that, as the boost pressure is increased, in any of the injection strategies considered, the combustion ends with a higher adiabatic flame temperature despite displaying a lower maximum during the combustion process. This trend, together with the fact that the Y_{O2} at the EoC increases as the boost pressure increases, explains the results found in terms of soot emissions and, more interestingly, how the increase of boost pressure permits to improve both soot and NO_x emissions simultaneously.

Similar results were found in a detailed study [6] (focused in conventional injection case only), in which the impact of the mixing process on the late soot oxidation is thoroughly detailed. The study shows how the improvement of the mixing process permits, via a reduction of the combustion duration, the improvement of the NO_x-Soot trade-off.

5.3.4 Cross-Effects of oxygen Y_{O2} and EPI Strategy at different Boost Pressure Levels

5.3.4.1 Case of the EPI at LP-TDC

Now explained that reduced combustion duration helps the improvement of the NO_x-Soot trade-off, the results and cross-effects between injection strategy, oxygen concentration and boost pressure level can be newly considered.

First cases considered are those of conventional single injection and EPI at LP-TDC strategies. For those two cases, first the NO_x-Soot and NO_x-BSFC trade-off are shown. Then, in order to understand the trends observed in emissions, the flame temperature during the diffusive part of the combustion is also shown.

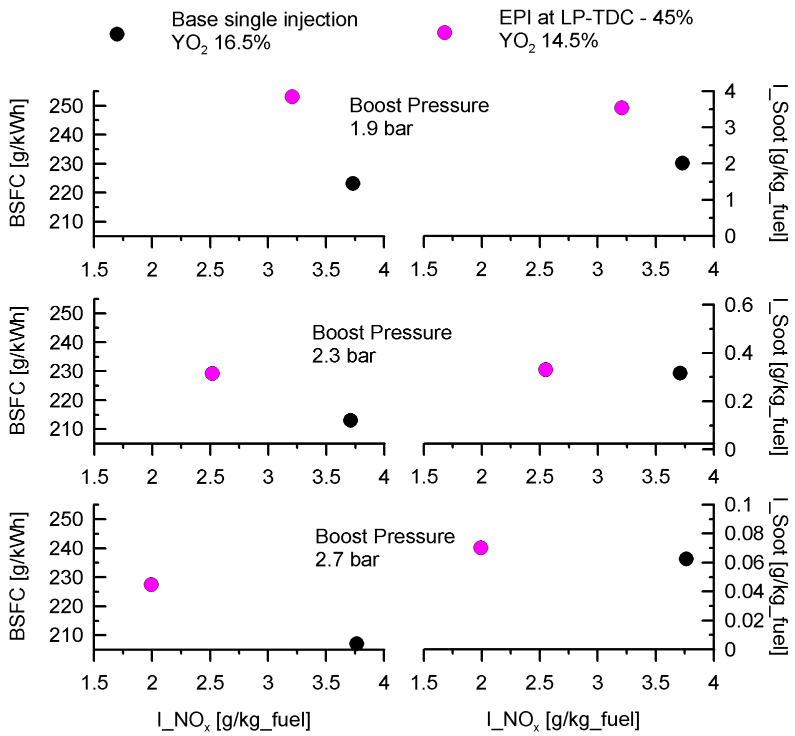


Figure 5.20. *Evolution of the NO_x-Soot and NO_x-BSFC Trade-Offs as the Boost Pressure is increased for two different Injection Strategies.*

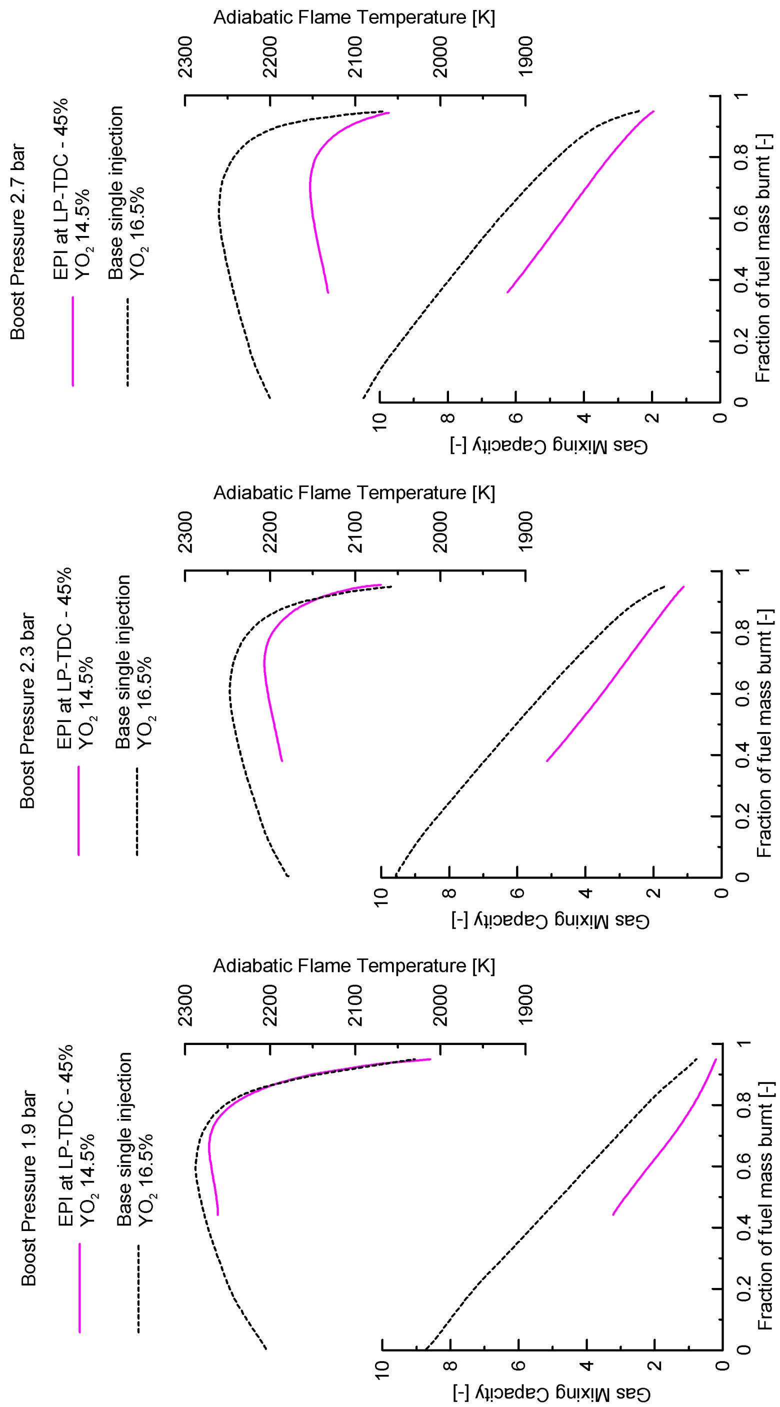


Figure 5.21. *Evolution of the Adiabatic Flame Temperature and Gas Mixing Capacity as the Boost Pressure is increased, for two different Injection Strategies.*

