Fundamentals of Thermodynamic for Pressure-Based Low-Temperature Premixed Diesel Combustion Control

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This work is the result of a three years PhD reseach program between 2009 and 2012 that took place at the Institute of Internal Combustion Engines and Automotive Engineering at the University of Stuttgart under the guidance of Prof. Dr.-Ing. M. Bargende.

I started my association with Prof. Bargende during my time as a visiting scholar at the Ohio State University Center of Automotive Research, where between 2007 and 2008 I had the possibility to work as a research assistant in the group of Prof. Dr. G. Rizzoni. I would like to express my thanks to Prof. Bargende for giving me the opportunity to work in a modern and competent research environment, providing me with all the support needed for the accomplishment of this project. I am also grateful to Prof. Rizzoni for welcoming me at the CAR and for the assumption of the co-examiner.

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Abstract

This thesis shows the results of investigations about the combustion mechanisms of low-temperature premixed processes, in particular those in which only a partial homogenization of the fuel mixture is achieved, since they represent a realistic solution for commercial engine applications. Measurements has been carried out on a modern 6-cylinder diesel engine equipped with a rapid-prototyping *ECU* and in-cylinder pressure sensors. The engine has been operated on a dynamic test bench. On the base of thorough combustion analysis a new closed-loop combustion control strategy has been defined. This has been completed by a procedure for allowing to switch the combustion mode to conventional diffusive combustion extending the drivability range up to engine full-load.

In a first step a sensitivity analysis of the system has been carried out in order to identify engine constrains and mayor system limitations. A first selection between useful actuating and target parameters has been made. The most relevant combustion mechanisms have been also highlighted. Matching these information, targeted measurements has been done under the variation of multiple parameters. In this way a clear target application area in the *EGR*-injection timing map could be defined. Optimal combustion phasing has been found for combustion beginning at *TDC* and large rates of *EGR*. This has been done for low engine operation mode where charge dilution did not represent a limitation. However, increasing the engine load, because of the larger injection mass by almost constant fresh air quantity, it was necessary to limit the *EGR* rate for avoiding soot production and a deterioration of the fuel conversion. Therefore, the combustion process has been phased later in the expansion.

After the identification of the combustion mechanisms and the application target, a new combustion control strategy has been defined. The goal is the accomplishment of the defined requirements concerning engine protection, i.e. rate of pressure rise and stability, fuel consumption and emissions. The proposed control strategy foresaw a control of the engine load over the injection duration, a control of the maximum rate of pressure rise over the injection timing (using a block-injection) and a control of the oxygen concentration in the charge over the

EGR valve. In this way a clear separation of the controller tasks was possible. When the engine load and the engine protection have been continuously provided by the first two controllers, which has been done indivudually for each cylinder, the charge quality could be varied in order to increase combustion efficiency and reducing emissions.

The control system has been completed by a switching strategy to and from conventional combustion, which was needed for extending the drivability range. Rapid changes of the charge dilution, the injection pressure and the injection strategy has been identified to be mainly responsible for high rates of pressure rise or, inversely, for the deterioration of the combustion process. On the base of measurements, dynamic parameters correction functions are proposed. This solution provided an effective switching procedure characterized by a monotonic engine torque signal and low combustion noise.

The proposed closed-loop control strategy has been tested under stationary and transient engine operation. The *NEDC* has been splitted into characteristics parts and adapted for test bench measurements. For a reliable comparison of the measured data, special attention has been given to a high reproducibility of the test cycle. The urban part of the cycle could be driven using solely the closed-loop controlled alternative combustion, whereas a switch to alternative combustion was needed in the extra-urban part of the cycle. The proposed strategy showed high robustness and optimal response even under high dynamic load changes.

Kurzfassung

Die vorliegende Arbeit zeigt die Ergebnisse einer Untersuchung der Verbrennungsmechanismen von homogenisierten Niedertemperatur-Prozessen, insbesondere in Bereichen in denen eine Teilhomogenisierung des Kraftstoffgemisches erreicht wird und darum einen realistischen Lösungsansatz für kommerzielle Motoren darstellen. Messungen wurden am einem modernen vollindizierten 6-Zylinder Dieselmotor, ausgerüstet mit einem Rapid-Prototyping Steuergerät, durchgeführt. Der Motor wurde mit einem dynamischen Prüfstandsystem betrieben. Auf der Basis einer gründlichen Analyse wurde eine neue Regelungsstrategie der Verbrennung definiert, welche, durch die Definition eines Umschaltvorgangs zur konventionellen diffusiven Verbrennung, vervollständigt wurde. Damit wurde der Ausbau des möglichen Applikationsbereiches bis zur Volllast erweitert.

In einem ersten Schritt wurde eine Sensitivitätsanalyse des untersuchten Systems durchgeführt. Dabei wurden motorspezifische Applikationsgrenzen und Limitierungen des Brennverfahrens identifiziert. Zusätzlich wurde eine Selektion der nutzbaren Regel- und Zielgrößen durchgeführt und die relevanten Mechanismen der Verbrennung beleuchtet. Auf Grund dieser Informationen wurden spezifische Untersuchungen durchgeführt, bei denen mehrere Größen gleichzeitig variiert wurden. Die Kombination der Ergebnisse in der Form von Isolinien, dargestellt in einem AGR-Einspritzzeitpunkt Diagramm, erlaubte die Definition eines klaren Applikationszielbereiches. Eine Wirkungsgrad optimale Lage der Verbrennung wurde bei einem Verbrennungsbeginn um den oberen Totpunkt und bei hohe Abgasrückführungsraten gefunden. Dieser Bereich ist bei niedrigen Motorlasten erreichbar, da hier die starke Ladungsverdünnung keine Limitierung darstellt. Bei höheren Lasten muss hingegen, auf Grund der erhöhten Kraftstoffmasse bei praktisch konstant bleibender Frischluftmenge, die AGR-Rate limitiert werden, um starke Ruß-Emissionen und eine Verschlechterung der Kraftstoffumsetzung zu vermeiden. Darum findet die Verbrennung später im Expansionstakt statt.

Nach Identifikation der Verbrennungsmechanismen und des Applikationsziel-

bereiches wurde eine neue Regelstrategie definiert. Anforderung an diese Strategie war, Grenzen bezüglich Motorschutz, d.h. maximale Druckgradiente und Verbrennungsstabilität, Kraftstoffverbrauch und Emissionen einzuhalten. Die vereinbarte Strategie sieht eine Regelung der Motorlast über die Einspritzdauer, eine Regelung des maximalen Druckgradienten über die Lage der Einspritzung (Blockeinspritzung) und eine Regelung der Sauerstoff-Konzentration der Frischladung über das AGR Ventil vor. Dieser Lösungsansatz erlaubt eine klare Trennung der Aufgaben der Regler. Während die Einhaltung der Last und die Motorschutzfunktion von den ersten beiden Reglern gewährleistet wurde, konnte die Qualität der Ladung variiert werden, um den Wirkungsgrad der Verbrennung zu verbessern und die Emissionen zu senken. Außerdem erlaubte diese Strategie eine zylinderindividuelle Anpassung des Verbrennungsvorgangs.

Die Regelung der Verbrennung wurde um eine Umschaltung der Betriebsarten erweitert, welche einen Ausbau des möglichen Applikationsbereiches bis zur Volllast erlaubt. Schnelle Variationen der Ladungsqualität, des Rail-Druckes und des Einspritzmusters wurden als verantwortlich für das Auftreten von starken Druckgradienten oder, umgekehrt, für das Erlöschen der Verbrennung identifiziert. Deshalb wurden aufgrund gezielter Prüfstandmessungen, Korrekturfunktionen entwickelt. Diese Korrekturfunktionen erlaubten eine wirkungsvolle Umschaltung der Betriebsarten, welche durch einen stetigen Momentenverlauf und niedriges Motorgeräusch gekennzeichnet ist.

Die vereinbarte Regelstrategie wurde im stationären und transienten Motorbetrieb getestet. Der $NEFZ$ wurde in für dynamische Messungen am Motorprüfstand relevanten Abschnitte aufgeteilt. Dabei wurde besonders auf die Reproduzierbarkeit des Testzyklus geachtet,um eine möglichst gute Vergleichbarkeit der Ergebnisse zu erzielen. Der Stadtteil des Zyklus konnte ausschließlich im geregelten teilhomogenen Betrieb gefahren werden, während für den außerstädtischen Teil eine Umschaltung in die diffusive Verbrennung notwendig war. Die Robustheit und das gute Führungsverhalten der vorgeschlagenen Regelstrategie konnten auch im stark dynamischen Betrieb bestätigt werden.

Contents

Nomenclature

Acronyms

$ACTION$	Advanced Combustion Technology to Improve engine-Out NOx, Ricardo Ltd.
$ADEC$	Advanced Diesel Engine Control, Ricardo Ltd.
$ATDC$	After Top Death Center
$BTDC$	Before Top Death Center
CA	Crank Angle
CCS	Combined Combustion System, Volkswagen AG
CI	Compression Ignited
CO	Carbon Monoxide
$CO2$	Carbon Dioxide
$DCCS$	Dilution Controlled Combustion System
DI	Direct Injection
DPF	Diesel Particulate Filter
DOC	Diesel Oxidation Catalyst
ECE-15	Urban Driving Cycle
ECU	Electronic Control Unit
EDC	Electronic Diesel Control
EGR	Exhaust Gas Recirculation
EVC	Exhaust Valve Closing
$EUDC$	Extra-Urban Driving Cycle
$HCCI$	Homogeneous Charge Compression Ignition
$HCLI$	Homogeneous Charge Late Injection
$HPCC$	Highly Pre-mixed Cool Combustion
$HPLI$	Highly Premixed Late Injection
HRR	Heat Release Rate

ICE	Internal Combustion Engine
ILC	Iterative Learn Control
IVO	Intake Valve Opening
LTO	Low Temperature Oxidation
$MIMO$	Multiple Input Multiple Output
MK	Modulated Kinetics, Nissan Motor CO. Ltd.
$NEDC$	New European Driving Cycle
NOx	Nitrogen oxide[1]
NTC	Negative Temperature Coefficient
PAH	Polycyclic Aromatic Hydrocarbon
$PCCI$	Premixed Charge Compression Ignition
PID	Proportional-Integral-Derivative controller
PM	Particulate Matter
PSG	Pressure Sensor Glow Plug, BorgWarner BERU Systems GmbH
RPC	Rail Pressure Control
RPM	Revolution Per Minute
SI	Spark Ignited
$SISO$	Single-Input Single-Output
$SULEV$	Super Ultra Low Emission Vehicle
UHC	Unburned Hydrocarbons
VGT	Variable-Geometry Turbocharger
VVA	Variable Valve Actuation

Latin symbols

$CA10$	10% of integral heat release, combustion beginning	$^\circ CA$
$CA50$	50% of integral heat release, center of combustion	$^\circ CA$
c_p	Heat capacity at constant pressure	J/kgK

[1]Can refer to a binary compound of oxygen and nitrogen, or a mixture of such compounds

c_v	Heat capacity at constant volume	J/kgK
E_A	Apparent activation energy	J/mol
$dp/d\theta_{max}$	Maximum rate of cylinder pressure rise	$bar/°CA$
$imep$	Indicated mean effective pressure	bar
l	Connecting rod length	mm
$nmep$	Net mean effective pressure	bar
$p_{Cylinder}$	Cylinder pressure	bar
p_{Exh}	Exhoust pressure upstream the turbine	bar
p_{Int}	Intake manifold pressure	bar
p_{max}	Maximum cylinder pressure	bar
p_θ	Cylinder pressure at defined piston position	bar
$\tilde{R}$	Universal gas constant	$J/molK$
r	Crankshaft radius	mm
s_φ	Piston position as a function of the crank angle	mm
$\overline{S}_p$	Mean piston speed	m/s
$T_{Cylinder}$	Cylinder temperature	K
T_{Int}	Intake manifold temperature	K

Greek symbols

ϵ	Engine compression ratio	$-$
λ	Air to fuel equivalence ratio	$-$
κ	Isentropic index	$-$
φ	Crank angle variable	$°CA$
τ_{id}	Ignition delay	ms
θ	Crank angle	$°CA$

1

Introduction

1.1 Motivation

Modern diesel engines has been shown to be a feasible solution for the light-duty and passenger car sectors. With the introduction and continuous improvement of the common-rail (CR), the direct injection (DI) system, and the use of multiple-stage turbo-charging in combination with cooled exhaust gas recirculation (EGR), diesel engines offer a greater efficiency compared to spark ignited (SI) gasoline engines by comparable comfort and drivability. A major drawback of this technology are the high particulate matter (PM) and nitrogen oxides (NOx) emissions, which mainly result from the diffusive combustion process.

