Universitat Politècnica de València
Departamento de Máquinas y Motores Térmicos

Evaluation of combustion concepts and scavenging configurations in a 2-Stroke compression-ignition engine for future automotive powerplants

Doctoral Thesis
Presented by:
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Directed by:
Dr. Ricardo Novella Rosa

Valencia, December 18^{th}, 2020

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Resumen. El trabajo de investigación presentado en esta tesis es el resultado de varios años dedicados al desarrollo, la implementación y la optimización de dos tecnologías combinadas: un concepto de combustión innovador y una arquitectura de motor de nuevo diseño. Esta investigacion se ha realizado en el marco de una colaboración con Renault SA, como continuación de las actividades realizadas en el proyecto europeo POWERFUL (POWERtrain for FUture Light-duty vehicles) por un lado, y en el marco del proyecto europeo REWARD (Real World Advanced technologies foR Diesel engines), devenido como continuación del proyecto POWERFUL en el marco del programa de investigación Horizonte 2020, por otro lado. Los principales objetivos de estos estudios eran evaluar el potencial del concepto de combustión parcialmente premezclada (PPC) operando con gasolina como combustible en un innovador motor de 2 tiempos de válvulas en culata, y luego diseñar una nueva geometría de motor de 2 tiempos utilizando la arquitectura Uniflujo para superar los principales problemas y limitaciones observados durante la primera etapa, que se pueden resumir principalmente en el rendimiento de barrido (especialmente trabajando en cargas elevadas).

La metodología diseñada para este trabajo de investigación sigue un enfoque teórico-experimental. La evaluación del concepto de combustión PPC operando con gasolina se llevó a cabo principalmente con un enfoque experimental con el apoyo del análisis en línea directamente en el banco de ensayo, seguido de un exhaustivo tratamiento posterior de los datos y de un análisis detallado del proceso de combustión utilizando herramientas de diagnóstico. Por el contrario, el desarrollo del nuevo motor Uniflujo de 2 tiempos consistió principalmente en iteraciones sobre modelado 3D-CFD, si bien las actividades experimentales fueron fundamentales para validar las diferentes soluciones propuestas y evaluar su sensibilidad ante diferentes parámetros de interés utilizando una metodología de Diseño de Experimentos (DoE).

La primera parte del trabajo se ha dedicado a la comprensión de los procesos termodinámicos involucrados en la combustión operando con el concepto PPC en un motor de 2 tiempos de válvulas en culata utilizando gasolina como combustible, y a evaluar su potencial en términos de emisiones contaminantes, consumo de combustible y ruido. Por último, se ha realizado un trabajo de exploración para ampliar en la medida de lo posible el rango de funcionamiento de este concepto de combustión en esta configuración específica del motor, investigando especialmente el rendimiento en cargas bajas en todo el rango de regímenes de giro del motor, y estableciendo también las principales limitaciones para la operación en cargas altas.

La segunda parte de la tesis se ha centrado en el desarrollo y optimización teórica de un motor Uniflujo de 2 tiempos de nuevo diseño, incluyendo su fabricación y validación experimental. El objetivo principal era optimizar, utilizando principalmente simulaciones 3D-CFD, el rendimiento de barrido de esta arquitectura de 2 tiempos mediante el diseño de nuevas geometrías de puertos de admisión, permitiendo un gran control sobre el flujo de aire hacia y a través del cilindro para barrer al máximo los gases quemados y minimizar el cortocircuito de aire fresco hacia el escape. Las soluciones óptimas se evaluaron experimentalmente siguiendo la metodología DoE, antes de comparar finalmente los resultados de rendimiento de barrido con la anterior arquitectura de motor de 2 tiempos con válvulas en culata.

Resum. El treball de recerca presentat en aquesta tesi és el resultat de diversos anys dedicats al desenvolupament, la implementació i l'optimització de dues tecnologies combinades: un concepte de combustió innovador i una arquitectura de motor de nou disseny. Aquesta recerca s'ha realitzat en el marc d'una col·laboració amb Renault SA, com a continuació de les activitats del projecte europeu *POWERFUL (*POWERtrain *for *FUture Light-*duty *vehicles) d'una banda, i en el marc del projecte europeu *REWARD (Real *World *Advanced *technologies *foR Dièsel *engines), esdevingut com a continuació del projecte *POWERFUL en el marc del programa d'investigació Horitzó 2020, d'altra banda. Els principals objectius d'aquests estudis eren avaluar el potencial del concepte de combustió parcialment premesclada (PPC) operant amb gasolina com a combustible en un innovador motor de 2 temps de vàlvules en culata, i després dissenyar una nova geometria de motor de 2 temps utilitzant l'arquitectura Uniflux per a superar els principals problemes i limitacions observats durant la primera etapa, que es poden resumir principalment en el rendiment d'escombratge (especialment treballant en càrregues elevades).

La metodologia dissenyada per a realitzar aquests treballs de recerca segueix un enfocament tant experimental com teòric. L'avaluació del concepte de combustió PPC operant amb gasolina es va dur a terme principalment amb un enfocament experimental, però sempre amb el suport de l'anàlisi en línia directament en el banc d'assaig, seguit d'un exhaustiu tractament posterior de les dades combinat amb una anàlisi detallada del procés de combustió utilitzant eines de diagnòstic. Per contra, el desenvolupament i el disseny del nou motor Uniflux de 2 temps va consistir principalment en iteracions sobre modelatge 3D-CFD, si bé les activitats experimentals van ser fonamentals per a validar les diferents solucions proposades i avaluar la seua sensibilitat davant una sèrie de paràmetres d'interés utilitzant una metodologia de Disseny d'Experiments (DoE).

La primera part del treball s'ha dedicat a la comprensió dels processos termodinàmics involucrats en la combustió operant amb el concepte de combustió PPC en un motor de 2 temps de vàlvules en culata utilitzant gasolina com a combustible, i a avaluar el seu potencial en termes d'emissions contaminants, consum de combustible i també de soroll. Finalment, s'ha fet un treball d'exploració per a ampliar en la mesura que siga possible el rang de funcionament d'aquest concepte de combustió utilitzant eixa configuració específica del motor, investigant especialment el rendiment en càrregues baixes en tot el rang de règims de gir del motor, i establint també les principals limitacions per a l'operació en càrregues altes.

La segona part de la tesi s'ha centrat en el desenvolupament i optimització teòrica d'un motor Uniflux de 2 temps de nou disseny, incloent la seua fabricació i validació experimental. L'objectiu principal era optimitzar, utilitzant principalment simulacions 3D-CFD, el rendiment d'escombratge d'aquesta arquitectura de 2 temps mitjançant el disseny de noves geometries de ports d'admissió, permetent un gran control sobre el flux d'aire cap a i a través del cilindre per a escombrar al màxim els gasos cremats i minimitzar el curtcircuit d'aire fresc cap a l'escapament. Les solucions òptimes es van fabricar i van avaluar experimentalment seguint la metodologia DoE, abans de comparar finalment els resultats de rendiment d'escombratge amb l'anterior arquitectura de motor de 2 temps amb vàlvules en culata.

Abstract. The research work presented in this thesis is the result of several years dedicated to the development, implementation and optimization of two combined technologies: an innovative combustion concept and a newly designed engine architecture. These investigations have been performed in the framework of a research collaboration with Renault SA following up the activities performed along the European POWERFUL project (POWERtrain for FUture Light-duty vehicles) on the one hand, and in the framework of the European REWARD project (REal World Advanced technologies foR Diesel engines), brought as a continuation of the POWERFUL project in the frame of the Horizon 2020 research program, on the other hand. The main objectives of these studies were to evaluate the potential of the Partially Premixed Combustion (PPC) concept operating with gasoline fuel in an innovative 2-Stroke poppet-valve engine, and then to design a new 2-Stroke engine geometry using the Uniflow architecture to overcome the main problems and limitations observed during the first stage, which can be mainly summarized to the scavenging performance (especially at high loads).

The methodology designed for performing these investigation is based on both experimental and theoretical approaches. The evaluation of the gasoline PPC concept was carried out mainly experimentally, but always supported by online analysis directly on the test-bench and followed by a thorough post-processing of the data combined with a detailed analysis of the combustion using combustion diagnostic tools. On the contrary, the development and design of the new 2-Stroke Uniflow engine consisted mainly of 3D-CFD iterations, but experimental testing was crucial to validate the different solutions proposed and evaluate their sensitivity to a set of parameters of interest using a Design of Experiments (DoE) methodology.

The first part of the work has been dedicated to the understanding of the thermodynamical processes involved in the combustion in a poppet-valve 2-Stroke engine operating with the gasoline PPC concept, and to evaluate its potential in terms of pollutant emissions, fuel consumption and also noise. Finally, a wide exploration has been performed to extend as much as possible the operating range of this combustion concept using that specific engine configuration, especially investigating the low loads performance throughout the full range of engine speeds, and also laying out the main limitations for high-to-full load operations.

The second part of the thesis has been focused on the development and theoretical optimization of a newly designed 2-Stroke Uniflow engine, leading to manufacture and experimental validation. The main objective was to optimize, using mainly 3D-CFD modeling simulations, the scavenging performance of this 2-Stroke architecture by designing new intake ports geometries and to enable a great control over the air flow into and through the cylinder in order to scavenge the burnt gases as much as possible while minimizing the fresh air short-circuit to the exhaust. The optimum solutions were then manufactured and experimentally tested following a DoE methodology, before finally comparing the results of the scavenging performance to the previous 2-Stroke poppet-valve engine architecture.

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Looking back a bit more than 6 years ago when I finalized my engineering degree and arrived at CMT - Motores Térmicos for a simple end-of-project internship, I would not then have imagined myself where I am today and all the experiences that came in-between those two moments, and all the knowledge and achievements those experiences have brought me, both professionally and personally. But nothing would have been possible without all the people surrounding me along those years, that believed in me and supported me –and also pushed me forward when needed– throughout every step. I will try to share it through these few lines, even if I know that words cannot express even a fraction of the gratitude I have towards these people.

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Finally, I would like to extend these acknowledgments and my gratitude to the people from outside the CMT and with who I had the opportunity to work during these past years. So I want to address a special thank you to the Renault SA research team, and especially to Pascal Triboté and Fano Rampanarivo who have been the managers and leaders of the projects that built my doctorate work. I want to address a special thank you to the IFPEN team, for their contribution to the success of the projects, but also and mostly for welcoming me into their team during an internship I realized inside the IFPEN facilities. So thank you to Florence Duffour, Gaetano de Paola, Jérémy Galpin, Emmanuel Rizzo, and a special thank you to Ludovic Nowak who became a friend more than a colleague from this internship forward. And of course I also need to thank the other project partners, Jan Macek and Pavel Brynych, from the Czech Technical University in Prague, who also played a big part in the development and success of the REWARD project.

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Chapter 1

Introduction

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1.1 Introduction

The present thesis describes the investigations performed during the development of two consecutive research projects focused on the 2-Stroke engine architecture, evaluating the potential of this technology to fulfill the upcoming requirements of the automotive industry. In this context, two main paths have been followed: the first focus of this work consists of the understanding of innovative combustion concepts such as the gasoline Partially Premixed Combustion (PPC), implemented in a poppet-valve High-Speed Direct Injection (HSDI) Compression Ignition (CI) single-cylinder 2-Stroke engine. The investigations performed during this phase lead to some limitations, which provided guidance for the second focus of the study presented here: the evaluation and optimization of the scavenging performance inside a newly designed Uniflow HSDI CI single-cylinder 2-Stroke engine.

As an introduction to all the investigations performed in this briefly described framework, this chapter approaches the current context of the automotive industry, the restrictions and regulations driving the future trends in engine research and development. An overview of the upcoming regulations and global expectations regarding the CO_2 and pollutant emissions reduction targets for the following years is presented at first, along with a 20-to-30 years vision about the different powertrain technologies available and with which the future of the internal combustion engine technology might need to be associated, or at least co-exist on different levels in order to provide a targeted global “well-to-wheel” energy balance. Then, an analysis of the short-term perspectives is discussed, mainly focused on the internal combustion engine as a part of hybrid solutions for automotive powertrains and implemented with new hardware technologies and new combustion concepts to help improving both efficiency and pollutant levels. Accordingly, the last part of this chapter presents the objectives of this thesis, along with an outline of the structure of the manuscript to guide the reader through its content.

1.2 Background and motivation

This description of the global context of the investigations performed during this thesis is particularly crucial, considering the worldwide environmental, economic, industrial and political challenges and their fast-evolving nature. For a long time, the trade-off between engine performance, fuel consumption and pollutant emissions has been a concern for most of the car and engine manufacturers. But a recent general change in the public mindset and awareness about environment challenges has encouraged political actions and industrial guidelines to push these challenges even further, and new technologies have been developed for every applications (mechanical such as transmission efficiencies or engine design, chemical such as new combustion systems or concepts, energetic such as energy recovery systems or hybridization / electrification, etc...). These development paths have been supported by political guidelines and commitments, as proposed during the latest COP conferences and supported by the most crucial countries around the world in this regard.

As a result, the automotive market has changed drastically in the last decade, and is bound to continue to change in the years to come. The following section then introduces the most important topics related to this context, and how the work performed during this thesis and presented in this document falls within these new perspectives.

1.2.1 Emission regulations for light-duty vehicles

In the last few years, global awareness accompanied by political measures have forced the public to reconsider the status of CI engines. Thanks mainly to its higher fuel efficiency, it has been for a long time the increasingly most popular choice amongst consumers in Europe (about 55% of the passenger cars market share in 2013), but also increasingly so in the rest of the world. However, the last few years have observed a general awareness regarding the pollutant emissions of CI engines has caused a change of mind in consumers, and this technology has become less and less popular, quickly falling to 35% of market share in 2018 according to the European Automobile Manufacturers Association (ACEA), being more and more replaced by petrol engines (share growing from 44% to 57% in just three years between 2015 and 2018). [1]

In spite of the significant diversification of the propulsion systems for road transportation applications, the ICE is expected to be still present in the future vehicle powerplants combined with different degrees of electrification. The general (and increasing) use of ICE's for road transportation in the current global warming context, started to raise both customers and governments awareness about air quality. Gaseous pollutants such as nitrogen oxides (NO_x=NO+NO_2), carbon monoxide (CO) and unburned hydrocarbons (HC), as well as particulate matter (PM) emissions, have well-known negative impacts on the life quality and environment. As a result, there is a growing social concern that is forcing the worldwide governments to promote increasingly restrictive vehicle emissions regulations.

In Europe, the Euro 5 and Euro 6 standards put in place from the end of the 2000's aimed to force the manufacturers to improve the design of their engines even more than before. And more recently, rather than focusing on reducing the absolute levels on pollutants, new methodologies have been introduced to test and control emissions. These new methods aim at harmonizing the regulations on a worldwide scale (until then, every region of the globe had different regulations, but also different testing cycles and procedures), and also at providing testing conditions much closer to the real-life utilization of the engine. Indeed, the New European Driving Cycle (NEDC) cycle used in Europe for the Euro 5 and early Euro 6 standards was a short cycle which was only limited to a small portion of the engine operating range. The Worldwide harmonized Light vehicles Test Cycle / Procedure (WLTC / WLTP) solves part of these issues, standardizing the methodology throughout the world and exploring a much wider range of engine operating conditions, getting closer to a real-life use. The absolute levels of pollutants are then unchanged between the various evolutions of the Euro 6 standard (Euro 6b, 6c, 6d), but those limits are getting harder to comply with as the testing methodology gets more stringent.

The next step towards cleaner and more efficient ICE's aims to propose the most realistic way of limiting the pollutant emissions, by measuring the exhaust outputs directly in real-driving conditions, directly on the street for a given amount of time or vehicle mileage. This Real Driving Emissions (RDE) procedure is much more demanding for the manufacturers, since it compels to optimize the entire engine operating range rather than focus only on part of it. It has been implemented since 2017 with the Euro 6d in addition to the WLTP, and controls the emissions of NO_x and PN with a conformity factor of 1.5 in reference to the WLTP levels. As a comparison, the table 1.1 provides the evolution of the emission limits for all the regulated pollutants. [2]

Table 1.1. *EU emission standards for passenger cars (category M_1).*

Stage	Cycle	CO [g/km]	THC [g/km]	NO_x [g/km]	$THC+NO_x$ [g/km]	PM [g/km]	PN [1/km]
Compression Ignition (Diesel)							
Euro 5	NEDC	0.5	–	0.18	0.23	0.0045	6.0×10^{11}
Euro 6c	WLTP	0.5	–	0.08	0.17	0.0045	6.0×10^{11}
Euro 6d	WLTP	0.5	–	0.08	0.17	0.0045	6.0×10^{11}
	RDE	–	–	0.12	–	–	9.0×10^{11}
Positive Ignition (Gasoline, NG, LPG, ethanol,...)							
Euro 5	NEDC	1	0.1	0.06	–	0.0045	–
Euro 6c	WLTP	1	0.1	0.06	–	0.0045	6.0×10^{11}
Euro 6d	WLTP	1	0.1	0.06	–	0.0045	6.0×10^{11}
	RDE	–	–	0.09	–	–	9.0×10^{11}

Additionally to the pollutant regulation, manufacturers of passenger cars are obliged to ensure a maximum average emission of CO_2 for the entirety of their new car fleet. This average, like the pollutant standards, is getting more and more stringent, going from 159 g/km in 2007 to 130 g/km since 2015, and to a targeted 95 g/km by 2021.

According to the previous scenario, the whole energy industry and more specifically here the automotive industry are evidently involved in a fast transition era with critical challenges in different aspects including highly stringent pollutant emission standards, implementation of new CO_2emissions regulations, fast-evolving markets and more exigent customer demands. Thus, it is evident how even greater R&D efforts are required since the long-term future of ICE's in the field of the transportation rely on their potential to adapt to the previous demands. These efforts are not being only focused on the further development of the well-established technological bricks already in use in ICE's, but other promising solutions with proven potential at research level such as alternative combustion concepts or engine architectures are also under evaluation.

1.2.2 Main R&D technology paths of modern powertrains for light-duty applications

Since 2017, several countries have announced their plan to eventually end the sales of fossil-fueled vehicle before 2040. This horizon may seems still far in the future, but this confirms the urge for manufacturers to develop new solutions for mobility and new powertrain technologies; at first to keep improving the cleanness of both SI and CI engines in order to offer solutions to comply with current and forthcoming regulations as a transitioning period; secondly to propose innovative solutions for mobility, such as electric vehicles, but also hybridization.

The main challenges for current technologies such as CI or SI engines is now to provide efficient and clean power to supply the required energy for today's mobility. Hybrid powertrains are a widely investigated solution, and the thermal part of the engine will keep using existing –but improved– systems. In that framework and focusing on CI engines, the injection and air management systems have been developed almost to their limits [3]. However, the possibility of further decreasing simulataneously $NO_x$and PMemissions is being doubted since the Conventional Diesel –mixing-controlled– Combustion (CDC) process characteristic of CI engines intrinsically promotes the formation of these pollutants due to the local temperature and air/fuel ratio stratification. Thus, a trade-off is observed between $NO_x$and PMemissions as the actions oriented to reduce one of them forces the increment of the other, while an additional trade-off can also be observed between NO_x and fuel consumption ($CO2$) as controlling NO_x usually requires countermeasures that affect negatively the combustion process and then the engine thermal efficiency.

Along the years great research efforts are being carried out in order to overcome certain aspects of the previous issues of the CDC concept, improving the characteristics of CI engines. The state-of-the-art package of solutions to control exhaust pollutant emissions include the combined use of internal (combustion based) and external (exhaust gas based) countermeasures. The most implemented internal strategies consist of introducing low to moderate external Exhaust Gas Recirculation (EGR) levels to control $NO_x$and increasing the boost and injection pressures to mitigate as much as possible the negative impact on PM, together with the use of external strategies as advanced after-treatment systems to further decrease these and all other pollutants in the exhaust line, before the gases reach the atmosphere. Another alternative lays in switching from the CDC concept to other advanced combustion strategies, known as Low Temperature Combustion (LTC) concepts, in order to control emissions from the source. These concepts are based on keeping the mixing-controlled combustion but decreasing the local temperatures below the $NO_x$and PMformation thresholds (mixing-controlled LTC concepts), or directly avoiding the mixing-controlled combustion by premixing the fuel with the air before the onset of combustion decreasing the local equivalence ratios below the PMformation threshold and controlling $NO_x$with additional measures such as high EGR rates if the local equivalence ratios are not low enough to avoid its formation (premixed LTC concepts). Some of the most investigated premixed LTC concepts are the Homogeneous Charge Compression Ignition (HCCI), Premixed Charge Compression Ignition (PCCI), Reactivity Controlled Compression Ignition (RCCI), Partially Premixed Compression Ignition (PPCI), and Partially Premixed Combustion (PPC) [4–6]. But even though the potential of these concepts has been widely highlighted, many of them either require specific conditions to be implemented, or present limitations in terms of load/speed operating range or mechanical constraints, such as the examples observed and displayed in this thesis, among others.

In the case of the SI engine, the key challenge is to reduce drastically CO_2 and then fuel consumption, keeping its advantages compared to the CI engine: lower NHV levels (Noise, Vibration and Harshness, synonym of comfort), simpler AFT systems, and lower costs for both manufacture and maintenance. Design solutions (such as downsizing and/or turbo-charging) combined to internal technologies (Variable Valve Actuation (VVA), Gasoline Direct Injection (GDI)...) are some of the keys that allow coping with this challenge by providing a much larger control on the engine operation. But those technologies also have drawbacks, as for example the particulate emissions generated by the GDI engines mainly during cold starts and acceleration (transitory conditions), that require specific AFT systems to comply with the latest regulations. To try and compensate those phenomena, in-cylinder management is implemented under various forms to keep the flame out the rich, high-temperature zones and thus avoid the PM formation. Among those strategies, fuel stratification and lean burn combustion are paths to keep improving the efficiency and performance levels of GDI engines, while a combination of Controlled Auto-Ignition (CAI) at low loads and SI at higher loads shows promising perspectives to reduce both CO_2 and pollutant emissions. However, this kind of advanced combustion strategies often rises issues in terms of engine control, having trouble managing the auto-ignition of gasoline in CAI mode, and the transition between the two modes.

As mentioned previously, dedicated AFT systems are now mandatory for both SI and CI engines in order to comply with regulation standards. Those systems have been developed and improved for the last 30 to 40 years, starting with two- and three-way catalysts in the 1970's to handle CO, HC, and NO_x. Then, specific systems were developed for specific applications, such as the Diesel Oxidation Catalyst (DOC) in the late 1980's or the Diesel Particulate Filter (DPF) in the early 2000's. With the increased requirements of emission regulations, the after-treatment of NO_x became one of the most challenging aspect to implement efficiently. Today's solutions rely mainly on the Lean NO_x Trap (LNT) often combined to a Selective Catalyst Reduction (SCR), and more recently an Ammonia Slip Catalyst (ASC). All those systems combined contribute to make the engine exhaust line –and thus the vehicle– more and more complex, and unavoidably more costly, expense-wise but also resource-wise, since those systems require specific and rare components. Even though these complex technologies apply mainly to CI engines, SI GDI engines also start to require them since the latest regulations of PM and PN. Thus, these dedicated AFT systems would contribute to lower the competitiveness of SI engines by increasing the complexity and costs of their pollutant emissions control.

Another path of investigation to reduce pollutants formation from the cylinder is to change the chemistry of the combustion itself, by using alternative fuels. SI engine platforms are particularly flexible to adapt the combustion process to the characteristics of other type of fuels. Compressed Natural Gas (either fossil or from renewable sources) is a good alternative example: among its advantages, being gaseous in standard conditions helps for the air-mixing process, reducing HC and CO by avoiding the liquid wall impingement and also decreasing extremely PM by the absence of locally rich regions associated to the evaporating liquid phase. For CI engines, the development of synthetic diesel fuels (Fisher-Tropsch process, vegetable oil conversion...) allows to tune the given fuel composition for any specific application, providing higher cetane number or lower aromatics content for example.

In any case, and as previously mentioned, the global trend for mobility leans towards new type of powertrains entirely. Electrification, with all its lot of limitations and restrictions, but hybridization, which is a more realistic approach in terms of accessibility in a mid-term scale. ICE's design will then follow different criteria, as the operating range could be reduced in some cases to be only used as an assistance to electric motors. Its compactness will also become a more predominant factor, since it will need to be associated to one or several electric motors and a pack of batteries often considerably big and heavy. In that framework, the potential for R&D of cleaner, more efficient, and smaller/lighter engines will be crucial to fulfill the requirements that will arise in the years to come from future regulations, politics, and public awareness discussed so far. Innovative combustion concepts and alternative engine architectures are thus one of the main paths to provide answers to these questions and cope with the hybrid powertrains increasing complexity.

1.3 Focus of the research work of this thesis

The activities and investigations presented in this thesis have been developed in the frame of two main programs: a research collaboration with Renault SA following up the activities performed along the European POWERFUL project (POWERtrain for FUture Light-duty vehicles), and the European REWARD project (REal World Advanced technologies foR Diesel engines) which was brought as a continuation of the POWERFUL project in the frame of the Horizon 2020 research program. Both projects targeted the development and optimization of the 2-Stroke engine architecture. The European POWERFUL project aimed to the evaluation of the potential of the Conventional Diesel Combustion (CDC) implemented on a single-cylinder 2-Stroke poppet-valve HSDI CI engine designed for passenger car application and was based on a commercial four cylinders 4-Stroke CI engine. 2-Stroke engines provide several well-known advantages compared to 4-Stroke engines, particularly their higher power density due to its doubled firing frequency, so the delivered specific power is theoretically doubled when operating on 2-Stroke cycles. This offers a more compact package and lighter weight for similar performance outputs, and also potentially lower mechanical losses [7].

As briefly discussed previously, one of the paths to increase engine efficiency and reduce contamination is the implementation of new combustion concepts to improve the global cylinder outputs. New combustion concepts mean new mixture preparation processes and possibly also new mixture compositions, to get more favorable conditions before the onset of the combustion resulting in a cleaner and more efficient chemical transformations. One of these advanced strategies was explored in detail during this thesis work: the gasoline Partially Premixed Combustion (PPC) concept, which consists in injecting gasoline fuel inside a CI engine and separate the injection events from the combustion to rich local conditions and thus prevent PM formation. This is achieved by properly phasing the injection events early during the compression stroke in order to control the auto-ignition of the mixture through its reactivity, which evolve accordingly to its degree of premixing [8].

Advanced combustion concepts, based on the auto-ignition in highly premixed conditions of either gasoline or diesel fuels, have been studied for several years on 4-Stroke engines, for both light- and heavy-duty applications. But the biggest limitation encountered then is the restricted operating range: diesel fuels, due to their high auto-ignition tendency, provide good results at low load conditions, but ignite too early in the cycle when the load is increased, then requiring to delay the injection events until recovering diffusive-like combustion. In the case of the gasoline premixed auto-ignition, the operational range is even more reduced, since low load operations does not usually provide the necessary pressure and temperature conditions for auto-ignition, while high loads are restricted by the extremely high cylinder pressures that rise over the mechanical engine limits, requiring also the delay of the injection event to recover the diffusive-like combustion.

However, the 2-Stroke engine architecture seems to be a suitable answer to some of those problems and to facilitate the implementation of advanced combustion concepts: the higher firing frequency compared to 4-Stroke engines and the lack of "cooling" stroke, combined with controlled levels of hot residuals, helps enabling the auto-ignition of gasoline at low load conditions through satisfying pressure and temperature conditions. Additionally, the naturally higher levels of Internal Gas Residuals (IGR) help reducing the reactivity of the mixture at higher loads, thus facilitating the control of the combustion process at these conditions. Moreover, the doubled firing frequency compared to a 4-Stroke engine delivers theoretically the same engine torque with only half the engine load (BMEP, Brake Mean Effective Pressure). Considering all these advantages, the 2-Stroke engine architecture then appears as a natural solution to implement and investigate the full potential of advanced combustion concepts. In that framework, the Chapter 4 of this thesis presents the study of the gasoline PPC concept implemented in a single-cylinder 2-Stroke poppet-valve HSDI CI engine.

But if the 2-Stroke engine architecture provides thermodynamic advantages for the implementation of new combustion concepts, it often also presents some important drawbacks, especially regarding lubrication issues, and most important concerning the scavenging performance usually resulting in high levels of fresh mixture short-circuit. These aspects, particularly critical when implemented as an automotive powertrain, have then restricted the use of 2-Stroke engine to very specific applications. Either very lightweight vehicles (such as motorcycles) or small-engined portable devices (chainsaws, outboards motors) in the case of SI 2-Stroke engines due to their high performance-to-compactness ratio; or very large applications (ship or locomotive engines, large electricity generators) in the case of CI 2-Stroke engines, due to the very high efficiency of their specific architecture, where brake efficiency can exceed 50% for large Uniflow low-speed engines used in marine applications, among the highest levels of all ICE's. Several automotive companies have explored the 2-Stroke engine potential in the 1990's, including Yamaha Motor Co. [9], Toyota [10] or AVL [11, 12], but none of them successfully brought it to production.

Most of the critical drawbacks of 2-Stroke engines are mainly caused by the architecture most commonly considered for road vehicle applications: the piston-controlled port scavenging, where the piston uncovers at the end of the expansion phase both intake and exhaust ports located at the bottom of the cylinder liner, facing one another. Those engines usually use port injection, so the intake mixture contains fuel, but also oil since the crankcase is used as a transfer case, the cylinder lubrication is then often achieved through oil mixed in the fuel. A large amount of this intake mixture is then by-passed directly to the exhaust by design, generating extremely high levels of unburnt HC emissions. This phenomenon requires specific intake and exhaust acoustic management to be minimized, but is therefore dependent on engine speed [7, 13–15]. Efficiency is then reduced since the fixed distribution timing reduces the optimization levers. Other aspects also need to be taken into account, as for example unstable part-load combustion, lower engine durability due to the lubrication issue and to the asymmetric heat flux caused by the flow through the ports, and increased piston ring wear [7, 13–15].

However, the 2-Stroke cycle can very well be implemented on different architectures, such as a poppet-valve engine similarly to conventional 4-Stroke engine. This solution solves the lubrication issue as crankcase lubrication can be used in that case, and facilitates the implementation of direct injection. It is also very cost-effective, considering the possibility of sharing most of the components and mechanical design with mass production 4-Stroke engines. But because of the intake and exhaust valves being both located next to each-other in the cylinder head, trapping difficulties appear [15], and specific solutions need to be investigated to avoid –or at least limit– the intake short-circuit to the exhaust, with VVA systems to control the scavenging timing, and new design of the cylinder head and piston to orientate the intake air flow [16, 17]. This specific engine architecture has been deeply investigated during the European POWERFUL project, evaluating the potential of such a design operating with the CDC concept, and confronting its performance and pollutant emissions to state-of-the-art 4-Stroke engines. The thorough knowledge of this specific engine architecture generated during these investigations, made it the most sensible platform for investigating the gasoline PPC concept described in Chapter 4, thus also benefiting from the mechanical technologies available to help control and optimize this combustion concept as much as possible (DI, VVA to manage the IGR level, EGR...).

Another possibility to implement the 2-Stroke cycle and help solving the scavenging issues observed with the poppet-valve architecture consists of being inspired by the most efficient existing ICE's –the long-stroke Uniflow scavenged engine– and design a passenger car engine considering the key factors of this architecture. Thus, in the framework of the European REWARD project, the main objective was to design a well-scavenged 2-Stroke CI engine in order to implement the CDC concept and evaluate another option among the 2-Stroke architecture possibilities. The objectives of this project was to design and optimize the complete scavenging process, in order to achieve in-cylinder composition as close as possible to a conventional 4-Stroke engine, but then to also benefit from the thermodynamic advantages of the 2-Stroke engine. The detail of this engine design, its manufacturing and its evaluation (both theoretical and experimental) is thoroughly described throughout Chapter 5.

The various technological paths highlighted here are of great interest as they represent cost-effective and efficient solutions for modern small-class vehicle CI engines, and could provide valuable alternatives to meet with the always more stringent pollutant regulation standards and keep competitive efficiency levels. The main paths of interest to be explored can be summarized as the following:

- The nature of 2-Stroke engines facilitates the implementation of advanced combustion concepts, which can bring new ways to reduce (drastically) pollutant emissions while also improving fuel efficiency.

- The increase power density of the 2-Stroke architecture allows extreme engine downsizing (either by decreasing cylinder unitary displacement, or by reducing the number of cylinders) while keeping equivalent levels of torque and power output.

- Different engine architectures can provide different optimization paths and enable a set of possibilities for solving the most critical issues that kept more conventional 2-Stroke designs from reaching production levels of cost, efficiency and cleanness.

These paths then rise some questions and problems, that the work presented in this thesis will aim to answer:

- What is the potential, in terms of both performance and pollutant emissions, of the gasoline PPC concept when implemented in a 2-Stroke poppet-valve HSDI CI engine based on a conventional 4-Stroke engine conversion, compared to CDC concept?

- What are the control limitations for this combustion concept, and can it be operated throughout a full engine map?

- In what measure can a different 2-Stroke architecture improve the scavenging process, and what level of complexity would it require in terms of design, manufacture and air management, to get as close as possible to 4-Stroke engines scavenging performance (IGR levels and Trapping Ratio)?

- What are the main limitations of these engine architectures, and could each of them be suitable for passenger car applications, whether it would be as a dedicated powertrain or a range-extender unit in a hybrid powerplant system?

These previous questions were addressed along the research activities carried out and they will be discussed in detail along the present dissertation.

1.4 Objectives and organization of the research

The present investigation was carried out according to the two main research lines described in the previous section. Thus, the present document is organized in six chapters, the present introduction being Chapter 1, followed by a thorough literature review of the most relevant research studies performed in relation with the various topics discussed throughout this thesis, especially concerning the reduction of pollutant emissions using different approaches of advanced combustion concepts, and the various engine architectures and their most common fields of applications.

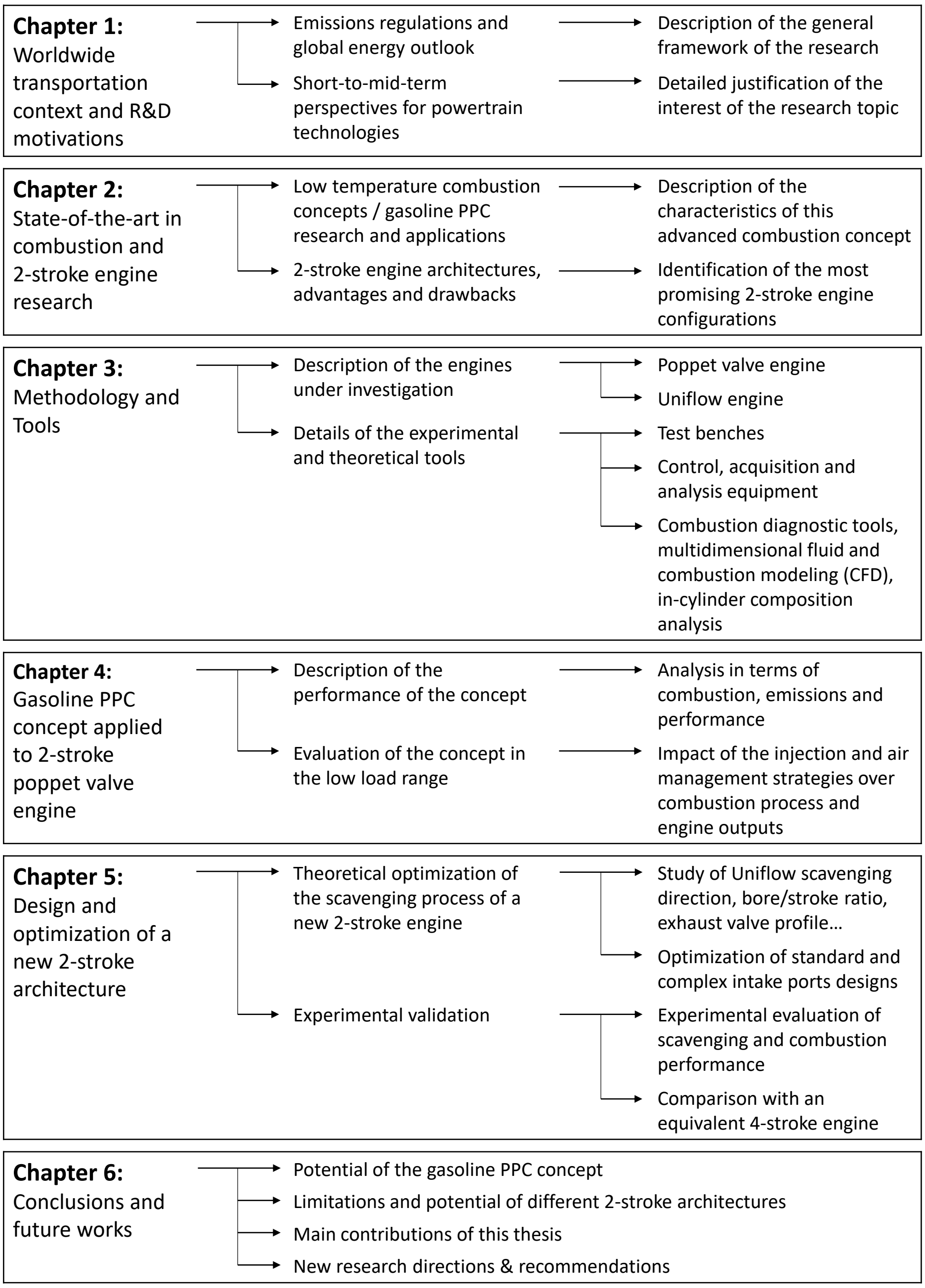

Figure 1.1. *Outline of the thesis.*

The outline of the thesis presented in figure 1.1 introduces the different parts of the document, each with a brief description of the main topics discussed.

After setting the general context of the work, the problems in play and the objectives to fulfill, and reporting the status of the research on those various topics, the Chapter 3 focuses on both the theoretical and experimental tools necessary to the studies performed, bringing a detailed description of the two engines and their experimental setup, as well as tools and programs used to post-process the data obtained throughout the different studies carried out. It also defines the methodology applied for each part of the investigations, such as the Designs of Experiments (DoE) methods used for the experimental campaigns of both engines (although with different objectives) or the complex CFD simulation optimization process followed during the design of the Uniflow engine design.

Chapter 4 and Chapter 5 describe the results obtained, both theoretically and experimentally, along the years of the doctoral thesis investigation work. Chapter 4 includes experimental measurements from the gasoline PPC concept implemented on the 2-Stroke poppet-valve engine, the steps towards understanding the effects of the air management and injection parameters on the combustion control and engine-out emissions, and the optimization path followed to improve the capabilities of this specific combustion concept and bring a contribution to the research literature on the topic. But it also describes the main issues and limitations encountered during these investigations, and tries to point out possible next investigation paths to follow in order to overcome those problematics.

Chapter 5 then comes in that continuity of work, introducing new engine architectures through the European REWARD project which aims to design a 2-Stroke power-unit that answer one of the main limitations pointed out at the end of the investigations described in Chapter 4: the 2-Stroke cylinder scavenging performance. This chapter presents the different designs considered, the very detailed 3D-CFD simulation campaign carried out to theoretically fully optimize the chosen geometry, up until the manufacture and the experimental testing and validation (using another DoE method) of the intake ports previously designed. As for the Chapter 4, a part of this chapter is also dedicated to the issues faced during the studies, more particularly the mechanical failures incurred during the experimental campaign and the hypothesis made both to explain them and to try to overcome them in the time-frame of the project.

Finally, Chapter 6 comes as a conclusion of the document, highlighting the most relevant results and observations obtained throughout these studies. Some optimization paths are proposed to further improve the performance and emission levels of the different technical solutions and concepts studied during this work, before trying to define new possible research directions and tasks to fulfill in order to further investigate and get better knowledge or control over the concepts under development.

Based on all the points highlighted throughout this document summarizing several years of investigations, the research team surrounding the PhD student hopes to bring a substantial contribution both to the understanding of advanced combustion concepts such as the gasoline Partially Premixed Combustion, their control and their potential for fuel efficiency improvement and emission reduction, and also to the renewed interest for 2-Stroke engines and the possibilities offered by innovative architectures when brought to wider application fields than those to which they are usually restricted.

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Chapter 2

Literature review

Contents

2.1 Introduction

After describing the general framework of the research carried out in this thesis, the detailed review of the most significant results reported in the literature on the targeted topics is necessarily interesting to define the scope of the study and set its objectives according to the state-of-the-art available knowledge in the corresponding scientific and technological fields. With that in mind, this chapter includes two main sections discussing the fundamental characteristics and the present challenges of the advanced combustion concept and the 2-Stroke engine architecture under investigation in the present thesis.

Section 2.2 establish the research context for the gasoline Partially Premixed Combustion (PPC) concept that is investigated in Chapter4. This section introduces the premixing-based Low Temperature Combustion (LTC) concepts developed for CI engines, providing an overview of the different options ranging from the fully homogeneous to the partially premixed heterogeneous implementation alternatives. The analysis of their current status and future trends includes a short description of mixture formation and combustion processes, followed by the evaluation of the pollutant emissions and efficiency levels, emphasizing the impact of fuel properties. The Conventional Diesel Combustion (CDC) concept is kept intentionally out of the scope of this literature review despite this concept is integrated as the reference combustion system in the new 2-Stroke engine designed and optimized in Chapter5. The main reason supporting this decision is the fact that this combustion concept is not a subject of investigation, which focuses on the understanding of the key aspects related to the 2-Stroke engine architecture under development, and a very complete description of the CDC combustion concept characteristics is already available for the interested reader in the works of GarcÃa [1] or Novella [2].

Section 2.3 surveys the different 2-Stroke engine architectures and their application for CI engines in order to define a suitable background for the study of a new Uniflow 2-Stroke CI engine concept included in Chapter 5. A wide range of architectures, each offering different scavenging processes with their corresponding benefits and drawbacks, can be considered when designing a 2-Stroke engine, and this section aims to summarize those geometries before focusing on the two of interest for the research work carried out in this thesis, which are the poppet-valve design (Chapter 4), very similar to conventional 4-Stroke engine simply converted to 2-Stroke operation through minor changes in the cylinder head and by adjusting the valve timing ratio, and the Uniflow design (Chapter 5), known for its high thermal efficiency in long-stroke CI engines mostly for marine applications.

2.2 Fundamentals of the LTC concepts in CI engines

The start point for discussing the basic ideas below the LTC concepts in CI engines is the understanding of the local equivalence ratio (ϕ) versus local temperature diagram explaining the evolution followed by the fuel along the injection-combustion processes reported by Kamimoto and Bae [3, 4] as early as in 1988. This diagram was initially developed to provide a consistent explanation for the NO_x-soot trade-off provided by the CDC concept that was already well-known at that time, since the regions where both pollutant emissions form are easily observable as two isolated contour zones. Thus, after this pioneer work the scientific community focused their research efforts on identifying suitable combustion processes avoiding the fuel traveling across both pollutant formation regions, originating a wide range of alternatives later known as LTC concepts, and with the first research works of innovative combustion concepts tracing back to the early 80's [5, 6].

In their Figure 2.1 Potter and Durret [7] illustrated different possible paths followed by a differential fuel mass pocket in the ϕ-T diagram. As described by Dec and Flynn et al. [8, 9], the CDC concept (top plot) based on sustaining a reacting spray structure forces the fuel that is traveling across the spray (A) to diverge from the inert mixing line as soon as it ignites (B) at a given distance from the injection nozzle known as lift-off length, while it is still in rich conditions. From that point the temperature of the rich air/fuel mixture pocket suddenly increases due to the exothermic combustion reactions until reaching ϕ-T combinations falling inside the soot formation region (C). Afterwards, the hot air/fuel mixture becomes progressively leaner as it mixes with the surrounding air completing the oxidation reactions while traveling along the chemical equilibrium mixing line, and evolving from the soot to the NO_xformation regions, with the peak temperature observed close to stoichiometric conditions (D). This mixing process still continues in the lean side of the chemical equilibrium mixing line (E), but without releasing noticeable chemical energy as the fuel is already converted into combustion products.

The regions of the ϕ-T diagram targeted by the premixing-based LTC concepts with the aim of decreasing simultaneously both NO_x and soot formation islands are included Figure 2.1 (bottom), showing two main alternatives differing in the level of homogeneity in the air/fuel mixture at the onset of the combustion process. Despite the fundamental idea behind both concepts is very similar, basically avoiding the soot formation region by decreasing the maximum local ϕ and the NO_x formation region by decreasing the maximum local temperature at which combustion develops, their implementation and performance are definitely different, and so are some of their challenges.