The combination of injection strategy and Y_{O2} were selected so that the NO_x and Soot emissions are on a comparable level, allowing then to give a clear view of the strategy that enhances the compromise. Regarding the fuel economy, the issue inherent to the EPI strategy described in section 5.2.3.5 are newly highlighted here. Although the enhancements brought by the increase of the boost pressure on fuel consumption is more noticeable in the case of the EPI strategy at LP-TDC than it is the case for base single injection, the level found in the EPI case remains way higher compared to the conventional strategy. It appears however that the NO_x-Soot emission trade-off is quite balanced between base single injection and EPI at LP-TDC strategies for the low boost pressure case. However, as the boost pressure increases, the balance is switched and the EPI at LP-TDC strategy appears to provide a much better compromise than the single injection strategy. Observing in figure 5.21 the corresponding traces of the adiabatic flame temperature during the diffusive combustion process for the cases considered clarifies how the introduction of the EPI at LP-TDC permits to enhance the NO_x-Soot trade-off.

In the case of low boost pressure, the introduction of the EPI at LP-TDC reduces slightly the combustion duration compared to the conventional injection case. As a result, the flame temperature difference between the maximum found during the combustion (that has a direct link to the NO_x emissions) and the level found at the end of the combustion process (that impacts the soot emissions) is approximately the same, leading hence to a similar NO_x-Soot trade-off.

As the boost pressure increases however, the larger enhancement of the mixing process for the EPI strategy at LP-TDC than for the conventional single injection combustion described in previous section 5.3.3.2 permits a more important reduction of the diffusive combustion duration in that case than it does for the conventional injection strategy. As a result, this permits that the large reduction in term of maximum flame temperature during the combustion can be obtained keeping an equivalent flame temperature at the end of the combustion process. The NO_x-Soot trade-off clearly reflects this trend with significantly lower NO_x emissions found in case of the EPI strategy at LP-TDC while the soot emissions are at the same level than those found for the conventional injection strategy.

5.3.4.2 Case of the EPI during Compression-Stroke

In the case of the EPI during compression-stroke, there is unfortunately no equivalent results with the conventional single injection case in terms of soot or NO_x for the high and the low boost pressure cases. For the middle boost pressure level, thanks to the fact that three Y_{O2} levels were tested instead of two in the other boost cases, several combinations of strategy and oxygen concentration levels offers very comparable results and are hence very interesting to detail in order to understand the breakthrough brought by the EPI strategy. Those cases were already presented in previous section 5.3.2, but they can now be more detailed considering the mixing aspect.

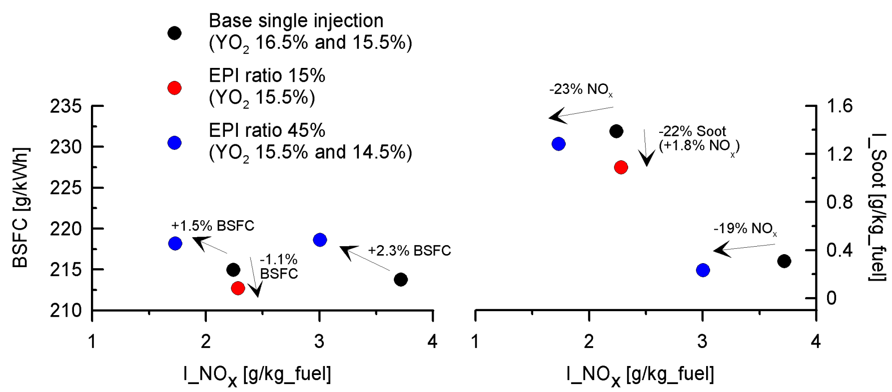


Figure 5.22. *NOx, Soot Emissions and Fuel Consumption Results varying Y_{O2} in Case of Single-Injection and EPI during Compression-Stroke Strategies.*

Figure 5.22 shows clearly that the introduction of the EPI during compression-stroke brings improvement to the NO_x-Soot trade-off compared to a conventional single injection strategy while the figure 5.23 presents the flame temperature and gas mixing capacity traces for the cases considered.

It is again shown that the introduction of the EPI strategy permits to reduce the difference between maximum flame temperature and the temperature at the end of the combustion process and that this difference diminishes as the EPI ratio increases, enhancing hence the compromise between NO_x and soot emissions.

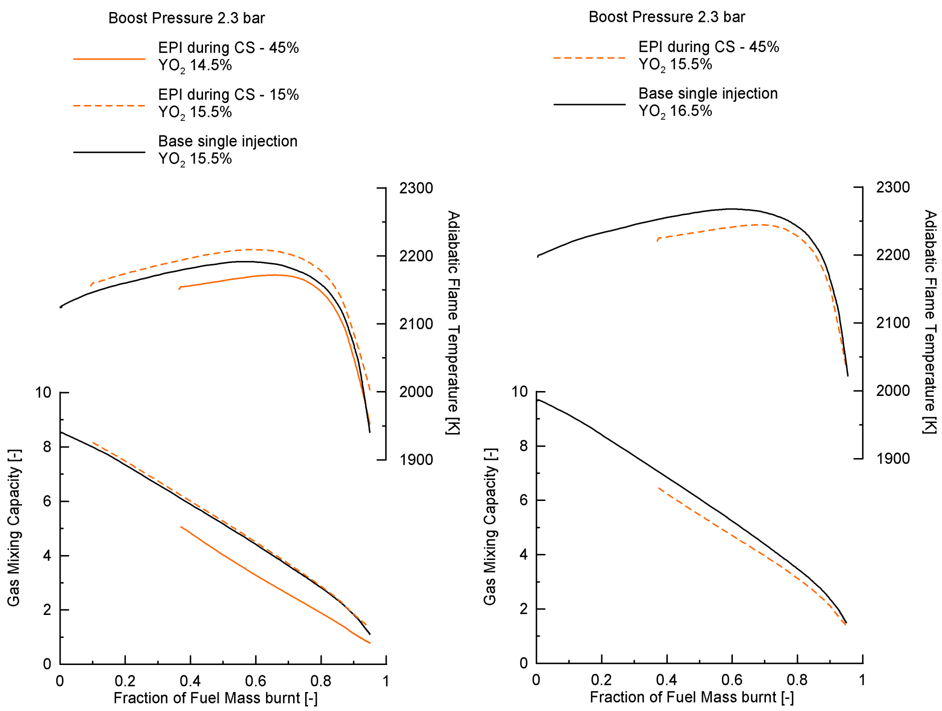


Figure 5.23. *Evolution of the Adiabatic Flame Temperature and Gas Mixing Capacity for different Y_{O2} Concentrations for Single-Injection and EPI during Compression-Stroke Strategies.*