Future legislative regulations require a reduction of the emissions produced by automotive combustion applications, and in particular, a significant reduction of the nitrogen oxides and soot emissions must be accomplished in the coming engine generation. Using conventional combustion processes however, such as the diesel diffusive combustion, this can be done only using cost expensive exhaust gas aftertreatment systems. In fact, the diffusive combustion process is characterized by a trade-off mechanism between these two pollutants and a simultaneous reduction of both components is not possible. Moreover, using exhaust gas aftertreatment systems, the complexity of the electronic control unit (ECU) software structures and the time needed for their application increases significantly, resulting in additional costs.

Low-temperature premixed combustion processes has been demonstrated to be a feasible solution for reducing NOx and PM engine-out emissions. Increasing the mixture homogenization and reducing the combustion temperatures provide a NOx and smoke-less combustion. This is the case of completely homogenous processes. However, the large attentions that have been given to this solution during the end of the last century has been lowered by drawbacks such as high combustion instability and poor fuel conversion. Large difficulties has been also encountered in the application of these processes in commercial engine applications. A compromise has been found by providing only a partial homogenization of the fuel mixture, timing the injecting event towards the end of the compression stroke. This solution has shown greater potential for series engine applications, but problems related to the control of the combustion process still remains the major drawback of this technology. This is mainly due to the high system sensitivity to small changes in the charge composition. Moreover the use of this combustion process is limited in the low to middle load range and a solution for switching to conventional combustion must be found. Particularly during transient engine operation is the realization of a torque-neutral switching procedure a difficult task.

The recent introduction of low-cost in-cylinder measurement sensors, together with the increased computation capacity of modern ECUs, has opened new perspectives for the application of alternative combustion processes. On the base of the pressure data measured in the combustion chamber, usually carried out with a resolution of one degree crank angle($^{\circ}CA$), valuable information about the combustion process can be determined, such as the combustion beginning and the indicated mean effective pressure (*imep*). These information can be implemented in a closed-loop control. This feature offers a solution to the most part of the above mentioned drawbacks related to the high sensitivity of the low-temperature premixed combustion.

1.2 Objectives

In the last decade, many authors have proposed new closed-loop pressure-based combustion control systems. However, most part of the described solutions rely on combustion models and statistical analysis of the system, and are not based on a thorough understanding of the combustion mechanisms. The object of this work is to show that a closed-loop combustion control strategy, based on a detailed investigation of the thermodynamic mechanisms taking place during the combustion process, can be a feasible solution for the application of low-temperature premixed combustion in modern diesel engines.

Based on an extensive system sensitivity analysis performed on a multi-cylinder turbo-charged diesel engine equipped with a rapid prototyping ECU and in-cylinder pressure sensors, the most relevant mechanisms of the combustion process must be indentified. On the base of these mechanisms, actuating and target variables as well as a new closed-loop control strategy must be defined. The strategy must take targets such as high fuel conversion and combustion stability, and low emissions and noise into consideration. To complete the control strategy, a procedure for allowing the combustion mode to be switched from alternative to conventional diesel combustion and inversely must be designed. This should provide a steady monotonic increase (or decrease) of the engine torque. Finally, the proposed strategy must be evaluated under stationary and transient engine operation.

2

State of the Art

New legislative emission requirements can be achieved increasing the quality of the combustion process as well by reducing engine tolerances. Near the possibility of drastically lowering engine-out emissions exploiting diesel alternative combustion processes, closed-loop combustion control seems to be a promising solution for the next engine generation. Aim of this chapter is to give an overview of the works that can be found in the literature about these topics.

2.1 Diesel alternative combustion processes

2.1.1 From diffusive to low-temperature premixed combustion

Nowadays, diffusive combustion, here called DI combustion[1], is still playing a dominant role in the diesel engine combustion process. Thanks to consistent improvements during the last decades, DI combustion is now characterized by high efficiency, a stable combustion, low emissions and low noise. Despite that, modern diesel engine are still trapped in the trade-off mechanism between NOx and PM emissions, which result from the combustion of high-temperature lean-mixture regions, and from cold and fuel-rich areas of the injection spray respectively.

[1]Direct Injection Compression Ignited *DI-CI* to differentiate from Direct Injection Spark Ignited *DI-SI*

Characteristic for DI combustion is a multiple injection strategy for the low-load operation range and a single injection event at higher loads. In the low-load range the main injection is anticipated by one or more short fuel injections, so called pilot injections, which are mainly responsible for increasing the combustion chamber temperature prior to the main injection event. The fuel, injected in small quantities, has enough time to mix with the gases in the combustion chamber and is than compression ignited. Therefore, a small premixed combustion take place. Due to the resulting temperature increase the combustion delay of the main injection, which is responsible for the pressure increase and thus the engine torque, can be drastically reduced. There is short or no time for the mixing process and the combustion of the main injection takes place almost completely diffusive. Main advantages of this solution are a low combustion noise and low engine vibrations. At higher loads this strategy is not adopted since the combustion chamber conditions by injection event, i.e. temperature and pressure, are sufficient to ignite the fuel spray rapidly. Also in this case, due to the lack of time for the mixture formation, the combustion has mainly a diffusive character.

The fuel/air distribution in a fuel spray is strongly heterogeneous. While in the core of the fuel spray any oxygen can be found, i.e. the air/fuel ratio (λ) is equal to 0, other regions of the combustion chamber are just filled with air, thus $\lambda \to \infty$. In the edge regions of the spray cone, where the drops[2] are smaller and fuel is partially evaporated, there is a wide range of fuel/air concentrations, going from 0 to ∞. If this mixture is ignited, e.g. by diffusive compression ignited processes, it results a very heterogeneous temperature distribution as well. This two phenomena, i.e. regions with a wide range of λ and the large temperatures distribution, are the main source for emissions in DI combustion. In fuel poor areas in the combustion chamber, where $\lambda \to \infty$, the temperature is too low for thorough oxidation causing UHC emissions. The highest temperatures are measured in the areas surrounding the spray, where $\lambda > 1$, and are mainly responsible for NOx formation. In fuel rich areas inside the spay cone, where $\lambda < 1$, CO and soot formation takes place [2].

The purpose by the development of modern diesel engines is to avoid NOx and PM emissions formation during the combustion processes. In the past years,

[2]As the liquid jet leaves the nozzle it becomes turbulent and spread out as it entrains and mixes with the surrounding air. The outer surface of the jet breaks up into drops [3].

different technical solutions has already proved their potential. The fuel mixing process can be significantly improved by increasing the injection pressure, reducing injector hole diameter and increasing the injector hole number. Beside a better distribution of the fuel in the combustion chamber the local λ range by combustion onset can be reduced between 0.8 and 0.9 with a significant decrease of the soot formation [4]. A large portion of the generated soot can be reduced toward the end of the combustion by late oxidation processes, which can be promoted increasing the turbulence in the combustion chamber during the expansion stroke. Soot oxidation up to 95% is possible [2]. An increased mixture homogenization has also benefits regarding lower NOx formation. A further significant reduction of NOx emissions can be achieved by lowering the combustion temperature. This can be done recirculating large amounts of exhaust gases or reducing the compression ratio. This last solution is always related to a reduction of the combustion efficiency. However, as long as the diesel combustion process takes place under (diffusive) heterogenous conditions, a trade-off mechanism between NOx and PM emissions can not be completely avoided.

Alternative solutions to the diffusive diesel combustion process have been a major topic in the engine combustion research and development of the last 40 years. Homogeneous and partial-homogeneous combustion processes have been found to be a promising solution in reducing NOx and PM emissions without significantly increasing fuel consumption, or even improving it. In consideration of what mentioned above, new combustion processes have to be based on the following two mechanisms:

- increased charge homogenization
- low combustion temperature

Injection strategy is the key factor for promoting mixture homogenization. The choice of a correct strategy mainly depends on the properties of the used injection system and the engine configuration, such as the piston bowl geometry for an instance. Large magnitude charge flow in the combustion chamber, such as high swirl or squish, may also promote homogenization if applied correctly. A longer ignition delay results in an increased time for mixture preparation. This can be achieved recirculating exhaust gas and cooling down the charge prior to enter the combustion chamber. In fact, reducing the oxygen content the mixture reactivity can be lowered prolonging ignition delay. On the same time EGR, which does not take part in the combustion reactions and has a larger heat capacity compared to fresh air, plays an important role in reducing the combustion temperature. Finally, both mechanisms can be further promoted by reducing the compression ratio.

Nowadays investigated alternative combustion processes mainly differ from the kind of fuel injection system and thus from the fuel/air distribution at ignition time. There exist two main categories. In the first one, a near homogeneous mixture is obtained exploiting an early fuel injection event. Combustion phasing is dominated by the kinetics of the chemical reactions. This mainly depends from the charge composition and is decoupled from the injection timing. The equivalence ratio λ at combustion onset is everywhere larger than 1. In the latter, the injection event is closely coupled to the combustion phasing, but only a partial homogenization of the mixture is achieved prior to combustion. In both cases, low combustion temperatures and low rates of heat release are obtained by charge dilution using EGR. Therefore, in this work a differentiation is made between full homogeneous and partial homogeneous combustion processes. An exhaustive explanation is given in the following sections. An overview of the most renowned combustion processes and their collocation in a temperature/λ map is shown in Figure 2.1.

2.1.2 Homogeneous combustion processes

Combustion processes in which thorough mixture homogenization is achieved prior to ignition are known under the names of Homogeneous Compression Charge Ignition ($HCCI$) and Premixed Charge Compression Ignition ($PCCI$). Characteristic is the simultaneous ignition of multiple areas within the combustion chamber, which are triggered by compression. The main objective of $HCCI$ combustion is to reduce NOx and PM emissions. The potential of a zero-emission concept using HCCI combustion has been demonstrated under laboratory conditions (using stationary and controlled boundary conditions). However, the realization of this concept in modern engines is not possible, yet, and for this reason, this work is not focusing on this kind of combustion process. For the sake of completeness, only a brief description is given in this section.