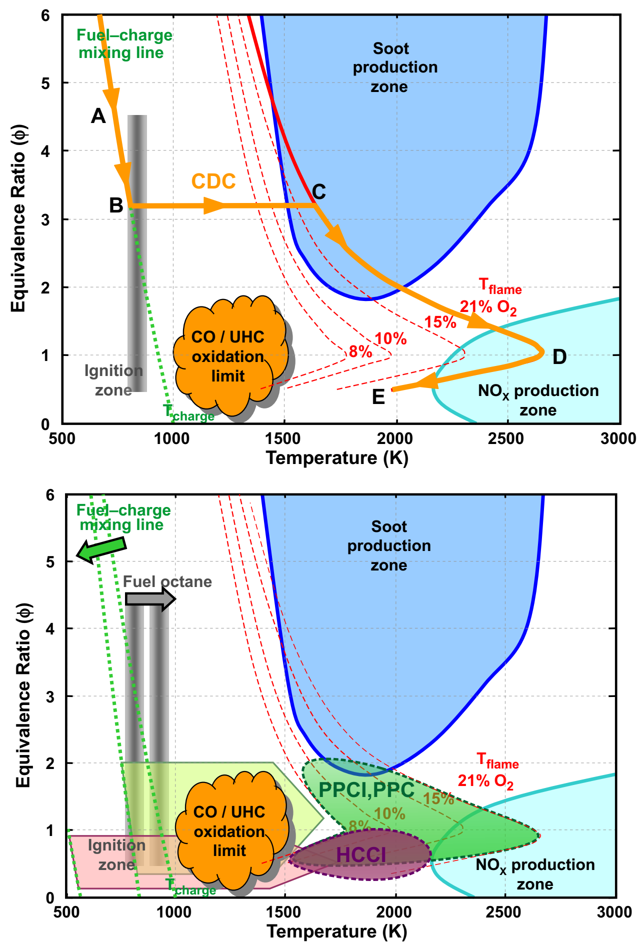

Figure 2.1. *ϕ-T diagram for CDC (top) and premixed LTC strategies (bottom). Adapted from Potter and Durrett [7].*

2.2.1 Description of the fundamental principles of the Homogeneous Charge Compression Ignition (HCCI) concept

With the need to reduce fuel consumption and CO_2 emissions, Low-Temperature Gasoline Combustion (LTGC) concepts are of particular interest as alternatives to conventional SI or CI combustions because they offer substantially higher thermal efficiencies, relying on two main reasons compared to CI engines, and two additional reasons if compared to SI engines [10, 11]:

- The lean equivalence ratios and low combustion temperatures of LTGC engines result in a significantly higher specific heat ratio (cp/cv) than for conventional combustions, thus increasing efficiency;
- Lower combustion temperatures help in reducing heat losses;
- LTGC engines typically have higher compression ratios, comparable to those of CI engines, thus directly improving thermal efficiency compared to equivalent SI engines;
- The charge of LTGC engines being typically controlled by injected fuel mass (similarly to CI engines), they are usually not throttled, thus minimizing the pumping losses.

However, in the fully homogeneous premixing-based LTC concept (generally known as Homogeneous Charge Compression Ignition (HCCI)), the fuel is injected indirectly into the intake port [12, 13] or directly into the cylinder but at the very beginning of the compression stroke [14–18], with the aim of extending the time available for obtaining the desired homogeneity between the injection and combustion events. Because of these types of injection strategies, controlling the combustion phasing is complex because of the absence of an explicit ignition-controlling event, such as a spark or a fuel injection near TDC. The onset of the combustion depends on the chemical kinetics of the charge during the cycle, which are very sensitive to the in-cylinder thermodynamic conditions, the oxygen concentration, the fuel molecular composition and, for some fuels, the equivalence ratio. The autoignition kinetics are particularly sensitive to temperature, so the most straightforward technique to control the combustion timing is by controlling the in-cylinder temperature at the start of compression. In a real engine, this can be achieved by heating the intake air using the exhaust gases. Alternatively, valve-timing strategies can be used to retain hot residuals [19, 20]. It should be noted that retaining hot exhaust products actually dilutes the mixture and decreases the oxygen concentration, but the high temperatures generally dominate over the decrease in oxygen content. However, this may not be true at low loads when the exhaust enthalpy of the retained hot residuals might not be sufficient to ignite the charge without supplying additional intake heat. One drawback of temperature control is that because of the thermal inertia of the engine, and because typical production cam profile switching is not particularly fast, controlling the temperature quickly enough to respond on cycle-to-cycle basis during transient operations is quite challenging. Using cooled EGR is another well-known strategy to control the combustion phasing [10, 21], which lowers the autoignition reactivity by reducing the oxygen concentration and the compressed-gas temperature (due to the lower ratio of specific heats of the EGR gases compared to air). However, it is also difficult to quickly change the amount of EGR, and at low loads significant intake heating or retained hot residuals are needed without EGR.

The combustion phasing can be also controlled by injecting fuel during the Negative Valve Overlap (NVO). The injected fuel undergoes some exothermic reactions before the main ignition event, raising the temperatures above what would occur from compression alone, altering the pressure-temperature trajectory towards the ignition point. Furthermore, highly reactive intermediate species are generated during the recompression, affecting the reactivity of the fresh air/fuel mixture. Thus, this technique controls the combustion through combined thermal and chemical effects that can be tweaked on a cycle-to-cycle basis. However, NVO fueling penalizes the efficiency mainly due to the heat losses during the recompression of the hot residual gases.

Spark Assist (SA) is another technique that can provide combustion-timing control on cycle-to-cycle bases. With SA-LTGC, a spark initiates a flame kernel and the subsequent flame propagation compresses the unburned mixture, increasing its temperature and pressure and ultimately, driving the main charge into autoignition. Because the spark acts as a trigger for the start of combustion, it allows direct control over the combustion phasing by varying its timing. However, for SA to be effective, the equivalence ratio and charge temperature must allow adequate flame propagation while avoiding the autoignition to occur at an earlier undesired timing. This limits the performance of SA, especially at low engine loads.

Varying the amount of fuel stratification is one of the most promising strategies for combustion-phasing control in LTGC engines. The level of stratification can be controlled by changing the settings of a direct-injection (DI) of fuel. The quantity of fuel and the injection timing can be adjusted from one cycle to the next, providing a fast response during transient operations, which often requires a rapid shift in the combustion timing to prevent the engine from knocking or misfiring. However, for this strategy to be effective, the different local equivalence ratios within the stratified mixture must have different ignition delays. This specific fuel capability is termed ϕ-sensitivity: a fuel is ϕ-sensitive if its autoignition reactivity varies with the fuel/air equivalence ratio (ϕ) for the same thermodynamic conditions. Under typical LTGC conditions, richer regions of the in-cylinder ϕ distribution autoignite faster, controlling the combustion phasing. Several authors have used fuel stratification strategies with the intent of controlling the combustion timing (with CA10 and CA50 generally used as indicators), such as Kalghatgi et al. and Johannsson and coworkers. These authors found that, for SOI earlier than a certain late-compression-stroke timing, CA50 became more advanced as the SOI was retarded, whereas for SOIs later than this particular timing, CA50 moved with SOI, as occurs when operating with the CDC concept. Sellnau et al. applied a double DI to control the combustion phasing of a LTGC engine fueled with regular gasoline. Keeping the timing of the first fuel pulse fixed, and sweeping the second DI timing starting from a relatively early point, the CA50 advanced as the second DI timing was retarded, providing about 7 CAD of CA50 control from the most-retarded to the most-advanced points.

Sjöberg and Dec developed a specialized fuel stratification technique, based on multi-zone kinetic modeling, called Partial Fuel Stratification (PFS), in which the majority of the charge (about 70 to 90% of the total fuel mass) was injected early in the intake stroke (or premixed) while the rest was injected during the latter half of the compression stroke. The early-injected fuel provided a background mixture that is rich enough to ensure good combustion efficiency, whereas the late-DI timing or fuel fraction was limited to avoid overly stratified conditions that can lead to unacceptable NO_x and soot emissions. Note that if mixtures are too lean or too diluted with EGR, combustion temperatures become so low (below 1500 K) that reaction rates are too slow for the bulk-gas CO-to-CO_2 reactions to go to completion before the piston expansion quenches the combustion. The potential of PFS to extend the engine load and control the combustion phasing was studied by Dec and coworkers in numerous studies.

An alternative strategy, capable of providing CA50 control for LTGC engines over a wide range, was recently developed by Dec and co-workers. This strategy, referred to as Additive-Mixing Fuel Injection (AMFI), is based on supplying a small amount of ignition improver (of the order of tenths of a milligram per cycle) into the fuel just upstream of (or inside) the gasoline-type direct injector (GDI) to adjust autoignition reactivity of the fuel. The amount of additive can be varied on cycle-to-cycle basis, providing fast control of the combustion phasing. Dec et al. showed that the AMFI system performed well over a wide range of conditions using 2-ethylhexyl nitrate (EHN) as the additive. EHN is a well-known diesel-fuel ignition improver that also enhances the autoignition of gasoline-like fuels. In addition, the amount of intake heat required to achieve autoignition is significantly reduced (or removed altogether), resulting in much higher charge densities, gross IMEP, and thermal efficiencies compared to the straight fuel. Furthermore, EHN-doped gasoline shows two-stage ignition and increased the intermediate-temperature heat release at conditions where straight gasoline does not. This improves the combustion stability, allowing for a more retarded CA50 and extending the maximum engine load.

A large part of the existing litterature demonstrates that HCCI (and more generally any well-mixed LTGC) operation is usually limited to low-to-medium engine loads. This limitation is due to the fast increase of the maximum cylinder-pressure rise rate (PRRmax), reaching knocking conditions and possibly damaging the engine, occuring when the fueling rate (thus the engine load) is increased. These high PRRmax values can be lowered by retarding the combustion phasing, but this delay is limited by combustion stability and possible misfire, since the combustion phasing window between misfire and knock conditions becomes narrower as the fuel mass (i.e. the load) is increased. Therefore, it exists a specific condition called "knock-stability limit", where autoignition is stable but no control over the phasing of the combustion is possible because of its high sensitivity to more fuel/load increase.

The wall heat transfer combined to the turbulent convection occuring in a real engine induce a thermal stratification, allowing a sequential autoignition of the mixture with the hottest regions burning prior to the coldest, somehow contributing to smoothen the combustion development with an extended duration and a reduced maximum RoHR, thus helping to extend significantly the load limit. This behavior can be amplified by the delayed combustion timing, further extending the load limit. On real conditions then, thermal stratification can be considered as the determining factor for the autoignition of HCCI operation. But if controlling thermal stratification could theoretically be used to control the RoHR profile on a wider load range, it has proven very difficult to achieve such control through practical methods, without damaging heat losses and thus fuel efficiency.

The knock/stability limit can also be extended through an additional method: the use of alternative fuels or operating conditions capable of greater amount of Intermediate Temperature Heat Release (ITHR). As it has been well described by Dec et al. [22]: *"ITHR results from the early reactions that ramp the combustion up into hot autoignition [23–25], and as the name indicates, it occurs at temperatures below the hot-ignition temperature and above those of the low-temperature heat release (LTHR) if the latter is present. The ITHR keeps the bulk-gas temperatures rising during the early expansion stroke, providing combustion stability as timing is retarded. Increasing the ITHR allows greater combustion delay with good stability, which shifts the knock/stability limit later in the combustion cycle, thus allowing higher loads. The importance of this effect on combustion stability and timing retard limits was first reported for PRF80 at naturally aspirated conditions [26]. Because PRF80 showed two-stage ignition, it was initially thought that LTHR was required to gain this benefit. However, it has been later showed that for a regular-grade (AKI=87) gasoline, the ITHR increased substantially as intake* *boost pressure was increased even though little or no LTHR was observed [27]. The increased ITHR allows to greatly increase the combustion delay to prevent knock for high fueling rates at high boost pressures. As a result, gross IMEP above 16 bar was obtained for well-mixed LTGC (i.e. HCCI). However, this required an intake pressure of 3.2 bar absolute, which may be unrealistically high for a turbocharger with this engine load."*

If both thermal stratification and ITHR strategies are practically too complex to achieve, then maybe another promising method to extend LTGC engine load ranges could consist of using fuel stratification as a lever for sequential autoignition, similarly to what thermal stratification could allow. But contrarily to this previous strategy, fuel stratification is much easier to control, since it can be managed directly through the DI-fueling strategy. For this method to provide an effective control over the RoHR, it must be calibrated in such a way that the different areas described by the local equivalence ratio gradient have different ignition delays, in order to ignite sequentially. This phenomenon is called ϕ-sensitivity, and is directly dependent of the fuel properties, and has again been well introduced by the work of Dec et al. [22]:

"Initially, ϕ-sensitivities were measured only at naturally aspirated conditions for a limited number of fuels, and it was thought that two-stage ignition was required for fuels to be ϕ-sensitive [28, 29]. However, later studies showed that two-stage ignition (i.e. LTHR) was not required and that ϕ-sensitivity was actually correlated with the ITHR. The ITHR is typically higher for two-stage fuels, but similar ITHR's can occur with single-stage ignition fuels under boosted conditions, as will be discussed further below.

Early attempts at fuel stratification simply tried various DI-fueling strategies in HCCI engines without knowledge of the importance of fuel effects and ϕ-sensitivity. Kalghatgi et al. [30] tried to extend the engine load of a single-cylinder DI CI engine fueled with conventional gasoline by applying different injection strategies with the intent of promoting fuel stratification. They obtained IMEP values up to 15.95 bar; however, the fuel stratification needed to reach such load led to NO_x emissions that were much higher than the US 2010 limit. This issue could potentially be resolved with more carefully controlled fuel stratification and ensuring that the fuel was ϕ-sensitive under the conditions studied. In fact, Sjöberg and Dec [28] experimentally evaluated the ability of PRF80 and PRF83 to increase the engine load by stratifying the mixture in a naturally aspirated LTGC engine. These fuels were selected because separate tests showed that they were ϕ-sensitive at this operating condition. Moreover, they [28] developed a specialized fuel stratification technique, based on multi-zone kinetic modeling, called partial fuel stratification (PFS), in which *the majority of the charge (80-90%) was injected early in the intake stroke (or premixed) while the rest was injected later during the compression stroke. The timing and fuel fraction of this last DI-fueling were varied to control the amount of stratification. With this approach, they obtained higher IMEP values compared to a well-premixed charge, without knock or misfire, and while keeping* NO_x *emissions at acceptable levels. These findings are supported by Yang et al. [24, 25, 31] who showed that PFS with* ϕ*-sensitive fuels could significantly reduce the PRRmax compared to premixed fueling for both PRF73 and a low-octane gasoline. Furthermore, they [25] showed that because PFS reduced the PRRmax at a given engine condition, combustion timing (*CA50*) could then be advanced to obtain higher thermal efficiency without exceeding the same PRRmax as the original premixed operation. These findings are also consistent with the works of Wada and Senda [32] and Dahl et al. [33] who studied the effects of fuel stratification using two-stage ignition fuels that were* ϕ*-sensitive under the naturally aspirated conditions of their tests.*

These early demonstrations of the fuel stratification advantages were conducted using two-stage ignition fuels that show ϕ*-sensitivity at naturally aspirated (*P_{int} *= 1 bar) conditions. Subsequently, Dec et al. [34] showed that regular-grade (87 AKI) gasoline which is not* ϕ*-sensitive at atmospheric conditions becomes progressively more* ϕ*-sensitive as the intake pressure is increased from 1 to 2 bar. In fact, it is more* ϕ*-sensitive at* P_{int} *= 2 bar than PRF 73 (a two-stage fuel) is at atmospheric conditions, even though it shows virtually no LTHR at these boosted conditions, only enhanced ITHR. Dec et al. [34] then applied PFS using this gasoline, showing that it could reduce the RoHR for intake pressure varying from 2.2 to 2.4 bar, which allowed them to obtain higher loads without engine knock and with good stability, and/or to advance* CA50 *for higher efficiencies. These changes in* ϕ*-sensitivity with intake boost correspond extremely well with the amount of ITHR enhancement with boost, which also allows greater combustion delay with good stability [27], as discussed above. Subsequent to this work, Yang et al. [25] examined ITHRs and* ϕ*-sensitivities of both a low-octane gasoline that exhibited two-stage ignition and this regular-grade gasoline under boosted conditions which showed only single-stage ignition, and they linked the* ϕ*-sensitivity to the intensity of the ITHR for both fuels. These authors found that the more intense the ITHR, the higher the* ϕ*-sensitivity. An initial attempt to understand the chemical kinetics behind the ITHR were performed by Mehl et al. [35]. They concluded that the chemical nature of the ITHR is significantly different from the chemical nature of the Low Temperature Heat Release (LTHR)."*

On the other side, some issues can also be observed with the LTGC operation when reaching low to very-low engine loads. Typically in a CI engine, the load is controlled by the equivalence ratio, and it has been demonstrated that when reaching values below 0.3 (usually around 0.28, but depending on various parameters), the combustion efficiency starts to decrease drastically until critical value around 60 - 70%. This is due to a too lean mixture, leading to an inability to initiate the oxidation reaction of CO into CO_2 through a lack of in-cylinder temperature. Moreover, this low temperature extends the combustion duration until exceeding the available time during the cycle, meaning that the combustion is extended towards the expansion stroke and the combustion reaction is stopped. When decreasing the equivalence ratios down to 0.24, it becomes impossible to initiate the combustion at all.

An easy solution to this issue consists of using a throttle valve (similarly to SI engines) in order to limit the air admitted into the cylinder and increase the equivalence ratio back again. But this solution is very inefficient because it can generate a large amount of pumping losses.

One other –and more efficient– possibility consists of generate once again some stratification of the mixture, in order to concentrate most of the injected fuel at the center of the cylinder and get a more reasonable local equivalence ratio, yet with a global equivalence ratio still lower than the previously detailed limit. This stratification can allow a much better combustion efficiency and stability, the reaction occurring in a virtually reduced volume.

2.2.2 Description of the fundamental principles of the gasoline Partially Premixed Combustion (PPC) concept

As described previously, most of the LTC concepts (using gasoline-type fuels) under study for the past decades by several research teams are limited by numerous factors, mostly when reaching high loads. Some solutions have been explored to expand the application range of such concepts, in terms of combustion development (suitable autoignition timing, limited detonation / knock, no misfire conditions) as well as in terms of pollutant emissions, to avoid the soot generation induced by high levels of EGR or fuel stratification before the onset of combustion, and also the NO_x generated in the locally high temperature areas or during knock-trending combustions.

One of the most promising solution seems to be brought by the partial stratification of an originally homogeneous charge, in order to control the mixture dilution and reactivity just before the onset of combustion, thus providing good control of the combustion development and outputs. This solution is relatively easy to set up, since it only requires a different injection strategy while being implemented on a completely standard platform: no hardware adaptation is needed, providing a complete flexibility for the implementation of such strategy on only part of the engine operating range if necessary, or to go back to operate with a more traditional combustion strategy for any reason (engine break-in or warm-up cycles for example). The lack of specific hardware requirement also contributes to make this strategy a cost-effective solution, along with the fact that a large variety of settings can be implemented for various outcomes simply by adapting the injection pattern as desired, as long as the mixture conditions are satisfactory for a proper ignition and combustion development.

With that flexibility in mind, the partial stratification of an homogeneous charge can be extrapolated to its extreme limit: injecting all the fuel during the intake stroke, most of it early enough to let the mixture premix before TDC, but without reaching full homogeneity as it is the case for the HCCI concept. This strategy is commonly called Premixed Charge Compression Ignition (PCCI) or Partially Premixed Combustion (PPC), and has being heavily investigated for the past decade using sometimes diesel fuels but more interestingly gasoline-type fuels, and implemented on various engine types (light- or heavy-duty, 2 or 4-Stroke architectures, etc...). This concept is based on the principles described previously and uses the variability of the mixture's reactivity depending on its homogenization degree to both phase the onset of the combustion and control its rate as desired (still within certain limitations that mostly depend on the operating conditions and the engine definition).

This control over the mixture reactivity is only possible using a HSDI system, allowing to set-up a finely tuned injection strategy: the largest part of the fuel mass is injected early during the compression stroke (usually between -60 and -30 CAD aTDC), creating a still stratified mixture by not giving it enough time to reach homogeneity before the onset of combustion, but yet without any liquid spray or liquid region left when this major part of the mixture ignites, thus separating the injection event from the combustion and avoiding a diffusive flame and the related soot generation. However, it is still often necessary to set a short but late injection event, closer to TDC, which provides a local reduction of the mixture temperature and reactivity in order to avoid too sharp combustion rates (i.e. knocking-like combustions). This late injection is in most cases very close to the onset of the combustion and the fuel injected then is burnt in a diffusive process. However this fuel proportion being quite low, the impact on the soot generation is either negligible or in the worst cases manageable to still presenting a viable compromise for the combustion control.

Of course, the control of the mixture reactivity alone is often not enough to provide a fine control of the combustion outputs, either in terms of thermodynamical and mechanical limits, or also in terms of fuel efficiency and pollutant emissions. In order to get finer tunings of the in-cylinder conditions, and also a wider operating range in some cases, the use of both external and internal gas recirculation is quite commonly associated with the gasoline PPC concept. The external EGR is a widely known way of controlling the combustion rate and reducing the NO_x emissions by lowering the combustion temperature; while the presence of IGR, which are essentially hot gases from the previous combustion cycle, can help to promote the ignition of the mixture (especially at low load conditions) by increasing the in-cylinder temperature during the compression stroke. Although those two parameters may also have drawbacks, mainly they promote soot formation by increasing the global richness of the mixture. Also, if the EGR rate is easy to finely set as a basic external parameter, the IGR rate can be much more difficult to manage, as it is mainly controlled by the scavenging performance of the engine.

On 4-Stroke engines, various valve-control strategies (Atkinson or Miller cycles for example), coupled to a good management of the intake and exhaust pressures can be implemented to control the amount of gas exiting and/or air entering the cylinder, thus allowing some control over the IGR rate, but the range of this parameter is usually quite reduced, and the separation of the intake from the exhaust (with only a small overlap in some cases) generally induce good scavenging performance and low residual rates. This is particularly a drawback for the gasoline PPC operation at low loads, since the in-cylinder temperature is then too low to properly ignite the mixture.

On the other side, the IGR rate of 2-Stroke engines is largely controlled by the engine architecture and its scavenging performance, which are usually known to be quite poor as it will be detailed in the next section. The large overlap between exhaust and intake is often favorable to high levels of short-circuits and bad scavenging efficiency, which generates high levels of residuals. This particularity, combined to an naturally hotter cylinder charge due to the lack of the empty cycle characteristic of the 4-Stroke architecture, can be a substantial advantage to promote the ignition of the mixture at low loads when operating with the gasoline PPC concept, but can become a critical drawback when increasing the load, for two different reasons:

- a high IGR rate implies less effective volume for admitting fresh air into the cylinder, thus quickly limiting the maximum attainable load before reaching excessive levels of richness;

- higher loads induce higher residuals temperatures, facilitating even more the onset of the combustion until reaching critical ignition conditions (too early, or excessive knock), thus requiring additional external EGR to reduced the gas temperature and then aggravating the first issue of limited effective volume (even more volume occupied by both internally and externally recirculated gases).

Therefore, the improvement of the scavenging performance of the 2-Stroke engine architecture can be one the key for the successful implementation of the gasoline PPC concept, as it could solve the main drawbacks and answer most of the issues for controlling this specific combustion concept on a wide operating range, by achieving the air management requirements necessary to complement the injection strategies that seem to provide very promising results when tested (both theoretically and experimentally) at suitable operating conditions.

Achieving a successful gasoline PPC combustion process can be of large importance considering the performance this specific strategy is able to provide, in terms of fuel efficiency but also in terms of pollutant emissions. Indeed, considering the current global trends towards very strict emission regulations, this objective is now more than ever a critical parameter to take into account when evaluating the potential of an innovative combustion process, and more generally of a new engine design as a whole. And the gasoline PPC concept has demonstrated a very promising potential through several studies, performed by numerous research teams and being implemented in different engine architectures for various applications.

The most important contribution on this specific topic has been brought for the last decade by Lund University (Sweden) in collaboration with various other authors depending on the topic treated, with jointed theoretical and experimental investigations of the combustion process and outputs. Their multiple studies of the effect of the gasoline PPC concept over the NO_x/soot trade-off [36, 37] confirm the results obtained by the author of this PhD thesis and the team around him [38, 39], proving that this combustion strategy can help to largely reduce –and in some conditions even break– this well known trade-off that has been one of the major issues of conventional Diesel combustions for many years. Their contribution regarding the impact of the gasoline PPC concept on HC and CO emissions also demonstrate an interesting aspect of this combustion and the importance of the injection strategy selected to achieve it [40].

One additional study is worth mentioning to complete the investigations of this combustion concept and better understand the thermodynamic occurring inside the cylinder. A detailed description of the phenomenon taking place in the engine is obtained using a complete optical setup coupled to multiple optical diagnostic tools. All the characteristic parameters usually used to describe the combustion process are identified and described: onset of combustion, flame front propagation, different combustion phases, and formation of the different chemical species [41].

Complementary studies have also been performed lately to try and combine the advantages of the gasoline PPC concept with the HCCI strategy, in order to benefit from the best of their outputs depending on the engine operating conditions [42, 43]. This kind of investigation is more focused on the transitioning between the two modes than the actual performance of the combustion, but the main objective here is to profit from the best outputs offered by the two combustion concepts, and the outcomes of these studies can be used to drive more commercially oriented objectives, with a possible complete mapping of an engine operation range both in stationary and transient conditions.

All these investigations are completed by other and more punctual researches performed by various investigation centers around the world which all confirm the potential of the gasoline PPC concept (whether it be for light- or heavy-duty applications) or also very similar combustion concepts such as the gasoline Premixed Charge Compression Ignition (PCCI) combustion, the Gasoline Compression Ignition (GCI) combustion...

Around this combustion concept, more in-depth investigations have been performed to evaluate the effect of the fuel on the mixture reactivity, the combustion development and outputs, and all the mixture related parameters. Indeed, the chemical composition and properties of the fuel have a direct impact on the mixture preparation process and therefore on the ignition delay and the possibility to control the separate between the injection events from the combustion onset (ignition dwell). The fuel properties (mainly characterized by its octane and cetane levels) also condition the dilution required for the mixture to control its global reactivity and its autoignition timing.

The work of Kalghatgi et al. performed on a heavy-duty CI engine [30, 44] introduced the first interest of the gasoline-like fuels for PPC applications, and demonstrated the potential of the combination of this type of combustion concept with higher octane fuels for improving performance and reduce pollutant emissions. Following this work, various researchers have investigated a wide range of fuel compositions, from the effect of low-octane gasoline on soot formation and emissions [45, 46] or on combustion development under various engine operating conditions, either in HCCI mode or PPC mode [47, 48], to high-octane or specific gasoline types (with an overview towards possible renewable fuels) under a wide range of operating conditions [49]. The use of alcohol as fuel for PPC combustion has also proven some interest to reach very low particulate emissions [50]. This alternate solution shows an additional path to break the NO_x/soot trade-off in a more radical way by allowing the increase of EGR levels without soot penalty and therefore reduce drastically the NO_x emissions simultaneously.

From the previous discussion, the potential of the gasoline PPC concept appears to be very interesting, partly in terms of fuel efficiency and performance, with still lower but yet competitive levels compared to CDC, but most of all in terms of pollutant emissions, with very low levels of both NO_x and particulate. But the control of the autoignition delay, the combustion stability, or the knocking tendencies, can become problematic when this combustion concept is applied to a wide range of engine speeds and loads. Several solutions are available to manage these issues, and the residuals management is one of the most efficient. But if the EGR levels are easily controllable, internal residuals require a lot more care, and depending on the conditions the valve management strategies may not be sufficient to fulfill the requirement for proper combustion. In this framework, the solution presented with the 2-Stroke engine architecture, if properly managed, might be a key to solve a significant amount of the issues observed when operating with a 4-Stroke engine.

Indeed, engines operating with a 2-Stroke cycle can present important advantages that may help extending the application range of some combustion concepts and exploring new optimization paths for engine outputs. One of the most important aspect of the 2-Stroke engine is its power density, allowing a similar effective torque with only half the IMEP compared to an equivalent 4-Stroke engine, thus relieving the engine from excessive thermomechanical stress and avoiding (or at least limiting) the detonation risks at high load operations. The 2-Stroke configuration also allows a higher flexibility for the control of the thermodynamical conditions inside the cylinder due to the high range of IGR rates, which helps the combustion initiation and development of high octane fuels at low load conditions by providing higher internal temperatures (which can be managed when increasing the engine load by adjusting the scavenge to reduce IGR and/or with the introduction of external cooled EGR). This previous discussion justifies the interest of evaluating the main characteristics of the 2-Stroke engine architectures which are detailed in the next section 2.3.

2.3 Two-Stroke engine architectures and scavenging processes

When designing an internal combustion engine, the first starting point to consider consists of the selection of its global architecture and cycling. Every engines operate following the same four phases, but they can diverge either on the development of those phases (durations relative to the complete engine cycle...), or on their repartition (timings of the different phases, possible overlap between them...). Those four phases are widely known as the following:

- The intake of fresh air or fresh air-fuel mixture (depending on the fuel supplying technology) into the combustion chamber. It is defined as the time during which the combustion chamber is connected to the intake manifold through open valves or ports. The air flow from the outside to the inside of the combustion chamber is generated either by a mechanical aspiration from the motion of one side of the combustion chamber (usually a moving piston, but other designs can be considered such as a rotor, etc.), or by a compression upstream the intake line using any charging technology (turbocharger, supercharger...), or both.

- The compression of the fresh air/mixture occurs inside the combustion chamber and starts as soon as the intake valves or ports are closed, defining a constant trapped mass that will generate, combined with the fuel already mixed within or injected during or at the end of this phase, the combustion and the power development. The compression of the gases inside the chamber is induced by the mobile parts of the engine (piston, rotor...) reducing the volume of the chamber down to a minimum. The combustion usually starts around the end of this phase.

- The expansion phase (also called power stroke) starts when the volume of the chamber begins to increase back. It is accompanied by the combustion inducing an increase of pressure and temperature, thus creating the power of the engine by transferring this thermodynamic energy to mechanical and motoring work. This phase lasts as long as possible during the volume increment of the chamber, until the opening of the exhaust valves or ports.

- The exhaust follows the expansion and starts when the valves or ports open, allowing the burnt gases to rush out of the chamber using the remaining overpressure compared to the exhaust manifold, plus the motion of the moving part of the chamber that will help push out the rest of the gases, before closing the exhaust valves/ports and opening again the intake ones.

As mentioned earlier, those four phases are common to every internal combustion engine, but they can be achieved in various ways. The vast majority of the terrestrial vehicles are equipped with a reciprocating 4-Stroke engine, in which a piston travels back and forth inside a cylinder, and each one of its strokes corresponds to one phase of the cycle, thus operating with one combustion every two engine revolutions. The scavenging process is almost exclusively performed using valves for both intake and exhaust. This engine architecture is very well known and can operate under any conventional injection-combustion processes (spark or compression ignition, direct or indirect injection...) and as been used for commercial applications for decades, as it evolved accordingly to the technologies and regulations.

However, some different approaches have been explored, for specific applications at first but also for investigation purposes, successfully or not. To illustrate this variety of architectures, it is worth to mention the rotary engine (also known as *Wankel* engine), which can be compared to a valveless 4-Stroke engine due to the motoring cycle positioning in regard of the chamber volume evolution. But this engine, instead of using a reciprocating piston inside a cylinder to perform the cycle, uses the continuous motion of an eccentric *Reuleaux* triangle-shaped rotor rotating inside an oval epitrochoid housing, each face of the rotor being a individual combustion chamber. It then operates with three combustion events for every rotor revolution (which gears with the crankshaft with a ratio of 3, thus one combustion per crankshaft revolution). The scavenging process is performed by ports located on the side of the housing, constantly open and switching for one combustion chamber to the next as the rotor rotates. This engine architecture is then drastically different from the first one presented, yet operates using similar thermodynamic processes to provide different performances. It is currently of heavy interest for range-extender application due to its high power density (very compact package compared to similar output-power from more conventional engines), where the combustion of hydrogen-enriched fuel proves to provide good pollutant emission levels with valuable fuel economy [51–53].

There are other engine architectures that are regaining interest in the last few years/decades, among them is the 2-Stroke cycle and the various designs in which it can be implemented. This section aims at describing the history of 2-Stroke engines applications as a first part, then the focus will be directed towards the renewed interest for this engine configuration in the recent years and the potential it may bring for some applications, depending on different architectural approaches.

2.3.1 Brief history of the 2-Stroke cycle

Historically, 2-Stroke engines were mostly used for very specific applications: either very small SI engines (chain saws, lawnmowers, small-engined motorcycles...) where the compactness of a cam-less engine, with both intake and exhaust performed through ports on opposite sides at the bottom of the cylinder wall, is a way to reduce packaging and weight; or in high performance, recreational vehicles (snowmobiles, off-road and racing motorbikes...) for the high specific power provided by such an engine with twice combustion processes per revolution compared to a more conventional 4-Stroke engine (although regardless of the fuel consumption) [54–57]; or also in very large heavy-duty CI engines (cargo boats) where the long-stroke Uniflow architecture operating with diesel fuel is used for its high efficiency (better volume-to-surface ratio inside the cylinder, thus less thermal losses during the combustion process) [58–60].

However, since before the 1950s, the major issue with the 2-Stroke engine configuration has been the scavenging process. Indeed, regardless of the specific architecture of the engine, the 2-Stroke cycle requires a high level of intake / exhaust overlap, since both these processes need to occur when the piston is near the BDC. The most common geometry has for a long time used port scavenging, whether they are cross flow (intake and exhaust ports on opposite sides of the cylinder wall) or loop flow (intake and exhaust ports on the same sides of the cylinder wall) engines. These configurations generate large amounts of intake short-circuit to the exhaust, thus considerable loss of unburnt fuel since the commercial engines were almost exclusively carburated (fuel mixed to the intake air prior to entering the cylinder). A large amount of literature report studies performed to investigate the scavenging process inside various engine architectures, for both SI and CI engines, and also in some cases the impact of scavenging performance on pollutant emissions [61–65].

Examples of exhaust acoustic management can be found from the 1950s, where Wallace et al. [66] investigated both theoretical and experimental ways to understand and improve the scavenging process in a CI opposed-piston engine. Although this architecture is very advantageous on several aspects (mostly by physically separating the intake and exhaust ports), the balance between scavenging efficiency and short-circuit has always been a struggle. The Uniflow architecture has always demonstrated similar advantages, and is largely used on cargo boats and other (very) heavy-duty applications, but was often discarded from smaller vehicles due to a larger packaging: despite operating under a 2-Stroke cycle and using intake ports, the exhaust is performed through valves in the cylinder head, thus increasing the total height of the engine.

Another major issue encountered throughout the years with 2-Stroke engines is the lubrication. Again, various articles report studies on that topic throughout the years [67–69]. In a conventional 4-Stroke engine, oil is bathing the crankcase and is projected or injected under the piston and onto the cylinder wall. However, when operating with ports, the crankcase is usually used as a transfer case for the intake air-fuel mixture before entering the cylinder. Lubrication is then often added directly to the fuel, but this also mean that not only part of this oil is short-circuit with the mixture, the part trapped inside the chamber is burnt with the fuel, generating a large amount of deposit and particulates, and even damaging engine components (spark plug electrodes in the case of SI engines, etc.).

Despite the main drawbacks depicted in the previous references, 2-Stroke engines have still been used massively in all the applications mentioned at the beginning of this section, thanks to their compact package, their high power density, and their high thermal efficiency in the case of very large CI engines. And these specific benefits of this type of engine have sparked a renewed interest for investigation purposes during the last decade or so as paths allowing the development of new combustion concepts, considering the different mechanical and thermodynamic aspects it offers compared to 4-Stroke engines.

2.3.2 New interests for the 2-Stroke cycle and most promising engine architectures

In the mid-90's, controlled autoignition combustion was already investigated by Ishibashi et al. [70, 71] in order to improve both fuel consumption and engine-out pollutant emissions of 2-Stroke engines. Their investigations were carried out using a loop-scavenged layout, with intake and exhaust ports both at the bottom of the cylinder with orientations inducing a loop path of the gases. In this case, the engine was still equipped with indirect fuel injection to keep the low cost advantage of the 2-Stroke architecture. But direct injection has also quickly been used to avoid fuel short circuit and reduce fuel consumption, and to provide some control over the mixing process and the combustion, considering that fuel efficiency and combustion stability/fluctuation were always the main drawbacks of the SI 2-Stroke engines.

However, the potential of this specific architecture is very limited in terms of scavenging performance. In that framework, some of the most noticeable and promising solutions have appeared through investigations widely performed using either poppet-valve configuration, Uniflow architecture, or opposed-piston design.

The first solution has the main advantage of being based on a conventional 4-Stroke layout, with intake and exhaust valves located in the cylinder head. This allows simple conversions from existing engine blocks, thus providing cheap solutions for potential large scale manufacturing based on existing production lines. A second advantage to this solution is the large knowledge of this architecture, including the control of the air flow through intake and exhaust lines, available technologies for turbo- and super-charging, possibilities for variable valve timings, and easy transfer of all the usual technological solutions commonly used on 4-Stroke engines. Conversion to 2-Stroke cycling is then very simple, and the only major drawback is the scavenging performance, with intake and exhaust valves located next to each other thus creating a potential for large amount of short-circuit or back-flow. This issue has been widely addressed by Tribotté et al. during the project preceding the work of this thesis [72], resulting in the issue of a patented cylinder head shape to orientate the intake air flow away from the exhaust valves (Patent Renault FR2931880 [73]).

A very similar architecture has also been heavily investigated by Fu and Zhao in a SI engine operating with CAI combustion [74–78]. The main objectives were to study and optimize the gasoline CAI combustion concept and widen its operating range as much as possible, very similarly to the work performed with the gasoline PPC combustion concept by the author of this thesis and described in Chapter 4.1. During these investigations, they used a poppet-valve engine equipped with direct injection and spark ignition assistance for portions of the engine map where CAI combustion cannot be implemented properly. Zhao and Wang, during parallel or later studies, have also investigated the scavenging potentials of two other 2-Stroke architectures: the Uniflow [79–82] and opposed-pistons [83, 84] configurations.

But the renewed interest for those engine architectures is not limited to a few teams across the world. More investigations are carried out to implemented new solutions and develop new combustion concepts: Mattarelli et al., from the University of Modena, have presented studies about the opposed-piston architecture used to combine good scavenging performance with the latest technologies of combustions systems developed through CFD modeling. Moreover, their work is admittedly based on heavy research and communications from Achates, who developed a fully viable opposed-piston engine that has been implemented on vehicle for further testings (driving cycles, etc...), operating with GCI combustion [85–90].

Among the various solutions available, the opposed-piston configuration appears to be the most appealing of them, mainly due to its inherent advantages:

- Its high scavenging performance: the intake and exhaust ports are located at each side of the combustion chamber, allowing a very low amount of short-circuit along with a high scavenging efficiency, when the internal air flow is managed properly.
- Its high thermal efficiency: two pistons sharing one combustion chamber means the elimination of the cylinder head, thus reducing the chamber wall area and consequently the heat transfers.

The main drawback to this configuration is the packaging, which is by design very large in relation to the engine displacement, compared to a conventional 4-Stroke engine or to a 2-Stroke poppet-valve engine. So the common alternative is often the Uniflow architecture, which present a similar first advantage (good scavenging performance), and can also provide good thermal efficiency in the case of long-stroke engines. These point were the main motivations and guidelines for the design of the engine presented in the Chapter 5.1 of the thesis.

Alternatively, some companies also develop completely new architectures, based on the 2-Stroke cycle, in order to further optimize either the scavenging performance, or the global efficiency of the engine. One of the most serious and radical solution of the last years has been the Z-engine, developed by Aumet [91], and which consists of reducing drastically the effective intake duration (thus avoiding the overlapping between exhaust and intake and allowing to delay the exhaust event itself to extend the effective expansion duration) by pre-compressing the intake air using a primary compressor-piston. This allows the 2-Stroke cycle operation while still separating the exhaust and intake events, by decomposing the four phases described in the introduction of this section the same way a 4-Stroke engine operates. This architecture, while requiring the addition of a compressor-piston, then combines the advantages of both 2-Stroke and 4-Stroke engines, allowing the specific power of the first (with one combustion per crank revolution) as well as the scavenging decomposition and performance of the second.

Since the first design of the engine, they have now started also to implement advanced combustion concepts such as HCCI or PCC, since this engine configuration allows good in-cylinder conditions to operate those type of combustion in a wider range compared to a 4-Stroke engine, while keeping an advantage over the 2-Stroke engine in the scavenging performance [92, 93]. More recently, they demonstrated the further potential of this engine by reaching high loads under HCCI operation, which is quite a promising step considering the difficulties usually encountered with the HCCI concept implemented on either 2-Stroke or 4-Stroke engines [94].

2.4 Summary and concluding remarks

This chapter describes the main most relevant phenomena related to the injection and combustion processes of a DI CI engine operating with advanced LTC concepts in high (HCCI, HPC) or moderately (PPCI, PPC) premixing conditions. It also aims to corroborate the interest of the 2-Stroke engine architecture for such combustion concepts through an overview of the advantages, drawbacks and uses of such design. An extensive review of numerous experimental and numerical investigations performed in various research center around the world brings an representative insight of the challenges required in terms of scavenging management and mixture preparation to allow satisfying autoignition timings and combustion processes, and also leading to understanding the formation of the main pollutants emissions (NO_x, PM, CO and HC) and paths to reduce them.

The detailed analysis of the literature about advanced premixed LTC concepts with application to CI engines discussed in section 2.2 confirms the simultaneous reduction of $NO_x$and PMemissions when the fuel is sufficiently mixed with the air before the onset of combustion, decreasing the local equivalence ratios and temperatures during the combustion process below these two pollutant emissions formation triggering levels. However, the operating region of the premixed LTC concepts is very restricted due to the lack of thermodynamic conditions at low loads to ignite the mixture (too cold for the autoignition of low cetane fuels), but also too abrupt heat release profiles at high loads, leading to highly knocking combustion and consequently high NO_x levels. Additionally, significant levels of CO and HC emissions arise mainly related to largely over-mixed mixtures failing to reach complete combustion and/or liquid fuel impinging onto the combustion chamber delimiting surfaces.

In order to solve some of the issues identified when operating with HCCI, investigations started to focus on partially premixed combustion concepts (PPCI and PPC) obtained by delaying the injection timings in order to switch from fully homogeneous to sufficiently stratified mixture conditions prior to combustion, still limiting soot formation by separating the injection and combustion events to avoid the presence of over-rich areas near the flame front and introducing moderate-to-high EGR rates to decrease local temperatures during combustion below 2200 K. Various solutions have been implemented to promote the ignition of low cetane fuels, like combining high injection pressures with also high swirl levels to enhance the fuel/air mixing process, integrating multi-injection events, and most of all the use of Internal Gas Residuals (IGR) to keep cylinder temperature after compression high enough to force the ignition. Compared to the HCCI concept, the partially premixed combustion concepts recover the link between the injection and combustion events up to a given extent since the stratification of the fuel/air mixture, which controls the combustion characteristics, is affected by the injection profile. However, the suitable combination of injection and air management settings (including EGR rates) required for keeping both a stable combustion –without misfire or knock– and simultaneous reduction of NO_x and PM is usually very limited, especially at low loads when cylinder temperature is barely high enough to ensure the ignition.

However, operating with the PPC concept using gasoline-like fuels provides better emissions and engine indicated efficiencies than those obtained using conventional diesel fuel no matter the combustion concept. Nonetheless, the load range for PPC operation strongly depends on the fuel octane number as different load conditions demand distinct fuel ignition properties. This suggests that the fuel reactivity should be adjusted to each operating condition, which is of course not possible with a single fuel. Thus, adjusting reactivity of the mixture to extend the engine load range is only (partially) achievable through the integration of an advanced variable valve actuation system, with its associated extra-cost, in addition to the accurate control of the boosting levels and EGR rates.

In this context, some 2-Stroke architectures can control the IGR rates in an extremely high range by adjusting the air management related settings, and significantly affect the cylinder gas composition and temperature evolution, therefore significantly extending the PPC operation range compared to the equivalent 4-Stroke engines. This intrinsic potential of 2-Stroke operation for adjusting the thermochemical conditions of the mixture and thus the combustion environment, is critical to control the combustion development and outputs (indicated efficiencies and final emission levels), especially when operating with advanced combustion concept such as the gasoline PPC. An additional aspect that justifies the renewed interest gained by 2-Stroke operation is its characteristic to provide equivalent Brake Mean Effective Pressure (BMEP) or torque response with just half the Indicated Mean Effective Pressure (IMEP) required in 4-Stroke operation for the same engine displacement. This drastically helps to solve the issues observed when operating with the HCCI and PPC concepts at high loads, reaching the target engine performance in terms of torque and brake power but limiting the IMEP to levels under 15 bar IMEP (medium loads), taking advantage of the previously discussed characteristics of the 2-Stroke operation.

Despite these significant advantages of the 2-Stroke architecture for the implementation of advanced combustion concepts, the scavenging management of this type of engine has always presented critical issues. This may be critical for engine brake efficiency, when taking into account the usually necessary over-scavenge of the cylinder when increasing the load, in order to exhaust as much residuals as possible and let enough fresh air into the cylinder to satisfy the charge requirements. This specific issue will reveal to be critical in the Chapter 4 of this thesis, and will be the main justification for the work describe in Chapter 5.

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Chapter 3

Experimental and theoretical tools

Contents

3.1 Introduction

Depending on the objectives defined for the present investigations and the application fields towards which they are aimed, the experimental research on combustion engines can be approached in different ways. The work described in this thesis is basically grounded on the theoretical and experimental results obtained from two different (yet comparable, and with a common purpose) single-cylinder research engines coupled with various diagnostic and modeling tools used to control the inputs to the engines, and analyze their performance and emission outputs. The decision of using these single-cylinder research engines instead of the corresponding multi-cylinder versions is supported by the need of generating fundamental knowledge about the combustion process and its sensitivity to the most relevant air management and injection settings, decoupling these two systems and finally eluding the uncertainties associated to the crossed effects. It is, in almost every cases, a necessary step in the investigation process before moving on to more specific studies of the combustion using more advanced techniques, as for example optical diagnostics of local in-cylinder processes.