5.4 Influence of the engine operation mode with EPI strategy

This section aims at giving an overview of the results obtained when introducing the EPI strategy in a different engine mode (i.e different engine speed or engine load) than that of the extended parametric study. Two modes were evaluated: first the full-load operation at low speed, that has a big relevance in terms of certification cycle, and second the mid-load operation at high engine speed.

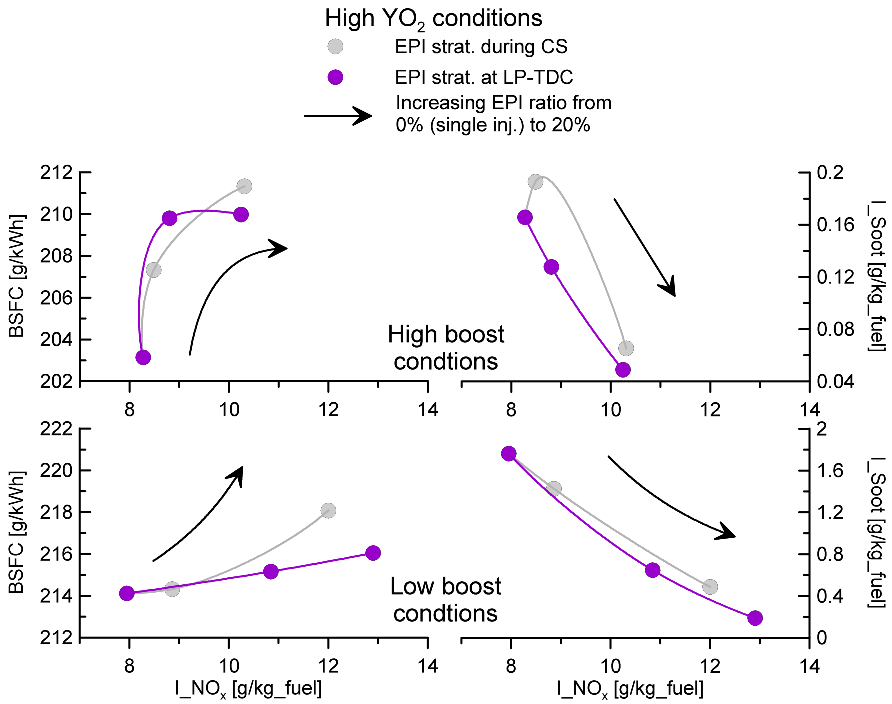
5.4.1 EPI Strategy in Full-Load

The introduction of the EPI strategy in full-load was evaluated for both EPI during compression-stroke and EPI at LP-TDC strategies. Besides the conventional injection case (0% EPI), two EPI ratios were evaluated, 10% and 20%, at two different boost pressure levels ($3.8bar$ and $4.6bar$), and two different Y_{O2} levels as it was described in sub-section 5.1.2.4.

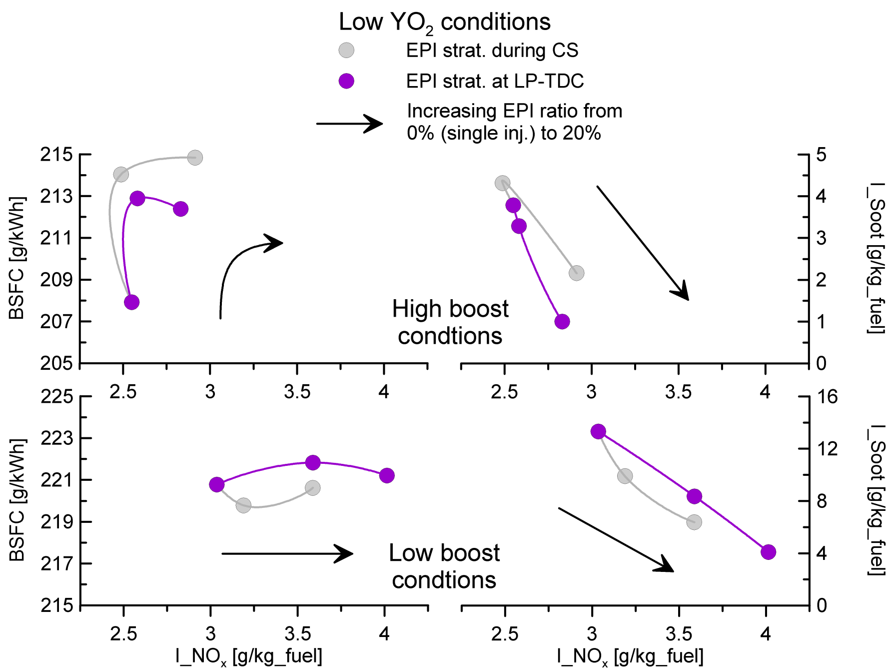
It is worse noticing that the introduction of the EPI strategy at LP-TDC for high load case is facilitated in terms of NVO enlargement compared to the mid-load cases. Indeed the higher exhaust gas density found in this mode permits to limit the extend of the NVO enlargement required to ensure the correct conditions at LP-TDC to avoid liquid fuel impingement onto the piston during the EPI event. As a result, the NVO was enlarged to $50°CA$ and $60°CA$ for the for the high and low boost pressure cases respectively, i.e. $30°CA$ to $40°CA$ less than in the nominal case at mid-load.

The introduction of the EPI strategy during compression-stroke is also helped by the use of higher engine load as the in-cylinder density during the compression stroke is increased and therefore limiting the spray liquid length during the EPI event and avoiding its impingement onto the piston bowl.