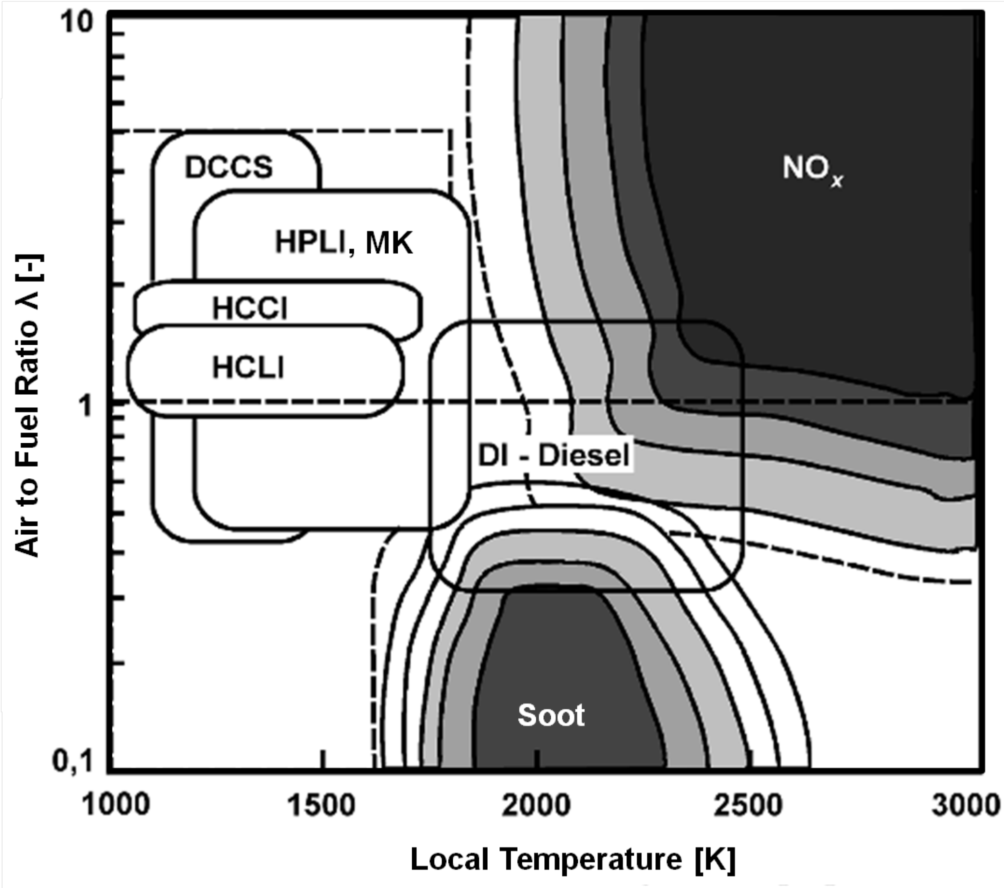

Figure 2.1: Collocation of the alternative combustion processes in the temperature-λ-map [2].

A complete mixture homogenization can be obtained using either a port injection system, such as a diesel atomizer [5], or injecting the fuel early in the compression stroke using an high pressure direct injection system. In the second case, for avoiding fuel wall impingement it is made use of a reduced injection cone angle. Due to the long mixing period a lean mixture is obtained in the entire combustion chamber so that a smoke-less combustion takes place. Moreover, the lean mixture burns at relatively low temperatures which results in less NOx formation. However, from the results of combustion observation and numerical simulation, the need to prevent the fuel spray from adhering to the cylinder liner and combustion-chamber wall has been identified [6] and still represent a major drawback of this combustion process.

$HCCI$ is usually operated only at part-load. Since the start of combustion is determined by the kinetics of the chemical reactions, influencing the in-cylinder conditions is the only way to control combustion phasing. Therefore, engine compression ratio, fuel properties and the degrees of freedom of the air path become the main limiting factors for achieving higher loads. Beside the need of assuring an appropriate auto-ignition timing over a large range of operation conditions, increasing the load, thus approaching stoichiometry, the combustion becomes unstable and knock-like cylinder pressure oscillations occur [7]. This is particularly difficult in transient operations, where the control of the desired cylinder charge compostition becomes an even greater challenge.

Another obstacle of HCCI engine operations are the relative high emissions of unburned hydrocarbon UHC and carbon monoxide CO which result from fuel wall impingement and the low combustion temperatures of the lean combustion, and are symptomatic for a low fuel conversion. Moreover, in $HCCI$ processes combustion may occur completely during the compression stroke, because of the very short combustion duration, with a consequent efficiency reduction and a higher combustion roughness.

2.1.3 Partial homogeneous combustion processes

A description of LTC processes in which only a partial mixture homogenization is attained is presented in this section. Partial homogenization mainly means that even if an equal fuel/air distribution is promoted, the mixture still shows a wide λ distribution at combustion onset. In other words, the combustion takes place under heterogeneous conditions but the occurrence of extreme fuel rich and lean areas is avoided.

Differently to $HCCI$, the fuel is injected towards the end of the compression stroke. Since the use of pilot injection is not made, the in-cylinder conditions by injection event are not sufficient to ignite the fuel and a mixing process takes place prior to combustion. Main advantage of this strategy is the possibility of phasing the combustion simply shifting the injection timing. Therefore, together with the air path, the fuel path can be used for controlling the combustion process. Another advantage of partial homogeneous combustion processes compared to the full homogeneous ones is the reduced need of engine hardware modifications. In fact, in the most cases, a conventional high pressure injection system can be utilized as well as a conventional air path. Fuel wall impingement is avoided injecting the fuel into the piston bowl[3].

Very low NOx and almost zero PM emissions can be achieved by the correct application of partial homogeneous combustion processes. To this aim, large exhaust gas recirculation is needed. Since high EGR and low combustion temperatures lead to combustion deterioration and higher UHC emissions, it results a target conflict which affects the application of these processes. Many different solutions has been found in the past years in order to overcome this problem. An overview of the most significant ones is given in the following paragraphs.

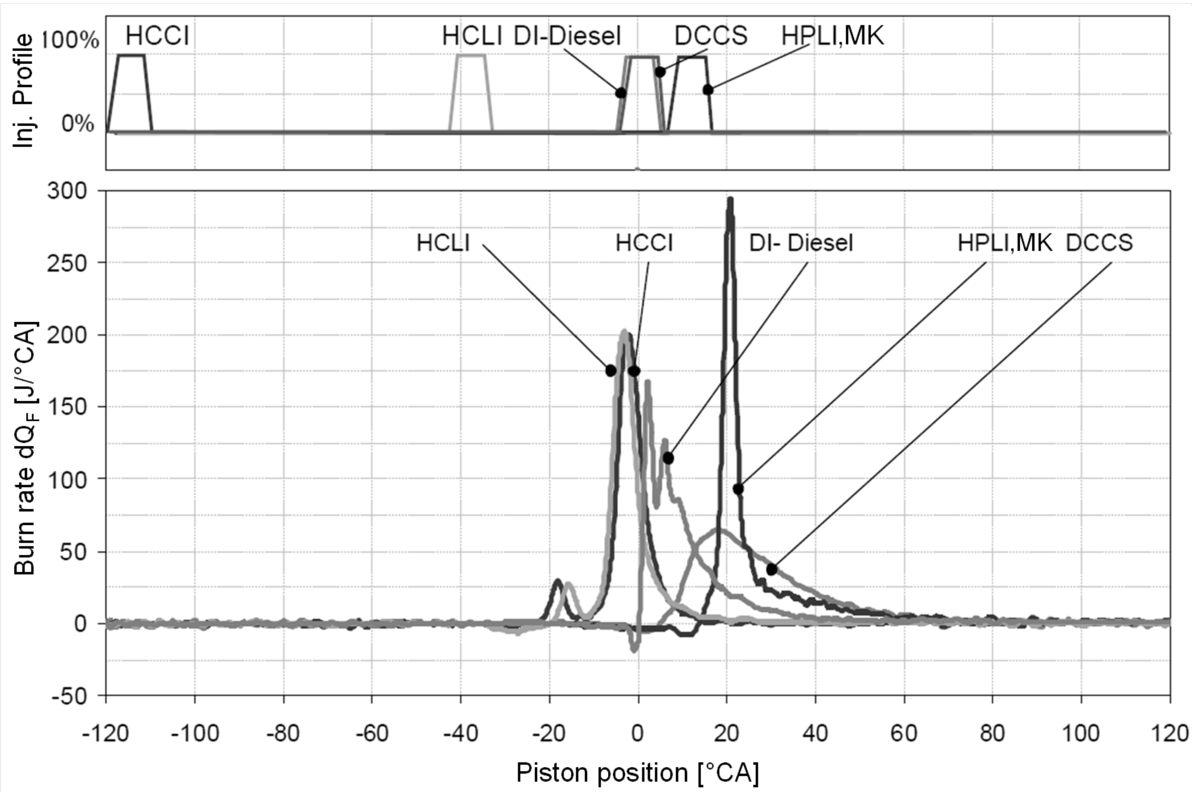

Figure 2.2: Chracteristic injection profiles and rate of heat release traces of the most significant alternative combustion processes [8].

[3]The correct interaction between injection spray and piston bowl must be always provided and represents a limiting factor for the mixture preparation in partial homogeneous processes.

Homogeneous Charge Late Injection

Characteristic for Homogenous Charge Late Injaction ($HCLI$) combustion processes is a high rate of mixture homogenization, which can be realized promoting long combustion delay timing the injection event late in the compression phase and using high rates of exhaust gas recirculation. Similar to the $HCCI$ combustion process, the mixture charge is well homogenized prior to ignition and the combustion takes place almost simultaneously everywhere in the combustion chamber [8]. A two phases heat release with cold and hot fuel reactions can be also observed. However, differently to $HCCI$, since the fuel is injected using a conventional high pressure injection system, wall wetting can be avoided.

Soot formation is avoided thanks to the high mixture homogenization. The typical λ distribution is enclosed between 0.9 and 2. High rates of EGR up to 80 % are responsible for low combustion temperatures and the suppression of NOx formation. On the other side, the high rate of mixture homogenization is the reason why the operation range of $HCLI$ combustion is usually limited at $imep$ below $4\,bar$. Since the stoichimetric mixture is characterized by fast fuel converstion, increasing the fuel quantity, thus approaching stoichiometry, the rate of heat release becomes too high.

In [9] is shown how combustion phasing can be varied increasing combustion efficiency. Measurements on a one-cylinder engine, supported by CFD simulation of the mixture formation, show an optimum between $35°CA$ and $15°CA$ before TDC, depending on the operation load. A sensitivity analysis of different engine parameters influencing $HCLI$ combustion using a 30 holes central injector in a one-cylinder light-duty diesel engine has been carried out in [10]. A similar approach is proposed by AVL List GmbH in [11], where, under laboratory conditions, the potential of reducing engine out NOx and PM emissions using $HCLI$ combustion is showed. A very similar approach is proposed by RICARDO under the name of Highly Pre-mixed Cool Combustion ($HPCC$). This is the result of an investigation named $ACTION$ (Advanced Combustion Technology to Improve engine-Out NOx) which target is a cost effective reducing of NOx emissions to achieve Euro 5 legislative requirements [16]. Using an advanced water coolant circuit to reduce EGR temperature and a twin stage turbocharger, optimized for low pumping losses, in a low compression ratio engine (ϵ 15.5:1) a NOx reduction of 85 % relative to Euro 4 limitations is achieved.