Aiming to improve fuel efficiency and reduce pollutant emission levels for modern IC engines through experimental engine testing and optimization campaigns typically requires a large amount of experiments changing sequentially different input parameters, which is extremely time consuming and therefore expensive. Thus, a proper definition of the experimental methodology associated with a correct selection and application of different post-processing and diagnostic tools will help to reduce the time spent for engine testing, and will also improve the quality of the information that can be extracted.

The key to improve the quality of the engine investigation and research strongly relies on a better comprehension of the processes involved inside the engine combustion chamber. Thanks to the high efficiency and reliability of CFD modeling, combined with the constantly increasing computational power, engine simulation is nowadays extensively used for engine outputs (performance and pollutant emissions) predictions and to provide optimization guidelines, either for new strategies exploration or implementation, or for engine design. In addition, a growing interest can be observed recently for advanced combustion concepts. The complex physical and chemical processes (as described in Chapter 2) involved with those concepts require advanced engine simulation tools (helped by improved mathematical models and faster computational and processing tools) to understand and control these processes and to propose sustainable solutions to answer the evolving problematics.

3.2 2-Stroke single-cylinder research engines

The first part of the work presented in this study was designed after the POWERFUL European Project. The philosophy of this project was to design a new 2-Stroke engine prototype, based on a 1460 cm^3 four-cylinder 4-Stroke Renault Diesel engine that downsized into a 730 cm^3 two-cylinder engine operating with a 2-Stroke cycle, with equivalent torque performance as the base configuration, but operating with gasoline to implement and develop a fairly new combustion concept: Partially Premixed Combustion (PPC). Considering a downsized (heavily loaded) CI engine, the poppet-valve scavenge type architecture, which presents a mechanically more robust solution, was the first direction selected for the project. It was coupled with the latest technologies of HSDI systems and up-to-date innovative boosting systems (with a combination of turbo- and supercharger) to supposedly accomodate for the scavenging and trapping difficulties inherent to a poppet-valve scavenge type configuration, as described in Section 2.3. From this point forward, this engine will be referred to as *poppet-valve engine*.

The second part of this thesis will be focus on a Uniflow engine designed in the framework of the REWARD European Project. The global objectives of this project was, consequently to the results obtained during the POWERFUL and Post-POWERFUL projects, to design a new 2-stroke engine architecture to improve greatly the scavenging of the cylinder, which would be proven to be one of the main limiting factors for reaching higher loads using a poppet-valve engine configuration (those results will be described in great detail in Chapter 4). The biggest part of this project and of the work described here, is then dedicated to the full design of a new engine, from the architecture selection to the scavenge and combustion optimization through theoretical means before manufacturing said engine for an experimental validation campaign. In order to differentiate from the poppet-valve engine introduced previously, from this point forward this new engine will be referred to as *Uniflow engine*.

3.2.1 Main engine geometric characteristics

The experimental investigation presented in this document were performed on single-cylinder versions of these two innovative 2-Stroke CI engines. The poppet-valve engine was specifically adapted (from the 4-Stroke version mentioned previously) and manufactured by Danielson Engineering to operate in 2-Stroke mode, while the Uniflow engine was designed from scratch by the author of this thesis and the project teams around him, before being also manufactured by Danielson Engineering. They are both equipped with a standard common rail HSDI system and a cam-driven Variable Valve Timing system which controls both the intake and exhaust timings in the case of the poppet-valve engine, only the exhaust timing in the case of the Uniflow engine. These two single-cylinder engines present the same layout and internal design as their corresponding two- or three-cylinder versions that are planted as project global objectives, but as mentioned previously the single-cylinder option allows an unaltered study of the fundamental physical phenomena taking place in the engine, related to both the air air-loop parameters and results as well as the combustion processes and outputs. Avoiding any interference or coupling with the other cylinders and the auxiliary systems helps in creating a more controlled and stabilized environment.

However, it is worth mentioning that if the poppet-valve engine design is well defined and fixed, it is not the case for the Uniflow engine, since the bigger part of the related project consists of the evaluation of various architectures, geometries and technical solutions for the engine and auxiliary design, and the whole engine geometry and characteristics will be presented as investigation results in the dedicated Chapter 5. But as a reference, some of the main final dimensions can be summarized and put in parallel with the poppet-valve engine in Table 3.1, which contains the main geometrical characteristics of both these single-cylinder 2-Stroke engines.

Two different pistons, one steel and one aluminum, with same geometric definition (compression ratio and bowl geometry) were available for the research activities on the poppet-valve engine. The two pistons were tested along the investigations to evaluate the impact of the material on the heat transfers, with experiments combined to theoretical combustion diagnosis tools.

3.2.2 Scavenge architecture and valvetrain system

The two engine general layouts are relatively similar regarding the block construction and auxiliary systems, but their internal geometries are drastically

Table 3.1. *Experimental engines main specifications.*

	Poppet-valve engine spec.	***Uniflow engine spec.***
Engine type	Single-cylinder 2-Stroke CI	Single-cylinder 2-Stroke CI
Displacement	365 cm^3	500 cm^3
Bore x Stroke	76 mm x 80.5 mm	76 mm x 110 mm
Connecting Rod Length	143.05 mm (steel piston) 133.75 mm (alum. piston)	208 mm (alum. piston)
Compression Ratio	17.8	17.58
Type of scavenge	Poppet-valve Scavenge Loop	Standard Uniflow Scavenge Loop (int. ports & exh. valves)
Number of Valves / Ports	2 int. valves + 2 exh. valves	Several geometries tested
Valvetrain	DOHC with VVA	DOHC with VVA

different. Thus, the description of their combustion chamber designs and the resulting scavenging paths can be described as follows:

- The poppet-valve engine, used in previous investigations and presented in much details in a previous doctoral thesis, was designed entirely during this past campaign. A detailed description of its design and construction can be found in Daniela DeLima's PhD thesis [1], and a synthetized version can be extracted for a brief presentation:

  "The combustion chamber of the first engine has four poppet valves with double-overhead camshafts, and the cylinder head has been specifically optimized and redesigned by Renault in the framework of the POWERFUL Project, to ensure a suitable in-cylinder flow pattern during 2-Stroke operation, in order to optimize the scavenging of burnt gases and to reduce the short-circuit losses of fresh air going directly from the intake into the exhaust [2, 3].

  The first definition of the engine architecture, boost system requirements, combustion chamber geometry and scavenging characteristics of this newly designed 2-Stroke engine have been reported by Tribotte et al. [2]. A preliminary work consisted in an iterative approach combining 1D and 3D multidimensional CFD simulations to evaluate and study the air charging and scavenging process during the cycle for different cylinder head designs using preliminary cam lift laws. Three different geometries were analyzed: a flat-roof cylinder head with a modified intake duct, an intrusive mask between intake and exhaust valves and a staggered roof.

The final geometry of the combustion chamber is based on the design patented by Obernesser et al. [3], and it presents a staggered roof for baffling the flow of air between the intake and exhaust valves, as can be seen in Figure 3.1. In that way, the fresh air flow is forced to follow the path of the cylinder wall towards the bottom of the combustion chamber, creating a tumble aerodynamics (instead of swirl) that improves the scavenging of the burnt gases while keeping short-circuit losses as low as possible during the valve overlap phase. This geometry provided the best compromise between scavenging efficiency, acceptable permeability, and convenient combustion chamber geometry."

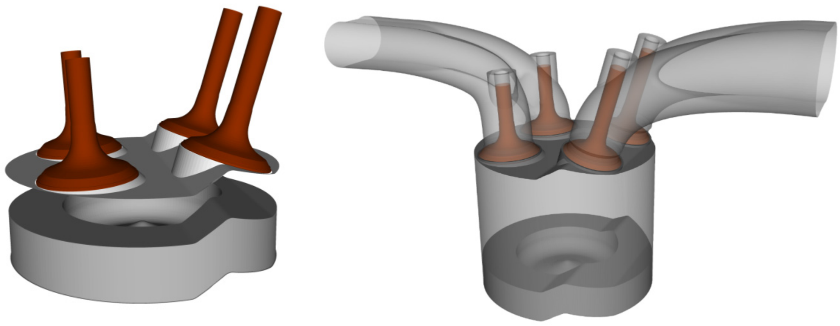

Figure 3.1. *Final cylinder head design of the poppet-valve engine. Source: Tribotte et al. [2].*

- Different scavenging solutions have been investigated during the second part of the research work, based on the Uniflow architecture: Standard Uniflow, with fixed intake ports placed at the bottom of the cylinder and exhaust valves in the cylinder head; Reverse Uniflow, with the same geometry but inverted flow, i.e. the intake enters the cylinder through the valves and the burnt gases are exhausted through bottom ports. Conventional stroke-to-bore ratio was also confronted to extended-stroke configuration and evaluated in terms of both scavenging and combustion efficiencies. All these investigations were performed collaboratively by all the project partners: Renault SA as a project leader, with the CMT - Motores Térmicos (Valencia, Spain), the IFP Énergies Nouvelles (Paris, France), and the Czech Technologic University (Prague, Czech Republic).

  The various iterations performed on those different initial steps of the project are detailed in Chapter 5, which presents the main results obtained through both 0D and 3D-CFD calculations. However to present a brief description of the engine for the rest of the investigations, it can be said here that the final definition lays on a standard Uniflow architecture associated with a long-stroke configuration as base geometry, providing a four exhaust valve engine with a stroke-to-bore ratio of 1.45. A typical diesel-like combustion chamber was also selected by default, with a piston bowl to be optimized by the means of CFD combustion tools and experimental validation depending of the final injector nozzle geometry (however, due to experimental issues generating a delay in the project planning, that last sequence of optimization could not be performed as expected).

Within these same frameworks, the optimum intake and exhaust camshaft lift laws were also determined for each engines, by testing different alternatives either numerically or experimentally in the single-cylinder engines, at representative load/speed conditions.

In the case of the poppet-valve engine, a total of 4 intake camshafts and 3 exhaust camshafts profiles to evaluate various opening durations and the maximum lifts through a Design of Experiment (DoE) methodology, and compare the air management characteristics as well as engine performance for every possible combination. The charging and scavenging processes of the 2-Stroke engines (mainly Trapping Ratio (TR) and Internal Gas Residuals (IGR) ratio) were characterized through detailed analysis of the most common air management parameters, as well as Indicated Specific Fuel Consumption (ISFC). An ideal scenario in terms of charging/scavenging correlation would be to decrease IGR rates by improves the scavenging of burnt gases without punishing the charging and trapping efficiencies. However, experimental results have demonstrated a trade-off between these two parameters, with the lowest IGR obtained for the best configuration is around 25 %, and with a very narrow range of variation (from 35 % to 25 %), trending towards a limit where the IGR cannot be decreased furthermore. In the case of the Uniflow engine, the exhaust valve lift profile was also drafted with the objective of providing maximum engine permeability. Later in the scavenging optimization process, two sets of intake port geometries, with different heights for different intake opening and closing events, will be selected for manufacturing and experimental testing, so a second exhaust valve lift profile will be designed to provide a different fitting with the intake timings.

A hydraulic cam-driven Variable Valve Timing system is mounted on both engines, allowing a working flexibility (independent from the mechanical cam phasing) of 30 degrees for both intake and exhaust timings. However, the intake events on the Uniflow engine must obviously be fixed and centered around the Bottom Dead Center (BDC), so one degree of parametrization is lost compared to the poppet-valve engine. The exhaust valves however will still benefit of the same 30 CAD range, thus allowing a significant flexibility to the scavenging design and the engine permeability. Figure 3.2 displays the flexibility of the VVT system (nominal phasing at 0°, intermediate at 15° and maximal angle at 30°) for the optimized intake and exhaust cam lift laws of the poppet-valve engine (left plot) and exhaust cam lift law for the Uniflow engine (right plot), alongside one intake port profile. The phasing of the exhaust and intake events, additionally to affecting the scavenging process through their individuals timings and the valve overlap, also modifies the effective compression and expansion ratio. The engine performance is therefore directly impacted by these parameters.

After the DoE campaign (for the poppet-valve engine) described in detail in Daniela DeLima's PhD thesis [1] and the numerical evaluation (for the Uniflow engine), the optimum and final definitions of camshaft configurations were selected, and the main characteristic values (maximum lift, opening duration, nominal timing for the intake and exhaust) are depicted in Table 3.2.

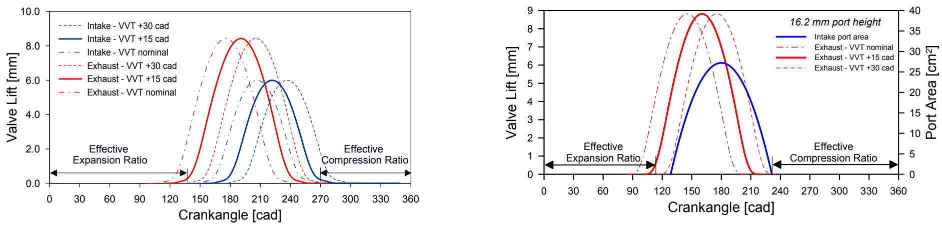

Figure 3.2. *Intake and exhaust valves lift laws on the poppet-valve engine (left), intake port area and exhaust lift law on the Uniflow engine (right).*

3.2.3 Fuel injection system

The injection system is a protoype common-rail DFI 1.5 (from Delphi) and allows any combination that includes up to two pilot injections, one main injection and two post-injections. The rail pressure is limited to 1800 bar for diesel fuel, but for the gasoline PPC investigations, the injection system was limited to 1000 bar to avoid cavitation issues inside the injector return line. Table 3.3 contains the main characteristics of this system, used with gasoline for the poppet-valve engine and with diesel fuel for the Uniflow engine.

Table 3.2. *Final intake and exhaust timings definition.*

	Poppet-valve engine	***Uniflow engine***
Intake camshaft / ports	Max Lift = 6 mm	Max height 1 = 16.2 mm
		Max height 2 = 10 mm
	Opening dur. = 80 CAD	Opening dur. 1 = 102.8 CAD
		Opening dur. 2 = 80.6 CAD
Exhaust camshaft	Max Lift = 8.5 mm	Max Lift 1 = 8.82 mm
		Max Lift 2 = 8.4 mm
	Opening dur. = 95 CAD	Opening dur. 1 = 92 CAD
		Opening dur. 2 = 82 CAD
Nominal intake timing	IVO = 161.9 CAD aTDC	IVO1 = 128.6 CAD aTDC
		IVO2 = 139.7 CAD aTDC
	IVC = 251.6 CAD aTDC	IVC1 = 231.4 CAD aTDC
		IVC2 = 220.3 CAD aTDC
Nominal exhaust timing	EVO = 122.6 CAD aTDC	EVO1 = 99 CAD aTDC
		EVO2 = 111.3 CAD aTDC
	EVC = 226.9 CAD aTDC	EVC1 = 191 CAD aTDC
		EVC2 = 193.3 CAD aTDC

The optimum nozzle geometry selected for CDC operation had 8 holes of 90 µm in diameter, with a 155° spray cone angle. When operating with the gasoline PPC concept, the nozzle geometry selected and used as a base case is similar to the optimum previously determined during the investigations with CDC, only with a slightly narrower spray angle (justified by the early injection strategies used with the gasoline PPC concept, as will be detailed in Chapter 4): 8 holes of 90 µm in diameter, with a 148° spray cone angle. It was assumed, for all this campaign, that the results should be considered mostly qualitatively as the injection hardware was not specifically designed to performed with gasoline, and therefore could never be optimal without a dedicated and adapted injection system.

The system used for the Uniflow engine is the exact same one, with the only difference that it is capable of reaching 2200 bar of rail pressure. Moreover, the campaign performed on this engine will use exclusively diesel fuel, so no lower pressure limitation is required for the tests performed during this work. The injector nozzle is also different, as shown in Table 3.3, as it has been defined to match the piston bowl design during a dedicated preliminary CFD optimization phase performed by the project partners. The base-case setup, used for the scavenging validation and optimization, will include a 8 holes injector with a spray cone angle of 155° and a hydraulic flow rate of 620 mL/min. Also, a wider set of nozzles was also planned to be tested during the combustion optimization phase of the project. Unfortunately, and as mentioned previously, this phase had to be drastically shortened due to experimental issues delaying the project planning, so this set of nozzles is provided here purely for informative purposes.

Table 3.3. *Injection system specifications.*

	Poppet-valve engine	***Uniflow engine***
Fuel injection system	Diesel common-rail HSDI	
Injector type	Centrally-mounted Piezoelectric injector	
Maximum injection pressure	1000 bar (with gasoline)	2200 bar (with diesel fuel)
Final nozzle definition	8 holes, 148°, 90 µm	8 holes, 155°, 114 µm
Injector mass flow rate	400 mL/min	620 mL/min
Additional nozzle options		7 holes, 155°, 450 mL/min
		8 holes, 155°, 450 mL/min
		8 holes, 155°, 520 mL/min
		8 holes, 155°, 590 mL/min
		9 holes, 155°, 520 mL/min

3.2.4 Fuel specifications

As mentioned previously, the two engines were each operating with different combustion concepts, the first one using the gasoline PPC concept, while the second one uses the CDC concept. Thus two types of fuels were used: a calibrated unleaded gasoline with 95 Research Octane Number (RON) was selected for the studies focused on the evaluation of the Partially Premixed Combustion (PPC) concept (except for the break-in phases, after a piston change or an engine rebuild for example, where conventional diesel fuel was used), and a conventional diesel fuel was used to operate the Uniflow engine under standard and well-know combustion behaviors, since the main focus of the study was pointed at the scavenging performance instead of the combustion development. However, since the injection system was originally designed to operate with diesel fuels, the lower viscosity of the gasoline requires the addition of a lubricity additive (approximately 30 ppm in volume of the total blend). The most relevant fuel properties are detailed in Table 3.4.

The study performed during the gasoline PPC study was initialized before the beginning of the doctorate program of the author, and thus a number of the preliminary investigations required for the design of the project had been performed and presented in the doctoral thesis of the previous PhD student working on the project. Among those investigations can be found the behavioral

Table 3.4. *Fuel specifications.*

Test fuel	***Unleaded gasoline RON95***	***Conventional diesel fuel***
Density (15 °C)	758.1 kg/m^3	836 kg/m^3
H/C ratio	1.76 mol/mol	2.01 mol/mol
O/C ratio	0 mol/mol	0.01 mol/mol
Oxygen content	$\leqslant$ 0.17 % m/m	$\leqslant$ 0.17 % m/m
Stoichiometric air/fuel ratio	14.37	14.55
Lower heating value LHV	42.82 MJ/kg	42.41 MJ/kg

variations between diesel and gasoline fuel during the injection event, which is critical when operating gasoline fuel in an injection system initially designed for diesel-like fuels. Then, the relevance in comparing the mass flow rate and spray momentum flux of those fuel for example is developed in detail in Daniela DeLima's PhD thesis [1]. The cavitation effect observed in the return line with gasoline when operating at high pressures was also pointed out during this work, highlighting the impact on fuel flow measurement precision and the risk of mechanical failure of the fuel pump.

Those preliminary studies were critical to improve the quality of 1D models (spray mixing calculations), multidimensional CFD models, and also the combustion analysis and diagnostic tools. Thus, the effects and/or limitations observed during this work helped defined some of the boundary conditions of the injection system operation range and the combustion capability of the engine.

3.2.5 Test cell characteristics and equipment

The poppet-valve investigation engine was assembled in a fully instrumented test cell in the engine laboratory at the Departamento de Máquinas y Motores Térmicos (DMMT) of the Universitat Politècnica de València (Spain), where all the experimental studies presented for the PPC investigations were performed. On the other side, the experimental campaign of the Uniflow engine was performed at the Énergies Nouvelles research center in Rueil-Malmaison (France), partners of the project, where the doctorate student joined the local teams in the frame of a research internship in order to follow and manage this phase of the project. Although the locations were different, the equipment in the two facilities was very similar: the control and measurement hardware were strictly the same, and most of the auxiliary systems in charge of the engine conditioning management, unless specified differently in the following sections, can be considered equal in both facilities.

Figure 3.3 shows a simplified scheme of the engine test cell layout, displaying the single cylinder engine along with the main sub-systems around it and the location of the most important sensors and probe measurements. The layout of the Uniflow engine test cell was strictly the same, with only a few differences in the equipment references, and the main differences being the control and data acquisition software, although this doesn't affect the control strategy nor the outputs.

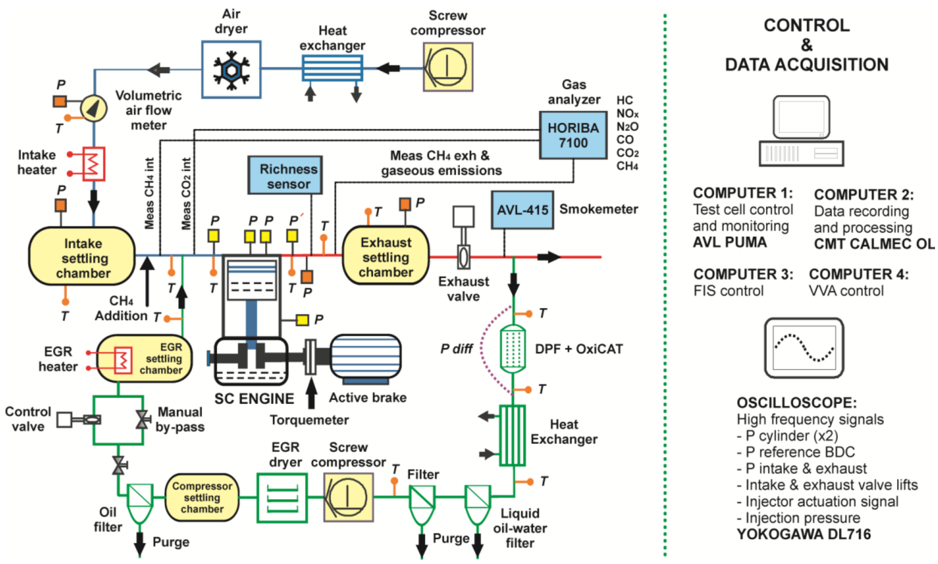

Figure 3.3. *Layout of the poppet-valve engine test cell.*

Since the work carried out during the first part of this project (all the investigations performed on the poppet-valve engine) comes in direct continuity of the previous POWERFUL European project developped by the preceeding PhD student, all the equipement relative to this engine is exactly the same as described in her PhD thesis. Thus, all the information regarding the equipement descriptions and references (engine dynamometer and all the auxiliary systems surrounding the engine), as well as all the sensors and acquisition systems and softwares (gas analysis, data acquisition, along with the post-processing tools used online on the test bench), can be found in much details in said document, and also in all the scientific communication related to this engine (both Diesel and HCCI operations, or gasoline PPC investigations), and published by either Daniela DeLima or the author of this thesis [1, 2, 4–12].

Then, considering that the vast majority of the equipment, acquisition and post-processing tools used during the investigations performed on the Uniflow engine are almost identical, the same referencing can be drawn for this part of the investigation also. The following section will be pointing out the few differences between the two setups and describe the equipment and processes that diverge from the ones presented in the previously mentioned references.

3.2.5.1 Equipment and acquisition systems specific to the Uniflow engine

3.2.5.1.a Air-loop systems for the Uniflow engine (intake, exhaust, EGR routes and equipments)

The intake air supply equipment represents the biggest difference between the two facilities. In the case of the Uniflow engine, the intake air is pressurized, filtered and supplied to the engine test bench by a global air network composed of two external industrial oil-free screw compressor (detailed along with the main characteristic of the air circuit in Table 3.5), providing a maximum pressure of 6.9 bar (absolute), sufficient to sustain the maximum intake pressure requirements for engine operations. Once compressed and filtered, the air is prepared (cooled down and dried) before being sent to a distributor which manages the air flow going to each test cell.

The air mass flow is measured by a multi sonic-nozzles (Venturi) system with various passing sections, which combinations are controlled via a bench supervisor integrated to the bench control software *MORPHÉE*. The air density, used to convert the volumetric flow units into mass flow units (in kg/h), is calculated by integrating the pressure and temperature measurements to the air circuit.

An identical procedure as for the poppet-valve engine ([1]) is applied here for the heating of the fresh intake air, the EGR mixing process, and the measurements of the intake conditions. The mean temperature of the intake mixture (fresh air + EGR) is measured once again in the intake plenum just before entering the engine, along with the instantaneous (absolute) pressure using two uncooled piezoresistive sensors (one on each entrance of the intake plenum) that provide a measurement range from 0 bar to 10 bar. The control of the pressure and temperature is achieved through dedicated PID controllers that oversight and adjust the intake charge conditions.

Table 3.5. *Intake air supply specifications.*

	Poppet-valve engine	***Uniflow engine***
Compressor Type	Oil-free screw air compressor	
Supplier and model	Atlas Copco ZA1-98	Atlas Copco GA90 & GA75
Maximum pressure	4 bar (abs)	6.9 bar (abs)
Flow rate at maximum speed	450 m^3/h	900 m^3/h & 760 m^3/h
Air dryer type	Refrigerant compressed air dryer	
Supplier and model	Atlas Copco FD 380W	-
Air temperature on the inlet	35 °C	-
Filtration	-	0.01 µm
Maximum flow rate	-	2520 m^3/h
Dew point temperature	3 °C	3 °C
Intake settling chamber volume	250 L	250 L
Exhaust settling chamber volume	50 L	50 L

While the intake system is partially managed differently from the poppet-valve engine, the exhaust is very similar: two water-cooled piezoresistive sensors (cappable to resist an exposure to gas temperatures in excess of 1100 °C, and providing a measurement range from 0 bar to 10 bar), are placed as close as possible to the valves to measure the instantaneous (absolute) exhaust pressure. The mean temperature is also measured in the same area. The design of the Uniflow engine requires two exhaust settling volumes (one on each exhaust manifold) to reduce acoustic phenomena, and the mean pressure and temperature are measured once again at these points. After this first part of the exhaust line, an electro-pneumatic valve regulates the back-pressure to the desired value.

A specifically designed method was developed to experimentally measure the trapping ratio in every test point, using methane (CH_4) as an external tracer gas. First, a controlled quantity of CH_4 (around 1000 *ppm*) has to be homogeneously injected in the intake flow prior to the mixing point of fresh air with EGR, when operating at the stabilized engine conditions. Next, the CH_4 concentration has to be measured at the intake and also at the exhaust manifolds with a dedicated CH_4 gas analyzer. The sampling probe in the intake side is located at a sufficient distance downstream the addition point, to assure that the CH_4/air mixture has been completely homogenized, but upstream the EGR mixing point. In the exhaust side, there is another independent sampling probe placed before the exhaust settling chamber. Assuming that all the CH_4 trapped in the cylinder will burn completely during combustion, an accurate estimation of the short-circuited mass from the intake flowing directly

Table 3.6. *Low frequency measurements.*

Variable	Sensor/equipment	Specification
Engine speed [rpm]	Optical angular encoder	1 to 6000 ± 1
Engine torque [N m]	Strain-gauges torque-meter	-200 to 200 ± 1
Intake pressure [bar]	Piezorresistive transducer	0 to 10 ± 0.001
Exhaust pressure [bar]	Piezorresistive transducer	0 to 10 ± 0.001
Intake temperature [°C]	Thermocouple K-type	0 to 1000 ± 0.5
Exhaust temperature [°C]	Thermocouple K-type	0 to 1000 ± 0.5
Fluid temperature [°C]	Pt100 thermoresistance	-200 to 850 ± 0.3
Air flow [m^3/h]	Rotary flow meter	0.05 to 160 $\pm 0.12\%$
Fuel flow [kg/h]	Dynamic Fuel Meter AVL 733S	0.05 to 150 $\pm 0.12\%$
Blow-by flow [m^3/h]	Blow-by Flow Meter AVL 442	0 to 4.5 $\pm 1.5\%$
Equivalence ratio [$-$]	UEGO NGK Sensor	0.01 to 2 $\pm 0.01\%$
Pollutant emissions	HORIBA MEXA 7100	Depends on species
Smoke level [FSN]	AVL Smoke Meter	0 to 10 $\pm 2\%$

to the exhaust is obtained by means of a set of mass balances in the intake, exhaust and in-cylinder gases.

3.2.5.1.b Data acquisition system

Some differences can be noted between the two setups also in the data acquisition system, mostly regarding the softwares used for sensors and ECU communication.

The low frequency system (related to the data related to the engine and test cell monitoring and the experimental average measurements) acquires data at a 1 Hz frequency over a predefined measurement period to be averaged. A summary of the most relevant mean variables is presentedd in Table 3.6, with the range and accuracy of the corresponding sensors. In the case of the poppet-valve engine, this acquisition process is performed by the *PUMA* program, which is also in charge of the global communication and control and the test cell equipment (dynamometer, air and EGR compressor, etc...). In the case of the Uniflow engine, this process is performed by the *MORPHÉE* program, which is the exact equivalent of the previous program described.

Regarding the high frequency acquisition system, it is also very similar between the two faclities with the same measured parameters and the same

Table 3.7. *High frequency measurements.*

Signal	Sensor/equipment	Specification
Cylinder pressure [bar]	Piezoelectric sensor	0 to 250 bar
Intake pressure [bar]	Piezoresistive sensor	0 to 10 bar
Exhaust pressure [bar]	Piezoresistive sensor	0 to 10 bar
BDC pressure [bar]	Piezoresistive sensor	0 to 20 bar
Injection pulse [A]	Clamp-on Ammeter	0 to 20 A
Injection pressure [bar]	Piezoresistive sensor	0 to 2000 bar
Camshafts reference [V]	Variable reluctance sensors	0 to 10 V
Angular Speed [−]	Optical angular encoder	1800 pulse/rev, resolution 0.2 CAD

sensors, the differences residing mainly in the acquisition device. Instead of using a Yokogawa DL716 digital oscilloscope with 16 channels linked to a computer for registring and storing the data, as it is the case for the poppet-valve engine, the Uniflow engine is directly linked to a computer and the *OSIRIS* program. This software performs the same operations as the Yokogawa oscilloscope, but has the additional advantage to display a continuous measurement of the various engine pressures (intake, exhaust, in-cylinder...) and camshaft strain-gauges, and can record and store these measurements for a given duration when requested. But the measurements are otherwise identical between the two methods, and the only variation between the two engines is the utilization of two intake pressure sensors and two exhaust pressure sensors for the Uniflow engine instead of one of each for the other, strictly due to the engine geometry.

3.2.5.2 On-bench fast post-processing code

If the on-bench fast post-processing of the poppet-valve engine in mostly focused on the combustion-related parameters (cylinder composition, combustion development, fuel efficiencies, etc...) as it is fully described in the aforementioned doctoral thesis and publications, the experimental investigations and thus the post-processing of the Uniflow engine are mainly oriented towards the scavenging parameters on one side (TR and IGR rate), and on the overall performance on the other side (fuel consumption and pollutant emissions). Thus, the data post-processing from the experiments does not need to be as detailed as described previously: for example, the in-depth combustion analysis, heat transfers, or fuel mixing parameters would not provide relevant information considering that a well-known Conventional Diesel Combustion (CDC) is operated for these investigations. However, a great attention will be paid to scavenging related parameters, especially Trapping Ratio and IGR rate, thus requiring good and consistent measurements and on-line calculations for the various instantaneous and mean pressures and temperatures, air and fuel flow measurements, and all resulting calculations such as equivalence ratios, mean effective pressures (IMEP, BMEP), indicated and combustion efficiencies (η_{ind}, η_{comb}), specific fuel consumptions (ISFC, BSFC), among others. Also, some combustion related parameters are still required as much for analysis purposes as for experiment controls, such as combustion phasing (SoC, CA50), combustion stability (CoV_{IMEP}, $CoV_{P_{max}}$), or mechanical limitations (P_{max}, dP/da_{max}).

All the on-line calculations are performed by the *MORPHÉE* and *OSIRIS* software through integrated macros running during the acquisition process, and by the HORIBA gas analyzer for the exhaust gas related parameters (emission levels and unit conversions, as well as oxygen concentration $X_{O_2,exh}$, among others). The post-processing of all those measurements gathered during the investigations performed on this Uniflow engine also includes Design of Experiments, which need to be prepared off-line upstream of the experiments, and then implemented using testing results to generate mathematical response of certain parameters in function of control settings of the engine. The detail of the methodology used for this work is described in depth in section 5.4.1.

3.3 Theoretical tools

To fully describe and understand the various processes contributing to the operation of the 2-Stroke engines under development, this investigation makes an extensive use of different diagnostic and modeling tools, to both process the experimental results and also extrapolate them through simulations (0D-1D or 3D CFD) for deeper understanding of the phenomenon taking place. This includes an in-depth combustion diagnostic and analysis along with a thermal and energy balance (presented in detail in the previously mentioned author's communications), and also a detailed description of the air management parameters, as well as the use of a multi-dimesensional engine model to complete the analysis of the experimental data and/or to evaluate specific non experimentally measurable quantities. To do so, several theoretical tools were used or implemented along this research work, and the main ones will be briefly described within the following sections.

3.3.1 1D models

Throughout the investigations presented in this thesis, various 1D models of the engines have been developed and used either to perform quick Designs of Experiments (DoE) and extract specific data (that might not be experimentally measurable, or only through specific installations), or to simulate engine behaviors as preliminary inputs for 3D-CFD models, as it will later be detailed in section 3.3.3.

One of these 1D-models, for the poppet-valve engine, has been through the *OpenWAM* software. *OpenWAM* is an in-house wave action model developed for the gas dynamics modeling of internal combustion engines. It solves the thermo- and fluid-dynamic equations in each part of the engine, by computing the different elements by the means of 0D or 1D objects connected by suitable boundary conditions. A 1D gas dynamics model performs the calculations of the flow properties along the intake and exhaust systems as well as the EGR paths. All the relevant phenomena taking place along the ducts are considered, including wave interactions, which have an important effect on volumetric efficiency (especially in the intake/exhaust manifolds), the species transport (the thermodynamic properties of the gas depend on the composition and the temperature), the heat transfer between the gas and the duct walls, and the friction between gas and walls. All these fluid-dynamic computations are implemented in detail through the method described by Galindo et al. [13, 14].

Additionally to the fluid calculations, the gas-dynamic model is coupled to a cylinder model that predicts the in-cylinder conditions based on the combustion process, where a detailed heat transfer model is used to obtain the heat rejection to the chamber walls. This combustion model is based on the Apparent Combustion Time (ACT) concept, which consists in a zero-dimensional approach to the combustion process in Diesel engines used to obtain a global and simplified description of the processes that take place in the combustion chamber [15–17]. This ACT model describes the relation between the running conditions of the engine and the Rate of Heat Release (RoHR). Those two models combined lead to the development of a reference virtual engine which allows the calculation of the Internal Gas Residuals (IGR) rate (among other relevant magnitudes) that can be used as a target for the estimator.

Once correctly calibrated, it is a well fitted tool to emulate the engine behavior, in order to run a large number of simulations much faster, less costly, and in more stable conditions (for repetitive iterations as in the DoE cases for example) than experimental tests.

3.3.2 IGR estimation model

As highlighted in several research works, in the case of predictive models, a small uncertainty in the temperature at IVC may affect the combustion delay [18], the combustion velocity [15, 16] or pollutant formation [19, 20]. Similarly, the amount of residual mass and its composition affect the combustion delay in Compression Ignition (CI) engines [21]. For the combustion analysis from in-cylinder pressure, uncertainties in the conditions at IVC were not usually as critical as in predictive models, because the key input is the experimental measured pressure which is substantially independent from any other conditions at IVC. However, the errors in the temperature evolution and the heat release law due to uncertainties in the initial conditions are not negligible [22]. The total trapped mass inside the cylinder, in that case, should be properly evaluated in order to minimize these errors. In that framework, a detailed method has been developed to estimate the residual gas fraction trapped inside of the cylinder by the beginning of the closed cycle, i.e. at IVC.

From the referenced absolute cylinder pressure signal, the trapped mass at IVC can be estimated as in (3.1):

$$m_{IVC} = m_{tot}^{int} + m_{res} - m_{sc} \tag{3.1}$$

where m_{tot}^{int} is the total intake mass (fresh air and EGR), m_{res} is the residual mass from the previous cycle and m_{sc} is the short-circuited mass. Both m_{res} and m_{sc} depend on the characteristic fluid-dynamic behaviour of the engine during the scavenging period, which is highly dependent on the operating conditions, on the manifold characteristics and, most notably, on valve timing. Short-circuited mass is usually small in 4-Stroke engines, where the intake and exhaust processes have dedicated strokes. The main contribution to the mass at IVC therefore is, apart from the intake mass (which can be measured), the residual mass. In the case of 4-Stroke CI engines, the residual mass is generally below 10%, usually about 5%, and cannot exceed 20% in 4-Stroke Spark-Ignition (SI) engines [23, 24]. On the contrary, in the case of 2-stroke engines the scavenging process is much shorter than in 4-Stroke engines and the overlap of the intake and exhaust periods (which represents about 20% of the engine cycle in the case of the poppet-valve engine, used here as a reference for the implementation of this methodology) can lead to high levels of both residual and short-circuited masses: depending on valve timing and intake/exhaust conditions, more than 50% of the mass at IVC can be residuals, and more than 35% of the intake mass can be short-circuited in the case of conventional 2-stroke engines [2, 5]. Moreover, a trade-off between trapping ratio and scavenging efficiency is often observed. In any case, the experimental determination of m_{res} and m_{sc}, even though it might be possible in some cases, is very rarely performed (unlike m_{tot}^{int}) and they are usually obtained from theoretical estimations.

The estimator presented here is conceived so as to require a limited number of inputs: instantaneous in-cylinder pressure, intake temperature, air and fuel mass flows, short-circuited mass m_{sc} (obtained from the experimental determination of TR) and a number of physical constants to be tuned as detailed below. The basis of the model consists of a mass and energy balance in the chamber, as in 0D filling and emptying models; however, its predictive capabilities are enhanced with respect to those models by means of the use of the experimental in-cylinder pressure as the main input of the model. In-cylinder pressure will be used to account for the expansion of the residual gas during the exhaust process and its later evolution during the intake period until IVC. A suitable calibration of the polytropic exponents during the expansion and intake periods will be performed by means of 1D modeling. The outcome of the proposed model is a precise m_{res} estimation that allows to accurately set the mass and temperature at IVC to be used in models dealing with the in-cylinder processes during the closed cycle, including (but not limited to) 0D models for combustion analysis purposes.

As mentioned previously, the development of this residual fraction estimator is based on the poppet-valve engine, and using all the associated experimental facility and equipment presented in detail in section 3.2. Note that even though the proposed method and its validation are both focused on a 2-stroke poppet-valve CI engine, the same thermodynamic approach with a specific calibration procedure would allow its use in other 2-stroke or 4-Stroke reciprocating engines, either diesel or gasoline. In parallel with the experimental measurement, the 1D-model of the engine previously introduced (using the *OpenWAM* program) is also used to run base-case operating conditions and DoEs for both calibration and validation of the IGR estimator and its polytropic coefficients.

3.3.2.1 Methodology

The work performed here was based on a reference engine operating point at 2500 rpm and 15 bar of Indicated Mean Effective Pressure (IMEP). However, all the methodology presented in the following study can be transposed to any operating point.

The 1D model *OpenWAM* was used to simulate the engine behavior after a fine calibration: CFD calculations were performed to get the scavenge profile (Fig. 3.4) and the boundary conditions, while the ACT combustion model was calibrated to reproduce the engine thermodynamics. The CFD results concerning the boundary conditions are very similar to the experimental data (Trapping Ratio of 80 % while the measurement stands at 81.8 %), except for the IGR that was experimentally measured through the simple model presented here (CFD estimated 21 % while experiments gave 30 %).

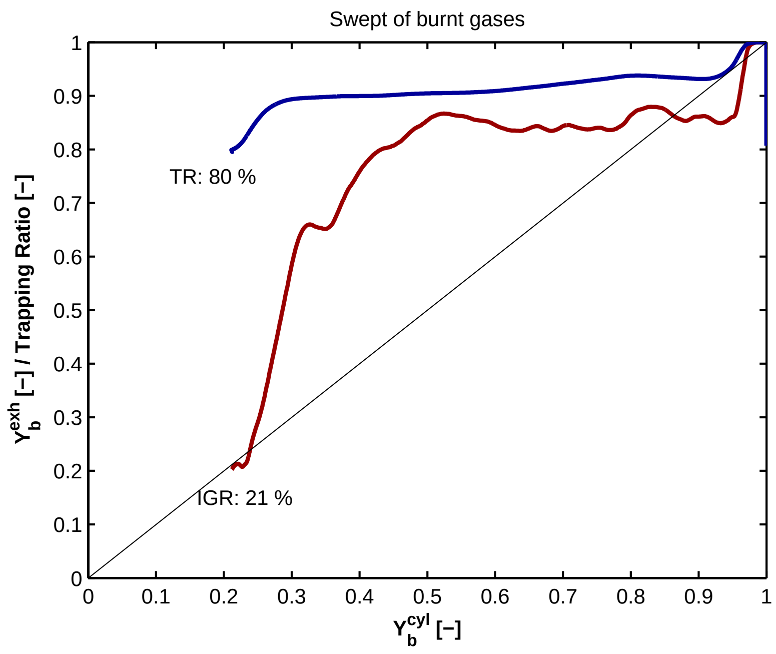

Figure 3.4. *Scavenge and Trapping Ratio of the engine (CFD results).*

By comparing the simulation with experimental results in terms of in-cylinder pressure (Fig. 3.5 left plot) and intake and exhaust pressures (Fig. 3.5 right plot), the WAM model seems to be a very reliable tool for the open cycle as well as for the closed cycle.

This 1D model could then be used to generate outputs which were not directly available experimentally, such as the residuals mass. In addition, such a model allows fast calculations, which is a non-negligible advantage considering the chosen methodology consisting of Designs of Experiments (DoE) for two operating points and four input parameters. The configuration and results of these DoEs are presented in detail later in this paper.

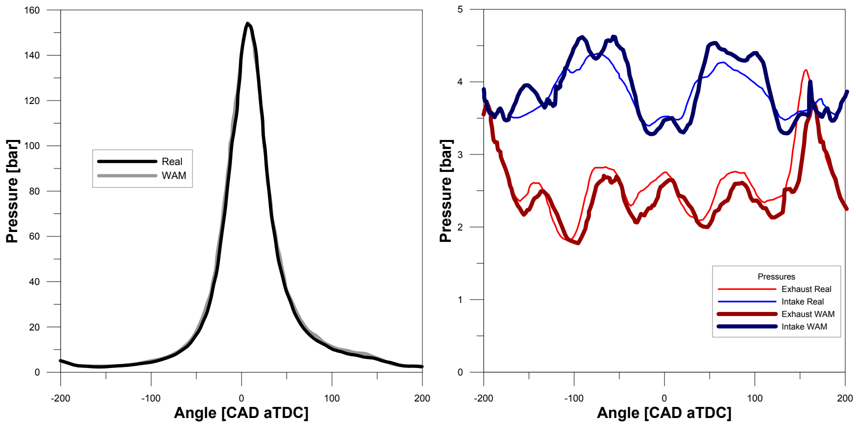

Figure 3.5. *Cylinder pressure (left) and Exhaust and Intake pressures (right) correlations between engine measurements and* OpenWAM.

Then, from the experimental results and the *CALMEC* calculations, the estimator was developed by solving the thermodynamic laws at several levels of complexity. The first level consisted of a basic solution for a fast application and online calculations on the engine test bench. In a second step, a more complex approach was considered, assuming polytropic processes during the scavenging. This method is much more realistic but also requires more calculation time, so it may not be easily adaptable for a direct application on a test cell.

Finally, the results of the different estimators were compared to *OpenWAM* simulations (IGR mass, temperature of the residuals, temperature at IVC. . .) and, in the case of the polytropic models, those simulations were used as reference to calibrate the polytropic coefficients of the estimator.

3.3.2.2 Model description

The IGR estimator is based on the application of two fundamental thermodynamic balances at IVC, the mass balance (3.2) and the enthalpy balance (3.3). The conditions at IVC are the result of combining the conditions of the intake trapped air and the residual gases retained from the previous cycle. In addition, all gases and particularly those retained inside the cylinder at IVC are considered ideal (3.4). Thus, the three main equations of the system can be expressed as:

$$m_{IVC} = m_{tr} + m_{res} \tag{3.2}$$

$$m_{IVC} \cdot h_{IVC} = m_{tr} \cdot h_{tr} + m_{res} \cdot h_{res} \tag{3.3}$$

$$P_{IVC} \cdot V_{IVC} = m_{IVC} \cdot R_{IVC} \cdot T_{IVC} \tag{3.4}$$

The enthalpy balance (3.3) is rewritten by expressing the enthalpies as a function of the corresponding heat capacities c_p and the temperatures (3.5) - (3.7):

$$h_{IVC} = \int_{T_0}^{T_{IVC}} c_{p_{IVC}} \cdot dT \tag{3.5}$$

$$h_{tr} = \int_{T_0}^{T_{tr}} c_{p_{tr}} \cdot dT \tag{3.6}$$

$$h_{res} = \int_{T_0}^{T_{res}} c_{p_{res}} \cdot dT \tag{3.7}$$

The three main equations, plus the expressions of the sensible enthalpies (no chemical reaction is considered), provide six equations and fifteen unknowns, which are three masses from (3.2) (m_{IVC}, m_{tr} and m_{res}), three enthalpies from (3.3) (h_{IVC}, h_{tr}, h_{res}), P_{IVC}, V_{IVC} and R_{IVC} from (3.4), and three heat capacities and three temperatures from (3.5) - (3.7) ($c_{p_{IVC}}$, $c_{p_{tr}}$, $c_{p_{res}}$, and T_{IVC}, T_{tr}, T_{res}). However, some of these unknowns are either defined directly by the geometry or measured, whence they are independent from the chosen resolution method.