The overall results provided by the introduction of the EPI strategies in terms of NO_x-BSFC and NO_x-Soot trade-off, for all tests performed in high load conditions are presented in the figure 5.24



(a) Case of high YO_2



(b) Case of low YO_2

Figure 5.24. *NO_x, Soot Emissions and Fuel Consumption Results in High-Load Conditions varying EPI Ratio, boost pressure and Y_{O2} levels.*

Several interesting trends can be noticed on the figure 5.24, both regarding fuel consumption and NO_x-Soot trade-off.

Regarding the fuel consumption trends first, it appears that the impact of the introduction of the EPI strategies has less negative impact in high-load than in the mid-load conditions. Comparing both EPI strategies between themselves it is also visible that strategy of injection at LP-TDC provides overall slightly better results than that of injection during compression-stroke (except in the case of low Y_{O2} and low boost pressure conditions) whereas the opposite was found at mid-load.

Going more in the details, it can be observed that the drawback of the introduction of the EPI strategies in terms of fuel consumption is more limited in low-boost conditions than it is for high-boost case (this trend is found for both Y_{O2} levels). The drawback is also more limited when considering the low Y_{O2} conditions compared to the high Y_{O2} case (again valid for both boost levels). Finally, and as it is shown in the figure 5.25, the fuel consumption penalty[1] found with the introduction of EPI strategies is getting less and less as the overall richness increases. To understand why this trends in found, figure 5.25 also displays the trend of the penalty in terms of CO emissions[1] compared to a conventional injection case and that of the premixed combustion center of gravity angle as function of the richness.

Two aspects of the combustion lead to a limitation of the fuel consumption penalty with EPI strategy compared a conventional injection case when the richness increases. First, the improvements brought by the EPI strategy in terms of combustion efficiency that is reflected by the CO emissions level in this case (as the HC emissions in high load operation are extremely low in any cases). As it was largely discussed in the previous section 5.3, the introduction of EPI strategies permit to limit the amount of fuel that burns after the end of the main injection process and to end the combustion earlier. That leads to the fact that soot and CO emissions of the main injection-combustion process is normally reduced with respect to the conventional injection case. In mid-load, it is not seen in the emission measurement as a large amount of CO is formed during the premixed combustion and later not completely oxidized while the conventional injection case displays very low amount of CO emissions thanks to a low overall richness. In high-load however, the richness is higher and the conventional injection case starts to display important amount of CO emissions. In this case, the improvement of the introduction of the EPI

[1]The fuel consumption penalty is calculated as a relative difference between the BSFC obtained in the considered EPI case and that obtained in the case of single injection strategy in the same conditions of boost pressure and Y_{O2}. For the CO penalty, the same type of calculation is performed but the absolute difference is considered

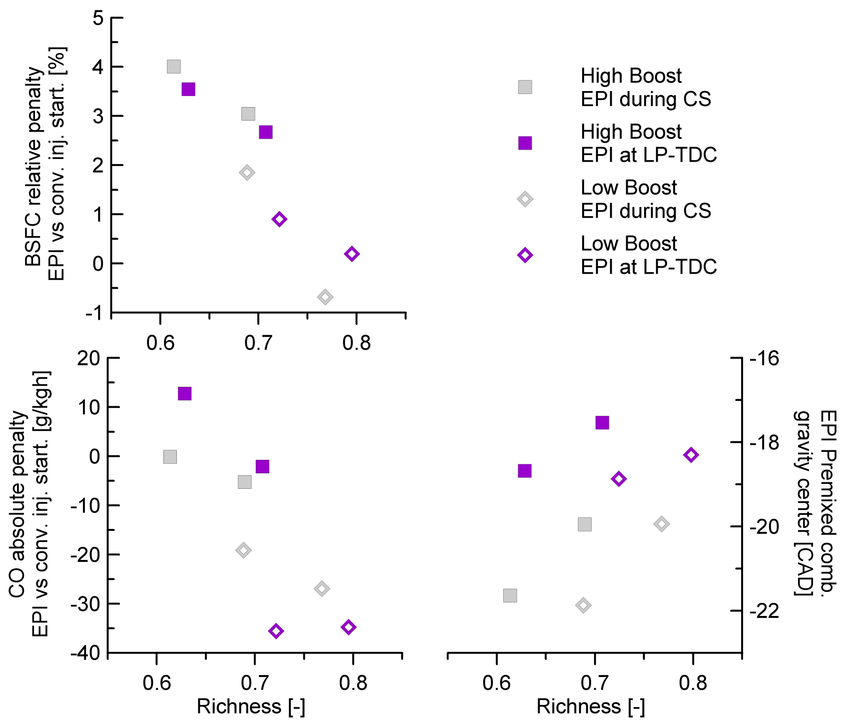


Figure 5.25. *Fuel consumption and CO emissions penalty of EPI strategies with respect to the conventional injection case, premixed combustion gravity center as function of the combustion overall richness. High load conditions.*

strategy normally found in soot emissions only, is also found in terms of CO emissions. It is interesting to notice that, in the middle richness conditions for the cases considered, the switch from high BP conditions to low BP conditions brings an additional improvement in the CO emissions, due to the fact that the lower amount of gas trapped in the cylinder in those conditions leads to higher the temperature during the premixed combustion phase.

Another aspect of the combustion that explains why the lower drawback in term of fuel consumption with respect to a conventional injection strategy is the center of gravity of the premixed combustion. Indeed, the premixed combustion appears to be more and more delayed and therefore closer to TDC as the Y_{O2} decreases due to the fact that the mixture reactivity is reduced as already mentioned in the sub-section 5.3.2 of the present chapter. It that case however, in opposition to what is observed for the CO penalty, the switch from high to low boost pressure conditions in the middle richness conditions has a negative impact, bringing the center of the premixed combustion more far away from the TDC. In case of the EPI during compression-stroke this is mainly due to the local equivalence ratio during the premixed combustion that are slightly higher in case of lower boost pressure (the lower density leading to a poorer mixing process during the EPI), while in the case of the EPI at LP-TDC this trend is more due to the NVO duration that is higher in case of low boost pressure, leading to higher temperature during the compression and therefore an earlier activation of the cool and blue flame of the premixed combustion.