An injection event near TDC by high injection pressure up to 1800 bar is used. Combustion phasing is optimized for improving efficiency and promoting long ignition delay.

Highly Premixed Late Injection

Increasing the load, and therefore the fuel injection quantity, it is required to retard the injection event phasing the combustion later in the expansion phase. This is necessary for avoiding high rates of pressure rise. Since the air to fuel ratio λ is already reduced by the large injected fuel mass, charge dilution using EGR can only be used moderately for promoting ignition delay and thus the mixing process. It results a different combustion process, the so called Highly Premixed Late Injection ($HPLI$). The mixing time is shorter compared to the $HCLI$ process and regions with λ lower stoichiometry are still present by combustion onset.

Optical investigation of the combustion process has shown that in $HPLI$ combustion soot formation is not avoided during mixture formation but a strong oxidation of the soot takes place in the second part of the combustion, which is still characterized by high temperatures [8]. However, phasing the combustion late in the expansion it results in a poor fuel conversion and higher CO and UHC emissions compared to DI combustion. NOx-less and smokeless combustion is still possible, but the importance of correct combustion phasing increases drastically. The high exhaust gas temperatures may also become a limitation for this process.

Using a commercial passenger-car diesel engine an optimal injection timing close to TDC has been found in [9]. In [11] it is shown how injecting after TDC it is possible to expand the operation range of LTC combustion up to 8 bar of $imep$.

Mudulated Kinetics

One of the first realizations of a partial homogeneous combustion processes has been published by Nissan Motor CO. Ltd. in [12]. The proposed concept is a low-temperature premixed combustion system that aims at simultaneously reducing NOx and PM emissions. The combustion temperature is lowered by applying heavy EGR rate, which also result in a slower combustion velocity and thus lower rates of pressure rise. Thorough premixed combustion is obtained prolonging the ignition delay by retarding the fuel injection short before or after the TDC. Essential condition is a complete injection of all fuel prior to ignition. Additionally, high swirl ratio improves fuel dispersion reducing both smoke and HC emissions.

Further investigation of the so called Modulated Kinetics (MK) concept, also known under the name of late $PCCI$, are shown in [13]. Using cooled EGR, increased injection pressure and a reduced engine compression ratio, a reduction of the NOx emissions up to 90 % compared to DI combustion is achieved simultaneously with a drastic reduction of the smoke level. Moreover, the process can be expanded to the entire load range of everyday driving.

Dilution Controlled Combustion System

By Dilution Controlled Combustion System ($DCCS$) NOx and soot formation are suppressed by applying up to 80 % of cooled EGR to a conventional DI combustion multiple injection strategy. Already by EGR rates of 40 %, which limits the combustion temperature below 2000 K, NOx emissions can be easily avoided. By a further increase of the charge dilution, reducing the oxygen content down to 9-10 % and limiting the combustion temperature by 1500 K, smokeless combustion can be realized regardless of injection timing or injection pressure [14], in other words, NOx and soot formation is suppressed independent of mixture quality by combustion onset (over-retarded injection may results in misfire). However, under near stoichiometric or even under fuel rich operating conditions, CO and UHC emissions increase drastically and the thermal efficiency is deteriorated.

By the combustion process developed by Toyota [15], the so called Low Temperature Oxidation (LTO), in a restricted operating range low noise combustion and sufficient exhaust gas temperature to activate catalyst reactions are obtained, even under idling conditions. Moreover, by introducing this high temperature and rich exhaust gas to a NOx storage reduction catalyst, NOx generated at higher load can be completely purified.

2.2 Pressure based diesel engine closed-loop control

For the accomplishment of the new legislative emission limits, substantial improvements in the engine control system are necessary. Together with the increasing complexity of new exhaust gas aftertreatment systems and their management, future engine control system must be capable of ensuring application targets and combustion stability over a long engine lifetime regardless of fuel quality.

Pressure based closed-loop combustion control represents a feasible solution for the coming ECU generation. With an evaluation of the indicated pressure, using dedicated algorithms, the combustion process can be adjusted in response of variations of disturbance variables, such as charge composition or temperature, in order to attain emissions, noise and efficiency targets. Moreover, closed-loop combustion control has the potential of enabling the use of NOx-smoke-less alternative combustion processes, which, as explained in the previous section, are affected by combustion stability problems due to the high sensitivity to the charge composition. For this reason, many different approaches for controlling LTC combustion have been published in the last decade. Thanks to the recent development of cost affordable pressure sensors which are integrated in the glow plug (PSG, BorgWarner BERU Systems GmbH) and are able to feed commercial ECUs with in-cylinder pressure inofrmation [17], the interest in a closed-loop combustion control is grown significantly for alternative as well as for standard diesel combustion applications.

Coherently to the aim of this work, i.e. the realisation of a pressure based closed-loop control strategy for the low temperature pre-mixed combustion, the following paragraphs mainly focus on the state of the art regarding closed-loop combustion control of lot-temperature premixed processes in which in-cylinder pressure sensors are used. For more information about other solutions, e.g. emissions based, noise based ore generally model based combustion control strategy, it is referred to [1].

To the best knowledge of the author, the first realization of a pressure based closed-loop combustion control focused on balancing the indicated torque between the cylinders adjusting the fuel injection quantity and has been presented in [18]. Since there is a direct physical correlation between indicated cylinder pressure and injection quantity, or simply the injection duration, this control can be realized using a single-input single-output (SISO) system. Together with a reduction of the engine vibrations, this solution provides a continuous compensation of engine aging processes. More complicated is however the control of other relevant combustion criteria, e.g. the combustion beginning, the center of energy-conversion or the maximum rate of pressure rise.

In [19], the authors propose a closed-loop control based on a pressure trace evaluation method called Pressure Ratio Management. Dividing the actual pressure trace of a fired cylinder by the stored motored pressure, the amount of heat release per unit charge mass is obtained. Information about the combustion phasing and the air/fuel ratio can be obtained and implemented in a closed-loop system for controlling the spark timing and the rate of EGR. This method has been demonstrated on a SI engine with lean combustion but can be also applied to DI or LTC combustion.

Exploiting the different autoignition properties of iso-octane, which is self-ignition resistant, and n-heptane, which on the contrary easily self ignites, and creating a mixture with a specific rate of these fuels adjusting the injected amount respectively, a closed-loop control of the $CA50$ in a $HCCI$ multi cylinder engine is shown in [20]. The same authors propose a concept which exploit the potential of a variable compression ratio (from 9:1 to 21:1) to phase the $CA50$ in a $HCCI$ prototype engine [21]. Since the compression ratio can not be adjusted individually for each cylinder, the $CA50$ is additionally corrected by means of λ, thus an equal distribustion of the $imep$ can not be provided. An further improvement of this concept using Fast Thermal Management is shown in [22]. Rapidly changing the charge temperature using an electrical inlet air pre-heater, the system response of the combustion phasing by step changes is reduced to only 8 engine cycles, 57% faster compared to the previous achievements. However, the intention of these works is mainly to demonstrate a functioning control system for $HCCI$ engines operated under laboratory conditions rather than to propose a concept for the use in commercial engines.

Under the name of $EmIQ$, for Intelligent Emission Reduction, AVL List GmbH proposes a combined DI-LTC strategy which aims in reducing emissions while maintaining good drivability [23][24]. Fundament of this concept is the pressure based closed-loop combustion control which provides targeted combustion phasing for both DI and LTC mode by shifting the injection timing, and maintains engine noise in a acceptable range by tuning the EGR rate and thus the maximum pressure rise. This solution has been first tested and validated using a one-cylinder engine equipped with a fully electro-hydraulic valve train, which is capable of rapidly change the rate of EGR, but is realized in the full engine only using the cooled EGR loop and the throttle valve. Although a cross-sensitivity between the control parameters and the non-respective disturbing variables is observed, a sufficient control quality is achieved via two independend $SISO$ controllers. A significant NOx and PM emissions reduction is achieved by applying $HCLI$ and $HPLI$ combustion at low-load and switching to DI combustion for higher torque requirements. In LTC mode consistently higher CO and UHC emissions are measured.

Rapid adjustments of the charge composition, i.e. pressure, temperature and rate of dilution, are not possible using a standard EGR loop with cooling bypass. A fully flexible Variable Valve Actuation (VVA) system has been demonstrated to be a feasible solution to this aim. In [25], using a single-cylinder research engine equipped with a VVA system and run by a 99% propane fuel blend, the potential of controlling $HCCI$ combustion by means of exhaust residual gas has been demonstrated. Near the possibility of controlling combustion phasing over the air/fuel equivalence ratio, an increase of the initial mixture temperature, for promoting the auto-ignition process at low-load, is also possible. Another concept exploiting the potential of the VVA technology is shown in [26] under the name of Combined Combustion System (CCS), Volkswagen AG. In LTC combustion mode, the mixture reactivity, and so the rate of pressure rise, is controlled setting the effective compression ratio and the residual gas fraction. Optimal combustion efficiency, by a $CA50$ short after TDC, is obtained controlling the injection timing of a single injection event late in the compression. High rates of EGR are obtained using both high and low pressure EGR. A similar concept has been proposed by the BMW Group on a gasoline engine equipped with dual VANOS (variable valve timing) and Valvetronic (variable valve lift) systems, water cooled EGR loop and direct injection [27]. The proposed closed-loop control provides a cylinder balancing of the $imep$ over the injection mass, and of the $CA50$ by means of residual gas fraction. To ensure auto-ignition, torque balancing and a low combustion noise during transients, an online adaptation of the controller output takes place in a neural network of type LOLIMOT[4] which combines the information from the pressure measurement with a cylinder charge model for optimal valve timing. Finally, to overcome the problems related to the high costs of a fully VVA system and its realization in commercial applications, Robert Bosch GmbH proposes a closed-loop control concept that works with a partly flexible valve train with cam phasers [28]. Differently to a fully VVA system, were charge composition can be adjusted individually for each cylinder, the cam phaser has a global effect on all cylinders. Based on this technology, a Multiple Input Multiple Output ($MIMO$) control strategy for gasoline $HCCI$ combustion is proposed. In a cascade control structure, the $imep$ and the $CA50$ (target value is set at $8°CA$ after TDC) are controlled using injection quantity and timing respectively. The residual gas fraction is than adjusted over the cam phaser in a secondary inner loop in order to additionally phase the combustion. Aim of the inner loop is to minimize the deviation of the averaged injection timing from its feed-forward value. Alternatively, to the aim of an exhaust aftertreatment strategy, exhaust λ can be also adopted as target parameter for the inner loop. The high temperatures required for the auto-ignition process are obtained by a marked negative valve overlapping (early EVC and late IVO).