3.3.2.2.a Definition of the geometric and measured parameters

The cylinder volume at IVC (V_{IVC}) is set geometrically knowing the position of the piston as a function of the angle. Also, P_{IVC} is obtained directly from the referenced cylinder pressure measurement. The trapped mass m_{tr} is decomposed into various components, each one being measured separately:

$$m_{tr} = \eta_{tr} \cdot m_{tot}^{int} = \eta_{tr} \cdot (m_a + m_{EGR}) \tag{3.8}$$

As indicated above, the trapping efficiency or trapping ratio (η_{tr}) is measured following the tracer gas methodology (injecting methane into the intake port [25, 26]), and the calculation process is detailed in Appendix A. The fresh air supplied to the engine (m_a) is measured through a volumetric flow-meter and the EGR mass (m_{EGR}) is calculated as a proportion of the total mass admitted:

$$m_{EGR} = m_{tot}^{int} \cdot T_{EGR} = m_a \cdot \frac{T_{EGR}}{1 - T_{EGR}} \tag{3.9}$$

The EGR rate (T_{EGR}) is defined, according to its common definition, as the CO_2 molar fraction ratio between intake and exhaust gases, and measured by the gas analyzer:

$$T_{EGR} = \frac{X_{CO_2}^{int}}{X_{CO_2}^{exh}} \tag{3.10}$$

The EGR mass is then expressed as:

$$m_{EGR} = m_a \cdot \frac{X_{CO_2}^{int}}{X_{CO_2}^{exh} - X_{CO_2}^{int}} \tag{3.11}$$

The number of unknowns is then reduced down to twelve. In order to solve the system, six more equations or hypotheses are required. Then and as decribed next, different thermodynamic models may be considered, depending on the assumptions considered for the thermodynamic properties of the fluids.

3.3.2.2.b ***Basic thermodynamic model***

In the simplest option, the mass balance is as defined by equation (3.2) while the perfect gas assumption is considered for all gases, so that the heat capacities in (3.5) - (3.7) are assumed to be constant and known. The equation (3.3) can be rewritten as (3.12).

$$\begin{aligned} m_{IVC} \cdot c_{p_{IVC}} \cdot T_{IVC} = m_{tr} \cdot c_{p_{tr}} \cdot T_{tr} \\ + m_{res} \cdot c_{p_{res}} \cdot T_{res} \end{aligned} \tag{3.12}$$

In addition, R_{IVC} is also considered constant in (3.4) and known.

This process generates a set of three equations (3.2) (3.4) (3.12) with five unknowns (m_{IVC}, m_{res}, T_{IVC}, T_{tr}, T_{res}), so that two additional hypotheses must be assumed in order to close this basic model. In this case, an attractive option for its simplicity consists of considering the temperatures of the trapped mass and the residuals equal to the intake and the exhaust temperatures, respectively.

$$T_{tr} = T_{int} \tag{3.13}$$

$$T_{res} = T_{exh} \tag{3.14}$$

Therefore, the problem is finally closed and it can be solved analytically giving (3.15).

$$m_{res} = \frac{c_{p_{IVC}} \cdot \frac{P_{IVC} \cdot V_{IVC}}{R_{IVC}} - m_{tr} \cdot c_{p_{tr}} \cdot T_{tr}}{c_{p_{res}} \cdot T_{res}} \tag{3.15}$$

Then, the mass at IVC (m_{IVC}) can be obtained directly from the mass balance (3.2):

$$m_{IVC} = m_{tr} + m_{res} \tag{3.16}$$

The temperature at IVC can also be calculated from the ideal gas equation (3.4).

$$T_{IVC} = \frac{P_{IVC} \cdot V_{IVC}}{m_{IVC} \cdot R_{IVC}} \tag{3.17}$$

As a final remark, this basic model has the clear benefit of being very simple and extremely fast, making it compatible with real-time applications such as control or on-line test cell data analysis during the experimental activities. However, some of the assumptions can introduce unacceptable errors depending on the particular engine characteristics, operating conditions, etc., and therefore its accuracy will be carefully evaluated in the next section.

3.3.2.2.c Detailed thermodynamic model

This model is based on a more realistic description of the thermodynamic properties of the fluids and of the processes occurring in the engine, but the calculations are also more complex and time-consuming. The new assumptions are expected to improve the accuracy of the estimate of the residuals and of the total trapped mass with respect to those generated with the previous basic model.

In this model, as in the simple model, the mass balance is as defined by equation (3.2), and the enthalpy balance corresponds to equation (3.3), with enthalpies expressed as a function of heat capacities (3.5) - (3.7). However, the previous hypotheses of perfect gases is not assumed anymore and the heat capacities are now expressed as a function of the composition of the gases and their temperatures.

The enthalpy balance (3.3) is then rewritten considering the previous dependencies to derive equation (3.18).

$$\begin{aligned} m_{IVC} \cdot \int_{T_0}^{T_{IVC}} c_{p_{IVC}} \left(T, Y_b^{IVC}\right) \cdot dT = \\ m_{tr} \cdot \int_{T_0}^{T_{tr}} c_{p_{tr}} \left(T, Y_b^{tr}\right) \cdot dT + \\ m_{res} \cdot \int_{T_0}^{T_{res}} c_{p_{res}} \left(T, Y_b^{res}\right) \cdot dT \end{aligned} \tag{3.18}$$

As a result, the heat capacities of the trapped mass, the residual mass or the total mass at IVC are calculated from their composition in terms of the fraction of fresh air and burnt gases in stoichiometic conditions, and their corresponding temperatures, according to the set of equations (3.19) - (3.21).

$$c_{p_{IVC}} = Y_a^{IVC} \cdot c_{p_a}\left(T_{IVC}\right) + Y_b^{IVC} \cdot c_{p_b}\left(T_{IVC}\right) \tag{3.19}$$

$$c_{p_{tr}} = Y_a^{tr} \cdot c_{p_a}\left(T_{tr}\right) + Y_b^{tr} \cdot c_{p_b}\left(T_{tr}\right) \tag{3.20}$$

$$c_{p_{res}} = Y_a^{res} \cdot c_{p_a}\left(T_{res}\right) + Y_b^{res} \cdot c_{p_b}\left(T_{res}\right) \tag{3.21}$$

For simplicity, the dependency of the heat capacities with temperature is defined by means of polynomial correlations only for fresh air with standard composition and burnt gases in stoichiometric conditions, following an approach similar to that suggested by Lapuerta et al. [27]. At this point, it is important to recall that the temperatures in equations (3.18) - (3.21), have already been defined as unknowns. Considering that the different gases are composed only by fresh air and burnt gases in stoichiometric conditions, their mass fractions are complementary according to $Y_a^x = 1 - Y_b^x$ with x standing for either IVC, tr or res.

The fraction of burnt gases in stoichiometric conditions in the trapped mass, the residual mass or the total mass at IVC are obtained solving the set of equations defined by the mass balances at the intake, in the cylinder at IVC and at EVO, and finally at the exhaust as discussed in Appendix B. Then, the expressions of these mass fractions depend only on variables that were either already included in the solution as unknowns (3.22) - (3.24) or obtained experimentally (not included in the set of equations to highlight the unknowns of the problem).

$$Y_b^{IVC} = f(m_{IVC}, m_{tr}, m_{res}, Y_b^{tr}, Y_b^{res}) \tag{3.22}$$

$$Y_b^{tr} = Y_b^{exh} \cdot T_{EGR} = f(m_{tr}, Y_b^{res}) \tag{3.23}$$

$$Y_b^{res} = Y_b^{EVO} = f(m_{tr}) \tag{3.24}$$

Concerning the ideal gas assumption, it is still accepted and expressed as equation (3.4), except that R_{IVC} now depends on the composition of the mass at IVC (3.25), assuming constant and known values for R_a and R_b:

$$R_{IVC} = Y_a^{IVC} \cdot R_a + Y_b^{IVC} \cdot R_b \tag{3.25}$$

Since no new unknowns are added to the problem, a total of nine unknowns (m_{IVC}, m_{res}, Y_b^x, T_x, R_{IVC}) and seven equations (3.2) - (3.4) plus (3.22) - (3.25) can be identified. In this way, two hypotheses must be formulated about the temperatures of the trapped mass and of the residuals in order to close the problem. With the purpose of retaining as much thermodynamic consistency as possible, polytropic evolutions were assumed to relate the intake manifold temperature to the temperature of the trapped mass at IVC, and the temperature of the residuals at IVC with the temperature at EVO.

For the trapped mass, only one polytropic process was considered to evaluate its temperature at IVC, as expressed by equation (3.26):

$$T_{tr} = T_{int} \cdot \left(\frac{P_{int}}{P_{IVC}}\right)^{\frac{1-k_{int}}{k_{int}}} \tag{3.26}$$

For the residual gases, the same approach with a single polytropic process between EVO and IVC was initially evaluated, but it was readily apparent that the temperature evolution of the residuals during this process cannot be well reproduced by considering a thermodynamic process with only one polytropic coefficient, as shown by the blue curve in Fig 3.6. Furthermore, in order to be able to reach the final residuals temperature predicted by the 1D model *OpenWAM*, this coefficient would need be set at a value of 20, which is thermodynamically incoherent.

However, by decomposing the exhaust process into two polytropic evolutions with different coefficients, the evolution of the gas temperature appears to be much better estimated (green curve in Fig 3.6). Therefore, this option based on two polytropic processes was considered, the first accounting for the blow-down stage between EVO and the minimum cylinder pressure (3.27), and the second accounting for the conventional scavenging exhaust stage (3.28).

$$T_{res,P_{min}} = T_{res,EVO} \cdot \left(\frac{P_{EVO}}{P_{min}}\right)^{\frac{1-k_{exh1}}{k_{exh1}}} \tag{3.27}$$

$$T_{res} = T_{res,P_{min}} \cdot \left(\frac{P_{min}}{P_{IVC}}\right)^{\frac{1-k_{exh2}}{k_{exh2}}} \tag{3.28}$$

In these polytropic processes, all the pressures are experimentally measured, as well as the intake temperature. Then, four new unknowns can be identified ($T_{res,EVO}$, k_{int}, k_{exh1}, k_{exh2}). By applying the ideal gas equation (3.4) at EVO, the pressure and volume are known from the geometry and measurements, the

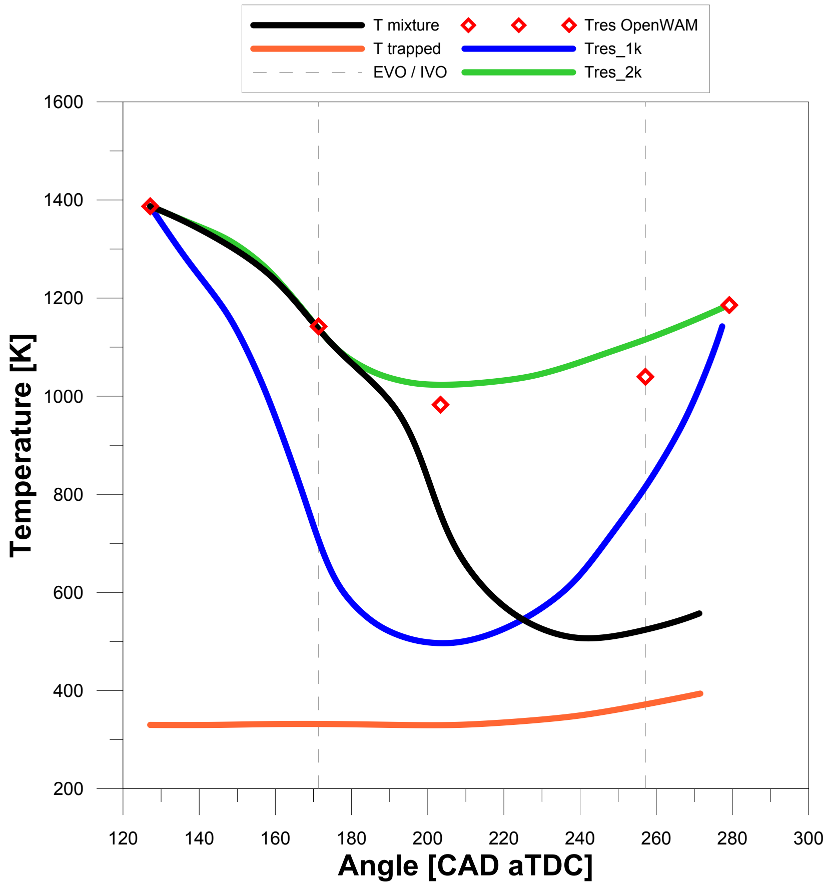

Figure 3.6. *Comparison between different estimator approaches.*

mass depends on the mass at IVC (already included in the equations system), the mass of fuel and the blow-by (both measured), and R_{EVO} is a function of the gas composition already expressed by equation (3.24). This new equation is then added to the system to solve $T_{res,EVO}$ (3.29).

$$T_{res,EVO} = \frac{P_{EVO} \cdot V_{EVO}}{m_{EVO} \cdot R_{EVO}} \tag{3.29}$$

Finally, the problem is formulated in terms of eleven unknowns (m_{IVC}, T_{IVC}, m_{res}, Y_b^x, R_{IVC}, $T_{res,EVO}$, k_{int}, k_{exh1}, k_{exh2}) when two polytropic processes are considered for the residuals, related by eight equations (3.2) -

(3.4) plus (3.22) - (3.25) plus (3.29). Then, some assumption about the values of the polytropic coefficients k_x was needed to close the model; however, it was expected that they could be easily calibrated (through 1D models, for instance) in view of their well-known order of magnitude and their relatively narrow interval of acceptable values.

3.3.2.3 Model validation

As discussed previously, a 1D-model of the engine was developed with *OpenWAM*. For the work presented here, the most critical part of the engine cycle is the open cycle, from EVO to IVC. In a 2-stroke engine, the pressure waves in both the intake and exhaust manifolds have a great influence on engine efficiency, and therefore careful consideration should be given to the correlation of the exhaust and intake pressures. Fig. 3.5 shows the results of this calibration, and it can be observed that the wave model is qualitatively very similar to the real conditions (the pressure waves are very well phased), even if there are some small quantitative deviations. Thus, this model can be used as a reference for the non-measurable parameters and to confront with the residual mass estimator results.

As shown in Table 3.8, all the measured parameters such as the intake or exhaust temperature and pressure, or the intake air mass, are very close to the values estimated from the model (less than 1.8% error). However, the non-measurable parameters (mass at IVC and IGR rate) are calculated with the simple estimator and appear to be substantially over-estimated, with an error above 40% , confirming the need to develop a reliable tool for the estimation of the residual mass.

Table 3.8. *Wave model validation.*

	OpenWAM	**Experimental**	**%Error**
Admitted mass *(kg/h)*	120.01	117.89	1.77 %
Retained mass *(g)*	0.61455	0.61528	0.12 %
IGR rate *(-)*	21.21 %	30.26 %	42.64 %
Mass at IVC *(g)*	0.78002	0.88225	13.11 %
Trapping Ratio *(-)*	76.81 %	78.29 %	1.92 %
T intake *(K)*	307.5	309.6	0.69 %
T exhaust *(K)*	807.8	797.0	1.33 %

In order to validate the IGR estimator, two DoEs, at different operating points, were used, one at medium load and low engine speed, and the other at high load and high engine speed.

These DoEs were created from experimental measurements performed on the engine, and the twenty-five corresponding points were calculated by means of the *OpenWAM* model. For each point, the calculated IGR rate was compared to the prediction from the simple estimator (used on the engine test bench). Simultaneously, the proposed detailed estimator (based on polytropic processes) was adjusted with the results from *OpenWAM* and from the measurements to properly estimate the residuals fraction and the trapped mass. In this way, direct comparison between the results obtained with the 1D model and those from the estimators was possible.

3.3.2.3.a Set up of the DoEs

The real operating conditions on which the two DoEs were based are presented in Table 3.9. Four parameters were then selected as inputs for the DoEs: the pressures at the intake and at the exhaust (P_{int} and P_{exh}), thus setting the pressure drop (ΔP), and the phasing of the intake and the exhaust camshafts (VVT_{int} and VVT_{exh}), thus setting the overlap ($Olap$).

Table 3.9. *DoEs reference points.*

	Point 1 (medium load / low speed)	**Point 2** (high load / high speed)
Speed *(rpm)*	1500	2500
IMEP *(bar)*	10.5	15.0
Torque *(N.m)*	50.2	76.4
Power *(kW)*	7.9	20.0
$\mathbf{P}_{int}$ *(bar)*	2.50	3.90
$\mathbf{P}_{exh}$ *(bar)*	1.80	2.55
$\Delta\mathbf{P}$ *(bar)*	0.70	1.35
$\mathbf{VVT}_{int}$ *(cad)*	256.7	246.7
$\mathbf{VVT}_{exh}$ *(cad)*	237.4	231.4
Overlap *(cad)*	70	74
Trapping Ratio *(%)*	70.3	78.3
IGR Ratio *(%)*	30.2	30.3
In-Cyl Richness *(-)*	0.54	0.64

The configurations of the DoEs are presented in Appendix C, where the parameters for each of the twenty-five points are detailed. For each DoE, each point was calculated with the *OpenWAM* model, and compared to those obtained from the simple estimator, providing a wide basis for comparison at various scavenging conditions.

3.3.2.3.b Results from the DoEs and comparison between the OpenWAM model and the estimators

As shown in Fig 3.7 (and detailed in Appendix D, see Table A.3 and Table A.4), the mean error between the *OpenWAM* model and the simple IGR estimator is around 4 % in absolute value, with a peak at more than 6 % for some conditions.

The calibration of the detailed estimator was performed by adjusting the polytropic coefficients for each operating point, thus obtaining: $k_{exh1} = 1.4$ and $k_{exh2} = 1.2$. As a first approach, the polytropic coefficient for the trapped mass is fixed at $k_{int} = 1.35$. As shown in Fig 3.7, the mean error is drastically reduced to around 1.4 % for DoE 1, and 0.5 % for DoE 2.

3.3.2.3.c Adjustments of the polytropic coefficients

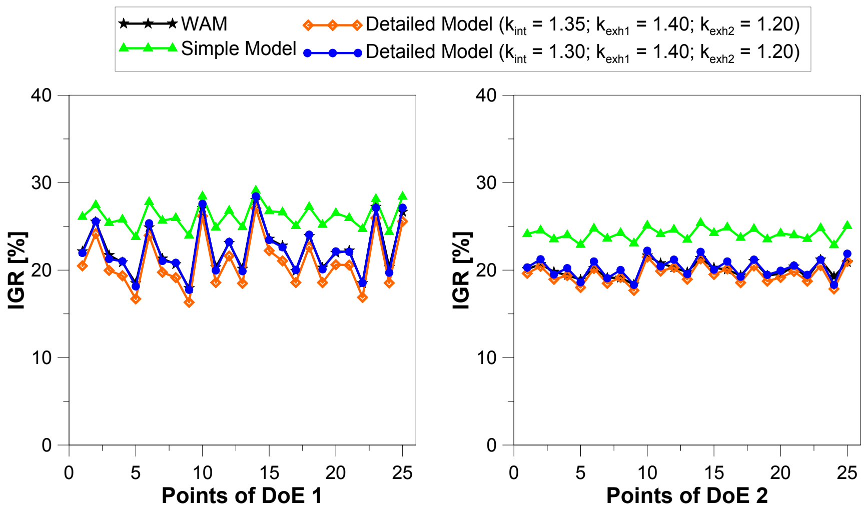

Figure 3.7. *Comparison between the IGR estimators for DoE 1 (left) and DoE 2 (right).*

The estimate of the residuals fraction is noticeably improved by the use of the detailed IGR estimator with respect to the simple one, as shown in Fig. 3.7, where the first version of the detailed estimator is represented by the orange plot. It can be observed that the trend from the WAM model is followed very closely, which was not the case with the simple estimator (for instance, between the third and fourth points of DoE 1, the IGR rate should decrease, but increases with the simple estimator). However, the precision can still be improved with some fine-tuning of the polytropic coefficient for the trapped mass (initially set to $k_{int} = 1.35$).

Indeed, the IGR rate is still underestimated due to a too high polytropic coefficient (hence a too low heat transfer) for the fresh intake air. A new value of $k_{int} = 1.30$ was selected to increase the heat transfer to the intake air, but ensuring that it should still be lower than that of the residuals (since the temperature difference between the residuals and the cylinder wall should be higher than that between the intake ducts and the fresh air). Then, the DoEs were repeated using $k_{int} = 1.30$, and the results are represented by the blue plot in Fig. 3.7.

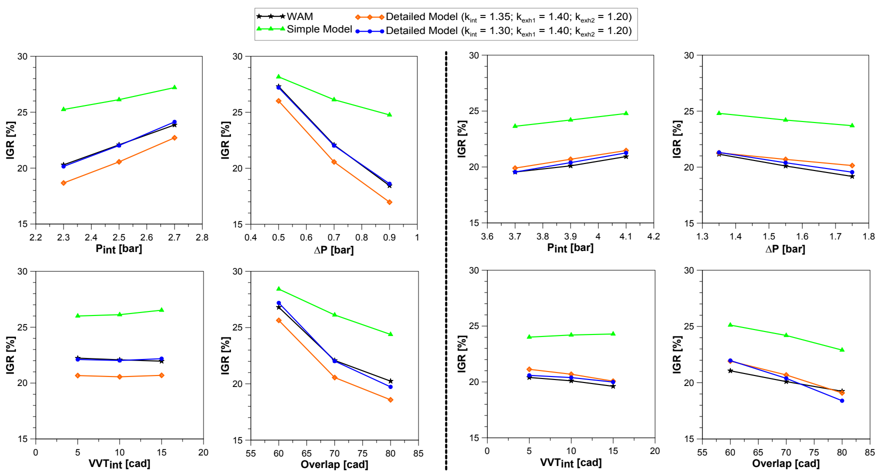

Figure 3.8. *Sensibility of the different versions of the estimator for DoE 1 (4 left plots) and DoE 2 (4 right plots).*

The previous study was then repeated for all the points of the two DoEs with a polytropic coefficient for the trapped mass of $k_{int} = 1.3$. In both operating points a slight improvement can be observed, with a mean difference of around 0.2 % (even 0.02 % for DoE 1) and a maximum difference below 1 %. The detailed results (data corresponding to Fig. 3.7) are given in Appendix D.

Another factor that needs to be considered in the validation is the sensitivity of the estimator to the engine parameters, as represented in Fig. 3.8 where the IGR rate is expressed as a function of the various parameters from the DoEs (each one being successively taken as a variable whereas the other ones are kept constant). Aside from the quantitative error already demonstrated in Fig. 3.7, it can be observed here that at low load (DoE 1), the response of the simple model is consistent with the WAM reference for the intake pressure variations, but loses sensitivity when changing the pressure drop or the valve overlap. At higher load (DoE 2), the sensitivity is also lost when the intake valve timing is modified. On the contrary, the detailed model keeps a good sensitivity compared to the WAM reference, regardless of the parameter studied.

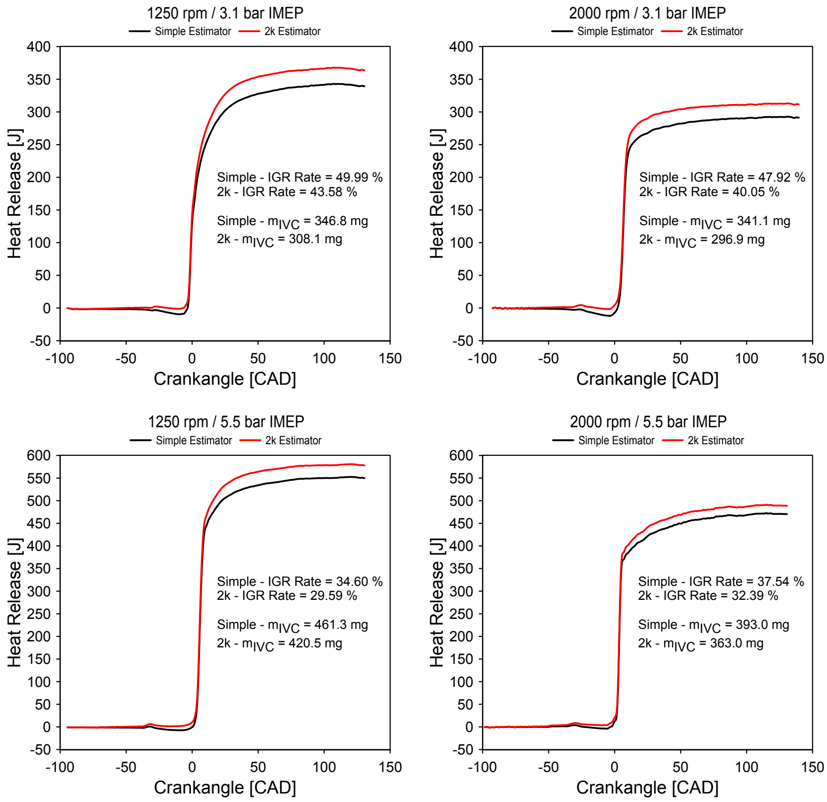

Figure 3.9. *Comparison between the Heat Release at two operating points, considering the residual mass estimated by the two models.*

As a final check, the detailed version of the residuals estimator was compared to the simple one used previously on the engine, and implemented into *CALMEC* to calculate the heat release for various operating points. As the heat release calculation is more affected by the residual mass fraction when low loads are considered, the validation was focused on points with IMEP ranging from 3 to 5 bar. As shown in Fig. 3.9, the residuals mass is not overestimated anymore, and thus a larger part of the energy released by combustion is transformed into indicated energy. This results in an apparent combustion efficiency (calculated as the ratio between the maximum cumulative heat release and the fuel chemical energy) more similar to the real combustion efficiency calculated from HC and CO measurements. Moreover, the detailed model is free from the unphysical phenomena observed with the simple estimator before the start of combustion (a negative heat release mathematically compensating for the over-estimated mass).

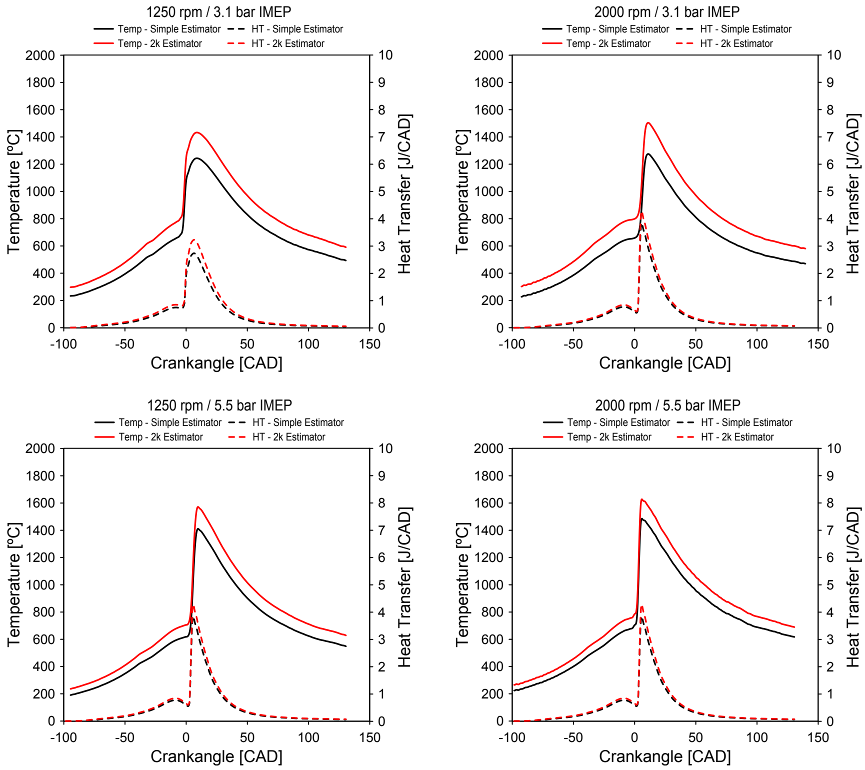

Figure 3.10. *Comparison between the in-cylinder temperature and heat transfer at two operating points, considering the residual mass estimated by the two models.*

The other parameters relevant for the validation are the in-cylinder temperature, which affects directly the heat transfer, and the global composition inside the cylinder. As it can be seen in Fig. 3.10, the variation of the IGR rate has a deep impact on the in-cylinder temperature thus affecting importantly the heat rejection to the chamber walls. As the heat transfer plays a key role in the energy balance of the engine [28–30], the behavior seen in Fig. 3.9 (that is, the negative heat release before the start of combustion) can also be justified mainly by the uncertainty in this term. According to these comments, it can be concluded that improving the residuals estimation has a global impact on the combustion diagnostic.

Finally, both the temperature and the cylinder composition are very important to better understand the link between combustion and pollutant emissions, as they determine the combustion conditions and the resulting adiabatic flame temperature.

3.3.3 Multi-dimensional air-flow and combustion modeling

The extensive use of comprehensive computer codes has nowadays become the norm in engine research, instead of empiricalof descriptions of engine processes through thermodynamic analysis. The multidimensional CFD modeling technology has enabled the detailed optimization of IC engines, whether it is for combustion investigations or internal air flow studies. This specific tool is particularly useful for the development of advanced combustion concepts or for the design of new architectures, since it allows a more accurate prediction and representation of the fluid dynamics, thermodynamical processes and chemical mechanism occuring inside the engine.

For the investigations presented in this thesis, a computational model of the Uniflow engine was built by means of the CONVERGE CFD code, developed by *Convergent Science* [31], to perform open cycle calculations during the design phase of the engine in order to evaluating the effects of intake port geometries on the air flow inside the cylinder. The model will be used to perform detailed analysis, and accordingly more accurate quantitative comparisons, of the resulting scavenging performance and the global in-cylinder conditions and the end of the open cycle. Those results will be used iteratively to select optimum intake port designs, which would then be experimentally reproduce for validation purposes before further experimental investigations on a wider range of operating conditions.

A set of computational models were also built for the poppet-valve engine during the very early phase of the project, in order to evaluate the detailed in-cylinder phenomenon occurring inside the engine cylinder during both open and closed cycles, and particularly the air/fuel mixing processes and their impact over the combustion development. Those results and the computational models used for these studies have already been presented and discussed by the previous PhD student in her thesis [1], thus will not be repeated here.

3.3.3.1 Set-up of the model for the Uniflow engine

The 3D-CFD investigations performed along the design process of the Uniflow engine were exclusively focused on the open cycle, using the complete intake/exhaust and cylinder geometries to be able to visualize all the details of the air flows inside and outside of the chamber. The mesh geometry is automatically generated by the CFD code within the surface geometry of the various domains similarly to the previous engine, using the same 3 mm base cell size. No AMR is used here, but local refinement is still allowed near the intake-cylinder and cylinder-exhaust domain interfaces, where flow velocity can reach very high levels at the beginning and ending of intake/exhaust events.

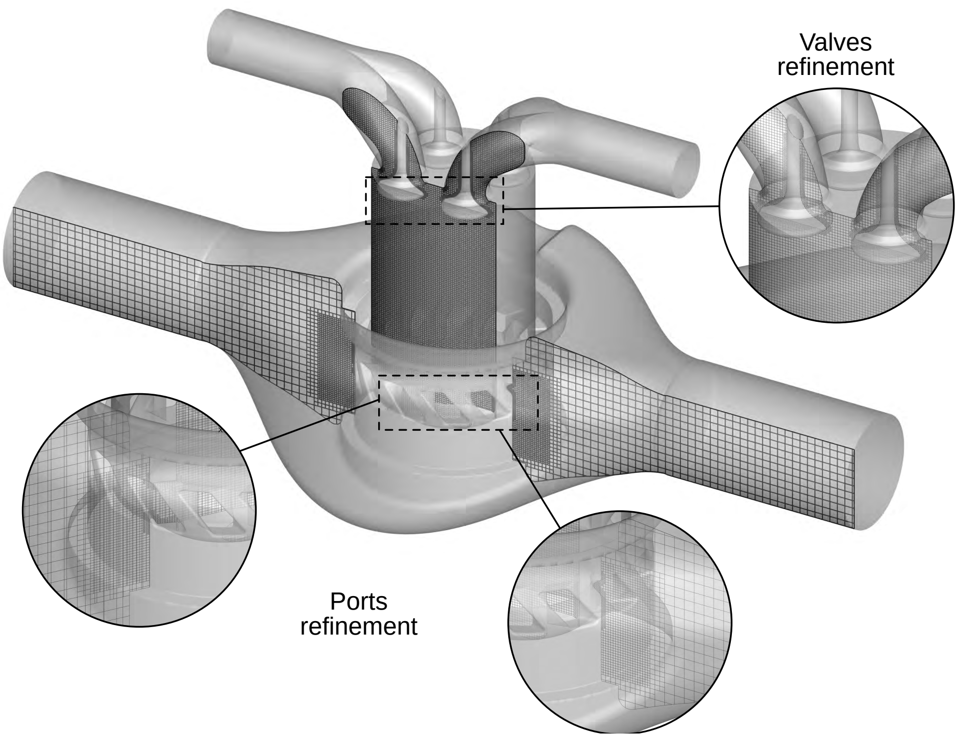

Figure 3.11. *CFD mesh of the REWARD engine cylinder.*

The baseline operating condition, in this case, was defined arbitrarily as a central point of the engine performance range: 2000 rpm and 7 bar IMEP. Here, since no experimental data is available during this engine design phase, all the data relative to the combustion (cylinder pressure, intake/exhaust pressures minimizing the ISFC as a first approach, temperatures...) are estimated through a detailed 1D model of the engine developed in parallel with the global design and performed by the Czech Technical University of Prague (REWARD Project partners), using standard diesel-like combustion models.

Once the global geometry of the model is defined (and updated with new port designs when necessary), the computation then needs to be finely calibrated with initial and boundary conditions, for the different regions in play (intake, in-cylinder, and exhaust).

3.3.3.2 Initialization, regions and boundary conditions

The general methodology of the initialization of the 3D-CFD model is well described by Daniela DeLima in her PhD thesis [1], with all the details about the various regions and domains. Although it is more specifically calibrated for closed cycle calculations (i.e mixing and combustion study), most of the specifications are still valid for open cycle simulations and air flow investigations. These main boundary conditions are summarized in Table 3.10.

Table 3.10. *Boundary conditions.*

Variable	Boundary condition	Value
Velocity	Law-of-the-wall	0 for fixed walls evolving values for moving surface (from law)
	Neumann	0 for inflow/outflow boundaries
Pressure	Neumann	0 for fixed/moving walls
	Dirichlet	fixed values for inflow/outflow boundaries (from 1D model)
Temperature	Law-of-the-wall	fixed values for fixed/moving walls (from 1D model)
	Dirichlet	fixed values for inflow (from 1D model)
	Backflow Neumann	fixed values for outflow (from 1D model)
Species	Neumann	0 for fixed/moving walls
	Dirichlet	fixed values O_2, N_2 for inflow (from 1D model)
	Backflow Neumann	0 for outflow
Passive	Neumann	0 for fixed/moving walls
	Dirichlet	1 for inflow
	Backflow Neumann	0 for outflow
Turbulent kinetic energy (tke)	Neumann	0 for fixed/moving walls
	Intensity	fixed values for inflow (from pipe dimensions)
	Backflow Neumann	0 for outflow
Turbulent dissipation (eps)	Dirichlet	0 for fixed/moving walls
	Length Scale	fixed values for inflow (from pipe dimensions)
	Backflow Neumann	0 for outflow

As previously mentioned, considering that the study was not focused on the closed cycle and the combustion phase, the calculation related to the poppet-valve engine are performed only for the open cycle, which takes place approximately between 110 CAD aTDC and 250 CAD aTDC, so the calculation range goes from 100 CAD aTDC to 300 CAD aTDC in order to leave some flexibility and stabilization time, and getting results which are valid for comparison between cases. Also, since no experimental data is available, all the initialization boundary conditions come from the 1D model simulations previously introduced, which provides generic wall mean temperatures and fluid mean thermodynamic states and compositions for the base-case operating condition. The simulations are then iterated between the 3D-CFD model and the 1D model to ensure the validity and stability of the scavenging results, and their coherence with the performance estimated by the 1D model.

Since the calibration of this computational model is based on the previously existing model introduced at the beginning of this section, the validation is assumed to be similar to the one described by the previous PhD student, for which the methodology can be found in her thesis [1].

3.4 Conclusions

This chapter introduced the most relevant features of two different engines used during this doctoral thesis: the poppet-valve 2-Stroke HSDI CI engine, developed in the framework of the POWERFUL / Post-POWERFUL European project, and the Uniflow 2-Stroke HSDI CI engine, entirely designed in the framework of the REWARD Project. The main geometrical descriptions of the single-cylinder research versions of these two 2-Stroke engine concepts and of all the sub-systems surrounding them and constituting the test cell, have been thoroughly described in section 3.2.

The section 3.2.5 details the layouts of the experimental facilities used during these investigations, while all the different auxiliary systems are referenced to the preceeding PhD student, as well as the required measurement equipment and sensors installed in the test bench for monitoring and acquisition purposes. Dedicated measurement and calculation procedures, especially adapted for the study of 2-Stroke engines and their specific air management process, are also described in detail in this chapter. This includes the experimental measurement of the trapping ratio based on the tracer gas method, and a very detailed methodology for the estimation of the IGR rate and the cylinder-charge composition. These two parameters are critical for the posterior analysis of both the internal air flow and of the combustion process.

Finally, the section 3.3 presents a description of the theoretical tools, with on the one side the combustion analysis and diagnostic programs used to post-process the results obtained from the gasoline PPC experiments, mainly decribed through the same references as the poppet-valve engine's test cell; and on the other side the modeling softwares and evaluation tools for cylinder scavenging performance, used to design the geometry and optimize the air flow through the intake ports, into the combustion chamber and out the exhaust valves of the Uniflow engine.

The following chapters will now focus on the practical results of the investigations performed during this doctoral thesis. Firstly, Chapter 4 will present the detailed analysis of the results obtained in the single-cylinder poppet-valve 2-Stroke engine operating with the gasoline PPC combustion concept to evaluate the potential of both this engine architecture and this type of Low Temperature Combustion, also highlighting the main issues either can raise. Then, Chapter 5 will try and bring resolution paths for some of these issues by adapting the geometry to propose new scavenging solutions to fulfill the objectives defined in the introduction of this thesis. Considering the reasoning and application processes are entirely different between the two following chapters, each of them will be introduced by a brief definition of the methodology used for the experimental and/or theoretical studies, and the associated analysis and optimization of their results.