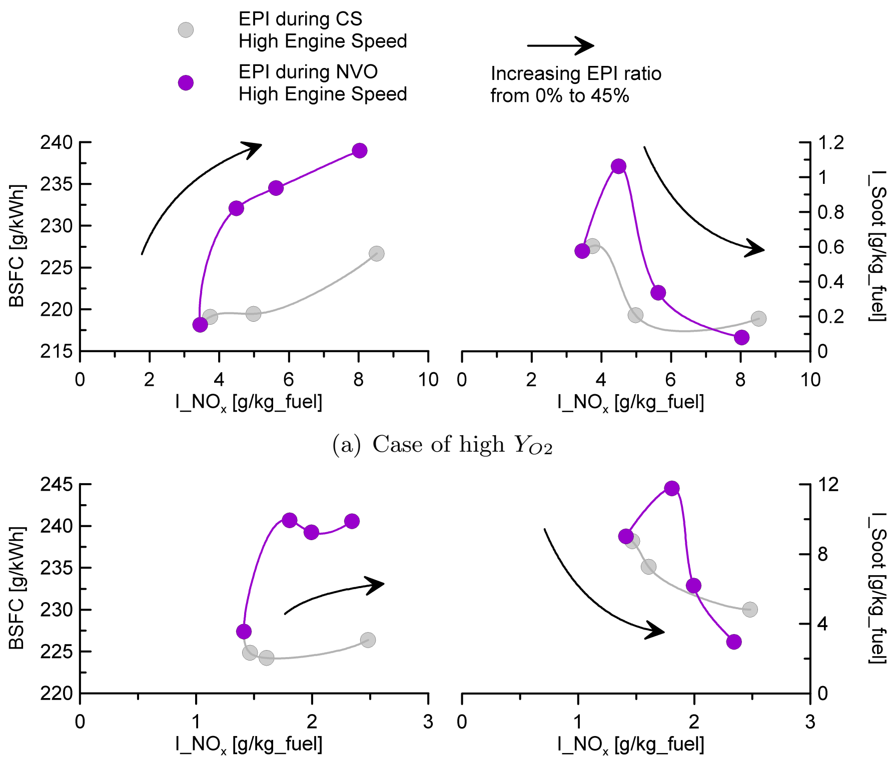
The combination of the two trends described above results in a quasi-steady improvement of the fuel consumption in comparison to the conventional injection strategy as the richness increases, leading, in the higher richness case, to a negative fuel consumption penalty (i.e. an improvement) when introducing an EPI strategy. It is fair to consider that a case (not measured in the present study) of even higher Richness would have displayed further fuel consumption breakthrough, describing a new and unexpected potential of the strategy.

Regarding the NO_x-Soot trade-off, the trends shown in figure 5.24 are similar to what was described in the mid-load cases. Unfortunately in this high-load cases, the combinations of input parameters do not offer cases where the soot or the NO_x emissions are the same between conventional and EPI strategies, it is therefore uneasy to clearly evaluated the breakthrough brought by the introduction of the EPI strategy.

It is nevertheless remarkable that the NO_x-Soot trade-off is generally better with the introduction of EPI at LP-TDC than it is for the EPI during compression-stroke (except in the low boost pressure and low Y_{O2} case) and, more interestingly, that the bigger NO_x increase that was found with the introduction of the EPI at LP-TDC compared to the EPI during compression-stroke in all mid-load cases is much more limited in the case of the high-load, especially in the case of the highest boost pressure where it is no longer existing. This trend is obviously found thanks to the shorter NVO duration that results in a global reduction of the temperature during the compression and combustion processes, hence lower NO_x increase compared to the mid-load case. The soot reduction brought by the introduction of the EPI strategy at LP-TDC appears on the other hand not affected by the reduction of the NVO duration. It finally results in a very "vertical" trend in the NO_x-Soot trade-off in case of EPI at LP-TDC in high boost pressure conditions with a division by 4 of the soot emissions obtained a limited increase of NO_x emissions (10% to 20%).

5.4.2 EPI Strategy in High Engine-Speed

The introduction of the EPI strategy in mid-load operation at higher engine speed was evaluated for both EPI during compression-stroke and EPI at LP-TDC cases. Like in the case of the mid-load low speed study, three EPI ratios were evaluated, 15%, 30% and 45% but this time no other variation except that of Y_{O2} (on two different levels) were evaluated as it was described in the test-plan shown in sub-section 5.1.2.4.



(a) Case of high Y_{O2}

(b) Case of low Y_{O2}

Figure 5.26. *NO_x, soot emissions and fuel consumption results in high speed conditions varying EPI ratio in case of both EPI strategies for two oxygen concentration levels.*

The overall emission results obtained show that the introduction of the EPI strategy at LP-TDC has a large negative impact on fuel consumption while the impact of the EPI strategy during compression-stroke has a limited influence on this parameter, and even provides improvements in the cases of low Y_{O2}. Regarding the NO_x and soot emissions, the EPI strategy during compression-stroke brings a better compromise compared to the EPI at LP-TDC strategy for the EPI ratios of 15% and 30% while the trend is inverted for the highest EPI ratio.

The trends are overall similar to what was found in the case of mid-load and low engine speed operation and it is therefore fair to assume that the conclusions risen from the previous section 5.2.2 are valid for the high engine speed conditions.

5.5 Impact of EPI strategy on NVH and oil-dilution

In this final section, it is intended to give insights of the consequences of the introduction of the EPI strategy regarding other aspects than the pollutant emissions or the engine efficiency.