Through the $ACTION$ project, presented in the previous section, Ricardo Ltd. has demonstrated the potential of highly low-temperature premixed combustion as a practical approach to significantly reduce engine-out NOx emissions. In a further collaboration with GM, which aims in achieving US Super Ultra-Low Emission ($SULEV$) and Tier 2 Bin 5 requirements without the use of NOx aftertreatment systems, which is designated under the name $ADEC$ (Advanced Diesel Engine Control), a closed-loop control of the combustion process is developed [29]. Using in-cylinder pressure sensors, the $ADEC$ coordinates fuel injection quantity and timing with the air path providing a $CA50$ balance between the cylinders, a limitation of the maximum pressure and combustion stability. This control strategy has been demonstrated on a 1.9-liter diesel engine with advanced air handling system, a two-stage turbocharger and an advanced exhaust gas recirculation with cooling bypass. The closed-loop control of the low temperature pre-mixed diesel combustion is also object of a study presented

[4]Local linear model tree (LOLIMOT) is a type of Takagi-Sugeno-Kang neural network fuzzy algorithm.

by Delphi Corporation in [30]. Using a rapid prototyping unit connected to a combustion analysis system on a multi-cylinder engine, the in-cylinder pressure traces are evaluated for the estimation of the *imep* and the $CA50$. A reduction of the cylinder-to-cylinder variations of these combustion parameters is shown under transient conditions. However, due to the pressure evaluation execution structure, which takes $720°CA$ for the calculations, real time control is not possible.

In recent works, [31], [32] and [33], the potential of an Iterative Learning Control (ILC) algorithm is demonstrated in a combined LTC-DI combustion strategy. In the $CA50$ control, the feed-forward value for the injection timing is extended by a physical model of the ignition delay. Input parameter for the model are charge dilution, cylinder pressure and intake manifold temperature. Moreover, model prediction accuracy is improved storing the actual value of the actuating variable in a so called offset-map. The $CA50$ control is than extended for attaining combustion noise and stability targets through flexible shaping of the injection characteristics and self-sustaining en- and disabling of the pilot injections. A similar approach is published in [34] and [35]. The response of the $CA50$ control is improved by a non-linear model-based correction of the feed-forward map which compensates changes in the ignition delay due to deviations of the charge composition. Information about the in-cylinder air mass and the rate of EGR are obtained evaluating the indicated pressure trace. Also in this case, system learning is done storing the actuating variable values in a map.

2.3 Alternative-conventional combined mode

Since the application of alternative combustion systems is limited in the part-load range, the ability of switching to conventional combustion becomes a relevant task of the control strategy. This has been also the topic of some recent publication.

The CCS concept of Volkswagen AG is able of switching the combustion mode in order to provide a wide range of engine drivability and optimal efficiency (8 % improvement in $NEDC$) [26]. The switching procedure must provide a neutral torque signal. In this process, the switch between different injection strategies and especially the large charge dilution difference is recognized as the greatest challenge. However, further details about the implemented switching strategy are not given. Stationary and dynamic switching procedures between $PCCI$ and conventional diffusive combustion are shown in [75]. In this work a limit fuel quantity has been chosen as mode switching criterion. It can be seen how, due to the different system response of the air and the fuel path, an improper strategy may result in poor combustion efficiency. The *imep* signal is not steady and an increased engine noise is measured during the switching procedure, as well as a significant increase in UHC emissions.

More reliable is the solution proposed in [28]. The combustion mode is switched from $HCCI$ to SI mode when the net mean effective pressure (*nmep*) reaches the limit value of $3.5\,bar$. To overcome the problematic of the different EGR rates between the two combustion modes, dynamic corrections of the closed-loop controllers have been implemented. In this way, the transition can be better synchronized and oscillations of the torque signal can be avoided.

In [36], only stationary measurements are shown. The switching procedures starts with an EGR rate increase or decrease depending on the switching direction. The injection pressure and the desired $max\,dp/dphi$ value are changed only when the injection timing has been shifted of $3°CA$ by the controller. At $6°CA$ after the beginning of the switching procedure the injection strategy is commuted. However, also this solution does not provide a steady *imep* signal and phases characterized by poor combustion quality or, inversely, high rates of pressure rise are measured.

3

Theoretical Background

The combustion process in an ICE is influenced by a large variety of parameters. An exhaustive explanation of all of them would be an enormous task. In order to better understand the investigation done in this work a selection of the most relevant ones is proposed in this chapter.

3.1 Piston dynamic

Among the mechanisms influencing the combustion process in a reciprocating ICE particular attention has to be given to the piston kinematics. The piston is not only the device needed to convert the energy released during the fuel combustion into mechanical motion but, another fundamental task of this device is to preset the conditions needed in order to induce the chemical combustion process. It must be considered that, differently to other ICEs (turbines, jet or rocket engines), in piston engines the combustion process is intermitted and has to be initiated by every new cycle. This and other mechanisms makes an ICE a very complex system. A selection of the most significant ones, which will be discussed in the present chapter, is shown in Figure 3.1.

The piston is connected to the engine crankshaft using a connecting rod. Its cyclical motion is guided by the cylinder walls and can be described by the following trigonometrical function:

$$s_\varphi = r\left(1 + \frac{l}{r} - \cos\varphi - \frac{l}{r}\sqrt{1 - \left(\frac{r}{l}\right)^2 \sin^2\varphi}\right) \tag{3.1}$$

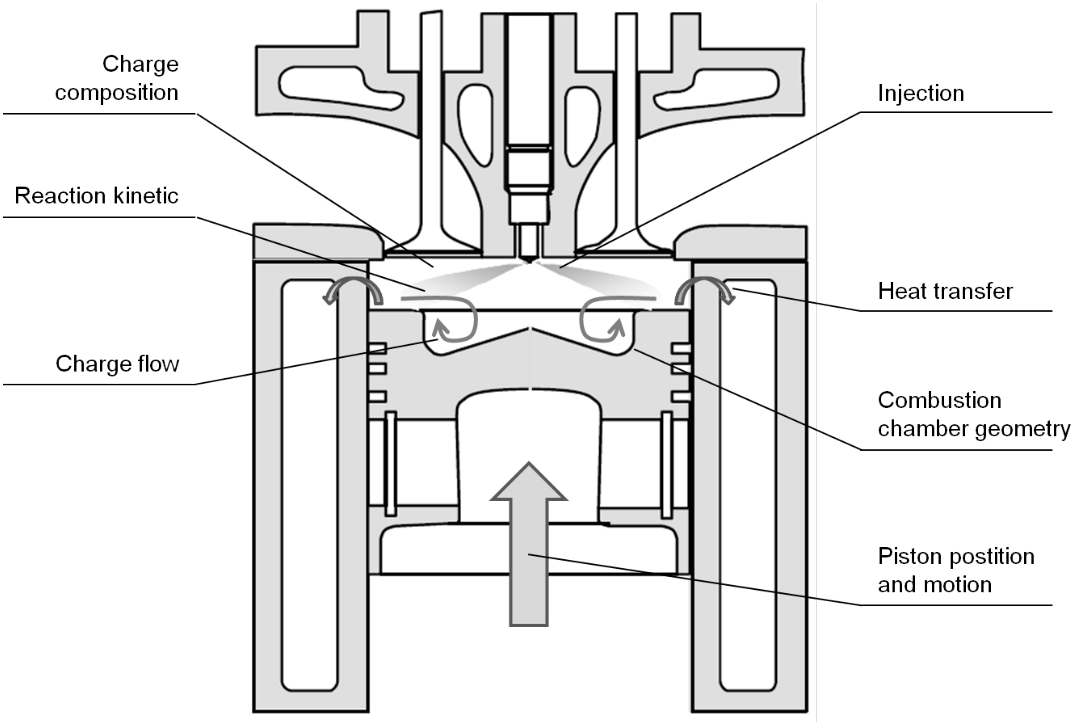

Figure 3.1: Description of the most relevant mechanisms influencing the combustion process in an ICE.

where r is the crankshaft radius and l the connecting rod length. The smallest volume in the combustion chamber, the so called clearance volume, is reached when $\varphi = 0$ and the connecting rod is perpendicular to the crankshaft rotation center. This is also the position at which the piston rest, for a very short moment, during every cycle, and it is known as Top Death Center (TDC). In fact the piston decelerates approaching the TDC during the compression phase and increases its speed again in the expansion moving in the opposite direction. Maximum piston speed is reached at $90°CA$ before and after TDC.

3.1.1 Charge heating process

More interesting is the effect of the piston motion on pressure and temperature. Once the fresh charge has been introduced in the combustion chamber and the intake valve closes, the compression phase begins. Due to the reduction of the combustion chamber volume pressure and temperature increase significantly. A simple way to describe the compression is to consider it as an adiabatic (and isentropic) process. This means that, following the first law of thermodynamic, there is no heat exchange with the system boundary and the complete energy resulting from the piston work W is transformed into internal energy of the charge U.

$$dU = dW \quad \text{where} \quad dU = m \cdot c_v \cdot dT \quad \text{and} \quad dW = p \cdot dV \tag{3.2}$$

Solving this equation using the ideal gas law $P \cdot V = m \cdot R \cdot T$ the dependency of pressure and temperature from the combustion chamber volume can be easily found:

$$p_{end} = p_{start} \left(\frac{V_{start}}{V_{end}} \right)^{\kappa} \qquad T_{end} = T_{start} \left(\frac{V_{start}}{V_{end}} \right)^{\kappa - 1} \tag{3.3}$$

where κ is the adiabatic index which results from the ratio of the heat capacity at constant pressure c_p to heat capacity at constant volume c_v. κ has, under standard conditions, a value of 1.4 for air and approximatively 1.36 for exhaust gas. The volumetric ratio between start and end of the compression can be approximated with the engine compression ratio ϵ. Considering this equation and the piston motion described by equation 3.1 pressure and temperature traces for the motored engine can be easily calculated. Following this idealized dependency, the maximum value for pressure and temperature is reached at TDC.

The charge temperature can be approximated as constant in the proximity of the TDC ($\pm 5°CA$), where the piston speed is very low, but decreases significantly departing from it. Between $30°CA$ before TDC and TDC a temperature increase of order $150K$ can be observed, as shown in Figure 3.2. Due to the stronger dependency from the volume ratio ($\kappa > 1$), the in-cylinder pressure usually duplicates during the last 40-30$°CA$ of the compression stroke. This means that the charge density increases drastically. As already mentioned, in-cylinder conditions are fundamental for the beginning of the combustion process (the fuel ignition process is explained in Section 3.3.2). In fact, especially in the case of compression ignited combustion processes, the energy needed for the activation of the chemical process has to be provided by the compressed hot charge. Thus, setting fuel injection timing and injection pressure the piston motion has to be considered carefully, especially applying alternative combustion processes with early injection timing.