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A Appendices

A.1 Tracer method for Trapping Ratio calculation

$$\eta_{tr} = \frac{m_{tr}}{m_a} = \frac{m_{CH_4}^{tr}}{m_{CH_4}^{int}} \tag{30}$$

$$m_{CH_4}^{int} = m_{CH_4}^{tr} + m_{CH_4}^{exh} \tag{31}$$

$$\eta_{tr} = \frac{m_{CH_4}^{int} - m_{CH_4}^{exh}}{m_{CH_4}^{int}} = 1 - \frac{m_{CH_4}^{exh}}{m_{CH_4}^{int}} = 1 - \frac{Y_{CH_4}^{exh} \cdot (m_a + m_f)}{Y_{CH_4}^{int} \cdot m_a} \tag{32}$$

$$\begin{aligned} X_{CH_4}^{exh} &\approx Y_{CH_4}^{exh} \\ X_{CH_4}^{int} &\approx Y_{CH_4}^{int} \end{aligned} \tag{33}$$

$$\eta_{tr} = 1 - \left(\frac{m_a + m_f}{m_a}\right) \cdot \frac{X_{CH_4}^{exh}}{X_{CH_4}^{int}} \tag{34}$$

$$\frac{X_{CH_4}^{exh}}{X_{CH_4}^{int}} = \frac{(1 - \eta_{tr}) \cdot m_a}{m_a + m_f} \tag{35}$$

$$\eta_{tr} = 1 - \frac{m_a + m_f}{m_a \cdot \left(\frac{X_{CH_4}^{int}}{X_{CH_4}^{exh}} + \frac{X_{CO_2}^{int}}{X_{CO_2}^{exh}}\right)} \tag{36}$$

$$\frac{X_{CH_4}^{exh}}{X_{CH_4}^{int}} = \frac{(1 - \eta_{tr}) \cdot m_a}{m_a + m_f - m_{EGR} \cdot (1 - \eta_{tr})} \tag{37}$$

A.2 Calculation of the burnt gas mass fraction at EVO

$$Y_b^{EVO} = \frac{m_{a,b} + m_{f,b} + m_{EGR} \cdot Y_b^{exh} + m_{res} \cdot Y_b^{EVO} - m_{sc} \cdot Y_b^{int} - m_{bb} \cdot Y_b^{bb}}{m_f + m_a + m_{EGR} + m_{res} - m_{sc} - m_{bb}} \tag{38}$$

$$Y_b^{bb} = \frac{Y_b^{IVC} + Y_b^{EVO}}{2} \tag{39}$$

$$Y_b^{EVO} = \frac{m_{a,b} + m_{f,b} + Y_b^{exh} \cdot (m_{EGR} - m_{sc} \cdot T_{EGR}) - \frac{m_{bb}}{2} \cdot Y_b^{IVC}}{m_f + m_a + m_{EGR} - m_{sc} - \frac{m_{bb}}{2}} \tag{40}$$

$$\begin{aligned} Y_b^{EVO} &= \frac{m_{a,b} + m_{f,b} + Y_b^{exh} \cdot (m_{EGR} - m_{sc} \cdot T_{EGR}) - \frac{m_{bb}}{2} \cdot \frac{Y_b^{exh} \cdot (m_{EGR} - m_{sc} \cdot T_{EGR}) + m_{res} \cdot Y_b^{EVO}}{m_{IVC}}}{m_f + m_a + m_{EGR} - m_{sc} - \frac{m_{bb}}{2}} \\ &= \frac{m_{a,b} + m_{f,b} + Y_b^{exh} \cdot (m_{EGR} - m_{sc} \cdot T_{EGR}) \cdot \left(1 - \frac{m_{bb}}{2} \cdot \frac{1}{m_{IVC}}\right) - \frac{m_{bb}}{2} \cdot \frac{m_{res}}{m_{IVC}} \cdot Y_b^{EVO}}{m_f + m_a + m_{EGR} - m_{sc} - \frac{m_{bb}}{2}} \\ &= \frac{m_{a,b} + m_{f,b} + Y_b^{exh} \cdot (m_{EGR} - m_{sc} \cdot T_{EGR}) \cdot \left(1 - \frac{m_{bb}}{2} \cdot \frac{1}{m_{IVC}}\right)}{m_f + m_a + m_{EGR} - m_{sc} + \frac{m_{bb}}{2} \cdot \left(\frac{m_{res}}{m_{IVC}} - 1\right)} \end{aligned} \tag{41}$$

$$\begin{aligned} Y_b^{EVO} &= \frac{m_{a,b} + m_{f,b} + \frac{(m_a + m_f + m_{EGR} - m_{sc} - m_{bb}) \cdot Y_b^{EVO}}{m_f + m_a + m_{EGR} - m_{bb} - m_{sc} \cdot T_{EGR}} \cdot (m_{EGR} - m_{sc} \cdot T_{EGR}) \cdot \left(1 - \frac{m_{bb}}{2} \cdot \frac{1}{m_{IVC}}\right)}{m_f + m_a + m_{EGR} - m_{sc} + \frac{m_{bb}}{2} \cdot \left(\frac{m_{res}}{m_{IVC}} - 1\right)} \\ &= \frac{m_{a,b} + m_{f,b}}{m_f + m_a + m_{EGR} - m_{sc} + \frac{m_{bb}}{2} \cdot \left(\frac{m_{res}}{m_{IVC}} - 1\right) - \frac{m_a + m_f + m_{EGR} - m_{sc} - m_{bb}}{m_f + m_a + m_{EGR} - m_{bb} - m_{sc} \cdot T_{EGR}}} \\ &= \frac{m_{a,b} + m_{f,b}}{(m_{EGR} - m_{sc} \cdot T_{EGR}) \cdot \left(1 - \frac{m_{bb}}{2 \cdot m_{IVC}}\right)} \end{aligned} \tag{42}$$

$$m_{a,b} = \frac{m_{f,b}}{AFR_{st}} \tag{43}$$

$$m_{f,b} + m_{a,b} = m_{f,b} \cdot \left(1 + \frac{1}{AFR_{st}}\right) = m_{f,b} \cdot \left(\frac{AFR_{st} + 1}{AFR_{st}}\right) \tag{44}$$

$$\begin{aligned} Y_b^{EVO} &= \frac{m_{f,b} \cdot \left(\frac{AFR_{st}+1}{AFR_{st}}\right)}{m_a + m_f + m_{EGR} - m_{sc} - \left(\frac{m_a + m_f + m_{EGR} - m_{sc}}{m_a + m_f + m_{EGR} - m_{sc} \cdot T_{EGR}}\right) \cdot (m_{EGR} - m_{sc} \cdot T_{EGR})} \\ &= \frac{m_{f,b} \cdot \left(\frac{AFR_{st}+1}{AFR_{st}}\right) \cdot (m_a + m_f + m_{EGR} - m_{sc} \cdot T_{EGR})}{(m_a + m_f + m_{EGR})^2 - (m_a + m_f + m_{EGR}) \cdot m_{sc} - (m_a + m_f + m_{EGR}) \cdot m_{EGR} + m_{EGR} \cdot m_{sc}} \\ &= \frac{m_{f,b} \cdot \left(\frac{AFR_{st}+1}{AFR_{st}}\right) \cdot (m_a + m_f + m_{EGR} - m_{sc} \cdot T_{EGR})}{(m_a + m_f) \cdot (m_a + m_f + m_{EGR} - m_{sc})} \end{aligned} \tag{45}$$

$$Y_b^{EVO} = \frac{m_{f,b} \cdot \left(\frac{AFR_{st}+1}{AFR_{st}}\right)}{m_a + m_f + m_{EGR} - m_{sc} - \left(\frac{m_a + m_f + m_{EGR} - m_{sc}}{m_a + m_f + m_{EGR} - m_{sc} \cdot T_{EGR}}\right) \cdot (m_{EGR} - m_{sc} \cdot T_{EGR})} \tag{46}$$

A.3 DoEs definition

Table A.1. *DoE 1 configuration.*

	P_{int} [bar]	P_{exh} [bar]	ΔP [bar]	VVT_{int} [cad]	VVT_{exh} [cad]	$Olap$ [cad]
1	2.5	1.8	0.7	10	15	70
2	2.4	1.8	0.6	7.5	7.5	65
3	2.4	1.8	0.6	7.5	17.5	75
4	2.4	1.6	0.8	7.5	7.5	65
5	2.4	1.6	0.8	7.5	17.5	75
6	2.4	1.8	0.6	12.5	12.5	65
7	2.4	1.8	0.6	12.5	22.5	75
8	2.4	1.6	0.8	12.5	12.5	65
9	2.4	1.6	0.8	12.5	22.5	75
10	2.6	2.0	0.6	7.5	7.5	65
11	2.6	2.0	0.6	7.5	17.5	75
12	2.6	1.8	0.8	7.5	7.5	65
13	2.6	1.8	0.8	7.5	17.5	75
14	2.6	2.0	0.6	12.5	12.5	65
15	2.6	2.0	0.6	12.5	22.5	75
16	2.6	1.8	0.8	12.5	12.5	65
17	2.6	1.8	0.8	12.5	22.5	75
18	2.7	2.0	0.7	10	15	70
19	2.3	1.6	0.7	10	15	70
20	2.5	1.8	0.7	15	20	70
21	2.5	1.8	0.7	5	10	70
22	2.5	1.6	0.9	10	15	70
23	2.5	2.0	0.5	10	15	70
24	2.5	1.8	0.7	10	25	80
25	2.5	1.8	0.7	10	5	60

Table A.2. *DoE 2 configuration.*

	P_{int} [bar]	P_{exh} [bar]	ΔP [bar]	VVT_{int} [cad]	VVT_{exh} [cad]	$Olap$ [cad]
1	3.9	2.350	1.55	10	15	70
2	3.8	2.350	1.45	7.5	7.5	65
3	3.8	2.350	1.45	7.5	17.5	75
4	3.8	2.150	1.65	7.5	7.5	65
5	3.8	2.150	1.65	7.5	17.5	75
6	3.8	2.350	1.45	12.5	12.5	65
7	3.8	2.350	1.45	12.5	22.5	75
8	3.8	2.150	1.65	12.5	12.5	65
9	3.8	2.150	1.65	12.5	22.5	75
10	4	2.550	1.45	7.5	7.5	65
11	4	2.550	1.45	7.5	17.5	75
12	4	2.350	1.65	7.5	7.5	65
13	4	2.350	1.65	7.5	17.5	75
14	4	2.550	1.45	12.5	12.5	65
15	4	2.550	1.45	12.5	22.5	75
16	4	2.350	1.65	12.5	12.5	65
17	4	2.350	1.65	12.5	22.5	75
18	4.1	2.550	1.55	10	15	70
19	3.7	2.150	1.55	10	15	70
20	3.9	2.350	1.55	15	20	70
21	3.9	2.350	1.55	5	10	70
22	3.9	2.150	1.75	10	15	70
23	3.9	2.550	1.35	10	15	70
24	3.9	2.350	1.55	10	25	80
25	3.9	2.350	1.55	10	5	60

A.4 DoEs detailed results

Table A.3. *Results of DoE 1 - IGR rates.*

	WAM	Simple Est.	*Diff.*	2k Est. k_{int} = 1.35	*Diff.*	2k Est. k_{int} = 1.30	*Diff.*
1	22.08 %	26.12 %	*4.04*	20.56 %	*-1.52*	22.02 %	*-0.06*
2	25.50 %	27.44 %	*1.94*	24.22 %	*-1.28*	25.62 %	*0.12*
3	21.72 %	25.46 %	*3.74*	20.05 %	*-1.67*	21.32 %	*-0.40*
4	20.93 %	25.82 %	*4.89*	19.44 %	*-1.49*	21.07 %	*0.14*
5	18.39 %	23.86 %	*5.47*	16.78 %	*-1.61*	18.15 %	*-0.24*
6	24.91 %	27.95 %	*3.04*	23.99 %	*-0.92*	25.41 %	*0.50*
7	21.29 %	25.66 %	*4.37*	19.81 %	*-1.48*	21.10 %	*-0.19*
8	20.66 %	25.96 %	*5.30*	19.24 %	*-1.42*	20.91 %	*0.25*
9	17.96 %	23.93 %	*5.97*	16.41 %	*-1.55*	17.83 %	*-0.13*
10	27.20 %	28.48 %	*1.28*	26.26 %	*-0.94*	27.63 %	*0.43*
11	20.27 %	24.95 %	*4.68*	18.64 %	*-1.63*	19.99 %	*-0.28*
12	23.13 %	26.85 %	*3.72*	21.67 %	*-1.46*	23.28 %	*0.15*
13	20.11 %	24.94 %	*4.83*	18.57 %	*-1.54*	19.93 %	*-0.18*
14	28.10 %	29.06 %	*0.96*	27.18 %	*-0.92*	28.50 %	*0.40*
15	23.59 %	26.78 %	*3.19*	22.28 %	*-1.31*	23.51 %	*-0.08*
16	22.81 %	26.67 %	*3.86*	21.13 %	*-1.68*	22.64 %	*-0.17*
17	19.96 %	25.14 %	*5.18*	18.63 %	*-1.33*	20.02 %	*0.06*
18	23.86 %	27.20 %	*3.34*	22.71 %	*-1.15*	24.13 %	*0.27*
19	20.29 %	25.24 %	*4.95*	18.67 %	*-1.62*	20.15 %	*-0.14*
20	21.96 %	26.52 %	*4.56*	20.69 %	*-1.27*	22.18 %	*0.22*
21	22.23 %	26.00 %	*3.77*	20.67 %	*-1.56*	22.11 %	*-0.12*
22	18.44 %	24.76 %	*6.32*	16.97 %	*-1.47*	18.61 %	*0.17*
23	27.30 %	28.16 %	*0.86*	26.01 %	*-1.29*	27.20 %	*-0.10*
24	20.24 %	24.39 %	*4.15*	18.58 %	*-1.66*	19.73 %	*-0.51*
25	26.80 %	28.42 %	*1.62*	25.64 %	*-1.16*	27.19 %	*0.39*

Table A.4. *Results of DoE 2 - IGR rates.*

	WAM	Simple Est.	*Diff.*	2k Est. k_{int} = 1.35	*Diff.*	2k Est. k_{int} = 1.30	*Diff.*
1	20.10 %	24.20 %	*4.10*	20.69 %	*0.58*	20.39 %	*0.29*
2	20.77 %	24.59 %	*3.82*	21.40 %	*0.63*	21.29 %	*0.52*
3	19.94 %	23.60 %	*3.65*	20.05 %	*0.10*	19.59 %	*-0.36*
4	19.63 %	24.06 %	*4.44*	20.76 %	*1.13*	20.28 %	*0.65*
5	18.95 %	22.97 %	*4.03*	19.45 %	*0.50*	18.70 %	*-0.24*
6	20.42 %	24.85 %	*4.42*	20.92 %	*0.50*	21.08 %	*0.65*
7	19.37 %	23.75 %	*4.38*	19.42 %	*0.05*	19.17 %	*-0.20*
8	19.31 %	24.31 %	*5.00*	20.32 %	*1.00*	20.10 %	*0.79*
9	18.51 %	23.15 %	*4.65*	18.93 %	*0.42*	18.41 %	*-0.10*
10	21.73 %	25.20 %	*3.48*	22.28 %	*0.55*	22.30 %	*0.57*
11	20.86 %	24.20 %	*3.34*	20.90 %	*0.04*	20.53 %	*-0.33*
12	20.56 %	24.65 %	*4.09*	21.62 %	*1.06*	21.25 %	*0.70*
13	19.80 %	23.59 %	*3.79*	20.28 %	*0.48*	19.60 %	*-0.20*
14	21.46 %	25.49 %	*4.03*	21.86 %	*0.40*	22.15 %	*0.69*
15	20.34 %	24.35 %	*4.01*	20.31 %	*-0.03*	20.15 %	*-0.20*
16	20.22 %	24.90 %	*4.69*	21.16 %	*0.95*	21.06 %	*0.85*
17	19.36 %	23.75 %	*4.39*	19.74 %	*0.38*	19.29 %	*-0.07*
18	20.93 %	24.78 %	*3.85*	21.47 %	*0.54*	21.26 %	*0.34*
19	19.55 %	23.64 %	*4.09*	19.90 %	*0.35*	19.55 %	*0.01*
20	19.61 %	24.29 %	*4.67*	20.07 %	*0.46*	19.99 %	*0.38*
21	20.40 %	24.01 %	*3.61*	21.13 %	*0.73*	20.59 %	*0.19*
22	19.17 %	23.70 %	*4.53*	20.14 %	*0.97*	19.55 %	*0.38*
23	21.15 %	24.80 %	*3.65*	21.28 %	*0.13*	21.31 %	*0.16*
24	19.24 %	22.90 %	*3.66*	19.09 %	*-0.15*	18.40 %	*-0.84*
25	21.06 %	25.12 %	*4.06*	21.92 %	*0.86*	21.97 %	*0.91*

Chapter 4

Investigation of the gasoline PPC concept in a 2-Stroke poppet-valve HSDI CI engine

Contents

4.1 Introduction

The first part of the doctorate work was focused on the implementation and understanding of the gasoline Partially Premixed Combustion (PPC) concept using a 2-Stroke poppet-valve HSDI CI engine. This engine had previously been the subject of another doctoral thesis [1] where a Conventional Diesel Combustion (CDC) implemented on a 2-Stroke (derived from a manufacture 4-Stroke Renault engine) and the potential for advanced diesel-like, Low Temperature Combustion (LTC) concepts were evaluated, within the framework of the European POWERFUL project. The work presented in the following chapter is then a continuity of these investigations, taking advantage of the possibilities offered by the thermal loop of the 2-Stroke architecture for LTC –and more particularly gasoline PPC– for control and stability compared to 4-Stroke engines.

The investigations carried out along this project were mainly focused on understanding the development of the gasoline PPC combustion, the influence of the injection and air loop parameters and their possible interactions, and learning paths for controlling both performance and pollutant emissions, mainly through the control of the Rate of Heat Release (RoHR). In this framework, an intensive campaign have been carried out to study the effects of the injection settings, performing experimental parametric studies to characterize the sensitivity and the potential of the combustion throughout a large range of operating conditions (1250 - 2500 rpm, 3 - 10.4 bar IMEP), and using CFD models to identify the local in-cylinder conditions before the onset of combustion. For the purpose of this thesis, only part of these investigations are presented here, as part of this work has been performed in the early stage of the doctorate and some of these results were already presented in the previous PhD student's thesis [1]. However, an extensive study (presented here) has been performed to map the lower part of the engine operating range, from 3 to 5.5 bar IMEP on a wide range of engine speed, to identify the best engine settings to lower fuel consumption to a minimum and compare the combustion outputs (performance and emissions) to an equivalent commercial 4-Stroke engine operating under conventional Diesel combustion.

Once the detailed principles of the combustion identified, a dedicated optimization process has also been performed to try and break the different trade-offs presented by this combustion concept (mainly the NO_x / soot trade-off, but also newly arisen ones, such as noise / emissions / efficiency trade-offs at some operating conditions). For those purposes, combinations between air management and injection settings had to be further investigated, and a backward optimization of the RoHR has been carried out, which would later be experimentally implemented and validated using the knowledge acquired beforehand over the combustion profile control.

4.2 Experimental equipment and methodology

The engine used during the studies performed along this chapter is a 2-Stroke poppet-valve HSDI compression ignition engine developed originally in the framework of the European POWERFUL project. As mentioned previously, a large amount of investigation works have been previously implemented on this engine under CDC conditions as the subject of a separated doctoral thesis [1], along with various technological patents and scientific publications describing both the design process of the engine itself and the CDC investigations [2–6]. The main characteristics of the engine are described in detail in Chapter 3, section 3.2.

The main objective of the project presented in this chapter is to evaluate the potential of the gasoline Partially Premixed Combustion (PPC) concept. As described previously in Chapter 2, the 2-Stroke engine architecture in which this concept is implemented provides adequate ignition conditions for low-reactivity fuels (such as gasoline) even at low loads, due to the higher combustion frequency inducing greater in-cylinder temperatures than what can be achieved in 4-Stroke engines. However, these conditions need to be well-managed in order to control both the rate and the phasing of the heat release. Indeed, too much premixing, or too high temperatures, can generate early and abrupt combustion damageable (or even fatal) for the engine; on the contrary, low premixed conditions may results in bad combustion performances (in terms of efficiency and/or pollutant emissions) or even misfire cycles, thus losing all the interest of the combustion concept itself.

The engine operating condition selected for the first step of these investigations corresponds with a medium speed (1500 rpm) and medium-to-high load (10.4 bar of IMEP) operating point. This corresponds to a baseline case at which most of the preliminary studies will be performed providing that the operating point offers sufficient combustion stability to allow extensive sweeps of the largest number of testing parameters; but it is worth mentioning that the behaviors and conclusions obtained here could be validated and in-depth understood using 3D-CFD as reported in [7], and experimentally confirmed on the majority of the engine operating range. The baseline case is set with a fixed fueling rate of 18.8 mg/stroke to achieve the targeted load with a CA50 of 5 CAD aTDC, and all the studies performed afterward at

Table 4.1. *Experimental test conditions and emissions and fuel consumption reference levels from CDC operation..*

Engine speed	1500 rpm
IMEP	10.4 bar (baseline)
Injected fuel quantity	18.8 mg/stroke
Intake air temperature	35°C
Coolant and oil temperature	90°C
NO_x reference (CDC)	2.13 mg/s
Smoke reference (CDC)	2.99 FSN
HC reference (CDC)	0.36 mg/s
CO reference (CDC)	13.02 mg/s
Noise reference (CDC)	86.4 dB
Indicated efficiency (CDC)	43.4%
ISFC (CDC)	196.6 g/kWh
$ISFC_{corr}$ (CDC)	238.8 g/kWh

this operating point were set according to the same inputs. The intake air temperature was fixed at 35°C, and the temperatures of oil and coolant were regulated at 90°C. Table 4.1 summarizes the main experimental test conditions and provides the pollutant emissions and fuel consumption levels obtained at this baseline case during prior investigations while operating with the CDC concept, with engine settings optimized for decreasing NO_x emissions [1, 3–6].

While previously operating this baseline case with the current engine hardware and performing with the CDC concept, a dedicated Design of Experiment (DoE) optimization methodology associated with mathematical models of several engine responses was used to select preliminary values for the most important air management settings [4–6]. The optimum air management settings were defined through the available statistical models, thus providing the desired O_2concentration at IVC and temperature profile required for a proper auto-ignition of the charge mixture around TDC.

All the studies presented in this research are performed using a triple injection strategy. The baseline case then presents a very small first injection placed around -60 CAD aTDC, a main second injection around -40 CAD aTDC providing most of the fuel quantity necessary to reach the targeted charge, and finally a small third injection close to TDC, shortly before the Start of Combustion (SoC). For this investigation, the injection timings refer to the Start of Energizing (SoE) current of the injector instead the actual Start of Injection (SoI), which happens a few crank-angle degrees (1.5 to 2 CAD) after the SoE due to the hydraulic delay cause by the needle lift.

Table 4.2. *Engine settings for experiments at 1500 rpm - 10.4 bar IMEP.*

Test	EGR (%)	P_{int} (bar)	ΔP (bar)	Overlap (CAD)	$VVT_{int,exh}$ (CAD)	P_{rail} (bar)	SoE1 (CAD)	SoE2 (CAD)	SoE3 (CAD)	Fuel Ratio (%)
1	43.5	2.75	0.71	78.4	(5,20)	850	**[-66 to -54]**	-40	-2	20/63/17
2.1	43.5	2.75	0.71	78.4	(5,20)	850	-60	**[-42 to -34]**	-2	20/63/17
2.2	43.5	2.75	0.71	78.4	(5,20)	850	-60	**[-44 to -36]**	-4	20/63/17
2.3	43.5	2.75	0.71	78.4	(5,20)	850	-60	**[-44 to -40]**	-6	20/63/17
3	43.5	2.75	0.71	78.4	(5,20)	850	-60	-40	**[-8 to 2]**	20/63/17
4.1	43.5	2.75	0.71	78.4	(5,20)	**750**	-60	[-44 to -36]	-2	20/63/17
4.2	43.5	2.75	0.71	78.4	(5,20)	**950**	-60	[-40 to -34]	-2	20/63/17
5.1	43.5	2.75	0.71	78.4	(5,20)	850	-60	[-44 to -40]	-2	**20/69/11**
5.2	43.5	2.75	0.71	78.4	(5,20)	850	-60	[-38 to -34]	-2	**20/56/24**

The range for sweeping the injection timing was predefined with the aid of the CFD model by performing parametric studies of each of the three injections. Later on, the trends observed in the calculations were experimentally validated by measuring the defined test plan directly in the engine. However, the range of SoE experimentally measured was reduced to a narrower operating window which was limited in one side by knocking-like combustion and in the other by misfire conditions.

Once the effects of the injection timings identified, the effect of injection pressure and fuel mass distribution over the performances of the PPC concept was evaluated by performing additional sweeps of the second injection timing, for two different pressure levels (higher and lower) and different fuel distribution (between 2nd and 3rd injection events) compared to the baseline case. The most relevant engine settings chosen for each parametric variation are detailed in Table 4.2 (top row represents the baseline case).

4.3 Analysis of the gasoline PPC concept

The main air management settings selected for the gasoline PPC operation are displayed in Table 4.2. It can be observed that these values correspond to considerably higher levels of EGR rate (43.5%), intake pressure (2.75 bar), ΔP (0.71 bar), and overlap duration (78.4 CAD) compared to those obtained when operating in CDC (presented in detail in previous works [3–6]). With this air management settings combination, a TR of 67%, a delivered flow of 67 kg/h and an IGR ratio of 35% are obtained, along with an in-cylinder global equivalence ratio (Φ_{cyl}) of 0.83, temperature and O_2concentration at IVC (T_{IVC} and $YO_{2,IVC}$) of 180°C and 12% respectively, and 4% of O_2concentration at EVO ($YO_{2,EVO}$). Experimental results have demonstrated that the impact of the injection settings over the air management characteristics is very limited, thus the most important gas cylinder conditions remain unaltered regardless of the SoE or the injection pressure.

4.3.1 Effect of 1st injection timing

For the baseline case (SoE1 -60, SoE2 -40 and SoE3 -2 CAD aTDC), CFD was used to confirm that injecting only a small fraction of the total fuel mass in the very early 1st injection can allow to avoid liquid fuel interaction with the cylinder walls, even when using injector with a 148° spray included angle combined to a relatively high injection pressure (850 bar). Moreover, the results displayed in Figure 4.1 (for SoE1 varying between -66 and -54 CAD aTDC) demonstrate the negligible effect that the 1st injection timing has over the combustion onset and RoHR. Additionally, even though the earliest SoE1 settings slightly increase the HC emissions, the rest of the exhaust emissions are barely affected by this injection timing sweep, as shown in Figure 4.2. Thus, the focus of the further analysis will be mainly put on the description of the effects of the 2nd and 3rd injections timings.

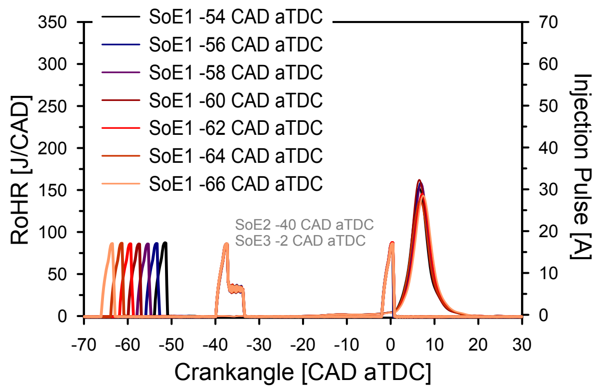

Figure 4.1. *Effect of SoE1: RoHR and injection pulse for experimental results.*

The main application of this early injection is then mainly to provide the required amount of fuel to sustain the demanded IMEP, avoiding to inject this fuel in either of the other events where it could interfere with the combustion conditions and pollutants formation. Its only side effect being the HC generation, the event then has to be optimized consequently inside the medium-to-high load range.

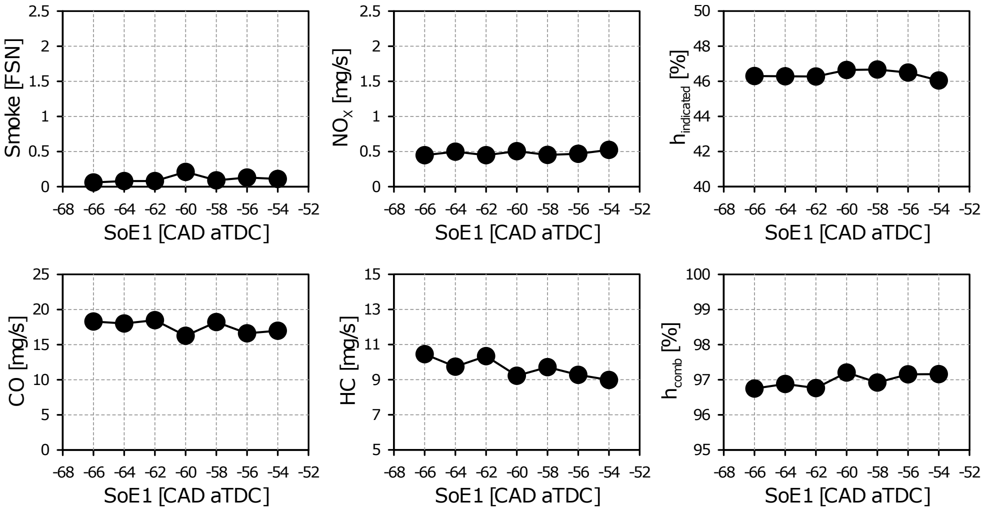

Figure 4.2. *Effect of SoE1: Exhaust emissions and efficiencies for experimental results.*

4.3.2 Effect of 2nd injection timing

The impact of the second injection timing (SoE2) on the combustion profile (RoHR) is shown in Figure 4.3 for CFD model (left plot) and experimental results (right plot). For the CFD calculation, SoE2 was swept from -34 to -50 CAD aTDC for the reference sweep with SoE3 in -2 CAD aTDC, while in the case of experimental results the parametric variation stops at SoE2 -42 CAD aTDC because of the rapid loss of combustion stability and appearance of misfire conditions for earlier timings. Both CFD and experimental results demonstrate that this main injection event controls both the onset of combustion and its phasing (represented by the CA50). Late SoE2 advances both SoC and CA50 toward TDC, as shown in Figures 4.3 and 4.4, rapidly approaching toward knocking-like conditions with very high pressure gradients and noise levels, respectively above 20 bar/CAD and 100 dB; on the contrary, early SoE2 shifts both SoC and CA50 towards the expansion stroke, causing combustion to become smother (low levels of noise and pressure gradients) but misfire trending with rapidly increasing values of covariance (CoV $P_{max} > 2\%$). These trends have been observed at different operating conditions and it is well-understood as it is explained by considering the impact of SoE2 over the local equivalence ratio distribution just before the onset of combustion [7, 8]. The potential of SoE2 to control the combustion profile and manage the combustion noise and applications of this behavior have been also investigated and will be exposed in details in section 4.4 on page 117.

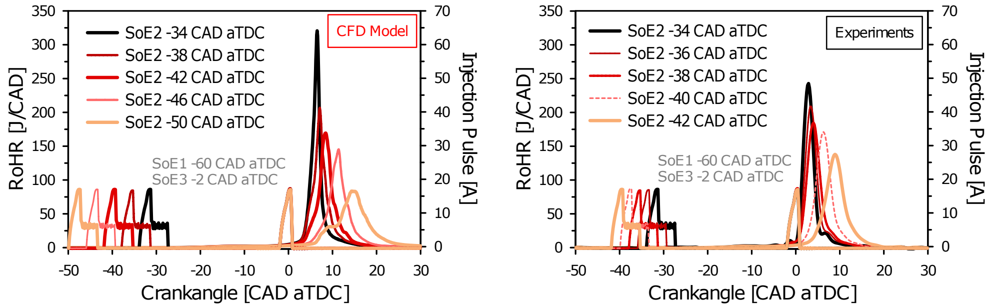

Figure 4.3. *Effect of SoE2: RoHR and injection pulse for CFD (left plot) and for experimental results (right plot).*

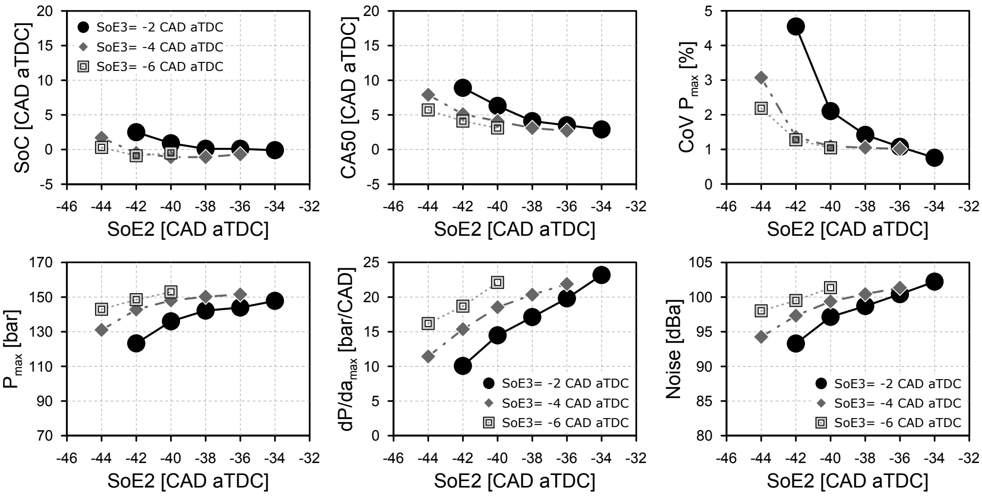

Figure 4.4. *Effect of SoE2: Global combustion parameters for experimental results.*

It is interesting to get a better understanding of how the local in-cylinder conditions along the combustion process affect the exhaust emissions formation / destruction. Therefore, Figure 4.5 shows the experimental results of the SoE2 sweep in terms of exhaust emissions. Comparing with a well-optimized CDC results (see reference values in Table 4.1), PPC proves to reduce significantly the NO_x and soot emission levels, while CO and HC emissions increase. In the PPC concept, NO_x formation is controlled by keeping the combustion temperature below 2400 K using high EGR rates. On the counterpart, high temperature and rich equivalence ratio regions promote the soot formation; therefore when using a triple injection strategy, the 3^{rd} event is responsible for most of the soot emissions by increasing local equivalence ratios just before the SoC.

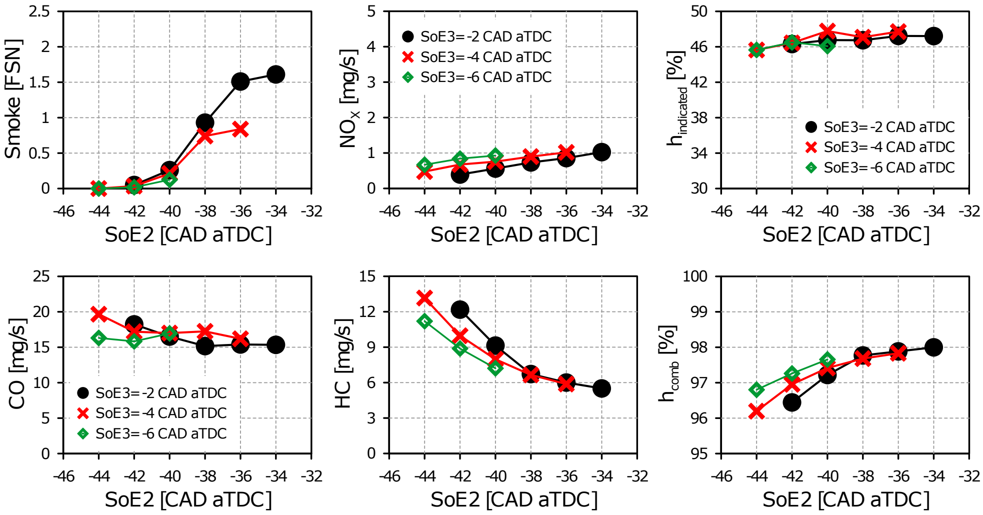

Figure 4.5. *Effect of SoE2. Exhaust emissions and efficiencies for experimental results.*

Then, advancing SoE2 helps in reducing the NO_x emissions by retarding and softening the RoHR profile with lower combustion temperatures; while the extended ignition delay –and therefore increased mixing time available for the 3^{rd} injection– induces lower PM emissions, separating this late injection from combustion being key to avoid soot formation. However, CO and HC emissions increase with early SoE2 and delayed CA50, as showed in Figure 4.5, so a clear trade-off appears between the levels of NO_x -soot and HC-CO that is worth to be further investigated.

To improve the comprehension of $CO$and NO_x trends, a dedicated CFD campaign has been performed early during the project to try and visualize the evolution (generation and oxidation) of those two species as function of the crankangle and the equivalence ratio, along with their spacial distribution inside the cylinder, for different cases of injection settings.

The experimental results displayed in Figure 4.5 also highlight a sharp increase of HC and CO emissions, and the consequent reduction in combustion efficiency, when advancing the SoE2. There again, the CFD simulations performed earlier help to understand how early injection events generate fuel-wall interactions and how they affects the combustion process and outputs [1, 7]. It is however worth to note that this triple injection strategy associated with this fuel distribution settings allow to keep most of the fuel from the 1st injection away from the cylinder walls thus avoiding impingement; whereas the late 3rd event is injected completely inside the bowl and does not contribute to the liquid fuel / cylinder walls interactions.

When comparing with the CDC operation, combustion efficiency from the gasoline PPC operation is relatively lower (thus higher HC and CO emissions), even if it still remains over 96% and 97% which is in a similar range compared to what can be seen in the literature [9, 10]. Finally, ISFC ranges between 181 to 178 g/kWh (46.5% to 47.5% in indicated efficiency), which represents a 10% decrease compared to the optimum point when operating with the CDC concept. Despite the clear benefits in indicated efficiency, they are counter-balanced by the increased mechanical power demanded by the air loop devices, resulting in an ISFC$_{corr}$ kept at similar levels to those obtained with the CDC concept (between 237 and 241 g/kWh) because of the additional supercharger work required to reach satisfactory equivalence ratio range and external EGR rate combinations necessary for proper PPC operation.

The knowledge acquired during these experimental and numerical investigations can be directly implemented when calibrating the engine, as a very fine tuning of the 2nd injection is mandatory to fully map the engine for real condition applications. It controls directly the overmixed lean zones where non-oxidizable CO is formed, and its impact on the SoC also influences the stratification of the 3rd injection.

4.3.3 Effect of 3rd injection timing

The effect of SoE3 on the RoHR profile is shown in Figure 4.6 for CFD simulations (left plot) and experimental results (right plot). In CFD simulations, the SoE3 was swept from -12 to +8 CAD aTDC, while in the experiments it was swept from -8 to +2 CAD aTDC, being limited by the appearance of knocking-like combustion on the early SoE3 side, and the very high soot emissions on the late SoE3 side. Both CFD and experimental results evidence that the SoC is mainly controlled by the SoE2, and barely affected by the SoE3. On the contrary, this last parameter has a large influence on the development of the combustion process, reflected by the RoHR profiles. The increased ignition delay and the mixing time induced by early SoE3 allow partial mixing of this last portion of fuel injected, thus increasing the reactivity of the global mixture prior to the SoC and promoting knocking conditions for the combustion, reflected by the sharp and fast RoHR profile observed in Figure 4.6. Whereas a delayed SoE3 has an opposite effect, shortening the ignition delay and the available mixing time. The fuel injected in this 3rd injection then burns similarly to a mixing-controlled process, instead of a highly premixed process observed previously, although with a critical impact on pollutant emissions that will be discussed later in this section.

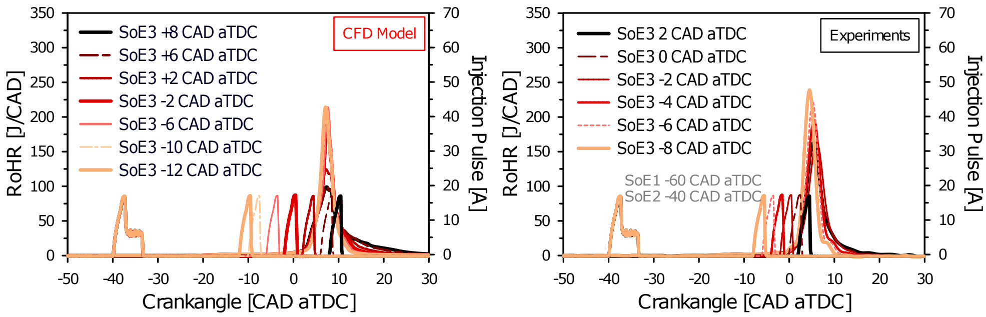

Figure 4.6. *Effect of SoE3. RoHR and injection pulse and for CFD (top plot) and experimental results (bottom plot).*

The experimental results shown in Figure 4.7 corroborates the critical impact of early SoE3 settings over P_{max}, dP/da_{max} and noise, reaching extremely high levels for the earliest SoE3 (150 bar, over 20 bar/CAD and 100 dB, respectively, in this case). However, the opposite trends is not observed: delayed SoE3 compared to the baseline case only has a moderate effect on those three parameters despite the extended combustion duration. Finally, the CoV P_{max} (which is a commonly used parameter for combustion stability diagnosis and misfire rate evaluation) is not affected by timing of SoE3 and the combustion remains stable in every cases, the cycle-to-cycle dispersion (in terms of both SoC and CA50) being mostly controlled by the 2nd injection.

Once again, the sensitivity of the combustion process to the latest two injection events has been confirmed by the CFD simulations, through the analysis of the local equivalence ratio distribution at CA10 (which is usually a good tracer of the SoC timing) [1, 7]. This distribution is an image of the global homogeneity of the mixture, and thus its global reactivity. Then, knowingly

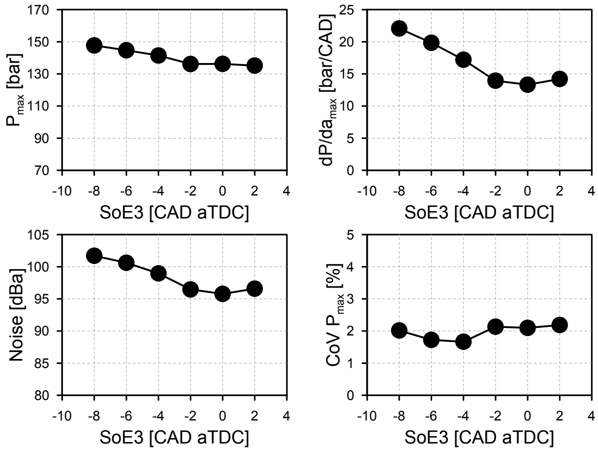

Figure 4.7. *Effect of SoE3. Global combustion parameters for experimental results.*

adjusting SoE2 and SoE3 allow to control this reactivity and the onset and sharpness of the combustion profile.

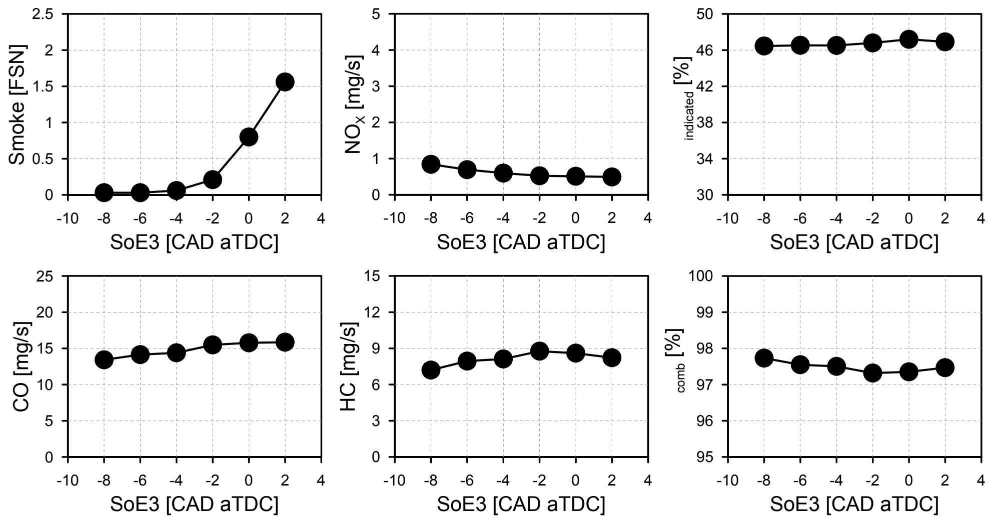

Figure 4.8. *Effect of SoE3. Exhaust emissions and efficiencies for experimental results.*

In terms of exhaust emissions, Figure 4.8 confirms the NO_x reduction induced by the slightly lower local temperatures (from a smoother and longer combustion process) observed when delaying SoE3. However, the extended mixing-controlled stage caused by the reduced mixing time for this delayed 3rd event promotes the soot formation by generating locally rich areas, thus recovering the NO_x-soot trade-off characteristic of the CDC concept. This is corroborated by the trend followed by CO emissions since their oxidation into CO_2 is worsened by the shifted combustion towards the expansion stroke when retarding SoE3. On the other hand, HC emissions remained almost constant since they are mostly influenced by the 1st and 2nd injections. Figure 4.8 also shows that neither combustion nor indicated efficiencies are significantly affected by SoE3, staying at very stable levels (respectively over 97% and 47%), which is also confirmed by the ISFC level: it only slightly punished by the fast and short (close to knocking conditions) combustion generated by SoE3, going from 178 g/kWh to 181 g/kWh, with a direct repercussion on $ISFC_{corr}$ which jumps from 236 g/kWh to 242 g/kWh.

The direct application for this late event is the control the NO_x-soot trade-off by adjusting the mixing time, and thus the local richness of the mixture. It also influences the development of the combustion by affecting the global mixture reactivity, which can help to manage the pressure gradient and noise, as it will be seen later in this document.

4.3.4 Effect of injection pressure

After discussing the critical impact of local mixing conditions on the combustion development and outputs (performance and emissions), two different levels of P_{rail} (+/- 100 bar compared to the baseline setting) were tested to investigate the effect of the mixing rate while operating with the gasoline PPC concept. Figure 4.9 shows the RoHR for the SoE2 sweeps corresponding to the two additional levels of P_{rail}, as well as a direct comparison of the RoHR and injection rate obtained at the same SoE2 (-40 CAD aTDC) for the three injection pressures (4.9-c).

According to Figure 4.9, misfire cycles were detected for SoE2 set as early as -40 CAD aTDC in the case of the higher P_{rail} of 950 bar. From the previously generated knowledge it is now clear how this misfire tendency is the result of the faster mixing rates that shift the local equivalence ratio distribution towards leaner conditions together with the higher spray momentum flux that pushes the spray further towards the squish region. The SoE2 sweeping window is then significantly shortened by this increased mixing rate generated by the high

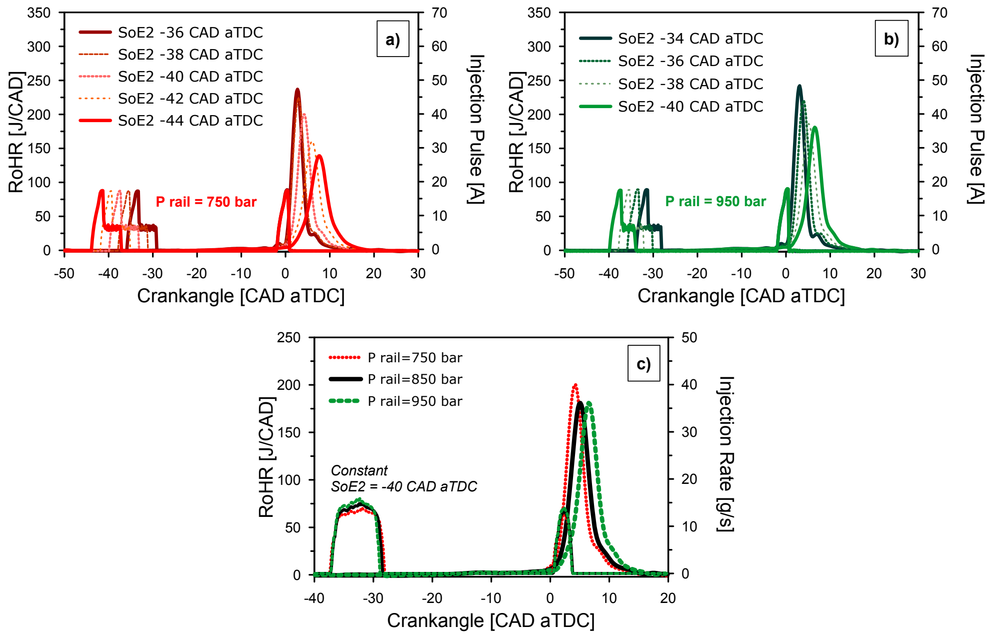

Figure 4.9. *RoHR and inj. pulse for SoE2 sweep at P_{rail} 750 (a) and 950 bar (b). Comparison between injection rate and RoHR profiles (c).*

P_{rail}, because of the reduced operation range available between knocking-like combustion and misfire.

On the counterpart, Figure 4.9-c shows the opposite effect induced when lowering P_{rail} to 750 bar, with both SoC and CA50 shifted earlier in the cycle, generating a shorter and sharper combustion profile. This is the direct consequence of the slower mixing rate of the 2nd injection which increased the portion of fuel mass located in the reactive equivalence ratio zone (between 0.6 and 0.9) consequently promoting the knocking conditions. Thus, Figure 4.10 confirms how P_{max}, dP/da_{max} and noise levels are increase for the lower P_{rail} level compared to the two other cases. Additionally, the trend followed by CoV P_{max} also confirms the combustion stability improvement provided by decreasing P_{rail} that allows to advance the 2nd injection up to -44 CAD aTDC without misfire. This result is clearly supported by the sharp CoV P_{max} increment observed at SoE2 -42 CAD aTDC from P_{rail} 750 bar to 850 bar, in fact, testing this SoE2 at P_{rail} 950 bar was not possible due to the high rate of misfiring cycles.

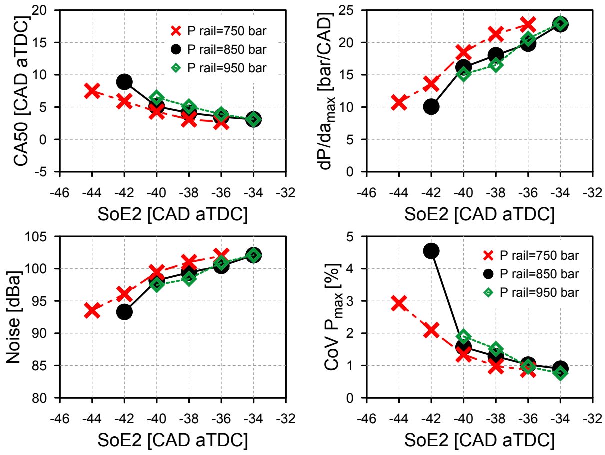

Figure 4.10. *Effect of injection pressure. Global combustion parameters for experimental results.*

Regarding the exhaust emissions, the effect of higher P_{rail} benefits to smoke formation compared to the baseline and low P_{rail} case, as displayed in Figure 4.11, allowed firstly by the faster mixing rate and secondly by the extended available mixing time of the 2nd and 3rd injections induced by the delayed SoC. On the counterpart, the increased spray penetration of the higher P_{rail} slightly punishes HC emissions, by worsening the fuel impingement into colder wall regions and squish area. The faster and sharper combustion profile obtained with lowering the P_{rail} induces higher combustion temperature, which has a negative impact on NO_x emissions, but benefits CO by improving its oxidation. Thus in the case of lower P_{rail}, reduced HC and CO emissions mean an improvement of the combustion efficiency. Finally, ISFC and indicated efficiency don't seem to be affected by the P_{rail} level and remain at stable levels for the three cases.