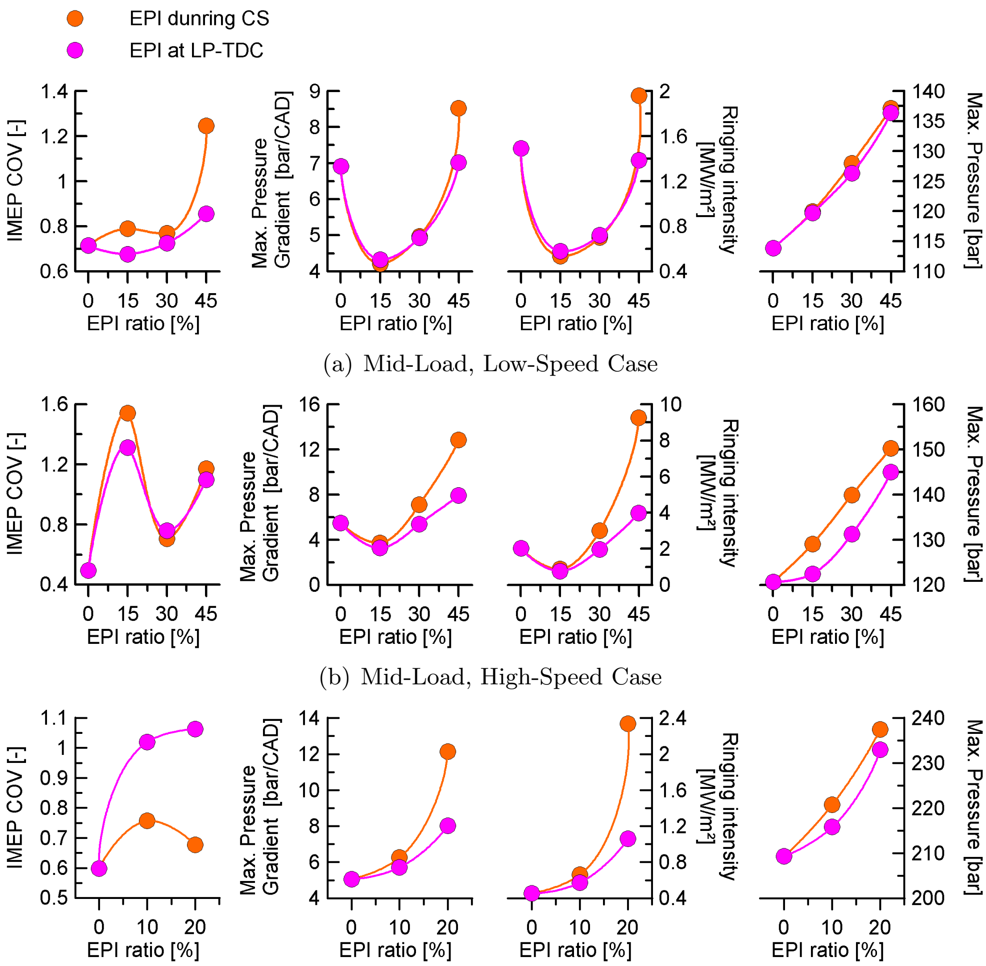
Instead of that, two aspects that were not discussed previously, as they are slightly out of the scope of the thesis, are addressed. Those two aspects are the impact of the EPI introduction on the Noise Vibration and Harshness (NVH) of the combustion on the one hand, and the impact of the strategy on oil-dilution on the other hand.

5.5.1 NVH evaluation

Four parameters were considered for the evaluation of the NVH results obtained: the coefficient of variation of the indicated torque, the maximum pressure gradients, the ringing intensity and the maximum in-cylinder pressure. Following figure shows the results obtained for these parameters for both partially premixed combustion strategies (in function of the EPI ratio) and conventional combustion for the three modes considered.

In all cases, even if it is negatively affected by the introduction of the EPI strategies, the coefficient of variation of the IMEP is kept at a low level (maximum 1.55). This shows how the EPI strategies are quite more stable than what is usually encountered in PCCI (around $2-4$) [11] or HCCI (around 15) [12] combustion strategies. This aspect therefore does not represent an issue for the introduction of the EPI strategy in an engine.

The second and the third parameters give an image of the noise emitted by the combustion process. The ringing intensity was calculated from the experimental data using the correlation proposed by Eng [13]. Both parameters are greatly affected by the EPI ratio for all considered modes. In mid-load conditions (low and high-speed), the results show that the introduction of the EPI strategy with low EPI ratio can have a positive influence on the maximum pressure gradient encountered during the combustion process. In those conditions the EPI indeed acts as a conventional pilot injection permitting a reduction of the pressure gradients at the beginning of the main combustion process thanks to the reduction of the ignition delay of the main combustion compared to a single injection strategy [14]. In the case of high-load, where the ignition delay is very limited even in single injection conditions, as well as in case of high EPI ratio for mid-load operation (especially for EPI during compression-stroke), the relatively large amount of



(a) Mid-Load, Low-Speed Case

(b) Mid-Load, High-Speed Case

(c) High-Load, Low-Speed Case

Figure 5.27. *Evolution of three NVH Parameters when introducing an EPI Strategy in the three Modes tested.*

fuel that rapidly burns during the EPI combustion leads to higher pressure gradients than the one found during the main combustion in case of single injection. In those conditions, it is expected that the combustion noise will increase with the introduction of the EPI strategy.

Eng [13] stipulates in his study of the correlation of the ringing intensity with combustion noise that a value around two represents a limit for acceptable combustion generated noise. In most of the EPI at LP-TDC cases this limit is respected, even for the highest EPI ratio case and this aspect should hence not be a blocking point for the introduction of the strategy in a serial engine, especially in the case of heavy-duty engine where the combustion noise is of less importance compared to the case of light-duty vehicles. For the EPI strategy during compression-stroke however, it appears that this aspect could impede the introduction of the strategy with the largest EPI rate, especially in the cases of high-speed mid-load (and in a lower extent for low-speed high-load case), where the ringing intensity level reached is too high.

Last parameter, the maximum in-cylinder pressure is also strongly affected by the EPI ratio. As shown in the figure 5.27, the maximum in-cylinder pressure reached during the combustion process increases in a close to linear way as the EPI ratio increases. It can also be remarked that the increase of maximum pressure is slightly bigger in the case of EPI during compression-stroke compared to the case of EPI at LP-TDC. Overall, this increase does not represent an issue in mid-load operation, but in the high-load case it could lead to overcome the maximum limit in terms of in-cylinder pressure that the engine can withstand in a sustainable way. This aspect could therefore also be an impeding criteria in high-load conditions, especially for high EPI rate and EPI strategy during compression-stroke.

5.5.2 Oil-Dilution Results

In order to evaluated the impact of the introduction of the EPI strategy on the oil-dilution by the fuel that goes from the combustion chamber to the crank-case, an oil sample was taken regularly during the tests and analyzed via the Fourier transform infrared (FTIR) spectroscopy technique that uses the capability of organic molecule to absorb infrared radiation in order to determine the composition of the different chemical elements present in the oil[2]. These analysis were performed only during the test-campaign at mid-load and low-speed, as the duration of the test campaigns at high engine load or high engine speed were too short to provide reliable results.

Following table shows the overall results, with analysis performed in the different EPI ratio and injection strategies tested.