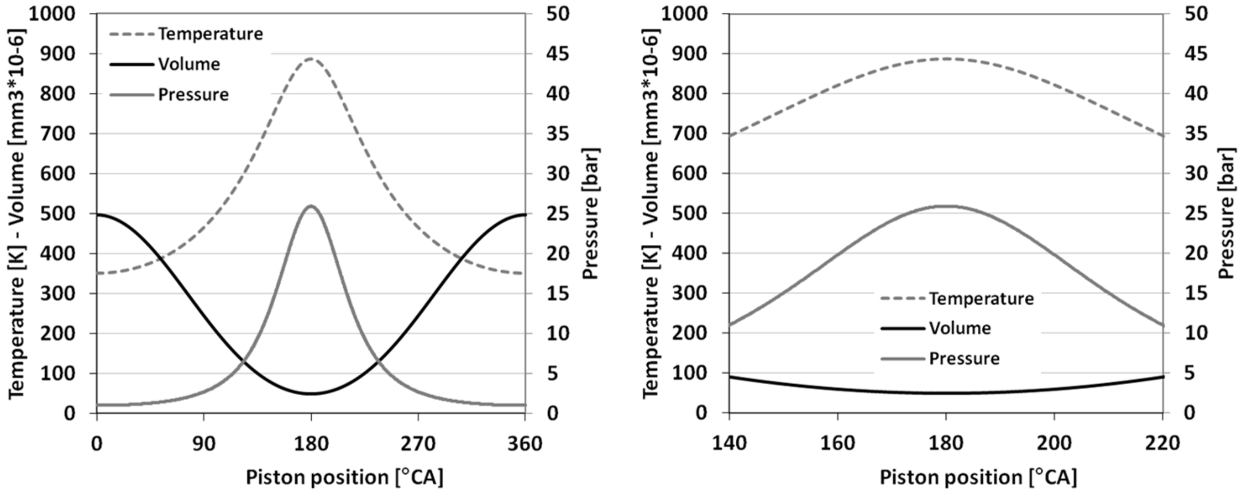

Figure 3.2: Adiabatic pressure and temperature traces for a motored engine (ϵ 15.5 : 1) as a function of the cylinder volume.

3.1.2 Wall heat transfer

The assumption of an adiabatic process proposed in the previous section is licit just for a qualitative description of the phenomena resulting from the piston motion. In fact, in a ICE an energy exchange between the charge and the combustion chamber walls can not be avoided. Together with the enthalpy of the exhaust gases, wall heat transfer is considered to be the largest energy losses in an ICE [37]. This is because a large percentage of the fuel heating energy is transferred to the coolant medium. Depending on engine speed and displacement it varies between 16 and 35 %. This is a complex process which varies with the location, the charge density and charge flow and finally with the mode of the heat transfer, i.e. conduction, convection and radiation. Generally it can be said that wall heat transfer increases with the temperature difference between charge and walls. Insulating the combustion chamber appears to be be a feasible solution but for many reasons not reasonable. On the other hand, increasing the combustion chamber wall temperatures would result in a lower volumetric efficiency.

Several investigations have been done during the past decades for better understanding the heat transfer mechanisms. Wall heat transfer is usually calculated considering the temperature difference between the in-cylinder charge and the surrounding walls, and multiplying it with the heat transfer coefficient using the Newton's approach (convection). The heat flux is not unidirectional. Entering the combustion chamber the cooled charge is first heated up by the hot walls[1] and only during the compression and later during the combustion process the heat flux is inverted and heat is subtracted from the charge. The highest heat transfer is reached during combustion where the temperature difference achieves its largest value. Heat flux values up to $10 MW/m^2$ during combustion can be achieved [3]. Energy transfers due to conduction and radiation also play a relevant role. The first one is related to the charge flow which, in diesel engines, is mainly dominated by swirl and squish, whereas the latter is mainly related to the soot production during combustion.

Of more relevance in the context of this investigation is the influence of parameters such as combustion phasing, charge dilution and mixture equivalence ratio on the wall heat transfer since these parameters are consistently varied during engine operation using alternative combustion processes. The combustion temperature of an air/fuel mixture is directly related to the ratio λ of the two reactants. For hydrocarbon fuels the largest combustion temperature is reached for slightly under-stoichiometry ratio [38], thus the highest heat losses are expected at this mixture ratio. Departing from stoichiometry, both increasing or reducing the fuel part, the temperature drops. Similar is the effect of cooled EGR on the wall heat transfer. Following a series of different mechanisms which are illustrated in section 3.5, EGR reduces the flame temperature and thus the wall heat transfer [14]. Wall heat transfer can also be reduced by means of late combustion phasing. If combustion occurrs in the expansion, where the combustion-chamber volume inceases, the combustion reaches lower peak temperatures and the wall heat transfer is reduced [3] [14]. However, reducing the wall heat transfer does not necessary means that the overall combustion efficiency increases. In fact the largest part of the energy which is not exchanged to the walls remains stored in the mixture mass and is not transformed in useful work. This energy leaves the combustion chamber together with the exhaust gases.

[1] The temperature of the combustion chamber walls is usually approximated around $350\,K$ but every single component has a different temperature. Cylinder walls are cooled by water and oil, piston by oil, valves and injector by the charge flow.

A characterization of the wall heat transfer in alternative combustion processes is carried out in [40]. Experiments shows a nearly steady and uniform distributed increase of the wall temperatures for highly premixed combustion whereas in DI mode a large inhomogeneity of the temperature increase is measured. This difference results in a lower heat transfer for the premixed combustion process.

Another interesting aspect related to the wall heat transfer is the so called thermodynamic loss angle which affects the measurement of the in-cylinder pressure. Due to the heat losses (and leakages) the pressure peak of a motored engine does not correlate with the piston position at TDC, as shown by the simplified correlations between piston motion and in-cylinder pressure shown in the previous section, but it is shifted a couple of degrees CA earlier in the compression. The thermodynamic loss angle has to be taken into account by the analysis of the indicated pressure data, first by calibrating the TDC reference respect to the motored pressure curve[2] and later by the interpretation of the data, such as emissions and combustion efficiency, in order to avoid interpretation errors.

3.2 Mixture formation

The goal by designing the mixture formation process is to achieve a large distribution of the fuel in the combustion chamber while simultaneously reducing the droplet size, in order to increase the exchange surface area. This is necessary for ensuring the fuel evaporation, which is an essential requirement for the onset of the chemical processes.

In high speed modern diesel engines the fuel is introduced into the combustion chamber using an injector nozzle. Exiting the injector through one ore more holes the fuel jet forms a cone-shaped spray. In the proximity of the nozzle the spray is mainly dominated by large drops and fluid ligaments, whereas in the outer regions disintegrated fuel structures can be found and evaporation takes place. The distribution of the fuel in the spray, the spray cone angle and the spray tip penetration are very important for the quality of the mixing process. Also the position of the injector in the combustion chamber and the spatial

[2] There are two methods for defining the exact TDC shift, using a capacitative TDC sensor or analyzing the pressure trace in the motored mode. In the second case a correction by the thermodynamic loss angle is needed.

arrangement of the sprays play an important role. These have to be chosen in accordance with the combustion chamber and piston geometry (bowl) for avoiding fuel wall impingement and providing maximum combustion efficiency.

The mixture formation process has a limited time which duration is limited by the ignition delay. As soon as combustion takes places a reaction of fast chemical processes occurs and the local mixture formation process is interrupted. Therefore, for avoiding poor mixing quality long ignition delay is favored.

Mixture formation can be also promoted by shaping the charge flow in the combustion chamber. This can be done designing the intake channels in order to create a rotational flow or exploiting the piston movement during compression which pushes the charge into the cylinder bowl. However these solutions may lead to increased heat losses or, if not applied correctly, to a deterioration of the mixing process such as over-swirl, for an instance.

3.2.1 Fuel injection

Nowadays, high pressure direct injection systems can be found in almost every passenger car diesel engine. Modern injection systems are capable of multiple injection events with different duration and injection pressures up to 2000 bar. This is realized using solenoid or piezoelectric valves controlled by an ECU. The fuel is stored under high pressure in a distribution pipe, also known as common rail[3], which is directly connected to the injection valves. The high pressure is provided by a piston pump driven by the engine camshaft [41]. Using these systems, a fine adjustment of the fuel injection time and quantity is possible independent of engine load and speed.

In the nozzle the fuel is accelerated before entering the combustion chamber. The pressure stored in the common rail is so converted into kinetic energy. During this phase cavitation[4] and turbulence take place in the nozzle internal

[3]The common rail system prototype was developed in the late 1960s by Robert Huber of Switzerland and further developed by Dr. Marco Ganser at the Swiss Federal Institute of Technology in Zurich. After research and development by the Fiat Group, the design was acquired by the Robert Bosch GmbH for completion of development and refinement for mass-production.

[4]When a liquid is subjected to rapid changes of pressure, e.g. due to constrictions and

flow. These phenomena play an important role in the breakup process of the fuel spray. Generally it can be said, that cavitation in the nozzle increases fuel atomization [42]. The geometry of the injector holes also determines the shape of the injection fuel cone and its penetration depth in the combustion chamber.

During injection the fuel is transported in the outer regions of the combustion chamber and is mixed with the charge. Liquid fuel is injected, usually late in the compression stroke, in a high-density mixture of air and exhaust gases. Therefore, high injection pressure is fundamental for achieving the required kinetic energy, which is the most important parameter for the mixture formation [2]. Exchange momentum and droplet size mainly depend from this parameter. Another advantage of using high injection pressures is the reduced need of swirl and squish movements of the charge, with a significant reduction of heat losses through the combustion chamber walls.

3.2.2 Spray penetration

Spray penetration has an important influence on the mixture formation. Impingment of liquid fuel on cool surfaces inside the combustion chamber may results from over-penetration of the spray and leads to undesired emissions due to unburned or partially burned fuel. On the other side, under-penetration reduces the air utilization and results in poor mixture homogenization.

Fuel spray penetration in the combustion chamber does not only depend from the injection pressure and the fuel properties. The number of injection holes and their geometry are also very important for achieving good mixture formation and high combustion efficiency, and they must be accurately designed in order to accomplish combustion geometry requirements. Together with the injection pressure, the number of injection holes has been continuously increased in the past years. Prerequisite for maintaining same injection duration is a reduction of the nozzle diameter [2]. Finally, charge movements may also influence the spray penetration. In fact, swirl does not only improve fuel spreading by deviating the spray cone but reduces the fuel penetration as well [3].

bends in the nozzle, the formation of vapor bubbles may occour. This phenomenon, called cavitation, can lead to excessive surface stress and material erosion.

3.2.3 Fuel spray breakup and evaporation

The ignition delay, i.e. the time between start of injection and start of combustion, can be divided in a physical and a chemical phase. The physical delay is the time needed for the preparation of the fuel into a reactive mixture. During this phase, fuel atomization (fuel spray breakup) and evaporation take place. The chemical delay is the time needed for the pre-combustion reactions between evaporated fuel and the charge components.

Fuel spray breakup

Fuel spray breakup is divided in two consecutive steps, a primary breakup phase, in which ligaments and both large and small drops are formed from the fuel jet and sheets, and a secondary phase, in which fuel atomization occurs as a result of aerodynamic effects. These processes are required for increasing the exchange surface between fuel and air and thus promoting fuel evaporation. The smaller the mean droplet diameter the larger will be this surface.

Primary fuel spray break up describes the initial disaggregation of the fuel stream into ligaments and primary drops. This process can begin already in the injector nozzle due to effects of cavitation and turbulence or once the fuel exits the injector holes and is subjected to the aerodynamic forces resulting from the speed difference between fuel spray and in-cylinder charge. There exist different breakup patterns which depend from the injection pressure, the nozzle geometry and the fuel properties. However, for a high pressure injection system like the one used in this work (injection pressures larger than $300\,bar$), a disintegration pattern of the fuel jet into drops, which diameter is smaller than the one of the injector hole, can be assumed [43].