The trends observed during this study are very similar to results highlighted during previous author's researches focused on the fuel repartition between the injection events [11]. Indeed, the objective was to control the RoHR profile in order to improve the noise / emissions / efficiency trade-offs. The injection pressure demonstrates an additional control over the combustion profile, through different paths: higher injection pressure influences greatly the effects of the second injection (generating higher levels of HC) while only affecting the third events by reducing the soot emissions. On the contrary, the fuel distribution proved to affect the main event in reversed ranges (helps reducing CO and HC levels), but also to influence the last injection by reducing both NO_x and soot emissions.

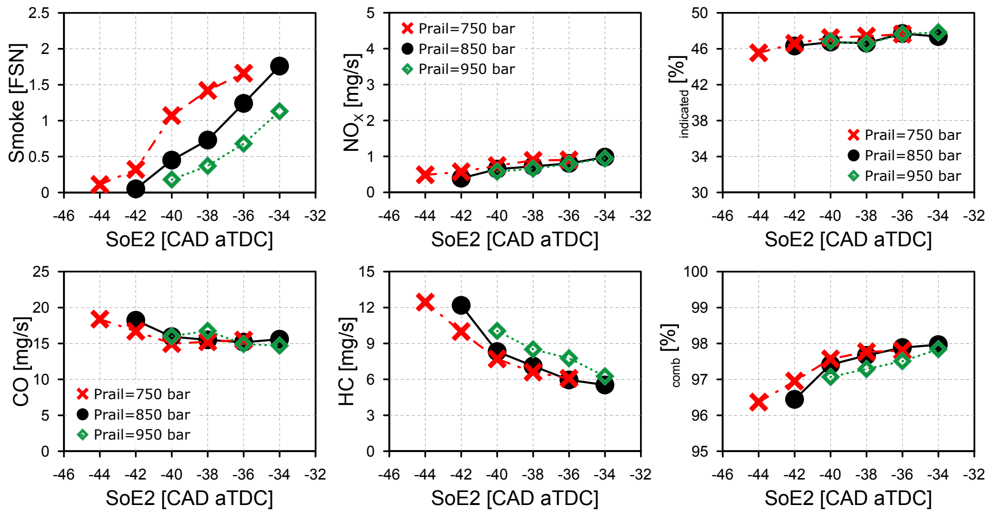

Figure 4.11. *Effect of injection pressure. Exhaust emissions and efficiencies for experimental results.*

The injection pressure effects can be considered as complementary to those of the 2^{nd} injection, as increasing the pressure has similar impacts on combustion and emissions as advancing SoE2. Then, as for the 2^{nd} injection, it needs to be carefully adjusted for an engine mapping application. The sensibility even increases at higher injection pressures, as the operating range between knock and misfire is narrowed.

4.3.5 Effect of fuel distribution

Similarly to the injection timings and rail pressure, the fuel distribution between the three injections might have interesting impacts on the combustion and its outputs. Thus, two additional distributions were experimentally tested to be compared against the baseline (fuel ratio = 20/63/17), and are introduced in Table 4.2. The first step consists on switching part of the fuel from the 3^{rd} injection to the 2^{nd}; while the second step consists on the opposite action and increases the fuel mass injected during the 3^{rd}.

Figure 4.12 shows the RoHR obtained during the SoE2 sweeps with the two additional fuel ratio cases, as well as the comparison RoHR with the baseline

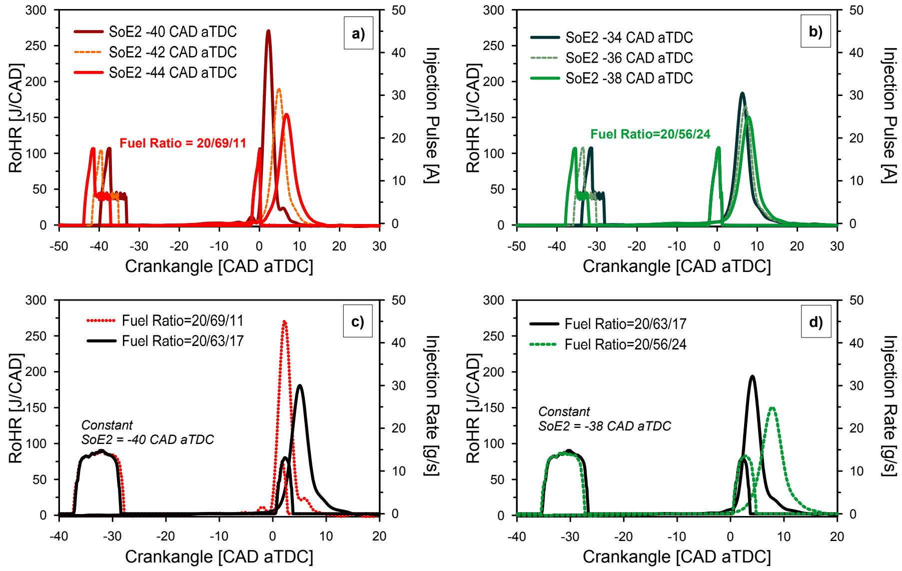

Figure 4.12. *RoHR and injection pulse for SoE2 sweep for fuel ratio equal to a) 20/69/11 and b) 20/56/24. RoHR and injection rate at constant SoE2 c) -40 CAD aTDC and d) -38 CAD aTDC.*

case for a fixed SoE2. In the first case (fuel ratio = 20/69/11), the SoE2 range was limited between -40 and -44 CAD aTDC, as shown in Figure 4.12.a, because of the increased knock trend induced by the higher fuel mass injected during the 2nd event and observed in Figure 4.12.c. Similarly to the effects of delaying SoE2 or lowering P_{rail}, a higher fuel mass in the 2nd injection increases the percentage of fuel mass in the reactive Φ zone, thus advancing both SoC and CA50 closer to TDC. As a consequence, excessive pressure gradients (over 25 bar/CAD) and noise levels (over 100 dB) limited SoE2 to -40 CAD aTDC at the latest.

On the counterpart, the additional fuel injected during the 3rd event when testing the second case of the study has the opposite effect on the mixture reactivity, lowering combustion stability and increasing the tendency to misfire, thus limiting the SoE2 range to -38 CAD aTDC at the earliest. This trend is confirmed by the RoHR profiles displayed in Figure 4.12.d, where the CA50 is noticeably delayed further into the expansion stroke in the case of fuel ratio equal to 20/56/24. As a consequence, the peak pressure, pressure gradient and noise level are substantially lowered, and the mixing-controlled stage of combustion is also extended, with a wider and softer RoHR due to the lower combustion rate.

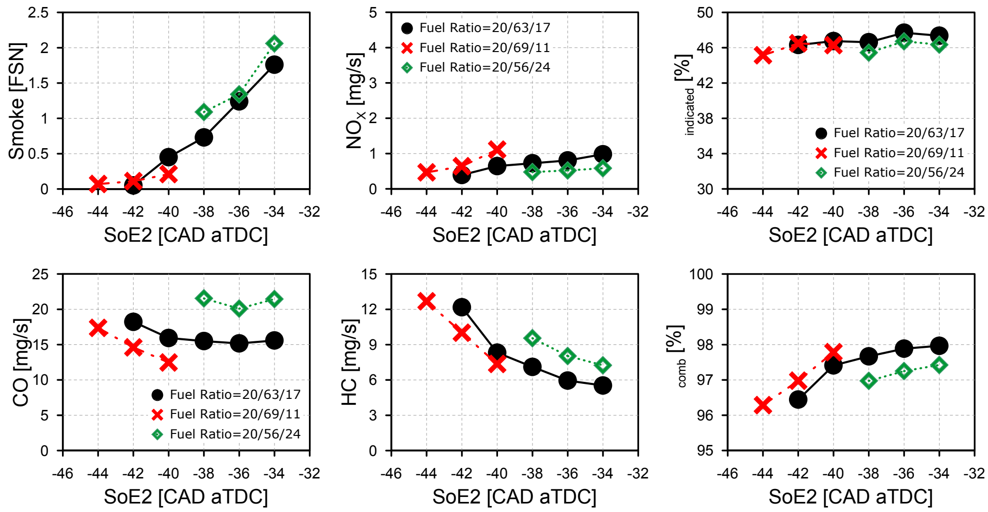

Figure 4.13. *Effect of fuel distribution. Exhaust emissions and efficiencies for experimental results.*

Figure 4.13 shows how the fuel distribution between the 2nd and the 3rd injection affects the pollutant emission levels: injecting more fuel into the 3rd injection promotes soot formation by increasing the local equivalence ratio distribution towards richer conditions before the onset of the combustion, combined with a worsened late soot oxidation process caused by the extended combustion duration; similarly, the delayed CA50 also lowers the energy conversion for CO and HC, thus punishing their oxidation process, resulting in higher levels of those pollutant emissions. On the other hand, the lower combustion temperature given by the later CA50 has the straight effect of reducing NO_x emissions. It is interesting to point out that when comparing at constant CA50, the first case (fuel ratio = 20/69/11) presents higher HC emissions compared to the baseline and the second case (fuel ratio = 20/56/24): this can be explained by the longer and earlier 2nd injection event generating higher fuel impingement into the piston top land and squish areas.

Finally, Figure 4.13 also confirms how the worsened combustion process and extended mixing-controlled phase obtained when more fuel mass is injected late in the cycle punishes both combustion and indicated efficiency.

4.3.6 Conclusions

In this research, a detailed analysis of the multiple injection PPC concept using gasoline as fuel has been carried out by combining experimental and CFD modeling activities. The most relevant conclusions obtained from this investigation can be summarized as following.

According to the results from high load operations, the gasoline PPC concept drastically reduces both NO_x and soot emissions compared to the levels provided by a well optimized CDC concept, leading to a large reduction of the after-treatment demands. Also, the faster and thus shorter combustion observed while operating with this concept, together with lower heat transfer losses, significantly improved the indicated efficiency by around 10%. However, these benefits obtained in indicated efficiency can be partially or even totally lost after considering the power demanded by the air loop devices to achieve the suitable EGR/Φ_{cyl} combination required to implement the gasoline PPC concept. As a counterpart, gasoline PPC operation provides worse results in terms of HC and CO emissions, which is translated in a decrease in combustion efficiency from 99.3% to 96.5% (which is still an acceptable level, especially considering the noticeable amount of HC from fuel-wall interactions induced by early injection events).

The 1$^{\mathrm{st}}$ injection helps to provide the required amount of fuel without affecting the combustion. The analysis carried out by means of CFD modeling confirms how the 2$^{\mathrm{nd}}$ injection event induces a liquid fuel spray/wall impingement for its most advanced timings, resulting in an increased HC formation. The other main drawback of the PPC concept is the high level of noise generated by the fast knocking-like combustion process in these medium-to-high load conditions, which can be controlled by fine tuning the 3$^{\mathrm{rd}}$ injection. Even though, in the optimum point operating with the gasoline PPC concept, noise level is significantly higher compared to that obtained operating with a well optimized CDC concept.

Additionally, the potential of injection pressure to control the local mixing conditions was investigated in detail. Increasing injection pressure shifts the combustion phasing towards the expansion stroke and softens its development, decreasing the cylinder pressure gradients and noise levels. In addition, the lower local equivalence ratios along the combustion process result in reduced soot emissions. As the main negative aspect, the longer spray penetration brought an increase in HC emissions. Similarly, adjusting the fuel distribution between the three injection events helps to control the shape and phasing of the combustion profile, affecting mainly the CO and HC emissions without punishing too much neither the $NO_x$and the soot emissions, nor the efficiencies.

Anyhow, this can provide an interesting paths for improving the pressure gradient and noise levels.

Finally, it is worth to point out that this test campaign was performed with the optimum engine hardware defined operating with the CDC concept, therefore, a detailed study of the piston and injector nozzle geometry to improve their compatibility with the gasoline PPC concept is expected to allow even further improvements. Moreover, future optimization work using a Design of Experiments (DoE) methodology could be useful not only to understand coupled effects that have influence over the combustion and emissions formation processes, but also to find the best injection pattern that could simultaneously fulfill a given set of targets and restrictions.

4.4 Optimization of the gasoline PPC concept

Detailed explorations have been performed on the engine to investigate the behavior of the gasoline PPC concept and gather an overview of the performance and emission levels achieved over the full operating range of the engine, from 1250 to 2750 rpm and from 3 to 10.4 bar IMEP. As discussed previously, to present all those results would be futile, but they can be found in the literature through the work of the research team [7, 8, 11–13]. However, the optimization paths described in the following section can highlight some of the most pertinent advantages and drawbacks of this particular combustion in this engine: here the focus is put on the point with 10.4 bar IMEP at 1250 rpm, to optimize fuel consumption and break some of the key trade-offs that have emerged along the investigations under certain conditions. The objective of is to evaluate the behavior of the combustion when the parameters of interest are changed. Nevertheless, and as commented, no fine optimization of the engine hardware or settings was performed: the engine hardware was optimized in previous studies to operate with the CDC concept [4–6]. The air management settings were selected using a mathematical model previously fitted by means of a specific Design of Experiment (DoE) at the current engine hardware in similar operating points [6, 14]. This allows to get preliminary results that can be directly compared with those operating with the CDC concept in terms of engine efficiency and emissions.

A triple injection strategy was used in most of the following studies, while the fuel rate required to get the IMEP target at the baseline case (with the optimum CA50) was maintained for the rest of operating conditions. The injection strategy used allows achieving the load target while mitigating knock tendency.

The first part of the research presented here was focused on the RoHR profile analysis with the objective of understanding the paths for noise reduction (through the pressure gradient $dP/d\alpha_{max}$), and observe the impact on efficiency and emissions. This was carried out with the in-house combustion simulation program *Siciclo*, which allows numerical redefinitions of the RoHR profile by changing either its shape (height and length) or its timing, keeping the same total cumulative heat release. It then provides the corresponding outputs such as pressure profile and all the derived parameters (pressure gradient, heat losses, efficiency...), in order to explore optimization paths through an unusual method using a desired heat release profile. Then, knowing how to control it from experience with the injection settings, the simulated trends were reproduced experimentally, mainly by changing the fuel distribution and injection settings.

During the last part of the research work, a study was dedicated to the air management, and more particularly to the exhaust pressure (and pressure drop). This came as an answer to a very low trapping ratio at certain operation conditions due to high pressure drop and low speed, resulting in a high fraction of wasted compressed air, and so a very high $ISFC_{corr}$.

The most relevant engine settings chosen for each operating condition are detailed in Table 4.3.

Table 4.3. *Main injection study at 10.4 bar IMEP: Engine settings.*

Parameters	T_{in} [°C]	$VVT_{(in,ex)}$ [CAD]	Olap [CAD]	EGR [%]	P_{rail} [bar]	m_{fuel} [mg/st]
All Tests	45	(5,20)	78.4	43.5	850	19.8
Parameters	P_{in} [bar]	ΔP [bar]	SoE1 [CAD]	SoE2 [CAD]	SoE3 [CAD]	$\%_{fuel}$ [%]
Studies	2.755	0.600	-60	**[-46 to -38]**	-2	17/66/17
	2.755	0.600	-60	**[-38 to -34]**	-2	**17/60/23**
	2.755	0.600	-60	-36	**[-8 to -2]**	**17/60/23**
	2.755	**0.725**	-60	**[-42 to -36]**	-2	17/66/17

4.4.1 Optimization of the RoHR profile to control the noise level

It has been demonstrated in the previous section how the injection timing (2^{nd} and 3^{rd} event) can easily be used to control the combustion phasing and emissions, with a low penalty in indicated efficiency and the air management parameters. From these conclusions, a theoretical study focused on the combustion profile (RoHR) was carried out to define paths to control noise. This was performed using *Siciclo*, which allows the change of RoHR shape (height and length) or combustion timing, keeping constant the cumulative heat release. A reference experimental RoHR is required for the initial model adjustment, and as the baseline test which main variables are described in Table 4.4. The reference RoHR profile is shown in Fig. 4.14 and the corresponding main characteristics in Table 4.5.

Table 4.4. *Reference point: Engine settings.*

n [rpm]	IMEP [bar]	P_{in} [bar]	ΔP [bar]	T_{in} [°C]	$VVT_{(in,ex)}$ [CAD]	Olap [CAD]
1250	10.4	2.755	0.600	45	(5,20)	78.4

EGR [%]	m_{fuel} [mg/st]	P_{rail} [bar]	SoE1 [CAD]	SoE2 [CAD]	SoE3 [CAD]	$\%_{fuel}$ [%]
43.5	19.8	850	-60	-40	-2	17/66/17

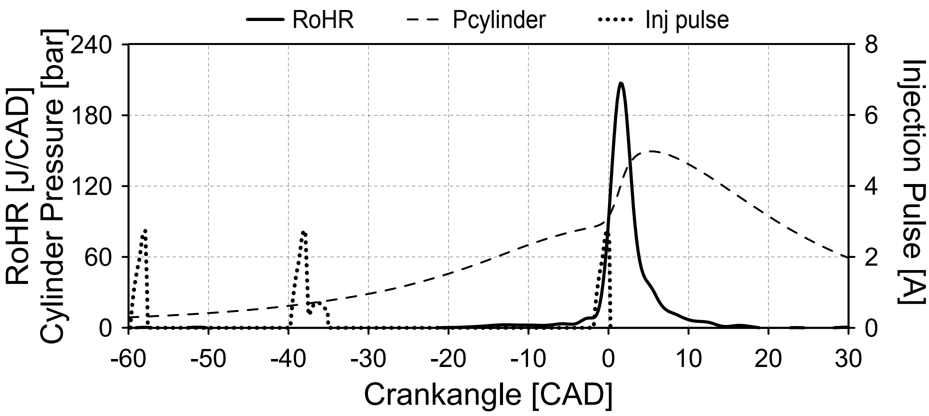

Figure 4.14. *Reference RoHR.*

The two paths investigated were focused on:

- **The RoHR shape:** extending the combustion duration (CA10 → CA90) from 100 to 200%, by 20% steps (100% being the reference). In order to keep the same cumulative heat release, the height of the profile is accordingly modified.
- **The combustion phasing:** moving the RoHR profile from −5 to +5 CAD from the reference, by 2.5 CAD steps (0 being the reference).

Table 4.5. *Reference point: Main characteristics.*

SoC [CAD]	CD [CAD]	η_{ind} [%]	η_{comb} [%]
-1.7	8.0	46.18	97.54

As discussed, the efficiency is hardly affected by changing the combustion profile or moving its onset (within a given but quite large range). The simulation is able to reproduce very similar results, as shown in Fig. 4.15. Despite the very different RoHR profiles considered in the present study, the efficiency keeps within a restricted range of less than 2 points (from 45 to 47%). In practice, this range is even more restricted due to the impossibility for the engine to reproduce some of these RoHR profiles: a too early SoC leads to hard knocking conditions, and a diffusion-like combustion is required to get a very wide RoHR profile, which generates too much soot.

It is worth highlighting that the combustion efficiency is not taken in account in this calculation. Although not negligible, its experimental variation is quite small and has a direct effect on the indicated efficiency, thus it was neglected in the simulations. Accodingly to this assumption, the results provided by the model are slightly optimistic and they must be validated with experimental data.

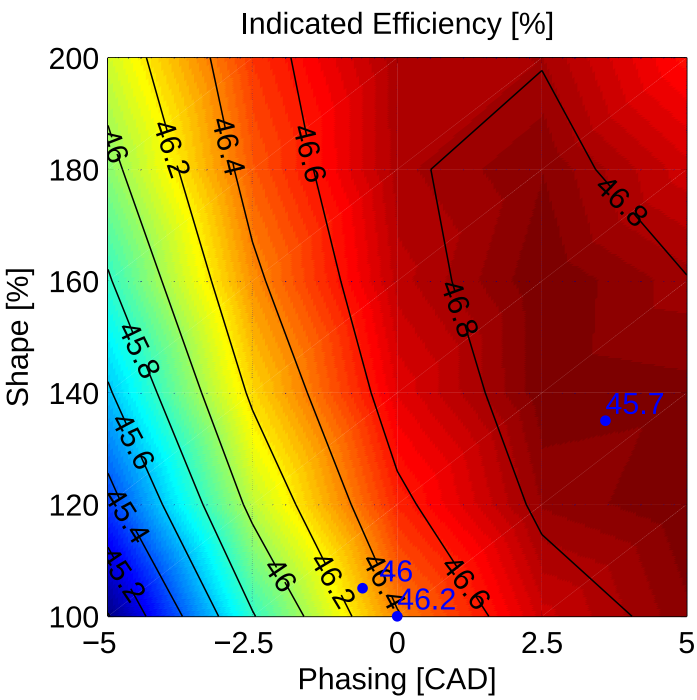

Figure 4.15. *Indicated efficiency.*

If Fig. 4.15 and 4.16 are compared, it appears clearly that the indicated efficiency is correlated with the heat losses. Indeed, except for late and wide RoHR profile, where the expansion work tends to decrease, the heat transfer is the main factor affecting the efficiency (aside from the combustion efficiency, as mentioned). This can be justified if it is taken into account that at this operating point, about 15% of the total fuel energy is lost due to the heat rejection to combustion chamber walls.

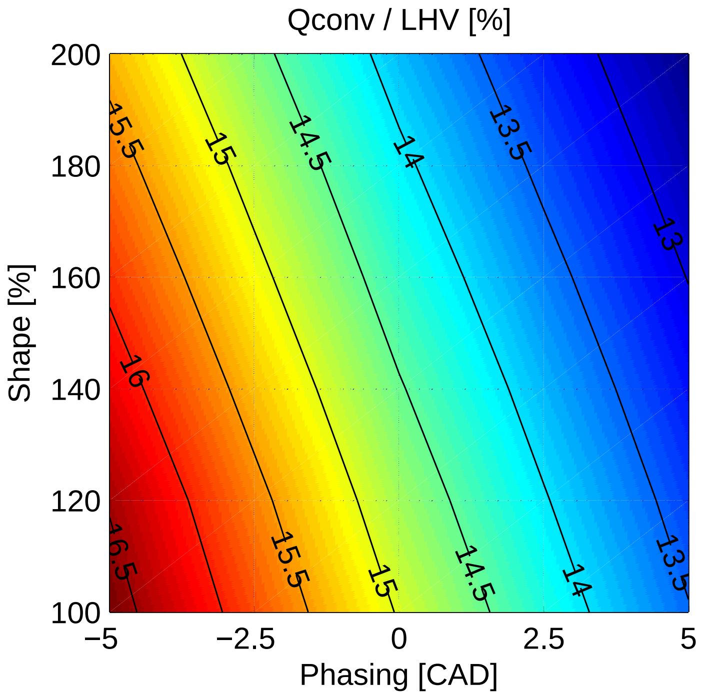

Figure 4.16. *Proportion of energy release compared to introduced energy.*

The second issue explored with the simulation is the noise reduction path. The noise level is directly related to the pressure gradient during the combustion process, which is mainly controlled by the shape of the RoHR and, in a lower extent, by the combustion phasing (at least inside the studied range). Fig. 4.17 shows the pressure gradient variation obtained with the simulations. It is interesting to remark that important reduction can be achieved by smoothing the RoHR profile, thus being a good option to overcome this intrinsic drawback of the gasoline PPC concept.

It can be observed that the effect of the RoHR shape on the pressure gradient (and thus on noise) is much higher than on engine indicated efficiency. However, the opposite trend is observed for the combustion phasing. The most interesting conclusion of this analysis is the lack of trade-off between noise level and indicated efficiency. However, there are two main restrictions to this strategy. First, generating a smooth combustion profile while keeping the premixed combustion process results in a strong deterioration of the combustion stability, even reaching the misfire limit, because the combustion must be shifted

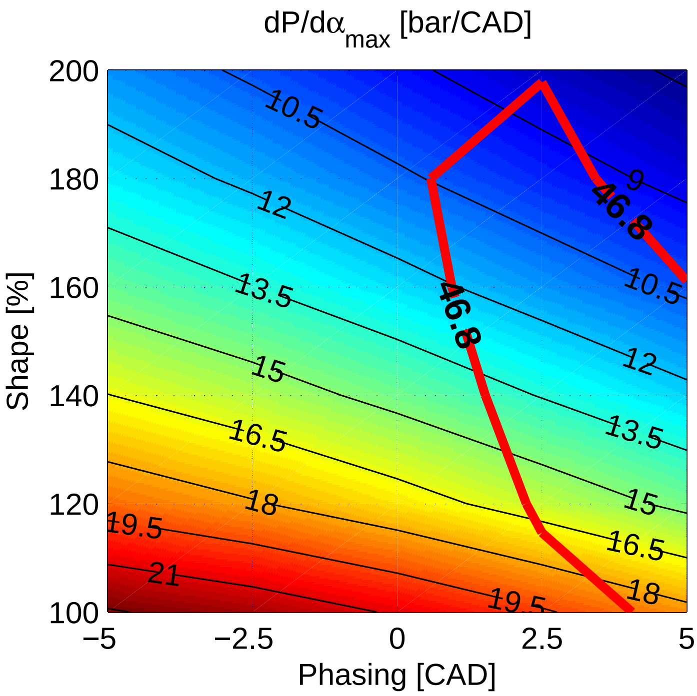

Figure 4.17. *Maximum pressure gradient.*

towards the expansion stroke to get a wide RoHR profile with a low peak. Second, optimizing the combustion profile by switching to a diffusive-like combustion while maintaining a wide RoHR profile impacts negatively on soot emissions and the NO_x/soot trade-off is recovered. The objective then is to find a strategy to generate an optimized combustion profile that allows having a low pressure gradient while keeping an acceptable combustion stability and soot emissions.

4.4.2 Strategy for improving noise/emissions/efficiency trade-offs

As seen along this chapter, a 3-injections strategy is used for this PPC concept (for most of the operating points). The 2nd injection timing controls the combustion phasing while the 3rd injection, close to TDC, controls its profile and final soot emissions. But the fuel quantity injected during this 3rd event can also be managed (in addition to the timing), in order to control the

	dP/dα [bar/CAD]	Noise [dB]	NO_x [mg/s]	Smoke [FSN]	CO [mg/s]	HC [mg/s]	η_{comb} [%]
SoE2 -40 - SoE3 -2 - 17/66/17 [1]	20.3	97.0	0.6	0.52	11.7	7.2	97.5
SoE2 -38 - SoE3 -2 - 17/66/17	23.8	99.2	0.7	1.36	12.6	6.2	97.8
SoE2 -38 - SoE3 -2 - 17/60/23	11.1	90.4	0.2	0.30	20.7	10.3	96.2

	SoC [CAD]	CD [CAD]	$RoHR_{max}$ [J/CAD]	ISFC [g/kWh]	$ISFC_{corr}$ [g/kWh]	η_{ind} [%]	MT 3rd inj [CAD]
SoE2 -40 - SoE3 -2 - 17/66/17 [1]	-1.7	8.0	207.3	182.0	244.2	46.2	4.3
SoE2 -38 - SoE3 -2 - 17/66/17	-2.3	8.4	234.2	182.9	243.1	46.0	5.0
SoE2 -38 - SoE3 -2 - 17/60/23	1.9	10.8	149.5	183.8	243.3	45.7	1.2

[1] *Siciclo reference point*

Table 4.6. *Results of the fuel distribution comparison.*

combustion profile to reduce noise. These hypotheses were checked on the engine, according to the parametric study presented in Table 4.6.

Fig. 4.18 shows the RoHR profiles obtained with the reference settings used for the previous *Siciclo* study, along with those adjusted for performing this analysis of the impact of the fuel distribution between the 2nd and the 3rd injection events. The test displayed in black represents the reference, in red corresponds to the 2nd injection event delayed 2 CAD, and finally in blue is shown a test with the same settings as the red one, but with a different fuel distribution. The key settings and results are shown in Table 4.6.

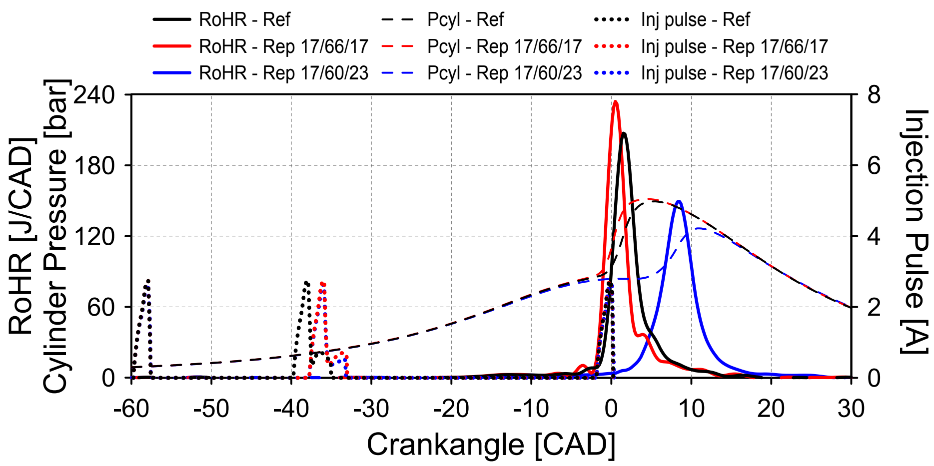

Figure 4.18. *Fuel distribution study.*

As shown, the impact of the fuel distribution is critical to tune the RoHR profile according to the trends observed with the previous theoretical study. Reducing the fuel quantity injected early during the 2^{nd} injection decreases the reactivity of the mixture, delaying the SoC and generating a later phased and smoother combustion process. Therefore, despite the corresponding increment in the fuel quantity of the 3^{rd} injection, with this fuel distribution the new trade-off between noise and soot can then be broken. Looking at the results shown in Table 4.6, most of the output parameters are improved (noise is reduced to the target of 90 dB, soot and NO_x are both reduced) or kept almost constant (efficiencies, ISCF/ISFC$_{corr}$), while only CO and HC levels increase.

These three points are represented as blue dots in Fig. 4.15. It should be observed that the combination providing the best indicated efficiency in simulations (46.8%) cannot be reproduced experimentally (45.7%): as discussed before, the indicated efficiency calculated through *Siciclo* does not take into account the impact of variations of the combustion efficiency since it is kept constant for all simulations. However, experimental results confirm how at this particular point combustion efficiency decreases by around 2% when fuel is shifted from the 2^{nd} to the 3^{rd} injections. This problem with combustion efficiency explains why the indicated efficiency level provided by the simulation cannot be experimentally, at least quantitatively, attained.

Additionally, during the experiments the main constraint observed when fuel is shifted from the 2^{nd} to the 3^{rd} injections (fuel repartition 17/60/23) is the combustion stability. Indeed, the range for the injection timings (2^{nd} and 3^{rd} events) is limited between knock and misfire. In fact, the combustion process is very sensitive to the injection settings and thus, a detailed study was carefully performed to determine the settings that provide a suitable combustion process.

A short preliminary study has been conducted on the effect of the 2^{nd} and 3^{rd} injection timings to observe the sensitivity and the trends obtained with fuel repartition 17/60/23 (see Table 4.3 on page 118). With this fuel repartition, the 2^{nd} injection timing has now a small impact on combustion phasing and maximum pressure, while pressure gradient (noise) is also slightly affected according to Fig. 4.19. The combustion stability also shown in Fig. 4.19 is quite correct in the testing range, but falls drastically (increment of the CoV IMEP) by advancing the injection event by 2 CAD more, to -40 CAD aTDC.

Additionally, as shown in Fig. 4.20 and Fig. 4.21, the indicated and combustion efficiencies are both affected by the 2^{nd} injection timing, together with emissions. The indicated efficiency increases about 2% by delaying the 2^{nd} injection, while CO and soot emissions also increase. However, HC production decreases by reducing the liquid fuel impingement onto the combustion chamber

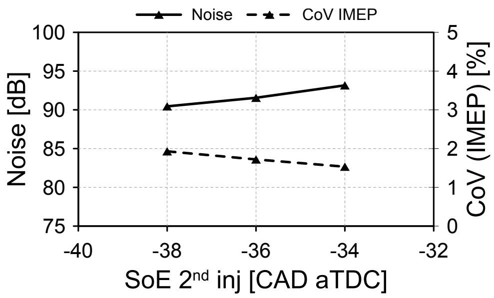

Figure 4.19. *Noise level with more fuel in the 3rd injection.*

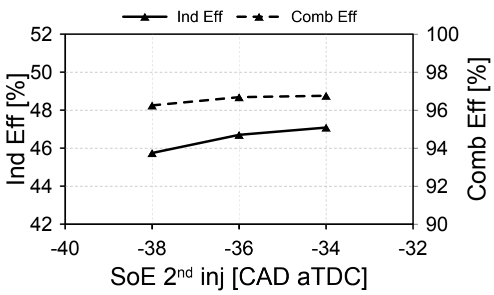

Figure 4.20. *Efficiencies with more fuel in the 3rd injection.*

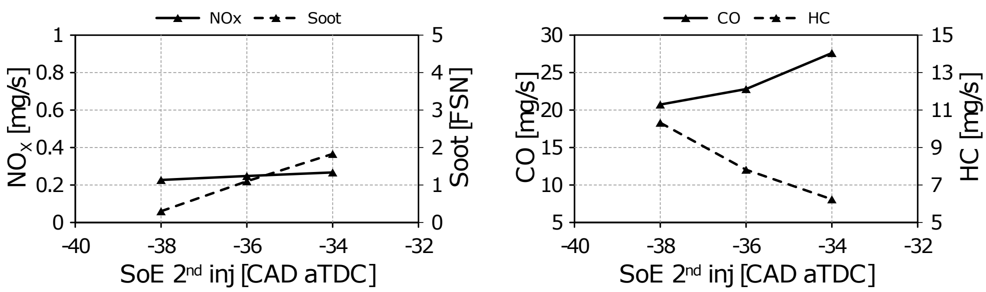

Figure 4.21. *Pollutants level with more fuel in the 3rd injection.*

walls, helping to keep the combustion efficiency constant. However, this 2nd injection timing has very low impact on the combustion profile or on its phasing

(Fig. 4.22). From these results, the 2nd injection timing can be set to control emissions, almost independently from the other parameters.

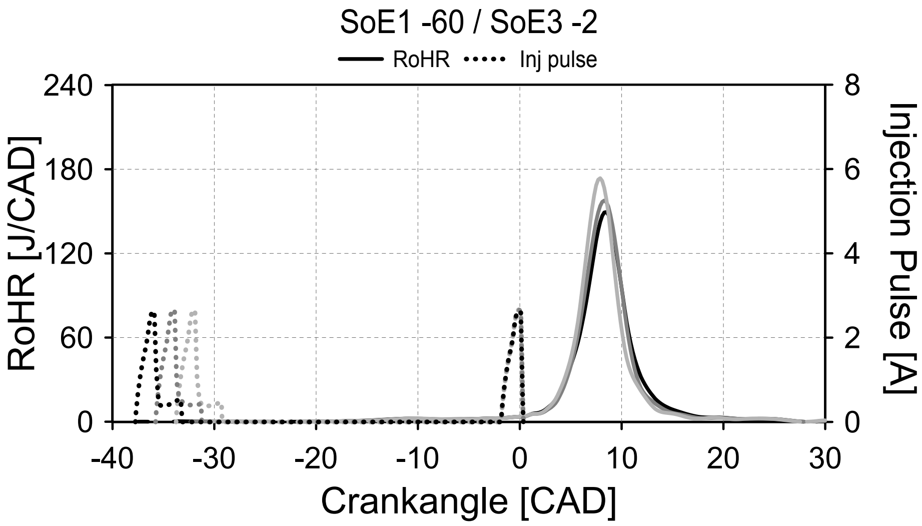

Figure 4.22. *RoHR profiles with the new fuel distribution (2nd injection sweeping).*

Focusing on the 3rd injection timing effects, delaying this event drastically decreases the maximum in-cylinder pressure, pressure gradient and noise. Indeed, it controls the combustion process as observed in previous experiments, and its influence is higher when its fuel quantity increases. Thus, according to Fig. 4.23 noise ranges from 95 to 85 dB within an injection timing variation range of only 6 CAD, while HC/CO levels moderately increase. Finally, NO_x/soot levels are kept constant.

The effect on the combustion profile is observed in Fig. 4.24, where the softening effect of delaying the 3rd injection is evident. Delaying further the 3rd injection generates a reverse trend, so the combustion starts earlier and overlaps with the 3rd injection. The explanation for this trend is the cooling effect of the 3rd injection event that decreases the reactivity of the mixture, delaying its ignition. When this event starts too late, the mixture reaches its ignition point without being affected by this cooling effect and then the onset of combustion advances. The final result is a sharp increment in soot emissions and noise levels caused by the earlier and faster combustion process.

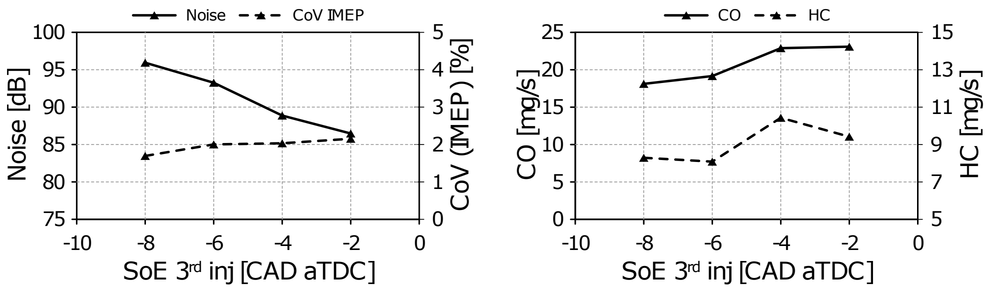

Figure 4.23. *Noise and pollutant levels with more fuel in the 3rd injection.*

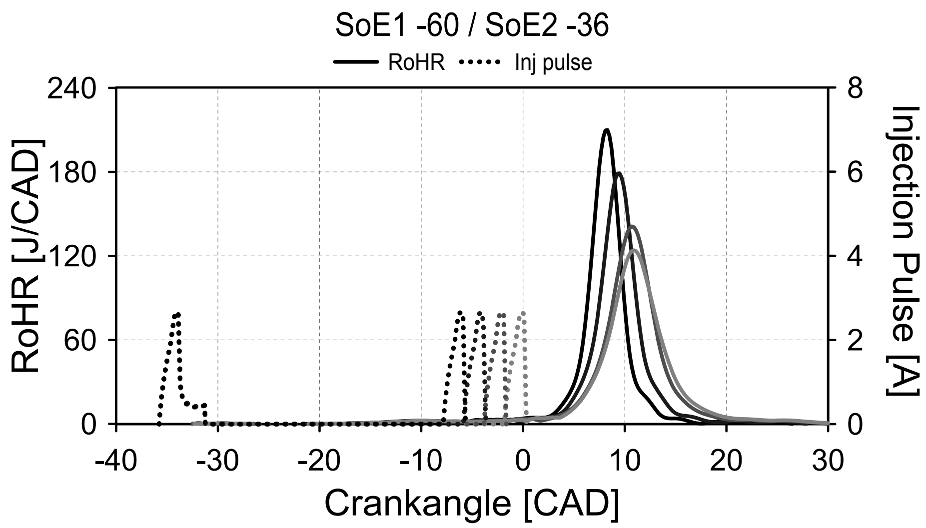

Figure 4.24. *RoHR profiles with the new fuel distribution (3rd injection sweeping).*

4.4.3 Strategy for improving the corrected efficiency

Finally, experiments were carried out to observe the impact of decreasing the intake-exhaust manifold pressures drop (ΔP) on ISFC and ISFC$_{corr}$. The reference value of ΔP was set at 0.600 bar, before being increased up to 0.725 bar. The impact on TR was significant, as well as on ISFC$_{corr}$, as shown on Fig. 4.25 and Fig. 4.26.

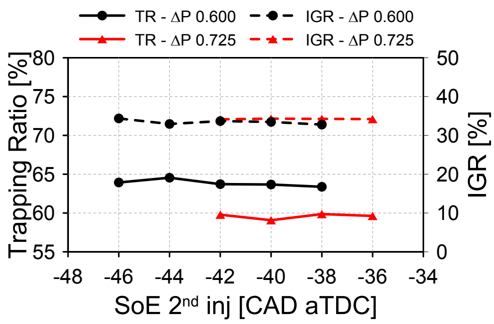

Figure 4.25. *Trapping Ratio & IGR.*

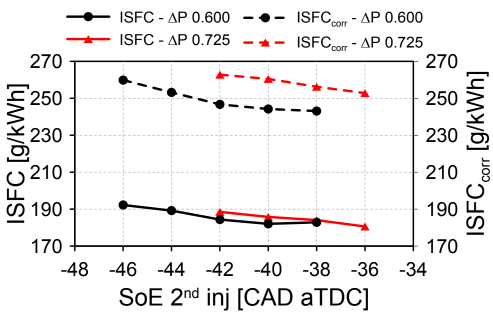

Figure 4.26. *ISFC & ISFC$_{corr}$.*

According to previous activities performed by the research team focused on air management optimization, the best TR considering the ISFC and ISFC$_{corr}$ trade-off is around 70-75% [1]. In this analysis, increasing TR from the previous 60 to 65% led to an important improvement of the ISFC$_{corr}$ around 10-15 g/kWh. Moreover, it must be highlighted that the impact on other parameters is very low. The combustion phasing and profile are slightly advanced, without damaging stability. This earlier onset explains the main drawback of this strategy: noise level increase by 1-2 dB. This effects was expected, as a reduced short circuit results in a higher IGR level, thus increasing T$_{IVC}$ and enhancing a more reactive mixture.

Soot emissions are slightly higher with a lower ΔP while the CO and HC emissions are improved (Fig. 4.27). This can be justified taking into account that a higher trapped mass results in a higher density: thus the spray penetration and the fuel in the squish region diminish. Additionally, the higher temperature along the closed cycle caused by the increment in T_{IVC} and the earlier onset of combustion (it is shifted towards the TDC) also helps to promote the conversion of CO into CO_2.

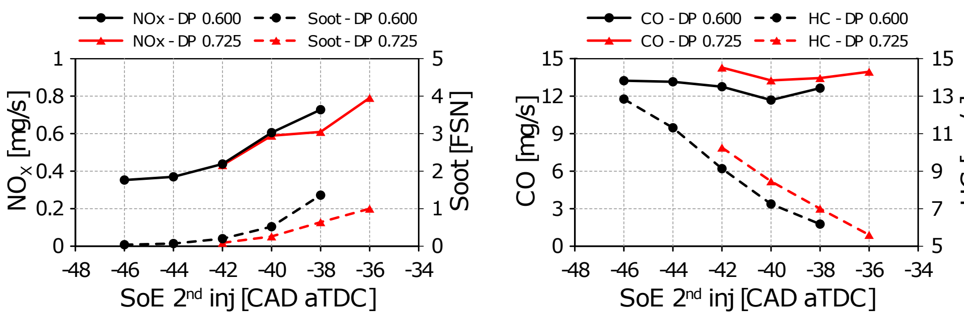

Figure 4.27. *Pollutants.*

Nevertheless, even though the benefits of reducing ΔP are not negligible, the $ISFC_{corr}$ level is still high and needs even more improvement. A ΔP of 0.500 bar was tested, but the stability was appreciably damaged at this point, and the combustion was too delayed (close to misfire) due to the deteriorated in-cylinder thermo-chemical conditions resulting from the poor scavenge process (too much IGR together with very low fresh air flow rate). Thus, results obtained can not be compared with that shown, however it is very important to highlight the interest of exploring paths to increase TR without punishing the scavenge process.

4.4.4 Mapping of the engine at lower load conditions and comparison with a CDC operated 4-Stroke engine

After understanding the effects of each parameter over the combustion process and engine performance at high loads, the study has later been focused on mapping the engine by applying those optimization guidelines in a set of operating conditions covering the low load region. A total of 10 engine speed / load combinations were measured, ranging between 1250 and 2500 rpm and between 3 and 5.5 bar IMEP, with the objective of optimizing the $ISFC_{corr}$, which is the parameter closest to real fuel consumption. The rest of the outputs depends on this particular optimization criteria, which means that they could be individually further optimized, i.e the engine has potential for more advantageous output compromises if they were specified differently than absolute best ISFC_{corr}.

Those 10 operating conditions were then compared to the equivalent 4-Stroke CI engine according to data provided by the project partner Renault SA. The comparison was made at iso-IMEP, which corresponds to the IMEP measured directly on the 2-Stroke cycle, and to the high pressure loop IMEP from the 4-Stroke cycle, thus disregarding the pumping losses for fair comparison. From these data is then extracted the corresponding high pressure loop ISFC. An equivalent calculation of the ISFC_{corr} is performed by taking into account the 4-Stroke pumping losses, considering they can be put in parallel with the work of the supercharger from the 2-Stroke engine, since they are the two additional works occurring between the exhaust and intake processes and allowing the base air-charge of the engine, which the turbocharger (similar in both engines) then completes. However, before presenting the detailed comparison of these two engine configurations, it must be mentioned that the poppet-valve 2-Stroke engine is a research engine, and as described in Chapter 3, its hardware is not optimized (using conventional Diesel injection system, standard turbocharger mapping model, etc). On the contrary, the 4-Stroke calibration provided is fully optimized for commercial implementation, both in terms of compromises between the engine output parameters, and in terms of hardware (injector nozzle / piston bowl matching, dedicated turbocharger matching, etc). Therefore, some of the outputs from the 2-Stroke engine could be further improved with dedicated optimized systems, especially in terms of fuel consumption (both gross and corrected).