Results globally show that the strategy of EPI at LP-TDC results in a larger oil-dilution than that of the EPI during compression-stroke. For both

[2]This is done via the analysis of the spectrum of an infrared light passing through a oil sample; all the frequencies of infrared spectrum are summed over time resulting in a time-dependent interference pattern called an interferogram. An algorithm called a fast Fourier transform is then used to convert this signal to an absorbency spectrum

Table 5.2. *Oil dilution by fuel during normal and EPI operation in mid-load and low-speed conditions.*

Injection Strategy	EPI ratio (%)	Dilution rate (% fuel in oil/h)
Single injection	0	0.063
EPI during CS	15	0.149
	30	0.181
	45	0.211
EPI at LP/TDC	15	0.167
	30	0.403
	45	0.625

strategies, the oil-dilution increases with the EPI ratio. It is however noticeable that the increase as function of the EPI ratio is much larger in the case of the EPI at LP-TDC (where it is almost linear) than it is in the case of EPI during compression-stroke, where the results showed limited influence of the EPI ratio but nevertheless a large "step" from the first EPI ratio level comparing it to a conventional injection strategy. Overall results point that the EPI strategy during compression-stroke may be introduced in any case, its contribution to oil dilution compared to an DPF regeneration event[3] is not out of range and it must not represent a dramatic issue for the oil change interval. In the case of high EPI ratio at LP-TDC however, the impact on oil dilution is much more marked (factor 10 compared to a conventional injection strategy, impact on oil dilution three times higher than the DPF regeneration events) and it is likely that this strategy combined with large EPI rates cannot be active constantly but must be limited to specific operation conditions and/or the oil change interval must be adapted accordingly.

5.6 Conclusions

This chapter presented a detailed analysis of the implementation of two different EPI strategies and put them in comparison to a conventional single

[3]The oil dilution rate during a DPF regeneration event is in the range of $2 - 3\%/h$, a regeneration event of fifteen minutes is usually found every five hours of driving, the contribution of DPF regeneration on oil dilution is hence in the range of $0.2\%/h$ overall

injection strategy. A parametric study was run in mid-load, low-speed conditions varying injection timing, Y_{O2} concentration and boost pressure while additional tests were also performed in full-load, low-speed and mid-load, high-speed conditions. The overall conclusions of this analysis are listed hereafter.

- The analysis of the conditions for the EPI showed that the issues of wall impingement were avoided or at least largely limited in any case of the introduction of the EPI at LP-TDC thanks to the adapted NVO duration. In case of EPI during compression-stroke, the impingement onto piston is not avoided while the limited premixing time in those conditions lead to substantial amount of fuel in conditions to form NO_x during premixed combustion for the high EPI ratio cases.

- The detailed analysis of the combustion process permitted to highlight that the improvement of the soot emissions results obtained with the introduction of the EPI strategy is related to the fact that the amount of fuel burned after the end of the main injection process is reduced while the flame temperature during that late combustion phase is increased. Further cross-analysis performed on the results obtained varying boost pressure and Y_{O2} showed that the introduction of EPI strategy has the potential to reduce both NO_x and soot emissions at the same time thanks to a reduced diffusive combustion duration and finally a limited drop of the flame temperature between its maximum (that impacts NO_x emissions) and flame temperature at the end of the combustion process (that impacts soot emissions).

- Impacts on HC and CO emissions are overall limited in terms of absolute values, which shows that the EPI strategies tested do not show the issues found generally with the introduction of HCCI or PCCI combustion.

- Fuel consumption is generally largely increased in case of the introduction of the EPI strategy, especially in the case of EPI at LP-TDC. It appears however that in certain cases, the fuel consumption can be reduced. This occurs mainly in very low NO_x conditions (by reduction of the Y_{O2} or main injection timing delay) and in case of low EPI ratio for EPI during compression-stroke in all conditions.

- The variations of commonly adjusted calibration parameters showed that the EPI strategies (especially that of EPI at LP-TDC) brings larger improvements compared to a conventional injection strategy when the boost pressure is increased. The results of EPI strategy appear also to be improved as the Y_{O2} in the intake is reduced while, on the other hand, the main injection timing delay appears to worsen the results obtained with EPI strategy compared to a conventional single injection strategy.

- Evaluation of EPI strategies in higher engine-speed conditions depicted overall similar results compared to those found at mid-load and low-speed, where the parametric study was performed. In condition of higher engine-load, the EPI strategy (especially at LP-TDC) appears to bring larger improvement in terms of NO_x-Soot emissions than it does at mid-load. It additionally was shown that the fuel consumption can be positively affected by the introduction of both EPI strategies in high Richness conditions.

- The final part of this chapter showed that the potential introduction of the EPI strategy in serial engine could be compromised by the issues it leads to in terms of oil-dilution and NVH in the cases of high EPI ratios. The high EPI ratio cases indeed lead, in the case of EPI during compression-stroke, to unacceptable pressure gradients (and therefore combustion noise) and, in case of full-load operation, to too high in-cylinder pressure. In the case of EPI at LP-TDC, it is mainly the oil-dilution rates that are found out of range (in case of high EPI ratio) compared to conventional injection strategy, and that will lead to an issue in terms of oil change interval for a serial application. The moderate and low EPI ratios on the other hand do not show those inconveniences.

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Chapter 6

Conclusions

Contents

In this last chapter of the thesis, the objective is first to synthesize the main conclusions of the study and put them in perspective with the current state of Diesel engine research and development. In a second part, several further investigations are proposed considering the conclusions of the current work.

6.1 Main Conclusions of the Thesis

Before listing the different conclusions of the thesis, it is important at this stage of the document, to remind that the main objective of the thesis is to determine and analyze the potential of an hybrid combustion mechanism, associating premixed and diffusive combustion processes, based on splitting the injection and also the combustion processes into two almost independent events.

A literature review was performed on Diesel combustion, both for diffusive and premixed combustion strategies. The literature review highlighted that, if the conventional diffusive combustion process is efficient and can be applied in the whole load range of the engine, it suffers an intrinsic issue in terms of pollutant emissions, the NO_x-soot dilemma. This dilemma lies in the fact that the conditions required to limit the formation of NO_x(that is mainly the reduction of the flame temperature) leads to a weaker soot oxidation during the late stages of the combustion process, while the condition of soot formation are always found in this conventional process.

Overcoming the NO_x-soot dilemma led to the introduction of advanced combustion strategies such as low temperature and/or premixed combustion. Those advanced strategies are efficiently leading to low NO_x-soot combustion however, they suffer usually from issues regarding the mixture preparation and above all, all of them are limited to the low load area of the engine operation range.