A further breakup of the primary drops is mainly determined by aerodynamic forces. As a consequence of the high injection pressure, the speed difference of the fuel drops relative to the in-cylinder charge is very high. This leads to a deformation of the drops shape, caused by the asymmetrical pressure distribution around the surface, and a further disintegration in even smaller units. Also in this case different breakup patterns are possible, which depend from the relationship of mass inertia force and surface-tension force for a drop, described by the Weber number. In the case of high pressure injection systems this number usually gains values between 100 and 350, which means that the drops are disintegrated following a stripping or catastrophic-breakup mechanisms [42]. Generally it can be said, that increased injection pressure leads to a reduction of the mean diameter of the drops during secondary breakup. Injection profile, injection spray geometry, charge density but also fuel properties like viscosity, surface tension or boiling point are other important parameters [44].

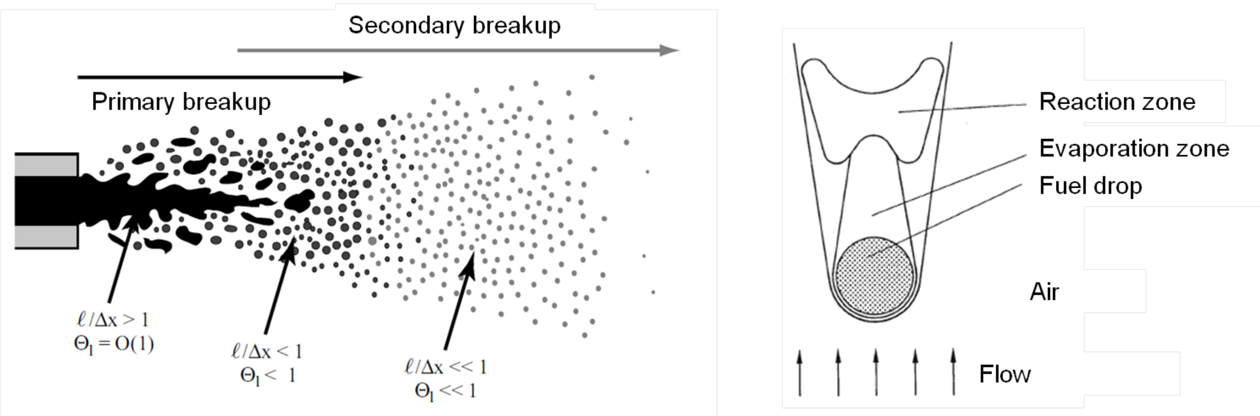

Figure 3.3: Illustration of fuel breakup and evaporation mechanisms [2].

In areas with high fuel drops density, i.e. in the proximity of the injection nozzle where $\lambda \approx 0$, interaction mechanisms between the drops occur. Collisions may result from the different velocity of the single drops and have a significant influence on their size distribution. Depending on the collision energy, the ambient conditions and the drop dimension prior to collision, different patterns can be observed, i.e bounce, coalescence or breakup [47]. The result is usually a change in the direction and in the speed of the drops as well as a change in their number, which increases in the case of a fuel drops breakup pattern.

Fuel evaporation

At injection timing, the charge has been already compression heated to temperatures around 1000 K (depending on engine compression ratio, intake temperature and boost pressure) which are much larger than the boiling point of diesel fuels (around 500 K). Convective heat transfer takes place between fuel drops and the in-cylinder charge leading to a fuel temperature increase. The partial pressure on the drop surface rises and a diffusive transport of fuel mass outside the drop boundaries takes place, i.e. the fuel evaporates [45]. Fuel evaporation is an essential mechanism for the onset of the chemical reactions.

Promoting heat transfer is fundamental for achieving high fuel evaporation. This can be done increasing the kinetic energy of the drops, e.g. increasing injection pressure. As already mentioned, high kinetic energy leads to smaller drop diameter and thus a larger surface area for the exchange process. Moreover, induced convection[5] can be promoted as well as a larger dispersion of the drops in the charge (so called air entrainment). Near the kinetic energy and the dimension of the drops, also the fuel properties and the charge composition play an important role in the fuel evaporation process.

During fuel evaporation a decrease of the charge temperature can be observed. In fact, the energy needed for heating the fuel at first, and for overcoming the enthalpy of evaporation at next, leads to a marked local cooling of the charge. This cooling effect has a significant influence on the chemical reactions that take place by combustion onset (see Section 3.3).

3.2.4 Charge movement

In-cylinder charge movement is an effective way for improving the mixture formation. Two solutions are known to be particularly interesting for diesel combustion processes, i.e. swirl, which is induced by the intake channel, and squish, which results from the forced charge flow into the piston bowl generated during compression. Charge flow improves the mixing process but, at the same time, may lead to increased heat losses and reduced volumetric efficiency, if not applied properly.

Swirl

Swirl is usually defined as an organized rotation of the charge about the cylinder axis [3]. A swirl motion results from the angular momentum impressed in the

[5] If there exists a positive speed difference between drops and charge the evaporated fuel can be transported away from the drop surface and new fuel can evaporate.

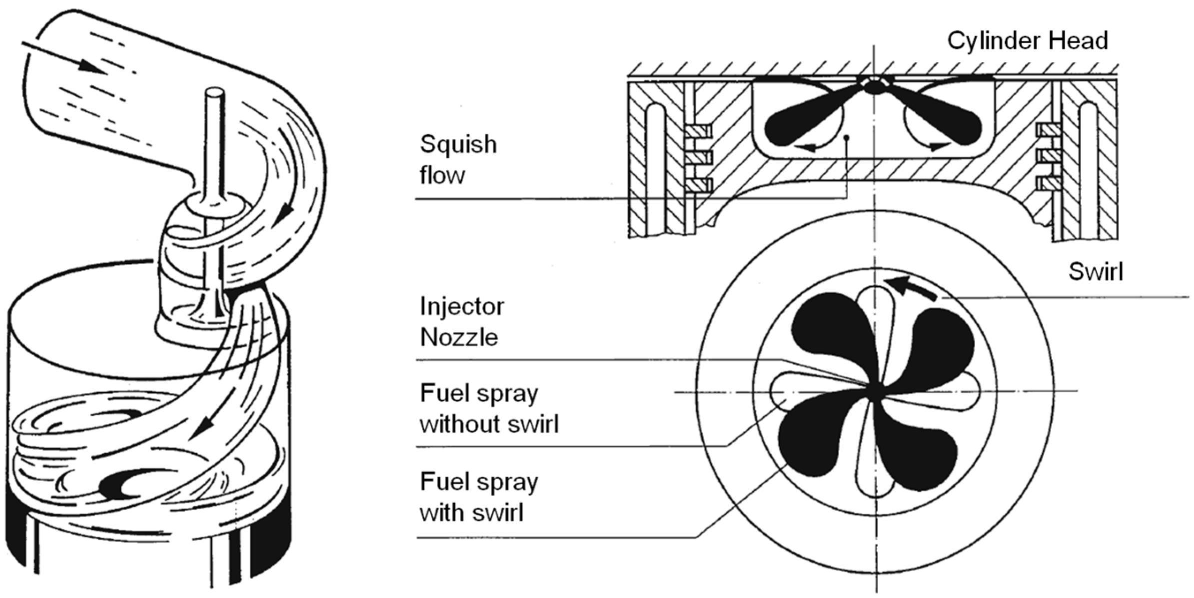

Figure 3.4: Schematic representation of charge flow mechanisms. On the left side swirl flow created in the inlet channel. On the right side in-cylinder charge motion and interaction with the fuel spray [2].

charge while flowing trough the inlet channel, i.e. before entering the combustion chamber.

Swirl results primarily from the geometry of the intake channel. Tangential or spiral channels are usually used. Increasing piston velocity swirl becomes stronger and the charge quantity moved pro degree CA increases. In order to maintain a constant relation between the charge motion and the injected fuel quantity, the injection duration (in degree CA) must remain constant over the engine speed range thus it becomes shorter for increasing speed. This can be partially realized with a proper application of the injection pressure map. However, since the adjustment range is limited, poor mixture formation by low engine speed or inversely in over-mixing by high speed may result. To overcome this problem a swirl valve (or flap) mounted in the intake channel can be used supproting the weaknesses of the injection system with swirl motion. In this case this is done by short injection duration, where mixture formation is supported by an increased swirl generated using the flaps. By modern high speed engines however, swirl is mainly exploited to promote soot oxidation during expansion.

Squish

During the compression stroke, the swirl produced using the intake channel is gradually replaced by a squish movement. Squish is the gas motion which results when the piston is approaching the cylinder head an pushes the charge portion placed in the circumference of the combustion chamber towards the middle creating a radial inward gas motion. By the presence of a ω-bowl in the piston, such as by modern passenger car diesel engines, this motion results in a toroidal-shaped flow. Moving from the combustion chamber into the piston bowl the charge flows in the opposite direction to that of the injected fuel spray. This circumstance increases the exchange-momentum improving mixture homogenization.

During cylinder expansion the flow direction is inverted. Also in this phase, squish is important for sustaining the combustion and helps late oxidation processes [2]. Squish increases with the engine speed. A larger cylinder bore diameter promotes swirl, on the contrary a wider piston bowl reduces it [3]. Swirl achieve its maximum short before TDC.

3.3 Combustion process

In alternative low-temperature premixed combustion processes the chemical kinetics plays a dominant role during the ignition and combustion phases. The heat release is characterized by a two step process in which the main combustion is preceded by a low-temperature reaction, the so called cool-flame. The oxidation of hydrocarbon fuels such as diesel occurs according to two ramification mechanisms which relevance depends from the ambient conditions. The formation and accumulation of metastable hydroperoxide intermediates at low temperatures delay the onset of the high-temperature reaction. Increasing the temperature the chemical equilibrium is shifted to the production of reactive radicals and a fast high exothermal oxidative reaction takes place.

3.3.1 Ignition delay

Once the fuel is mixed with the charge and evaporation has occurred some time is still needed till combustion can be detected. This time (or CA) interval is defined as chemical ignition delay. During this period pre-combustion reactions between the fuel and the charge components (air and exhaust gas) take place.

These processes depend on the fuel characteristics, the charge composition and the ambient conditions, i.e. in-cylinder temperature and pressure, as well as their distribution in the combustion chamber. Among these parameters auto-ignition properties of the fuel plays a large role. This are usually defined by the cetane number, which describes the ignition delay of a fuel compared to the one of a reference fuel mixture, as described in [3]. A low cetane number correspond to a higher resistance to ignition, thus a longer ignition delay. For engine operation strategies in which the combustion process is controlled by the rate of injection, i.e. by diffusive diesel combustion, a high catane number is favored. On the contrary, in alternative combustion processes fuel homogenization can be prolonged using low cetane fuels.

For given fuel composition however, in-cylinder temperature and pressure are the most important variables influencing the ignition delay. This dependency can be described by the empirical equation developed by Hardenberg and Hase in [48]:

$$\tau_{id} = (0.36 + 0.22\overline{S}_p) \exp\left[E_A \left(\frac{1}{\tilde{R}T_{Cylinder}} - \frac{1}{17.19}\right)\left(\frac{21.2}{p_{Cylinder} - 12.4}\right)^{0.63}\right] \tag{3.4}$$

where τ_{id} is the ignition delay in CA, E_A is the activation energy for auto-ignition, $\tilde{R}$ is the universal gas constant and $\overline{S}_p$ the mean piston speed. Differently to the usual Arrhenius correlation, this equation takes into account the cooling effect induced by the fuel evaporation as well as the change in temperature and pressure due to the piston movement during the delay. τ_{id} refers to the entire ignition delay, i.e. the time interval between injection and combustion beginning. During this time physical and chemical processes run almost simultaneously and a clear separation is not possible. However, research into the duration of the various phases of the ignition delay (physical and chemical) have shown a dominance of the chemical delay [46].