Figure 4.28 displays the main results of this comparison, with the top plot representing the ISFC_{corr} for both engines. As expected, fuel consumption, even with a selection based on the absolute lower value, is higher that of the 4-Stroke engine in almost all cases. However, one point that highlights the potential for improvement (either through hardware optimization or calibration, or both) is the wide amplitude of both ISFC and ISFC_{corr} variation observed in the case of the 2-Stroke engine, while the general fuel consumption levels are much less dispersed in the 4-Stroke engine.

The two following plots describe in two different ways the impact of the supercharger (for the 2-Stroke engine) or of the pumping work (for the 4-Stroke engine) over the ISFC: the difference between ISFC and ISFC_{corr} is a direct image of the work required by these two elements (with only the fuel mass as coefficient, equivalent for both engines); the ratio between the two ISFC values represent the net work (left after retaining the losses from the two previous elements) proportionally to the gross work produced during the combustion. In

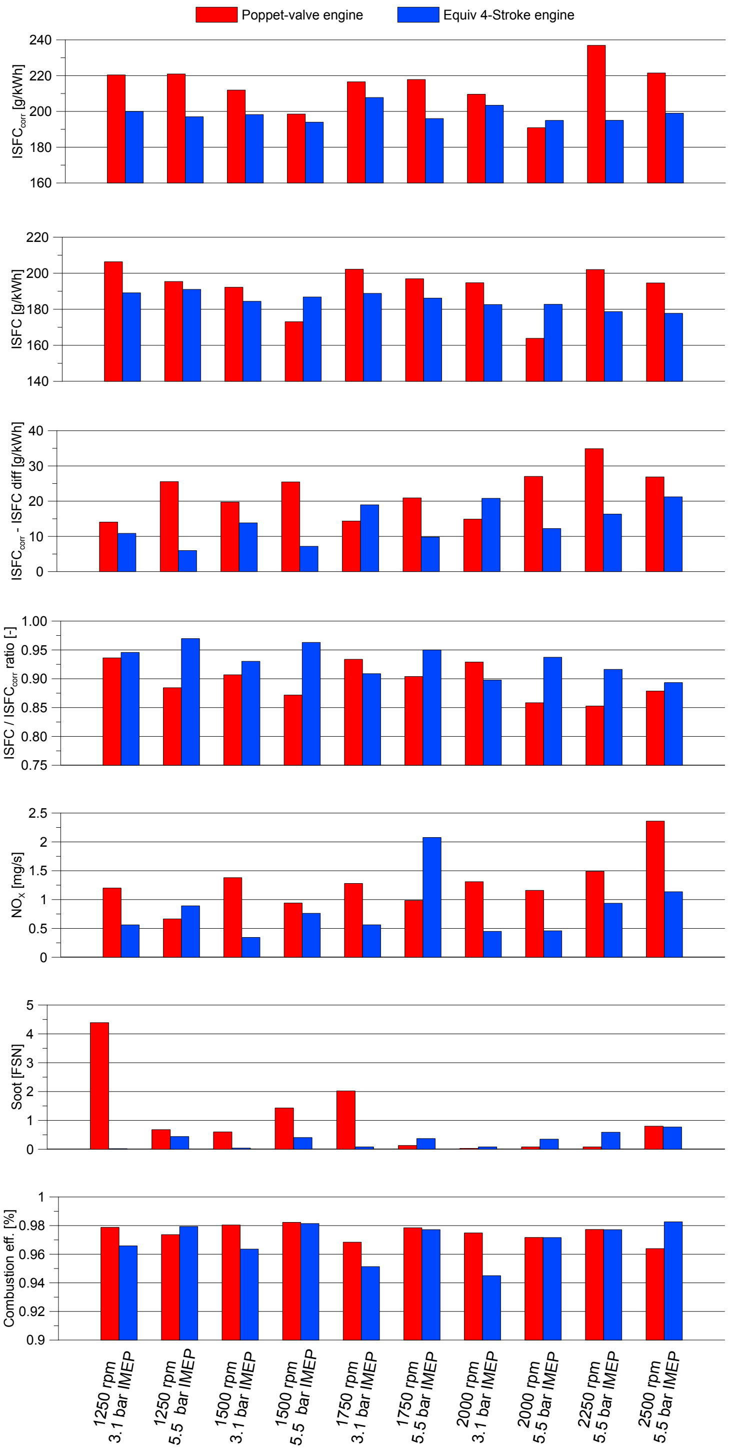

Figure 4.28. *Outputs comparison between gasoline PPC concept in poppet-valve 2-Stroke engine and commercial CDC 4-Stroke engine.*

both cases, the global trend shows that the work required by the supercharger is greater than the pumping work. The consequence of this factor is that the combustion provided by the gasoline PPC concept from the 2-Stroke engine then needs to be much more fuel efficient than the CDC concept in the 4-Stroke engine to be able to compensate this additional work and obtain equivalent levels of ISFC$_{corr}$.

However, the specific trends show a very different picture between the two engines. Indeed, while the pumping losses seems to be decreasing slightly with higher loads and lower engine speeds (both in gross values and proportionally), the work required by the supercharger increases drastically with higher loads, but does not seem to be affected by the engine speed. Considering the degree of optimization of this specific engine, this demonstrates a promising potential for low load operations, where ISFC$_{corr}$ could be easily matched after slight improvements of the turbo- and supercharging systems (or even better than the 4-Stroke engine at very low loads if pumping losses keep increasing while supercharging losses would keep decreasing). Moreover, the penalty at higher loads could also be considerably reduced with a better matching of the turbo-system to decrease the load of the supercharger and thus its induced losses. Also, a substantial improvement of the scavenging performance to improve the TR of the 2-Stroke architecture would help to drastically reduce its ISFC$_{corr}$ over the whole engine operation range.

Regarding the pollutant emissions, the CDC concept of the 4-Stroke engine tends to produced globally less NO_x than the gasoline PPC concept if the target focuses on the lowest ISFC$_{corr}$ possible, although the specific trends are once again opposed: in the first case they appear to increase with load (with an increment with engine speed only in the higher range, no particular effect at lower speeds), while the new combustion concept shows an opposite tendency regarding the load (and similar effect with engine speed). This could be explained by the large amount of EGR introduced during PPC operation at higher loads, while its rate might be more limited when increasing the load during CDC operation to keep soot emissions at reasonable levels considering the diffusive combustion (prone to generate particulates with higher equivalent ratios). This is confirmed by the soot plot of Figure 4.28, where for the 4-Stroke engine they range from 0.5 to 1 FSN for every higher load points whereas there is almost no soot emissions at lower loads. The trend is different and much more difficult to evaluate on the 2-Stroke engine since the data shown here are not optimized for soot emissions and thus their levels do not seem to follow any specific behavior. Moreover, the large amount of measurements performed show a wide range of emission levels within very small variations of ISFC or other parameters, justifying the previous mention of potential better setting combination by penalizing very slightly the fuel consumption. And the levels displayed at higher engine speed prove this potential, where almost no soot can be reached regardless of the load.

Finally, the two other critical pollutant emission levels, HC and CO, are represented by the combustion efficiency levels, which are more relevant for direct comparison sake. And despite being a partially homogeneous charge with early injection events (inducing potential fuel-wall interactions), the combustion efficiency is similar –if not better– compared to the CDC 4-Stroke engine, with levels consistently over 97%. This is a huge step forward for premixed combustion concepts, especially at low loads, allowed by the 2-Stroke operation cycle. Indeed, these concepts when implemented on 4-Stroke engine can't reach suitable in-cylinder conditions (in terms mainly of temperature) to generate proper combustion, and usually resulting in either misfire or at best in bad combustion and very high levels of HC and CO.

4.5 Conclusions

The research work reported in this section was focused on analyzing and optimizing the gasoline PPC concept in a 2-Stroke poppet-valves HSDI CI engine. This particular engine architecture offers a large operating range for this combustion concept and also high flexibility on the different settings, allowing the quasi-independence between air management (influencing in-cylinder conditions) and injection settings (controlling mixture stratification). The complementarity of the wide amount of experiments (combined with 3D-CFD simulations described by the research team in [7]) allowed to understand in detail the behavior of the combustion and how to influence and control it for different purposes. The ISFC obtained was in a satisfactory range, but the ISFC$_{corr}$ taking in account the power demanded by the supercharger to provide the required intake pressure, especially at low load, was too high. Also, the other observed drawback was the combustion noise level, unacceptably high, because of the sharp knocking-like combustion process especially at high load.

In order to try and reduce it, the multiple injections strategy adopted allows some flexibility concerning the fuel distribution. Then, the study carried out focused on the mass injected during the 3rd event shows very promising results, providing control over the noise, by switching only a little quantity of fuel from the main 2nd injection to the 3rd event. It also helps to reduce significantly both soot and NO_x emissions, but deteriorates combustion efficiency by generating more CO and HC emissions. But the indicated efficiency and ISFC/ISFC$_{corr}$ trade-off are hardly affected from one distribution to another. This confirms the possibility of controlling NO_x/soot emissions keeping attractive fuel consumption levels in a relatively wide range of injection settings, while noise control at high loads demands a fine tuning of both injection and air management settings to generate a suitable combustion process. These experimental observations were possible thanks to a numerical study performed using a predictive thermodynamic model, revealing trends that were not –and could not be– observed experimentally and allowing the researchers to define new investigations paths.

Then, it has been highlighted that the ISFC$_{corr}$ can be reduced by influencing the trapping ratio. Being able to increase it would mean a reduction of the short-circuit flow from the intake to the exhaust ports during the valve overlap interval, keeping more intake air trapped in the cylinder and finally wasting less work from the supercharger. This was obtained by increasing the exhaust pressure to reduce the pressure drop across the engine (ΔP). The gains were significant, but not sufficient to reach the expected trapping ratio levels without impacting negatively combustion stability and pollutant emissions. However, this strategy is suitable in the range where the air management and the combustion process are reasonably independent. The next objective would be to further improve ISFC$_{corr}$ trying to decrease the work of the supercharger by increasing the trapping ratio, but without worsening the scavenge to keep a suitable combustion process.

A similar methodology to the one used so far at high loads has then been applied at lower loads (from 3 to 5.5 bar IMEP) to confirm that similar engine behavior and output trends can be reproduced, and more importantly to compare the engine outputs to an equivalent commercial 4-Stroke engine operating with the CDC concept. At these lower loads, the 2-Stroke gasoline PPC concept displays a very interesting potential compared to this 4-Stroke CDC concept. Indeed, despite the levels of gross and ISFC$_{corr}$ being noticeably higher, the behavior of this innovative combustion concept in regard with the engine load and speed is very different from what can be observed with the 4-Stroke engine and highlights potential optimization paths, which could be reached by some improvements of the super- and turbo-charging systems, or of the scavenging performance. Also, the fact that the gasoline PPC concept could be reliably implemented at such low loads while allowing a wide range of flexibility for air loop and injection settings and still high levels of combustion efficiency (over 97%), is a big improvement compared to previous investigations on premixed combustion concepts, demonstrating the high potential of the 2-Stroke architecture for this type of operations.

Finally and after all those dedicated investigations and optimizations, it appears that despite the very promising potential of such combustion concept, the main limitation comes from the engine architecture itself, with the poppet-valve configuration making it physically impossible to improve both the trapping ratio and the scavenging efficiency, without punishing drastically the $ISFC_{corr}$ (especially when increasing the load, but a noticeable margin has also been observed on that aspect at lower loads conditions). Thus, maybe the answer to that scavenging problematic lays in a completely new engine design, keeping the 2-Stroke cycle to maintain its thermodynamic benefits, but using a more unconventional design such as the Uniflow architecture in order to physically separate the intake from the exhaust in the cylinder layout, thus avoiding as much as possible the short-circuit of the fresh air provided by the supercharger.

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Chapter 5

Theoretical design and experimental validation of a 2-Stroke Uniflow HSDI CI engine

Contents

5.1 Introduction

Following the conclusions of the previous chapter concerning the necessity of improving drastically the scavenging performance of the engine, a new campaign has been defined to design a fully new engine. This study has been carried out in the frame of the European REWARD project, which main objective is to develop a new 2-stroke engine architecture for automotive applications and complying with upcoming emission regulations. In the previous chapter, the engine used was designed following the guidelines of the POWERFUL project, which included industrial restrictions: it should be based on an existing engine block in order to be easily manufactured using the current production lines. This meant that the only available option as for engine design was the poppet-valve architecture.

For this new project, that limitation has been removed and a total freedom was allowed for the engine architecture and dimensioning. The main objective is, as explained throughout the previous chapter, to improve drastically the scavenging performance of the 2-stroke architecture, in order to be able to evaluate its potential when fully optimized. In this framework, a complete engine dimensioning will be performed from scratch, defining a new engine displacement, bore-to-stroke ratio, and of course scavenging pattern. Concerning this last topic, the option of the Uniflow architecture very quickly imposed itself because of its potential for a clean scavenging process (as described in Chapter 2, section 2.3) while conserving a relatively compact package, usually barely larger than conventional poppet-valve engines, but still easy to implemented for automotive applications.

As a first step of the implementation and experimentation of the engine, all the tests and comparisons with previous references will be performed under CDC conditions, as this type of combustion is very well known in terms of behavior and control, and also allows a larger panel of references. However, it should be kept in mind throughout the engine design that one further goal would be to implement the gasoline PPC combustion concept in this engine, in order to fully benefit the advantages of such combustion highlighted in the previous chapter combined with an improved and optimized engine scavenge, if successfully achieved.

This chapter will present all the steps followed along the design of this new engine, from its dimensioning (displacement, bore-to-stroke ratio, ports designs...) to its experimental validation and optimization of both scavenging and combustion processes.

5.2 Engine dimensioning

5.2.1 Design methodology and theoretical tools

The very first step of the design of this new engine was the definition of the targeted power output, depending on the vehicle class it was supposed to fit. The objective was set to the B-class and C-class vehicles, with a power target of respectively 60 and 90 kW. Based on these objectives, the solution of 1 L bi-cylindrical and 1.5 L tri-cylindrical engines seemed obvious: this meant that the unitary displacement of the single-cylinder research engine would be of 500 cm^3, and should be able to produce 30 kW. From this base-case definition, all the theoretical dimensioning described in the following sections was developed, using mainly 3D-CFD simulations to study and optimize the scavenging process, combined with CAD design (for cylinder / piston geometry, intake and exhaust manifolds...) and 1D-models (to set preliminary boundary conditions) provided by the project partners.

All the simulations performed in CFD along this study were set for a predetermined mid-speed / mid-load operating point (2000 rpm / 7 bar IMEP), in order to get a generic representation of the evolution of the air flow pattern while optimizing the design of the engine. Some speed and load variations have later been performed with the optimum geometries to verify the viability of the design along the operating range.

The boundary conditions were also initially set to generate over-scavenged processes, with a higher pressure drop between intake and exhaust than what would normally be calibrated on the experimental version of the engine. This deliberated choice comes from the fact that the induced over-scavenging only affects the speed of the process, therefore showing more of it within the same time-frame, without affecting significantly the process itself (i.e. providing very similar scavenging profiles). This will be demonstrated later in this chapter during the above-mentioned validations of geometries, using more realistic pressure drops for various engine speeds and loads.

Considering these aspects, the final figures coming from the 3D-CFD simulations (mainly TR and IGR rates) can only be seen as indicative and not quantitatively valid as optimization parameters. Therefore, relative versions of these parameters have been considered and set as targets: the TR at 10% of IGR, and the IGR rate at 90% of TR are two of the most important criteria considered in this study. Also, the final swirl level is an important aspect to consider, as a too high level of swirl during the combustion phase can lead to deteriorated burning conditions by over-diluting the mixture. The target for this parameter was then set at a maximum swirl number of 2 at the end of the simulation (300 cad aTDC), the end of the scavenging process (Intake Closing) varying depending on the intake port height, between 215 and 240 cad aTDC.

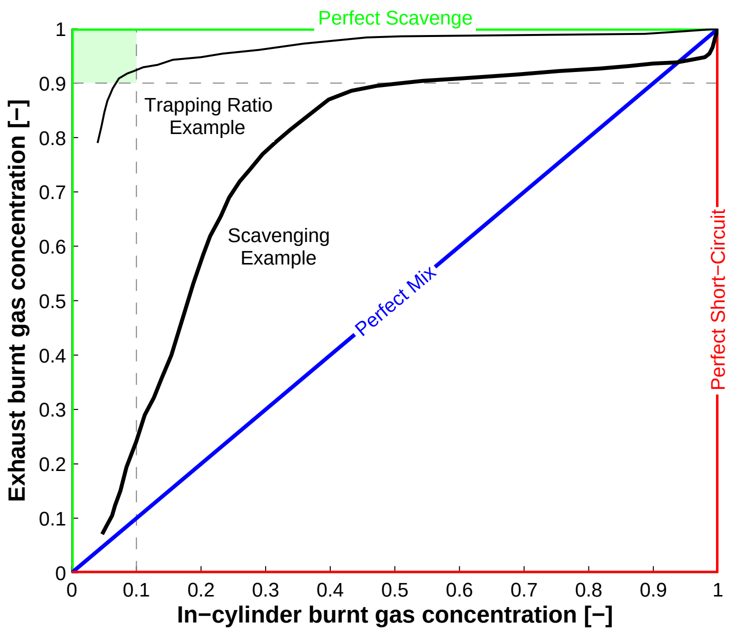

Figure 5.1. *Scavenging profile example.*

The Figure 5.1 shows an example of a typical scavenging profile, along with a typical TR curve. Also on this figure, the main targets previously mentioned are displayed by the green box: over 90% of TR and below 10% of IGR. If the TR curve passes through this area at some point, this means that the target is attainable, by reducing the over-scavenge to stop it at the desired moment. This can be done either by lowering the pressure drop between intake and exhaust, or by shortening the exhaust duration (or advancing its phasing).

5.2.2 Architecture definition

As briefly mentioned previously, the design of this engine is wildly inspired from the high-efficiency 2-stroke CI engines widely extended in marine applications. One of their key aspects is their characteristic high stroke-to-bore ratio, usually around 2 which is the optimum for providing the most favorable surface / volume ratio and therefore minimizing as much as possible the heat losses through the cylinder walls. In the case of the engine development presented here, it was impossible to reach this optimum stroke-to-bore ratio due to height restrictions, but some studies have been performed in that framework in order to take advantage as much as possible of the selected architecture nonetheless, and investigate the potential benefits that a long-stroke engine might have on the scavenging performance.

Thus, three stroke-to-bore ratios have been studied: S/B = 1.0, 1.2, 1.4 (i.e. from most conventional ratios to longest stroke possible in our case, maintaining the unitary displacement of 500 cm^3). To achieve so, the same geometry is used fro all cases, simply stretched and squeezed according to the S/B ratio (including the ports height, increased proportionally). The obtained results are displayed in Figure 5.2 and clearly underline that:

- An increase of the S/B ratio leads to a mutual increase of the trapping and scavenging efficiency ratios (see Figure 5.2-left);
- An increase of the S/B ratio leads to an increase of the trapped mass of the fresh air (see Figure 5.2-right).

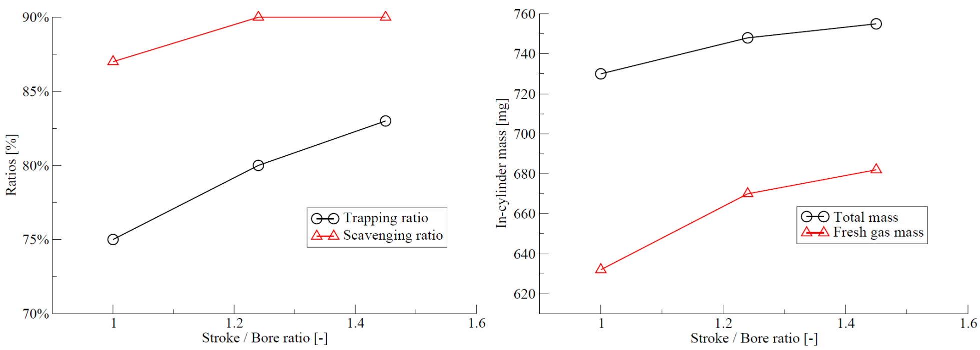

Figure 5.2. *Effect of B-S ratio over TR and Scavenging Efficiency (left) and over the in-cylinder trapped mass (right).*

The scavenging is clearly improved with large values of S/B ratio, which can be explained by the following elements:

- The by-pass of air to the exhaust is delayed with a large S/B ratio due to the larger distance to be covered by the fresh air from the bottom of the cylinder up to the exhaust at the top.

- The mixing between fresh and burnt gases is reduced with a large S/B ratio because of the decrease of the mixing volume (as a first approximation it depends only on the bore diameter, but this hypothesis will be later confirmed by CFD visualizations).

- With a large S/B ratio, the apparent equivalence of the piston top land height in CAD is smaller, which helps reducing the by-pass of fresh air due to later effective opening of the intake, and also the back-flow from the cylinder to the intake plenum which occurs at the end of the scavenging when the exhaust valves are closed (favorable to the charging ratio, by trapping a larger amount of fresh air).

These parameters and their effects are mentioned here because they are crucial aspects of the engine design philosophy, but as they were not specifically part of the doctorate's work, they will not be fully investigated in detail here. In order to fully understand the impact of each of these parameters, these investigations are described in more details by the project partners in the literature [1, 2]. The investigations performed on the piston bowl and top land height are also described in these papers.

Once the dimensions of the engine were fully defined, the next step of the investigations focused on the orientation of the scavenging path of the Uniflow architecture, as two options were available: a standard configuration, with intake ports at the bottom of the cylinder liner and exhaust valves in the cylinder head, or a reversed configuration, were the flow is inverted going through the cylinder top to bottom, from intake valves to exhaust ports. Attention was primarily shifted towards the reversed case, and this study quickly highlighted the relevance of optimizing the intake geometry to improve the scavenge process, the exhaust port design having a very low impact over the air flow through the cylinder (being at the end of the path). This geometry showed good scavenging performances, with a very low residual gas levels (less than 5%). However with this architecture, and as it can be observed in the scavenging sequence displayed in Figure 5.3, the air flow entering the cylinder is directed by the intake valves either onto the cylinder wall then directly towards the exhaust ports or along the center of the cylinder until impacting with the piston and then flowing horizontally towards the exhaust ports. Thus, burnt gases are trapped under the valves, displaying a TR / IGR trade-off, where TR needs to be quite worsened to be able to reach suitable IGR levels due to the important short-circuit observed.

Various valve geometries and intake manifold shapes have been tested out to try to direct the air flow as desired, but even using complex designs (partial

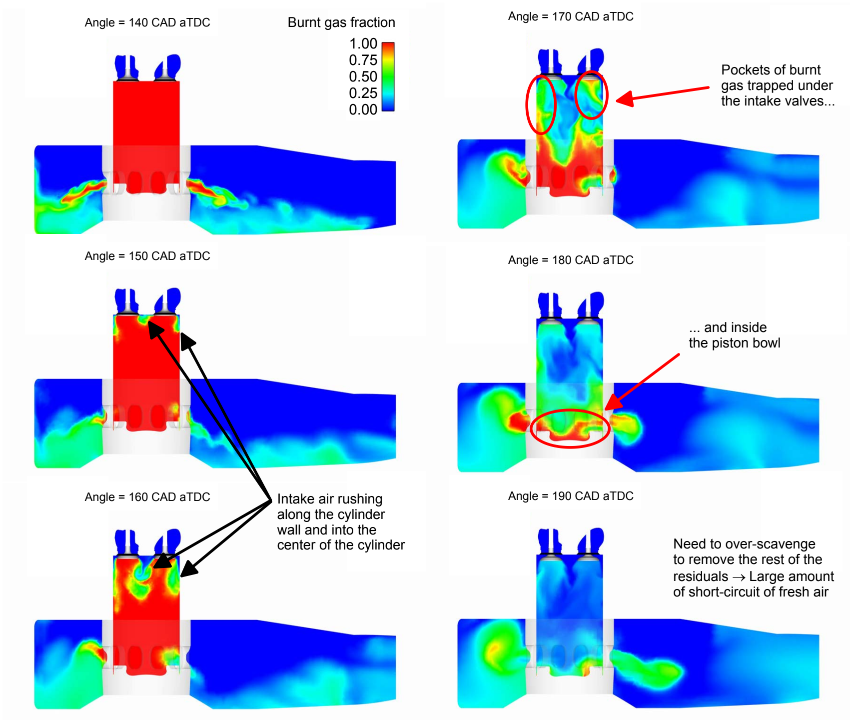

Figure 5.3. *Scavenging performance of the reverse Uniflow configuration.*

masking of the valves, twisted ducts...) the impact on the scavenge performance was very low. This Uniflow pattern orientation thus was proven to provide both poor performance and very little flexibility on the air flow path, therefore it was quickly discarded and the research efforts were diverged towards a more conventional pattern orientation with intake ports at the bottom of the cylinder, providing much more control parameters over the entrance of the fresh air into the cylinder.

5.3 Intake port study

5.3.1 Simple ports: parametric investigations

Once the engine configuration has been selected and the most critical factors identified, a detailed parametric study has been performed on the intake ports geometry in order to understand how to best control the air flow pattern inside the cylinder. The main philosophy of this study is to identify the influence of each of the individual geometrical parameter over the intake air flow and its mixing process with the burnt gases already in the cylinder (from the previous cycle), in order to best control it and get the best scavenging profile possible, in terms of scavenging efficiency (push out as much of burnt gases as possible to the exhaust, i.e low IGR rate) and TR (avoiding short-circuit of fresh air to the exhaust, i.e high TR). The ideal pattern (providing a perfect scavenging curve, see Figure 5.1) would be to get a flat separation between fresh air and burnt gases, push it towards the exhaust while fresh air is entering the cylinder, and close the exhaust valves just as this perfect flat separation reaches them.

With this pattern in mind, general design of the intake is based on a circular row of 16 rectangular ports, from which the geometrical parameters to be investigated are the horizontal and vertical orientation of the ports, their height and their width (while respecting a minimum mechanical constraint of 2.5 mm between each port). A summary of these parameters, along with their ranges used for this study, is presented in Table 5.1.

Table 5.1. *Summary of the port parameters selected for the simple ports investigations.*

Parameter	**Range**
Horizontal angle	8 to 18°
Vertical angle	10 to 18°
Port width	8 to 12 mm
Port height	12 to 20 mm

For these investigations, the first parameter studied was the horizontal angle, thus aiming to the generation of swirl motion and promote the separation between fresh air and burnt gases. From the purely radial ports, a tangential angle was introduced, increasingly from 8° to 18°. CFD calculations were then performed to evaluate the effect of this angle over the scavenge main characteristic numbers. The results showed that increasing this angle had a critical impact on swirl levels (from 1.5 for 8° angle to 3.5 with a peak at 4.5 for 18° angle), but only a slight, yet positive impact on the scavenging curve, as shown in Figure 5.4. However, the resulting scavenging efficiency and TR are still very poor, with IGR rates at 90% of TR around 30 to 40%, and never reaching the 10% mark even with high over-scavenge. These results evidence how the horizontal angle alone is not sufficient to provide satisfactory results.

The best configuration here in terms of scavenging performances (with an angle of 18°) will then be used as base case and reference for the next studies,

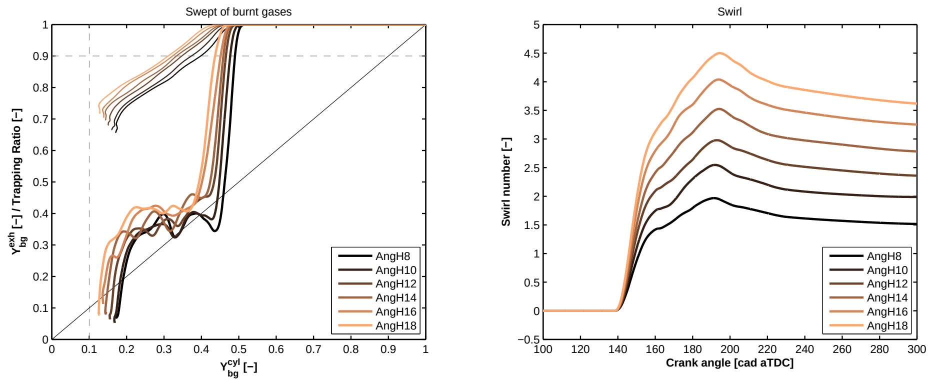

Figure 5.4. *Scavenging curves and Swirl profiles for various horizontal orientations of the simple intake ports.*

carrying the high levels of swirl from that point and trying to control it through other parameters.

Results from CFD calculations showed how a significant part of the residuals that are still in the cylinder by the end of the scavenging process originally comes from the piston bowl area, as the fresh air flow entering the cylinder is mainly going upwards, away from the piston and towards the exhaust valves. To help overcoming this trend, a downwards vertical inclination of the intake ports has been tested out, in order to orientate part of the intake air flow towards the piston bowl and help to remove those burnt gases. Starting from the reference geometry previously selected (with an horizontal angle of 18°), downwards angles from 10° to 18° have been tested. The results presented in Figure 5.5 show a slight improvement going from 10° to 14°, the vertical angle helping the separation between fresh air and burnt gases by injecting more of the fresh air under the burnt gases therefore boosting the push-up effect of the one over the other, additionally to improve the scavenging of the piston bowl. Also, the impact on swirl levels is negligible.

The effects of port angles have been investigated and results show some improvement paths, but the scavenging performances are still not satisfactory enough. Some additional parameters have to be considered and optimized, such as the ports width and height. These two parameters will change the port area, hence the intake section, modifying the general permeability of the engine. The first parameters to be investigated here is the port width: using the previous reference (which used 10 mm wide ports), the width was swept from 6 mm to 12 mm, but keeping the same number of ports. The total intake

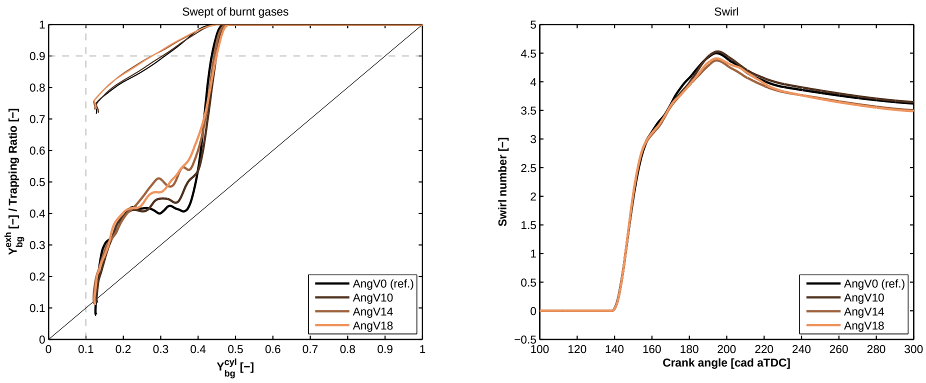

Figure 5.5. *Scavenging curves and Swirl profiles for various vertical inclinations of the simple intake ports.*

area was then changed by a factor 2 from one extreme to the other. The results in Figure 5.6 show a significant improvement when reducing the port width. This can be directly linked to the critically high swirl levels, and explained by the increased entry speed of the fresh air inside the cylinder combined with the horizontal angle of the ports. This combination helps to create this ideal separation between the fresh intake air and the burnt gases.

Narrower intake ports can then be identified as a key for improvement when combined with an horizontal angle, as the main drawback that is the reduction of permeability can be easily overcome by increasing the number of ports to still get maximum total intake area.

The last geometrical parameter left to investigate is the port height. However, while the width is a non-problematic parameter, one critical factor has to be taken into account when considering the port height: varying this parameters will not only influence the scavenging process (by affecting the total intake area), but it also mainly defines the timing of the opening and closing events of the intake. Therefore, it also affects both the effective expansion ratio (indirectly, as earlier intake opening might induce earlier exhaust opening) and the effective compression (directly). This will have a drastic impact on the resulting IMEP, and therefore the engine ISFC.

To further investigate these crossed effects, the scavenging 3D-CFD studies carried out in this workflow should be implemented into a 1D-model of the engine to evaluate the pros and cons of each port dimensioning and select the best definition(s). 1D-modeling have been carried out for this project by Macek et al. to study these crossed effects between scavenging performances and

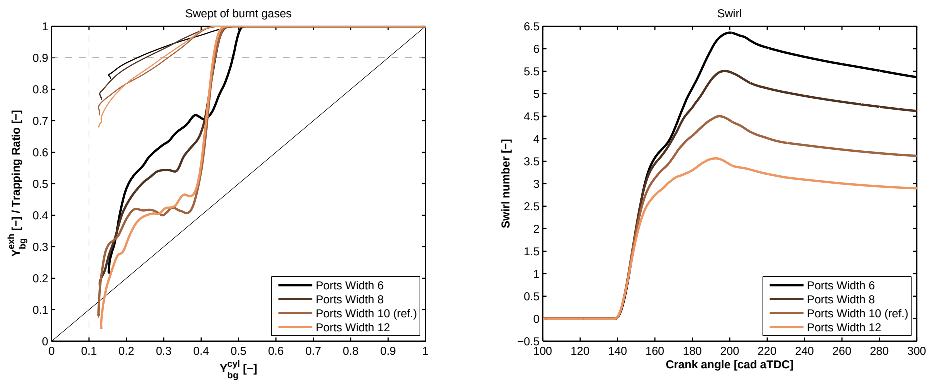

Figure 5.6. *Scavenging curves and Swirl profiles for various widths of the simple intake ports.*

engine outputs (performance and fuel consumptions) [3], and also to evaluate the turbo-machine needs to sustain the air-loop requirements for the whole engine operating range.

Bearing these considerations in mind for further analysis, the impact of the port height over the scavenging performances is however very low, as shown in Figure 5.7. While the swirl levels are slightly reduced when increasing the port height (because of the slower entry speed of the fresh air), the scavenging profile is almost unaffected by this parameter. The only aspect worth considering is the extended over-scavenging allowed by the larger amount of air passing through the cylinder (slightly lower IGR rate and much lower final TR).

Considering these results against the main effect of this parameter highlighted earlier, it was later decided to experimentally investigate two different port heights, thus recovering some flexibility for the intake timing that was intrinsically lost when selecting the port-scavenge Uniflow architecture definition. On the exhaust side, this flexibility is provided by the Variable Valve Timing (VVT) system, but two valve lift profiles were still designed (one for each intake port height) in order to be in accordance with each intake duration and permeability. This exhaust design process has been introduced by Galpin et al. [1] and will be summarized later in this paper, along with the related experimental validations.

Given all the results provided by the parametric investigations using simple port geometries, it appears clearly that such designs cannot provide satisfactory scavenging performances: high levels of swirl need to be quickly generated right at the beginning of the scavenging process in order to avoid either to

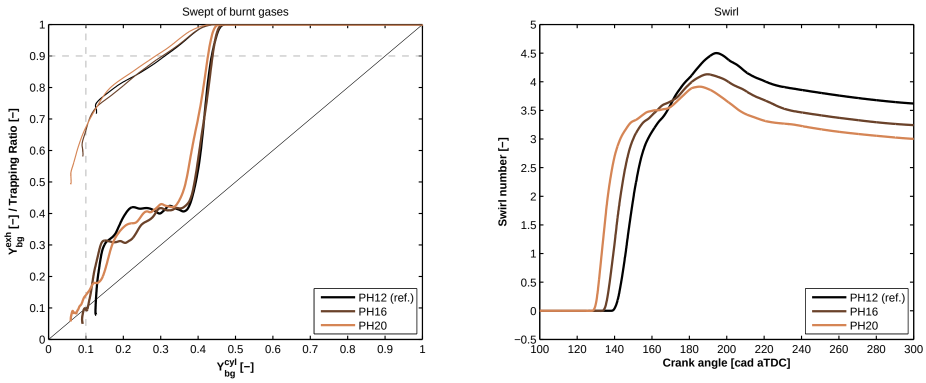

Figure 5.7. *Scavenging curves and Swirl profiles for various heights of the simple intake ports.*

mix the fresh intake air with the burnt gases, or to directly short-circuit it through the exhaust and leave large amounts of residuals inside the cylinder. But somehow during the process, the swirl level has to be drastically reduced to reach sustainable values by the end of the scavenging.

5.3.2 Complex ports: different geometry propositions and CFD optimization

The best solution to control swirl motion in such a way is to direct the flow to rotate in one direction first, then inject air in the opposite direction to counteract this motion and slow it down. Two design solutions can then be considered to achieve this goal:

- A two-stage port corona, with a first row oriented in a clockwise angle, and a second row with a counter-clockwise orientation (or vice and versa) (Figure 5.8 - left)
- A twisted ports corona, with a port gradually changing orientation along the cylinder axis, guiding the air flow progressively from a clockwise to a counter-clockwise direction (or vice and versa) as the piston goes down and unveils the ports (Figure 5.8 - right)

Both solutions have advantages and drawbacks, as the first one is quite standard in its design being the superposition of two conventional ports but reduces the permeability because of the vertical separation between the two stages, and the second one may offer higher flexibility in the design given the large number of geometrical parameters allowed but this also increases the complexity of the manufacturing process.

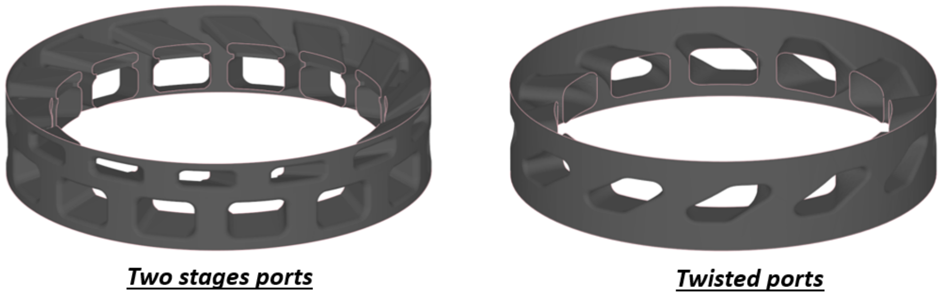

Figure 5.8. *Two propositions of complex intake ports designs: two-stage ports (left) or twisted ports (right).*

Considering that one of the primary objectives of this research is to achieve maximum scavenge performance, priority was given to higher permeability and the twisted ports geometry has then been thoroughly investigated. With that option of port opening, more than 110 CFD simulations have been performed to investigate in detail various top and bottom angle combinations, with different port heights or widths, taking into account that the height would also define the intake opening (IO) and closing (IC) timings.

Finally, the port height had to be fixed accordingly to the exhaust camshaft definition, which was performed by a project partner as described later in this section (see page 152). An intake duration of 102.8° was determined as optimum with the exhaust profile defined, which sets the port height at 16.2 mm. According to the previous studies of simple ports, a 14-ports corona with a width of 12 mm each seemed to provide a suitable base-case for this more complex design, with a reasonable level of swirl and good scavenging performance. Also, the downwards angle of 10° was conserved, as it seemed to help scavenging the piston bowl quicker even if it is not clearly noticeable on the final numbers in section 5.3.1.

Then, a DoE-like methodology was first followed to evaluate the order of magnitude of the angles needed to generate both swirl and counter-swirl. The settings used for these simulations are listed in Table 5.2 and the resulting trends, presented in Figure 5.9, demonstrate a clear logical path: the higher the top angle is to generate swirl as fast as possible, the higher the bottom angle needs to be to counter that effect and restore balance. As soon as this path is broken, not only the swirl levels are well over the limits, but also both IGR and TR are worsened, by too high short-circuit combined with either not enough or too much mixing between fresh air and burnt gases, thus exhausting not enough residuals and too much fresh air. Two cases then seem to fulfill all criteria: 65°/-60° and 75°/-70°. However, smaller angles would provide a better permeability due to a wider cross-section. The first case then seems to be a very good design to be further investigated, first under different conditions through CFD, then experimentally.

Table 5.2. *Top and bottom angles tested for the twisted port designs.*

		Top angle		
		55°	65°	75°
Bottom angle	−50°	X		X
	−60°		X	
	−70°	X		X

As a comparison and a display of the progress made throughout this optimization process, Figure 5.10 compares the fresh air paths for the most basic intake ports design (straight ports, with only a horizontal angle of 8°) with this last and most advanced geometry, during half of the scavenging / charging phase (from 140 to 180 CAD aTDC). In both cases, the exhaust valves are open and the cylinder is already decreased when the intake opens, but still slightly higher than the intake plenum pressure at the very beginning of this process, generating a small back-flow (which is then admitted back into the cylinder).

The optimized twisted ports are slightly higher, thus advancing somewhat the IC which explains the earlier entrance of the fresh air into the cylinder. But it can be observed that from the very beginning of the scavenging process, the fresh air enters the cylinder tangentially due to the swirling motion and thus avoiding to collapse at the center of the cylinder, unlike what can be seen with the basic geometry where the fresh air forms a “mushroom” shaped pattern and rapidly rushes through the center of the cylinder towards the exhaust. Within 20 CAD, fresh air already begins to be short-circuited, leaving a large amount of burnt gases inside the cylinder along the cylinder walls, which will then be mixed and impossible to exhaust without even more short-circuit. On the contrary, the optimized ports, by primarily generating this tangential motion of the fresh air, induce a much clearer separation between intake and exhaust gases, avoiding mixing them as well as the central rush observed previously. The swirling motion is then reduced by the counter-angle of the ports, and

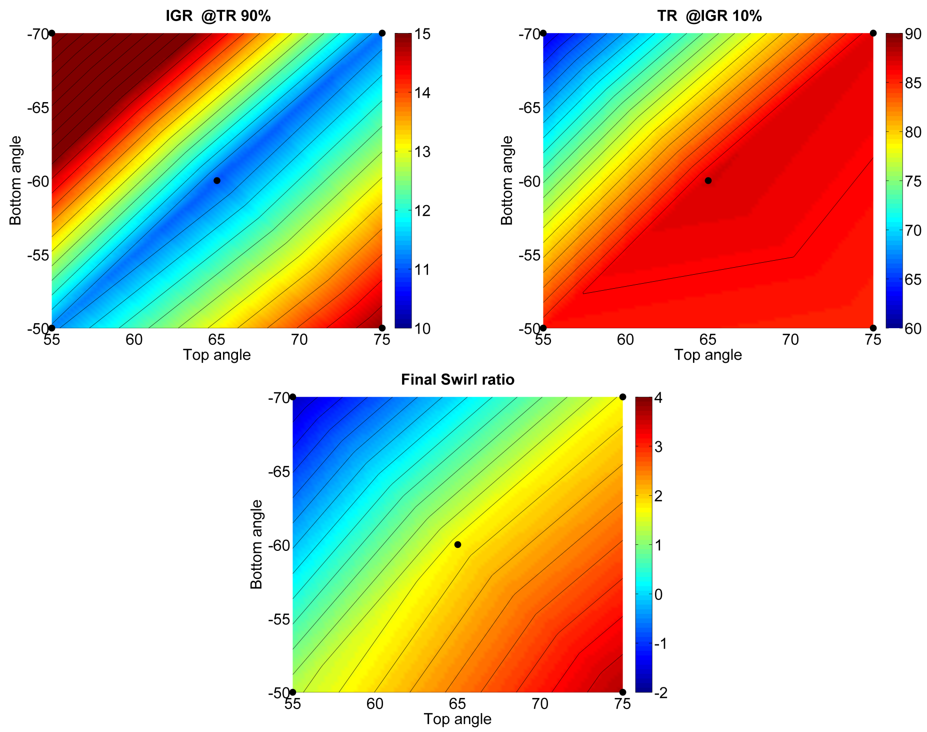

Figure 5.9. *DoE for the top and bottom angles and the twisted ports: Impact on IGR, TR and swirl.*

the fresh air that keeps entering the cylinder simply pushes the burnt gases up towards the exhaust valves, acting like a gaseous piston. By the time fresh air reaches the top of the cylinder, about 30 to 35 CAD after the start of the intake phase, almost all the burnt gases have been expelled from the cylinder, and what follows is only the over-scavenge phase done on purpose as described in the methodology of this study.

The associated scavenging / TR curves and characteristic numbers of these two cases are displayed in Figure 5.11 and Table 5.3.