The association of premixed and conventional strategies permits to extend the load range application of advanced combustion strategies. The resulting combustion process appears to produce less soot emissions for a given NO_x emission level, overcoming hence the NO_x-soot dilemma at higher loads than the range of application of the low temperature and/or fully premixed combustion. Among those strategies, the Early Pilot Injection (EPI) strategy shows important potential for improvement in terms of pollutant emissions compared to a conventional strategy. On the other hand, the EPI strategy suffers from an important issue of hardware wear and oil-dilution due to liquid fuel spray impingement onto cylinder liner during EPI event.

Considering the conclusions of the literature study, two EPI strategies were proposed. The two strategy tested permitted to avoid the fuel spray impingement onto cylinder liner during EPI event by either using a narrow angle nozzle to allow injection very early during the compression stroke but still inside the piston crown, or by injecting with wide angle nozzle at the LP-TDC and take advantage of a VVA system to enlarge the NVO in order to obtain the required density at the LP-TDC for avoiding liquid fuel impingement onto the piston.

The strategy with narrow angle nozzle was found to be unsuccessful regarding the reduction of pollutant emissions mainly because of the non-optimized jet path and important spray bowl interaction during the main injection (around TDC). Also, the liquid fuel deposition onto the piston during the EPI event, and the suggested interaction between main injection and premixed combustion lead to largely worsened results in term of pollutant emissions and fuel consumption compared to a conventional single injection strategy.

The strategy of injecting fuel at the LP-TDC using enlarged NVO on the other hand showed potential in terms of emissions reduction compared to a conventional injection/combustion strategy and was hence further investigated in an extended parametric study. Together with this strategy, the EPI during compression stroke with wide angle nozzle, injecting late enough to avoid injection onto the cylinder wall, was also included in the parametric study.

The main conclusions of the thesis came out of the parametric study. The detailed analysis of the combustion process permitted to highlight the fact that the improvement of the soot emissions results obtained with the introduction of the EPI strategy is related to the fact that the amount of fuel burned after the end of the main injection process is reduced while the flame temperature during that late combustion phase is increased. Further cross-analysis performed on the results obtained varying boost pressure and Y_{O2} showed that the introduction of EPI strategy has the potential to reduce both NO_x and soot emissions at the same time thanks to a reduced diffusive combustion duration and finally a limited drop of the flame temperature between its maximum (that impacts NO_x emissions) and the flame temperature at the end of the combustion process (that impacts soot emissions).

Impacts on HC and CO emissions are overall limited in terms of absolute values, which shows that the EPI strategies tested do not show the issues found generally with the introduction of HCCI or PCCI combustion.

Fuel consumption is generally largely increased in case of the introduction of the EPI strategy, especially in the case of EPI strategy at LP-TDC. It appears however that, in certain cases, the fuel consumption can be reduced with the EPI strategy compared to the conventional injection case. This occurs in mid-load conditions mainly in very low NO_x conditions (by reduction of the YO_2 or delay of the main injection timing) and in case of low EPI ratio for EPI during compression-stroke in all conditions. In high-load conditions this occurs when the overall richness is increased to a very high value.

The variations of commonly adjusted calibration parameters showed that the EPI strategies (especially that of EPI at LP-TDC) bring bigger improvements in terms of pollutant emissions compared to a conventional injection strategy when the boost pressure is increased. The emission results of EPI strategy appear also to be improved as the Y_{O2} is reduced while, on the other hand, the main injection timing delay appears to worsen the emission results obtained with EPI strategy compared to a conventional single injection strategy.

Evaluation of EPI strategies in high-speed conditions depicted overall similar results compared to those found at mid-load and low-speed where the parametric study was performed. In condition of high engine load, the EPI strategies (especially at LP-TDC) appear to bring larger improvement in terms of NO_x-Soot emissions than it does at mid-load. It additionally was shown that the fuel consumption can be positively affected by the introduction of both EPI strategies in high richness conditions.

Finally, it was shown that the potential introduction of the EPI strategy in serial engines could be compromised by the issues it leads to in terms of oil-dilution and NVH in the cases of high EPI ratios. The high EPI ratios indeed lead, in the case of EPI during compression-stroke, to unacceptable in-cylinder pressure gradients (and therefore combustion noise) and, in case of full-load operation, to too high in-cylinder pressure. In the case of EPI at LP-TDC, it is mainly the oil-dilution rates that are found out of range in case of high EPI ratio compared to conventional injection strategy, and that will lead to an issue in terms of oil change interval for a serial application, especially in the case of heavy-duty application where this topic is of major importance. The moderate and low EPI ratios on the other hand do not show those inconveniences.

6.2 Perspectives and Future Works

The main conclusion of the thesis highlighted the fact that the association of premixed and diffusive combustion strategies have the potential for emission reduction and may be considered to be implemented in a serial application in some cases.

Putting in perspective the results with the current status of engine development, it appears that the enhancement in terms of pollutant emissions and fuel consumption brought by the strategy are still rather limited and will most likely do not lead any manufacturer to introduce new technology such as variable valve timing permitting to adjust the NVO duration based on the engine load or the boost pressure. That hence leaves the biggest technological potential to the EPI strategy during compression-stroke.

Considering the results and the consequences on NVH and oil-dilution, the lower EPI ratio have the most potential for introduction in serial engine, but it appears nonetheless necessary to perform a more profound study of the impact of the introduction of this strategy on oil wear. However, even considering a unacceptable impact on the oil wear in case of constantly implemented EPI strategy, a partial integration of the strategy for specific conditions where it showed an interesting potential can also be considered. This could be for instance proposed to activate the strategy only in high richness conditions where the strategy has shown potential of reducing both fuel consumption and soot emissions. Typically those high richness conditions are found in fast load transient (tip-in) conditions, especially at low engine speed, where the introduction of EPI strategy may hence offer the possibility to increase in a certain extend the richness-limitation (smoke limitation) and finally offer better tip-in response of the engine, important aspect in the case of small diesel engine implemented in light duty vehicles. It appears overall hence relevant in future works to investigate the potential of the EPI strategy in a small, high speed Diesel engine and to extend the study not only to stationary but also to transient conditions.

Considering now the main limitation of the extension of the EPI strategy to large EPI ratio and to the fact that the fuel evaporation and mixing process during the EPI event is possibly the most challenging aspect of the introduction of partially premixed combustion, another investigation track that appears relevant is that of implementing the strategy with a different fuel type better fitting to premixed conditions (Diesel with low Cetane number, Gasoline, mixture of Diesel and gasoline...). Also the association of two type of injection systems, one for the EPI and another for the diffusive combustion appears to be a sensible path to follow in future investigations.

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