As demonstrated earlier, the simple ports curve quickly falls to a perfect mix profile (and even below this line), generating high short-circuit and inducing a drop in TR, with an already very low value at 20% of IGR (not even reaching the targeted 10% or 15% of IGR at the end of the scavenging process). However, the optimized twisted ports scavenging curve confirms the good performance of this design, following the perfect scavenge line for the major part of this phase, only dropping for the over-scavenging phase. As a result, TR maintains a

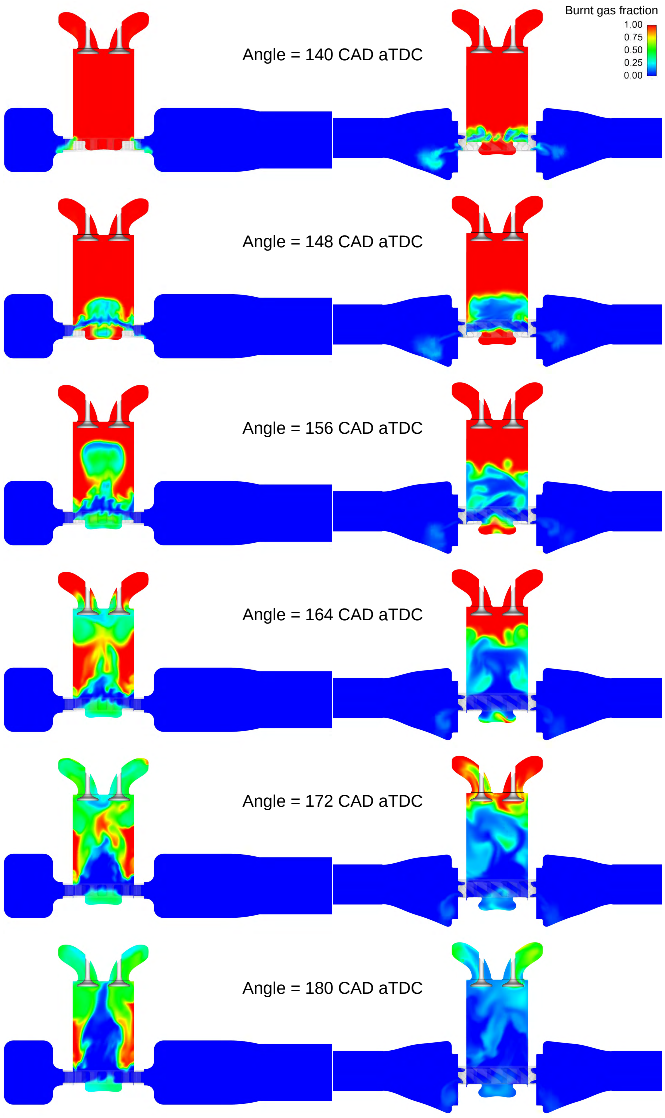

Figure 5.10. *Comparison of scavenging performance between initial simple port and final twisted port designs.*

100% value for the major part of the scavenging, and by the time the IGR level reaches 10%, TR is still around 90%, thus complying with the most restrictive targets of the ports design.

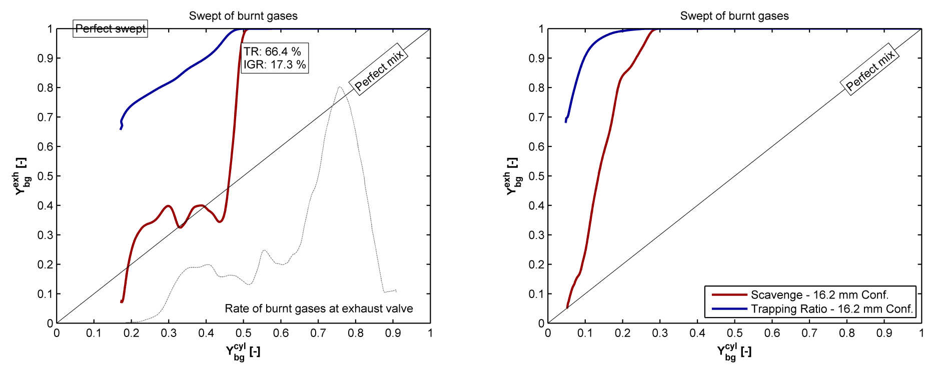

Figure 5.11. *Scavenging and TR curves for the initial simple intake port design (left) and the final optimized twisted port geometry (right).*

Table 5.3. *Characteristic numbers for the initial simple intake port design and the final optimized twisted port geometry.*

	Basic ports design	**Optimized ports design**
Trapping Ratio	66.41	68.71
IGR	17.29	4.83
Delivery Ratio	146.92	174.90
Charging Eff.	97.57	120.17
Trapping Ratio @10% IGR	-	90.81
IGR @90% TR	39.86	9.74
Delivery Ratio @90% TR	66.61	134.00
Charging Eff. @90% TR	59.95	120.60
Swirl	1.52	1.30

As mentioned previously, the exhaust valve lift profile also needs to be determined very carefully, as the dynamics of the scavenging depends on the time evolution of the in-cylinder pressure and on the pressures in the intake plenum and in the exhaust ducts, two of which are directly controlled by the exhaust valve lift law: the in-cylinder and the exhaust ducts pressure. Three phases can be distinguished:

- The blow-down: the exhaust valves open before the opening of the intake ports, allowing a quick decrease of the pressure in the cylinder.

- The scavenging phase: the ports and the valves are simultaneously open and the burnt gases need to be swept, as cleanly as possible, by the fresh air filling in the cylinder.

- The supercharging phase: the exhaust valves are closed while the ports are still open, the cylinder is then overfilled with fresh air thanks to the turbo- and super-charging systems.

The exhaust diagram then requires a fine definition of its timings and lift in order to find the best compromise between these three phases.

However, other parameters must be taken into account, such as maximum acceleration and maximum velocity of the valves, as those criteria are known to be critical for the valve-train reliability. It is worth to remind at this point that the valve-train will be equipped with a Variable Valve Timing system (VVT), allowing flexibility on the opening and closing timings. However, the lift needs to be well defined as it cannot be adjusted here and thus has to be viable for the whole engine operating range, and can generate either under- or over-scavenge when faulty. As an illustration, Figure 5.12 shows the difference in performance for two lifts of 6 mm and 8.5 mm.

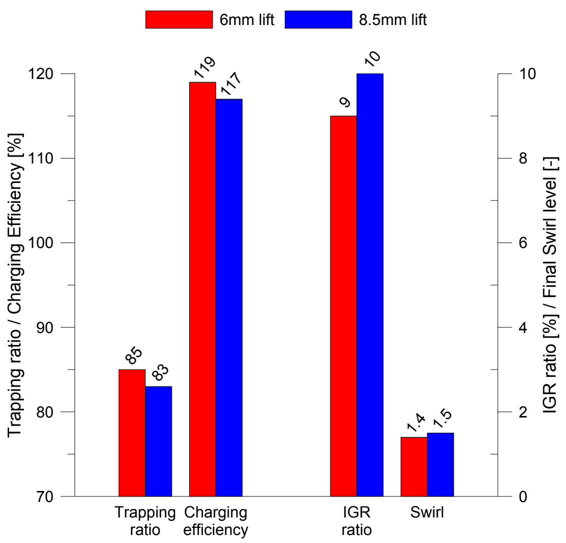

Figure 5.12. *Scavenging performance for two different valve lifts.*

Considering the small differences in performance in that case, mainly caused by the default timings used for this study (which has a larger impact than the lift, influencing the air flow by switching the cylinder pressure at opening and closing events), the lift can be defined following the maximum permeability target. Then, after various iterations, two exhaust law proposals are suggested for testing in accordance with the intake timing induced by the ports definition, providing two different maximum valve lifts (8.4 mm and 8.8 mm), and two different opening durations. The proposed diagrams are described in Table 5.4 and the corresponding the valve lift laws are plotted in Figure 5.13.

Table 5.4. *Exhaust camshaft definitions & characteristic angles.*

	Camshaft *C1*	**Camshaft *C2***
Max lift *[mm]*	8.4	8.8
Opening angle *[CAD aTDC]*	127.6	98.5
Closing angle *[CAD aTDC]*	209.6	190.5
Duration *[CAD]*	82	92

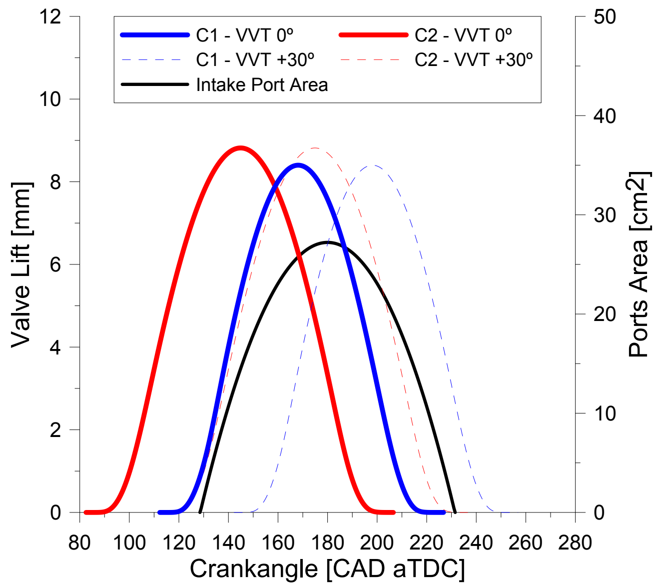

Figure 5.13. *Exhaust camshaft profiles.*

In parallel with this global definition of the main engine composition and structure in terms of scavenging, some studies have also been performed on the topic of the combustion chamber definition (mainly piston bowl shape and injection patterns). However, this work was not directly part of this research and was performed exclusively by the project partners. That is why it will not be detailed here, but all the necessary information and corresponding investigations are presented in the literature [1, 2, 4–6].

5.4 Experimental validation and optimization

5.4.1 Testing methodology

All the theoretical investigations and structural definitions now need to be experimentally tested and validated. This was performed during an doctoral internship at *IFP Énergies Nouvelles*, one of the project partners. The complete test cell and its instrumentation have been described in detail in Chapter 3, Section 3.2.5.

The objectives of this experimental study is to validate the potential of this engine architecture and the design of its components. Based on the experience and knowledge gained during the previous project, it is known that the scavenging and the combustion processes may have cumulative effects, but don't interact directly with each other. Therefore, they can be studied separately, and the first step here will be optimize the scavenging phase for best performance in terms of TR, IGR, and also monitor closely the fuel consumption, both indicated and corrected to carefully evaluate the supercharging cost and be able to compare to known references and targets (i.e. equivalent 4-stroke engine).

Those experimental investigations are performed using three-factor Central Composite Design (CCD) Design of Experiment (DoE) using the basic air-loop control parameters as inputs: the intake pressure (P_{int}), the pressure drop between intake and exhaust (ΔP), and the valve overlap ($Olap$). All the desired outputs were then measured experimentally before being linked to the inputs via response models. Those models were then used to generate two or three optimum input configurations, each one for optimizing a different response (ISFC, ISFC$_{corr}$, TR...), which were then repeated on the engine to validate the DoE response model and get the corresponding results for every output.

This methodology was repeated at 5 operating conditions, selected previously to be fairly representative of the engine operating range on the WLTP driving cycle (simulated with the engine coupled to a base-case gearbox and assembled in a base-case chassis / body). Those 5 operating conditions are presented on the engine map in Fig 5.14.

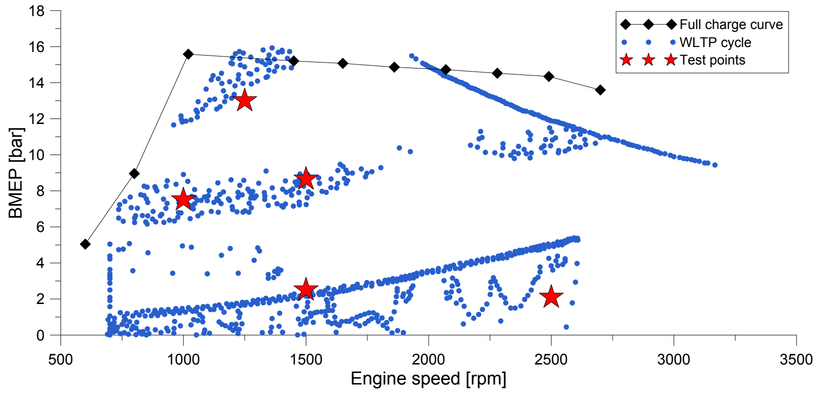

Figure 5.14. *REWARD engine map with 5-points WLTP representative operating conditions.*

5.4.2 Scavenging performance: results and discussions

5.4.2.1 Validation of the 16 mm twisted ports

For this part of the study, the engine is equipped with the 16 mm twisted intake ports developed throughout section 5.3.2 and with the C2 camshaft (92 CAD of opening duration).

Despite the experimental problems encountered along the tests, mainly due to lubrication issues causing engine damages and delaying the research, some interesting results could be obtained nonetheless during this study. The engine equipped with the newly designed complex intake ports proved from the very first measurements significant improvements over the poppet-valve engine in terms of scavenging performance, providing a larger flexibility in the air management settings and widening the operating window between knocking-like combustion and misfire conditions (higher ranges of intake and exhaust pressures, more gradual sensitivity to VVT position).

The DoEs performed at the different testing points demonstrated a drastic improvement in IGR rate (i.e. scavenging efficiency) from one engine to the other. For the 1500 rpm / 4 bar IMEP point as an example, Figure 5.15 shows the field of reachable points (here in terms of TR, IGR, ISFC and ISFC$_{corr}$) generated by the DoE methodology by varying the three input parameters defined earlier (P_{int}, ΔP, $Olap$) and by limiting this field to the points providing particulate emissions below a reasonable target.

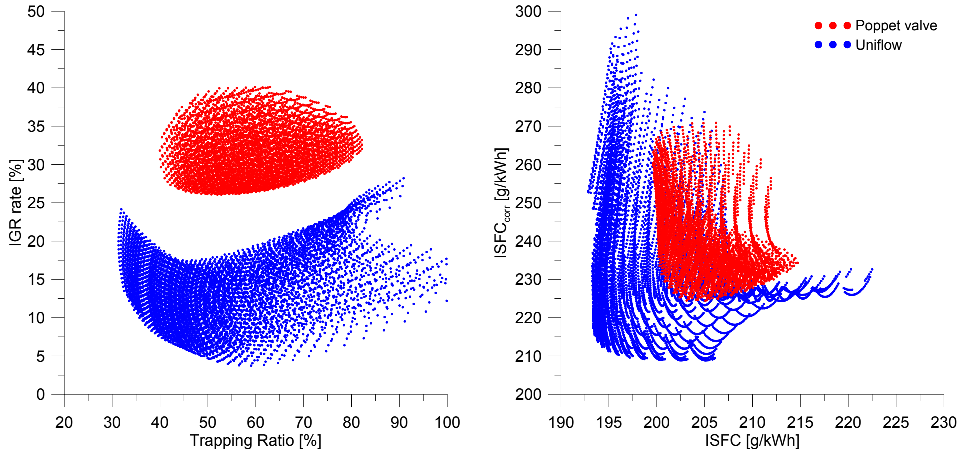

Figure 5.15. *DoE fields comparison of TR / IGR (left) and ISFC / ISFC$_{corr}$ (right) between Uniflow and poppet-valve engine at 1500 rpm / 4 bar IMEP.*

The scavenging improvement is clearly observable on the left graph of Figure 5.15, with the worst levels of the Uniflow engine being already better than the optimum of the poppet-valve engine. Also the TR range is widened, being able to reach high levels even with good levels of IGR (below 10%).

Those two effects combined have a direct benefit on both ISFC and ISFC$_{corr}$ (taking into account the work of the supercharger) shown on the right graph, the first one being improved through better cylinder composition (less residuals thus better and fresher air-fuel mixture, allowing a more efficient combustion and heat release with less heat transfer), the second one gained through the reduced work required by the supercharger, as more fresh air is retained inside the cylinder.

The Uniflow architecture along with this port design seems to prove great possibilities for good scavenging performance, allowing to reach levels much closer to those of a 4-stroke engine even with a very reduced scavenging time compared to them. However, even if this port geometry seems to have a lot of benefits (great control of the air path, good permeability), its main drawback for a potential production application is its manufacturing complexity. Indeed, the twisting geometry does not allow simple forging and machining processes and requires specific equipment and technologies. In this framework, and in order to simplify the design while keeping the same control over the air path towards the cylinder, a second design has been developed in parallel by the project partners from IFPEN: a two stages geometry.

5.4.2.2 Design and testing of the 16 mm two stages ports

This second design of the intake ports has been performed in parallel of the work previously described for the complex geometry evaluations (see section 5.3.2) by the project partners form IFPEN. The design process has been described in more details in previous publications [4–6], but as a summary the objective was to combine the simplicity of the basic port design and the control of the twisted profile. The superposition of two basic port coronas, each with a dedicated angle to follow a similar strategy as for the twisted profile described previously, then arose as an obvious option.

An optimum version of this intake port configuration is then selected for manufacturing and testing. The manufacturing process is easier with this design, as the orientation of the each port is constant along its height (and not transitioning from one extreme to the other as for the twisted definition), thus the horizontal angles do not need to be as pronounced to have a similar guiding effect.

The main drawback of this design is the loss of permeability due to the cylinder strip left to separate the two coronas and split the ports in two parts, which represents more than 6% of the total area (1 mm of the 16 mm total port height). To evaluate the impact of this reduced permeability criticality, this geometry then has to be experimentally tested and compared with the twisted design.

This comparison was performed using the same methodology and at the same operating conditions. And as it was partly expected, the permeability reduction combined with a less abrupt swirl peak at the beginning of the scavenging process (due to the lower horizontal angle of the ports) have a significant impact on the scavenging efficiency, as demonstrated by Fig 5.16, allowing only a small range of IGR rates. TR is also worsened by this design, even if maintained at reasonable levels, confirmed by the still competitive levels of ISFC (slightly improved) and ISFC$_{corr}$ (similar to the previous geometry).

However it was observed that this performance was improved at high loads, the higher intake pressure required helping to counter-balance the permeability reduction, helping to generate higher initial swirl and scavenge better the cylinder by recovering an air flow similar to the previous design, thus reaching IGR rates below 10% with still high TR levels. The high loads being one of the most critical points observed during the poppet-valve engine study, with

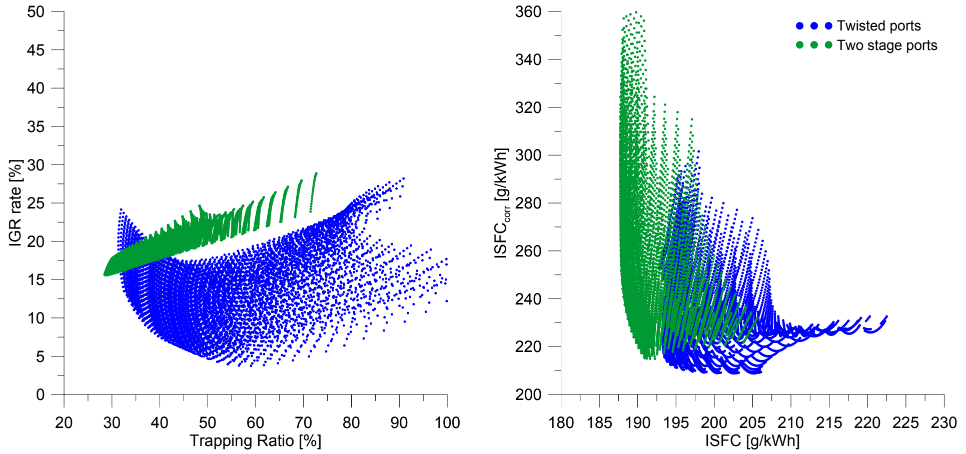

Figure 5.16. *DoE fields comparison of TR / IGR (left) and ISFC / ISFC$_{corr}$ (right) between different intake port designs at 1500 rpm / 4 bar IMEP.*

load limited by the cylinder composition, the main objective for the two stage port design can be considered fulfilled, even though some optimization is still required to reach its full potential.

This more conventional geometry, even though proven slightly less efficient when it comes purely to scavenging performance, is nonetheless a suitable alternative providing good in-cylinder conditions and allowing an efficient and clean combustion operating with the CDC concept. On this basis, the impact of intake timing was explored by designing new intake ports to adjust their total height.

5.4.2.3 Evaluation of the performance of 10 mm height ports

The main parameter of interest for this study was the increase of the expansion stroke with the aim of recovering mechanical work and thus improve the indicated efficiency. On the counterpart, this implies a reduction of the intake port height and a significant decrease of permeability. A port height of 10 mm was selected (switching the intake opening duration from 102.8 CAD down to 80.6 CAD), and the two stages design was scaled down according to this new definition taking advantage of its easy manufacturing compared to the twisted port design. To accommodate the reduction of the intake duration, the C1 camshaft (82 CAD of opening duration) was also installed in the engine.

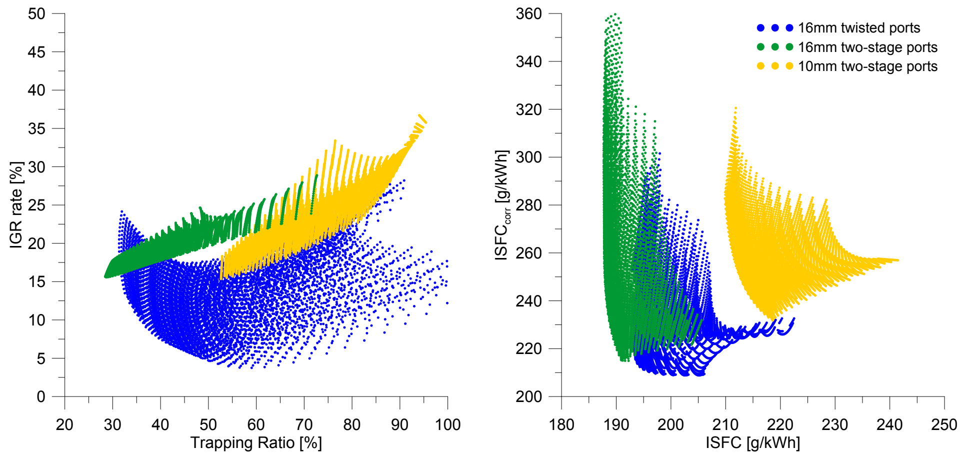

Figure 5.17. *DoE fields comparison of TR / IGR (left) and ISFC / ISFC$_{corr}$ (right) between different intake port heights at 1500 rpm / 4 bar IMEP.*

3D-CFD simulations performed during the re-design of these intake ports showed that scavenging performance would be similar to that of the 16 mm version, and as shows Figure 5.17 this trend was confirmed experimentally, even with an expected improved TR field due to the lower permeability combined with shorter scavenging duration, allowing less air mass flow through the cylinder thus less potential for promoting short-circuit. However, the effect on fuel efficiency was not as expected, with a significantly increased ISFC (by around 10%) despite the longer expansion stroke.

However, this loss in ISFC is not directly projected into ISFC$_{corr}$, which stays globally at similar levels compared to the 16 mm. This can be explained by the improved TR, which means that the work required by the supercharger consequently decreases. Unfortunately, due to critical mechanical failures (related to lubrication issues), those shorter two stages intake ports were not tested at higher loads. But it can be assumed that, contrarily to the higher ports, even if the air flow generating swirl inside of the cylinder might be partially recovered, this effect would not be sufficient to improve the IGR rate and restore good in-cylinder conditions to increase significantly the efficiency, while the lower permeability resulting from the reduced intake duration will turn into higher demands on intake pressures, thus supercharger works and finally increasing ISFC$_{corr}$ levels.

5.4.3 Benchmark of engine technologies

As a general overview of the final results of this study, the optimum case operating at 1500 rpm and 10.4 bar IMEP has been selected and compared to the results of the poppet-valve engine as well as to an equivalent state-of-the-art 4-Stroke CI engine.

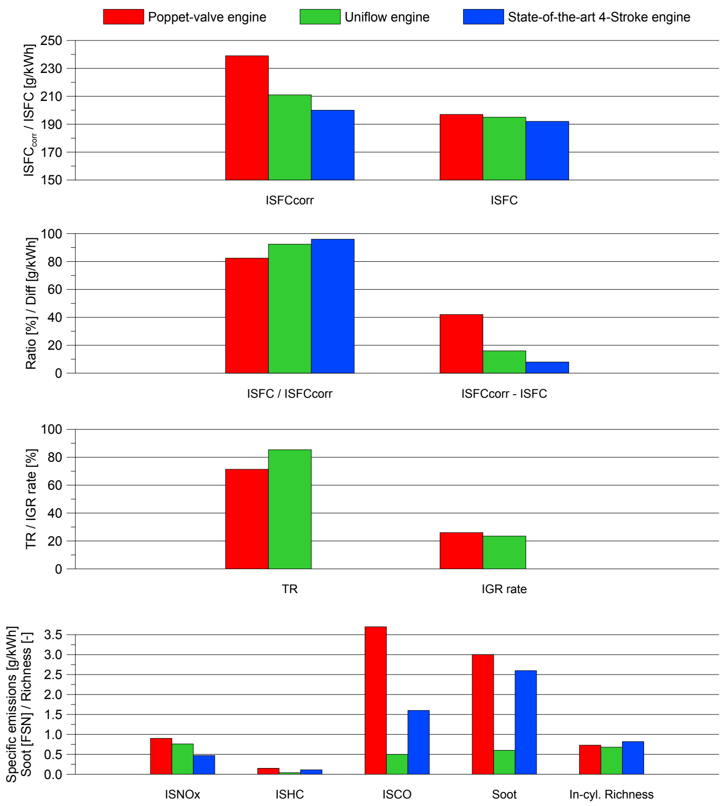

Figure 5.18. *Comparison of the poppet-valve 2-Stroke engine, the Uniflow 2-Stroke engine and an equivalent state-of-the-art 4-Stroke CI engine.*

Those results are displayed in Figure 5.18, and highlight the strong potential of the Uniflow 2-Stroke engine over the two other engines in terms of pollutant emissions, and also a large improvement of the engine efficiency compared to the poppet-valve 2-Stroke engine. Focusing on comparing the 2-Stroke engine architectures, it is evident how most of the benefit is observed in ISFC$_{corr}$, proving the lower requirements of the Uniflow configuration to reach similar scavenging performance, and its further potential to reach significantly better

TR and IGR levels if needed, which is particularly interesting for reaching higher loads.

It is important to remark that the Uniflow 2-Stroke engine provides in general better results in terms of emissions while keeping competitive fuel consumption levels compared to the fully optimized 4-Stroke engine, even without a dedicated combustion optimization in terms of related hardware and settings. This is an additional proof of the interest of following up with the research efforts that should focus on the full optimization of the combustion system operating with the CDC concept until reaching the best fuel consumption and emissions trade-off in order to confirm if this Uniflow 2-Stroke architecture is a feasible alternative to the traditional 4-Stroke engines.

5.5 Conclusion

The work presented in this chapter reports the design and optimization of a 2-Stroke engine architecture, both theoretically via 1D and 3D-CFD simulations and experimentally through engine testing on a test bench, with the objective of improving significantly the 2-Stroke architecture scavenging performance, as it was demonstrated to be a limiting factor for the implementation of LTC concepts such as gasoline PPC.

The research activities, carried out in the context of the European REWARD project and in collaboration with various project partners, were oriented towards the Uniflow architecture, and the work was focused mainly on the design of the combustion chamber definition and its surroundings, including the air flow direction, the ports design and the camshafts definition. Other aspects have also been investigated by the other project partners, such as piston and injector definitions on the engine design side, as well as multi-cylinder and driving cycle extrapolations on the theoretical projection side.

The engine design phase has provided a deep understanding of the scavenging process and the importance of the air flow through the cylinder. Indeed, it quickly appeared essential to control as good as possible the intake flow in order to scavenge the largest amount of burnt gases from the cylinder without short-circuiting fresh air to the exhaust. In this framework, the importance of air motion, swirl, and species separation could be clearly identified and the ports design process was carried out accordingly, and various port geometries have been selected for manufacture and experimental testing.

The experimental campaign, even though it was limited by lubrication and mechanical failures thus reducing the original expectations, allowed nonetheless the validation of the theoretical work performed upstream, confirming the drastic improvement obtained in terms of scavenging performance compared to the previous poppet-valve architecture. The target intake port designs each demonstrated some advantages and drawbacks, with sometimes a compromise arising between scavenging efficiency and manufacturability (mostly for low load conditions), or between expansion duration and engine permeability.

In any case, the preliminary experimental results (and theoretical extrapolations [2, 3]) obtained with this new engine definition and with the selected intake port designs have confirmed the potential of such architecture for implementing LTC concepts. It would be very interesting to further investigate the full potential of this engine operating with the CDC concept (as it was the case during the work described here) by optimizing more in-depth the combustion hardware (piston, injection system) and strategies (injection settings, EGR rate, intake and exhaust pressures fine tuning). Finally, performing a similar investigation under gasoline PPC operation and being able to compare with the work performed on the poppet-valve engine, and also compare with state-of-the-art 4-stroke engine, would be the final accomplishment strongly suggested for a future continuation of this research work.

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Chapter 6

Conclusions and future works

Contents

6.1 Introduction

The work presented in this thesis represents a significant contribution to the development and optimization of both the 2-stroke architecture in various forms and advanced combustion concepts such as the gasoline PPC concept, which have both the potential for being implemented in small passenger vehicles, either as a dedicated powertrain, or as part of a hybrid unit. The demonstration of the potential of these two distinct –and unconventional– paths towards cleaner and more efficient combustion highlights the possibilities offered by new concepts in general for light-duty vehicles, but also possibly for wider applications if looked upon with a more global point of view.

This last chapter of the thesis brings the main conclusions and contributions of the work performed during the investigation process described along the document. The most relevant results are outlined, and the main optimization paths explored so far are highlighted and extrapolated to suggest new directions which might allow further improvement of the performance and emission levels of internal combustion engines.

Finally, the last step of this chapter proposes possible research orientations based on the knowledge and experience gained during this work, together with recommendations for future tasks and studies to pursue not only the investigations and development of the already studied concepts, but also to combine ideas to generate new solutions for current and future challenges.

6.2 Summary and main conclusions of the thesis

The work presented in this doctoral thesis and performed by the research team surrounding the PhD student was motivated by various objectives and technological requirements from both the industrial and research worlds. Those objectives consisted of exploring optimizations paths for internal combustion engines for small passenger car applications, through both the development of advanced combustion concepts, and newly designed 2-Stroke engine architectures. All this work has been performed in the framework of two project: the post-POWERFUL private project between Renault SA and CMT – Motores Térmicos, and the European REWARD project within the Horizon 2020 research program; both of them have been designed as a continuity of the European POWERFUL project.

Through these investigations, and as described in the Chapter 1 of this document, the main objectives are focused on the reduction of pollutant emissions (mainly HC, CO, NO_x and PM) as well as the improvement of fuel efficiency, compared to state-of-the-art 4-Stroke CI engines for the same small passenger car application. The main guidelines of these objectives are imposed by the evolution of the European and worldwide regulations, which are more and more restrictive along the years. The first chapter briefly describes the methodology followed during the investigations to answer the problematic presented.

In order to summarize the most acknowledgeable works performed along the years on related topics, Chapter 2 presents a bibliographic review of the existing literature, focusing on the two main aspects of this thesis: advanced combustion concepts (insisting mostly on LTC and the different strategies available), and 2-Stroke engine architectures. This literature review allowed the identification of the main tendencies followed throughout the years and the most promising solutions for one specific application or another. This analysis lead to the hypothesis of combining the gasoline PPC concept with the 2-Stroke engine cycle, to try and overcome some of the main issues identified with the implementation attempts of HCCI and/or CAI combustion concepts on CI engines, both 2-Stroke and 4-Stroke. Furthermore, a wide number of investigations performed on 2-Stroke engines highlighted the scavenging limitations of this architecture for most of its forms, which were confirmed by the experimental results obtained during the first phase of the work described in this thesis, and then allowed the identification of a better design in order to overcome this restriction and set up the second phase of the work.

After analyzing the literature and identifying the main paths to follow, the work methodology of the different projects carried out during this thesis is described in the Chapter 3. This includes a detailed description of the tools, both theoretical and experimental, used during the investigations: the engines and their test cells are extensively described, since a very large part of the results depend on the pertinent exploitation of the available tools and setups. The control of the engines themselves, but also of the various components and parameters (air management, fuel settings, their flows through the engine), is critical for obtaining consistent and relevant measurements and results. In addition, various post-processing tools have been developed to facilitate the analysis of the data obtained and extract the most useful conclusions.

Also, some non-measurable parameters had to be estimated in order to fully evaluate the potential of both engines under certain conditions, as well as identify and understand some critical limitations. This was the case of the IGR (Internal Gas Residuals) levels for example, for which a dedicated evaluation process has been developed and implemented, with several levels of complexity / precision depending on the available computation capability and the required response timing. This process proved to be a key element for the evaluation of the scavenging performance of the poppet-valve 2-Stroke engine, allowing the characterization of the high levels of IGR in the cylinder, thus limiting the effective charge of the engine especially at full load operation.

This phenomena has been particularly described in Chapter 4, where the full results of the poppet-valve 2-Stroke engine investigations are extensively presented. This part of the work had several objectives, and was focused on the following items:

- To understand how to implement and control the gasoline PPC concept on a wide range of both engine speed and load. This work was performed mainly experimentally (combined to preliminary theoretical 3D-CFD calculations, performed by the research team and referenced throughout this document, that highlighted the fuel –and by extension the local richness– distribution), which lead to understanding the controlling factors of this type of combustion (ignition onset timing, combustion development) and its possible outputs (IMEP and ISFC, but also pollutant emissions).

- To try and optimize these parameters, by exploring experimentally the limits of the potential of this new combustion concept. Using either a DoE methodology or a parametric study depending on the parameters and their ranges, the investigations performed here allowed to identify the influence of various air management parameters (intake and exhaust pressures, valve opening and closing timings, EGR ratio...) and injection settings (number of injections and their individual timings and mass distribution, injection pressure...). This lead to identifying the main optimization paths for both fuel efficiency and pollutant emissions reduction, and some of the most common trade-offs, such as the NO_x / PM, could even be broken in some cases, which is a groundbreaking potential regarding the future of the automotive industry and its regulations.

- To evaluate in more detail the global potential of the 2-Stroke architecture for small passenger-car applications. The poppet-valve engine used during these investigation was designed to be cost-effective on a manufacturing point-of-view: based on the conversion of an existing 4-Stroke Diesel engine, one of the objectives of the POWERFUL project from which this engine is sourced was to be easily industrialized, avoiding the redesign of an entire production line. Thus, after the extensive evaluation of its potential under CDC operation allowing a direct comparison with equivalent 4-Stroke engines in terms of efficiency and pollutant emissions, the transfer towards gasoline PPC operation provided additional operating conditions and different, yet complementary outputs. Indeed, the engine efficiency was reduced when operating with this new combustion concept compared to CDC, but pollutant emissions reacted differently, reducing drastically some of them (mainly NO_x and PM) while other were increased (mainly HC due to the advanced injection timings generating wall impingement in some cases).

  However, the main issue was observed independently of the combustion mode operated: scavenging efficiency was the main limiting factor for proper operation at high loads. High levels of IGR are sometimes beneficial and even necessary, especially at low load were the in-cylinder pressure and temperature are lower: having hot residuals in the cylinder helps the onset of the combustion as well as NO_x reduction without having to inject EGR, and helps also the combustion stability by avoiding knocking-like conditions. But when the load increases, bad scavenging performance restricts the fresh air trapped inside the cylinder and the combustion performance is negatively affected, both in terms of efficiency by increasing heat losses due to higher in-cylinder temperatures along the cycle, and in terms of pollutant emissions due to the high in-cylinder richness induced. At a certain point, it even becomes impossible to reach full load, simply because of the lack of air inside the cylinder (global equivalence ratio quickly reaches stoechiometric levels).

The main conclusion of this Chapter 4 is then quite mixed: a promising potential for the gasoline PPC concept, which is allowed and helped in most of the operating range by the 2-Stroke cycle inherent characteristics; but this specific 2-Stroke architecture also induces critical limitations due to the bad scavenging performance of the poppet-valve configuration.

This issue was then one of the main motivation of the next European REWARD project, for which the objectives and content are fully described in the Chapter 5 of this thesis. The main goal was to design a fully new 2-Stroke CI engine starting from scratch (contrarily to the previous configuration) for small passenger car applications, with the objective of further improving fuel efficiency and pollutant emission, and also extending the operating range compared to previous results, which necessarily means improving the scavenging performance. Taking inspiration from the most efficient existing 2-Stroke CI engines –extremely large, marine engines– the Uniflow architecture was selected as a base architecture, and after some evaluations of both the potential and the sizing constraints, a high stroke-to-bore ratio was selected to further increase thermal efficiency (once again inspired by marine engines).

Once the global architecture had been defined, the full design process was then articulated around three successive main axis to determine and optimize the detailed geometry of the engine, firstly theoretically before being experimentally validated:

- The first investigations were focused on the general design of the engine, to select between the standard (intake through ports at the bottom of the cylinder, exhaust through head valves) or reversed Uniflow configurations. Using 3D-CFD simulations to test out various ports and pipes geometries for both configurations, the higher flexibility offered by the standard layout in terms of air flow control was quickly identified, therefore providing a clear selection. Even though this result was expected, this preliminary study was a first demonstration of how critical the air flow management is on this type of engine to control and improve the scavenging performance.

  In addition to these investigations, calculations were also performed to select the exhaust timing: knowing that the intake would have to be fixed and symmetrical in reference to the BDC and with only the ports height defining its duration, the exhaust had to be timed accordingly in order to provide good performance on the whole operating range of the engine (integrating a VVA system to get the required flexibility).

- The second part of the theoretical study was focused on the intake ports geometry, and the effect they have on the air flow paths inside the cylinder. A very intensive campaign of 3D-CFD calculations was performed, firstly with simple ports to investigate the effects of isolated parameters such as port number, width, height, vertical and horizontal orientations. These effects on the global scavenging performance were characterized according to specific output quantities easily available in CFD, such as trapping, scavenging and delivery ratios, charging efficiency, and also swirl levels, since this parameter is crucial to control at the end of the open cycle to provide good in-cylinder conditions before the combustion.

  Once the effects of these individual geometric parameters had been quantified and understood, some more complex port designs were proposed and tested through a large number of simulations. In order to obtain good scavenging performance with high trapping ratio and suitable swirl levels, two main port layouts were selected: a two-row geometry consisting of a first small crown of ports horizontally angled to generate swirl and a second crown of higher ports counter-angled to control the swirl level while filling the cylinder with fresh air; a second geometry consisting of single crown of twisted ports, with its top horizontally oriented to generate maximum swirl, progressively transitioning to a counter-angled bottom to help controlling the flow inside the cylinder. Both geometries provide different advantages and drawbacks, but the two of them display promising and comparable results. Two different heights (corresponding to two different intake timings) were then selected for each port geometry, and the four resulting liner designs were manufactured to be experimentally tested and validated.

- A DoE methodology was developed for the experimental campaign in order to extract as much information as possible from the various designs available. The section 5.4 of chapter 5 describes in detail the steps followed for this experimental validation, as well as some of the challenges encountered during this campaign. Despite a relatively low improvement of the fuel efficiency and pollutant emissions (mostly due to a lack of optimization of the combustion system and calibration), the most significant outcome of this experimental phase is the confirmation of the tremendous improvement of the scavenging performance compared to the poppet-valve architecture, almost reaching 4-Stroke levels in some cases. The four port geometries tested here provide different levels and ranges of performance, and the potential for improvement with optimized combustion parameters accordingly to a selected geometry appears quite clearly.

All these results, both theoretical and experimental, represent a critical breakthrough for 2-Stroke engines, and demonstrates the potential of this layout for reaching more restrictive targets of engine efficiency and pollutant emissions, while maintaining a compact packaging with a high power density. The gasoline PPC concept has demonstrated the possibility of breaking the NO_x / PM trade-off in some part of the operating range, and the full control of the combustion offered by this concept combined with the 2-Stroke cycle allows a very precise optimization, proving a wide potential when implemented under the right conditions. The new engine architecture proposed through the Uniflow design and its deeply optimized ports have demonstrated a significant step forward in terms of scavenging performance, but also in a more general way, in terms of in-cylinder air flow control, allowing an in-depth understanding of ways to keep the fresh intake air separated from the burnt gases and thus to extend the "perfect scavenging" phase as much as possible. Operating with the CDC concept in the framework of this project, the results in terms of fuel efficiency and pollutant emissions were already quite competitive compared to state-of-the-art 4-Stroke engines, but the range of improvement still available at the end of this project is significant, and expectations can be optimistic on this side.

6.3 Future works and further investigation paths

As highlighted all along the development of the thesis, both 2-Stroke engines and innovative combustion concepts present an important potential for the future of automotive transportation, weather they are used as a dedicated powertrain or as part of a hybrid unit in the vehicle. The compactness and high power-density offered by the 2-Stroke architecture, whichever its layout, opens to a wide range of possible applications for the automotive industry but also to other markets where 2-Stroke engine are already widely used but their technologies could be improved: high performance motorcycles; motorsport (Pat Symonds, chief technical officer of *Formula1*, recently declared an renewed interest for the introduction of 2-Stroke engines in *Formula1* as part of the hybrid powertrain and running with synthetic eco-fuels); even the already very efficient long-stroke CI engines on heavy-duty boats –from which the work described on the chapter 5 of this thesis was inspired– could be even further improved by implementing new technologies and new combustion concepts.

Following this path of further optimization, the work performed during this thesis can –and should– also lead to further development of the different ideas suggested along the investigations. Among these ideas, those which appear to be the most promising and pertinent could be listed as such:

- *Pursuing the work of this thesis by combining its two main topics:*

  The most obvious path to follow, after analyzing the results from chapters 4 and 5, is to combine the two main topics investigated: implementing and optimizing the gasoline PPC concept on the 2-Stroke Uniflow CI engine. Indeed, the main achievement of the gasoline PPC concept observed in chapter 4 is the reduction of both NO_x and PM emissions on a wide range of the operating map of the engine, and especially on the most critical parts regarding the regulations and their associated driving cycles as well as general real driving conditions. Whereas the biggest limitations here occurs through the engine architecture preventing proper scavenging at higher loads. On the contrary, the newly designed Uniflow architecture proposed in chapter 5 solves this scavenging issue by showing tremendous performance in terms of air control through the cylinder, but the CDC concept used during these preliminary tests was not conclusive in terms of fuel efficiency and pollutant emissions.

  So even though fuel efficiency might be the critical factor for this next investigation step (since gasoline PPC results are more focused on emissions than efficiency compared to CDC, speaking in a global point of view), implementing this new combustion concept and optimizing it on a properly scavenged engine offering higher flexibility on in-cylinder conditions and thus more range for further improvement of the combustion outputs, can be a clear follow-up investigation path.

- *Extending the exploration of partially and highly premixed combustions:*

  The gasoline PPC concept, and in a more general way the Low Temperature Combustion concepts implemented in CI engines, whether they are partially or highly premixed, are the focus of lots of investigations throughout the world and they demonstrate very promising results in a wide range of possible applications: light- and heavy-duty transportation (either as dedicated powertrains or parts of hybrid units), light and heavy marine applications, industrial and energy solution as generators... However, they often present limitations preventing their full development. As an example, the complexity of their calibration in the case of automotive applications due to their high sensitivity to ambient conditions is one of the limits for a wide industrialization and use. Also, in the case of the HCCI concept for example (but similar phenomenon are observed for most of the advanced combustion concepts), even if some manufacturers try to implement this solution on commercial vehicles (Mazda proposes this technology on the latest generation of their Skyactiv-X engine) they are usually restricted to implement this solution only at low engine speeds and low loads due to heavy knocking trends appearing when increasing one or the other.

The main objectives for next investigations towards this path would then be the implementation of new solutions to extend this operating range by an improved control over the combustion development, combined to a better adaptation to variation in the ambient conditions (those topics could even be complementary, with an improved control providing a higher general flexibility). This type of further development could be done through self-adaptive solutions such as closed-loop monitoring and control systems that could enable this possibility of real-time re-calibration and direct combustion control; or even solutions like artificial intelligence, which are nowadays becoming a predominant technology in a very wide range of applications, could be considered to predict and adapt the air management and injection calibration depending on the ambient conditions, but also on previous driving experiences of the vehicle and the driver. Those kind of solutions could be implemented on any type of engine calibration and combustion concept, but their contribution to advanced combustion concepts such as gasoline PPC or HCCI could be very valuable to further improve the potential of these technologies and reduce even more both pollutant emissions and fuel consumption.

- *Expanding the variety of engine designs:*

  Continuing on the idea initiated by the European REWARD project described in chapter 5, getting inspiration from the most efficient technologies available in different application fields can be a source of drastic improvements. The Uniflow architecture considered here is one example of this possibility, but other recent research have shown promising results with different innovative engine layouts: the opposed-piston geometry is one of the most fashionable solution from the last few years, and presents numerous advantages in terms of thermal efficiency –among others– due to its high cylinder volume-to-surface ratio, generating high indicated work with reduced thermal losses.

  There is a number of other architectures that could also be worth investigated and could provide further improvement in various areas, some very exotic such as the 5-Stroke engine that combines two 4-Stroke cycled cylinders with one 2-Stroke cycled cylinder to double the expansion of the burnt gases and recover more of their enthalpy; the Wankel rotary engine also presents some interest as a improvement path for compact and simple engines, due to its very limited amount of parts in motion but yet very high power density; as a last example, the Z-engine presents a similar configuration as a poppet-valve 2-Stroke engine with one combustion event for every crankshaft revolution, but with the scavenging capability of a 4-Stroke engine due to an extended exhaust event passed BDC and a very reduced intake event where the air is admitted into the cylinder at high pressure due to an upstream dedicated piston-compressor.

All these configurations have advantages and drawbacks up to some extent, being either their complexity for manufacture or control, or their constraining packaging. But some elements from each of them (or others) could inspire new designs and find a place in a fully innovative engine architecture, thus providing a good compromise and allowing the implementation of the next most efficient combustion technology.

The author, along with all his project partners, hope that this thesis and all the investigations carried out during the projects that built it have contributed to the development of new generations of internal combustion engines along with new advanced combustion concepts. Hopefully this work and the proposed investigation paths associated with it will inspire new projects and new solutions to keep improving the performance, efficiency and cleanness of future engines, thus extending the path towards transport solutions always more respectful with the environment. This way, the work presented here can be included as part of a more global overview and be considered useful for society on a larger scale, which is the primary objective for any engineer and researcher.

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