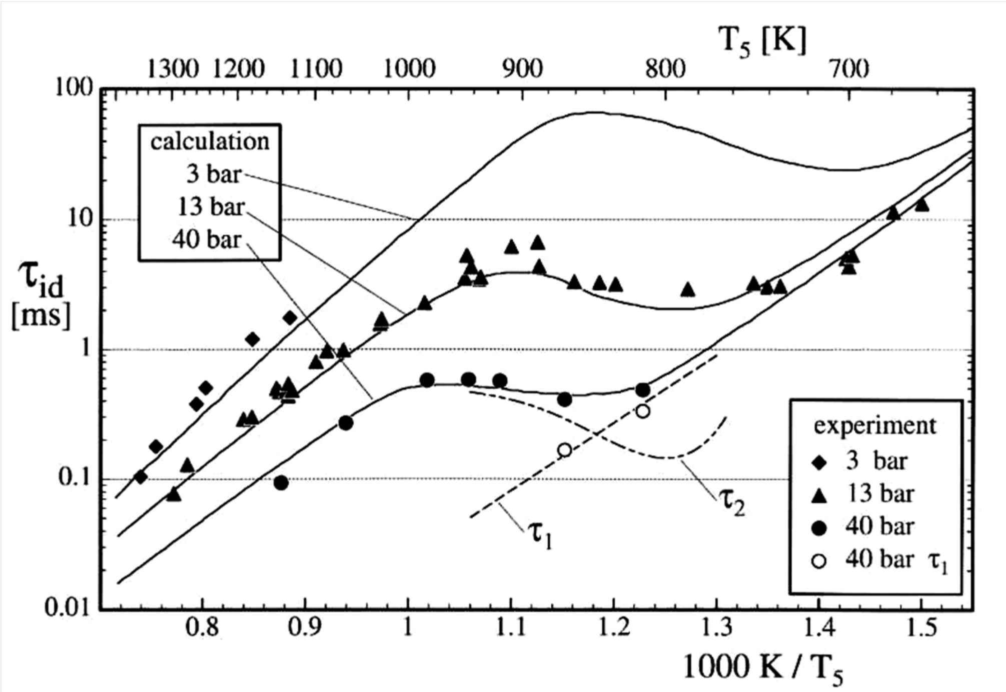

Figure 3.5: Dependency of the ignition delay from pressure and temperature for n-heptane fuel at λ 1.

Other physical parameters which affect the charge state at the time of injection, such as compression ratio, engine temperature or engine load, also influence the ignition delay. During engine operation however, especially the injection timing has a direct effect on the delay. Since in-cylinder temperature and pressure are subjected to the piston motion the delay decreases approaching TDC at first, but increases again if the combustion is phased later in the expansion, where temperature and pressure decrease again.

Another parameter of relevance is the oxygen concentration in the charge which is usually varied by recirculating exhaust gas in the combustion chamber. If the oxygen concentration is reduced, ignition delay increases. This is mainly due to the sparse distribution of the oxygen molecules in the charge and the reduced probability of an interaction with the hydrocarbons. Accordingly to the ignition limits, the local air to fuel ratio, must remain in a range of 0.6$\leq \lambda \leq$1.5 so that oxidation occurs [2].

3.3.2 Fuel ignition and combustion

The oxidation of the fuel develops in a chain of different processes. It can be differentiated between thermal processes, where the energy released manifests itself as a thermal energy (exothermal reaction), and chemical processes, in which the number of species varies. The latter are known as chain-branching mechanism and are typical for hydrocarbon-chain reactions. Chain-branching is necessary for the formation of the reactive radicals, and thus the start of the thermal reactions. On the other side, radicals are destroyed by degenerate-branching reactions or by diffusion to the combustion chamber walls.

The oxidation of the hydrocarbon molecules initiates the chain reaction process. During this first step, reactive radicals $R^{\bullet}$ are produced by mean of the reaction:

$$RH\, +\, O_2\, \rightarrow\, R^{\bullet} + HO_2^{\bullet} \tag{3.5}$$

The radicals react with other reactant molecules and the chemical reaction process propagates forming other radicals, such as CH_3 and C_2H_5, or intermediates. During this phase chain-branching reactions take place increasing the number of reactive radicals. In the other case, i.e. by degenerate-branching, stable molecules result from the reaction of the intermediates. The dominance and rate of these two paths depend on the ambient conditions. The combustion process ends than with the degradation of the radicals in so called termination reactions [3].

Mixture temperature and pressure define the kind of reactions that take place during ignition delay and later during combustion. Depending on these parameters, four type of behavior can be differentiated: slow reactions ($\leq$500 K), single or multiple cool flames, two stage ignition (cool flame followed by hot flame) and single stage ignition (hot flame) [3]. Since in internal combustion engines in-cylinder conditions at injection timing are characterized by temperatures above 800 K and pressures larger than 50 bar, two stage ignition or single stage ignition with hot flames are usually observed. Near temperature and pressure, the appearance of a cool flame vary enormously with the molecular structure of the fuel. In the case of long hydrocarbon-chains, i.e. by conventional diesel fuels, cool flames may arise.

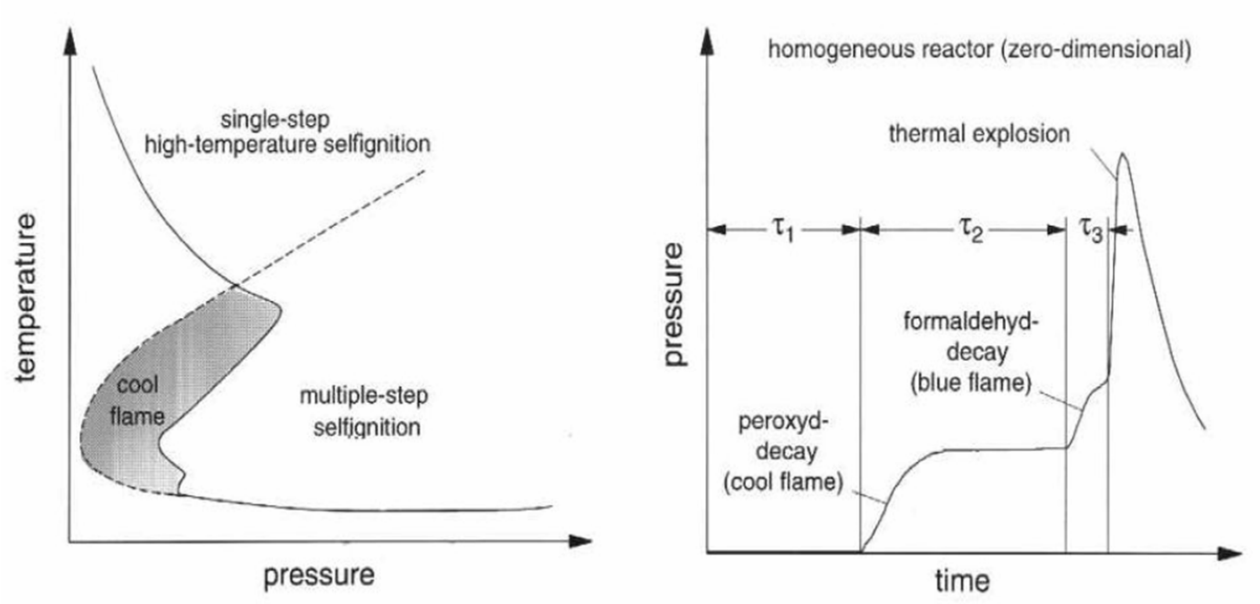

Figure 3.6: Explosion p-T-diagram showing the ignition limits of hydrocarbon fuels on the left and the pressure increase during self-ignition process on the right [51].

Cool-flame

Figure 3.6 shows a representation of the ignition limits for hydrocarbon fuels. In the low-temperature region ignition takes place in a multiple-step process that can be divided in a cool-flame, a blue-flame and a hot-flame (explosion). The production of metastable hydroperoxide intermediates with the form $ROOH$, as a result of the chain-branching reactions mentioned above, plays here an important role retarding the self-ignition process and splitting it in many stages. In fact, only after having reached a critical concentration (end of τ_1), the hydroperoxides degenerate in an exothermal reaction building new radicals and in particular formaldehyde. This reaction is usually know under the name of cool-flame (τ_2). The energy released during this phase usually corresponds to 5-15% of to the total injected fuel energy. Due to the temperature increase, the chain reaction proceeds with the oxidation of the hydrocarbons, which is now sustained by the formaldehydes, and the creation of carbon monoxides as an intermediate product. This phase, which correspond to τ_3 in the diagram, is usually of very short duration and it is characterized by blue radiation (blue-flame). Only after this multiple-step chain reaction the main combustion takes place. The carbon monoxides react with the oxygen in a high exothermal process know as hot-flame. During this phase the main part of the energy is released [42].

The total chemical ignition delay results from the sum of τ_1, τ_2 and τ_3. As explained in the previous section, the delay usually decreases with increesing ambient temperature. In the combustion of long chains hydrocarbons, however, a phenomenon called Negative Temperature Coefficient, short NTC, can be observed. In a certain temperature region the correlation between delay duration and temperature is inverted. This phenomenon occurs when, due to a temperature increase, the formation of the hydroperoxide intermediates is reduced in favor of the production of reactive radicals. This results in a mitigation of the cool-flame process. The time τ_2, needed for the temperature increase during the cool-flame, is prolonged resulting in a longer ignition process. Indeed, by the discussed temperature range, the energy level needed for the activation of the chain-branching reaction of the radicals can not be achieved and the process speed is limited by the rate of the cool-flame reaction. This phenomenon can be observed in Figure 3.5.

Cool-flame phenomena are usually observed in alternative combustion processes where the ignition temperature is reduced by means of early combustion phasing or strong charge dilution.

Multiple-step and single-step self ignition

Increasing the temperature in the combustion chamber, more radicals are produced in the beginning phase of the chain-branching reaction and $ROOH$ is no longer the major products of the process. Instead it is hydrogen peroxide, H_2O_2, which reacts building OH radicals following the chemical equations:

$$HO_2^{\bullet} + RH \rightarrow R^{\bullet} + H_2O_2 \tag{3.6}$$

$$H_2O_2^{\bullet} + M \rightarrow HO^{\bullet} + HO^{\bullet} + M \tag{3.7}$$

where M is an inert impact partner. By temperatures above $900\,K$ the ignition process occurs in a single-step path. The energy is enough to immediately activate the disintegration of the reactive radicals $R^{\bullet}$ into small alkyl radicals with the form CH_3 and C_2H_5. The chain branching occurs primarily by way of the reaction:

$$H^{\bullet} + O_2 \rightarrow O^{\bullet} + HO^{\bullet} \tag{3.8}$$

Conventional diffusive diesel combustion is predominated by the high-temperature single-step reaction. Multiple-steps reaction occurs only in the premixed burning part of the injected fuel which, however, represents only a small portion of the total. For this reason, the detection of cool-flames during this phase is very difficult.

3.3.3 Combustion temperature

The flame temperature of an air/fuel mixture mainly depends on the concentration of the reactants (charge dilution) and the rate over which the system conditions change, i.e. temperature and pressure. Generally it can be said that for an adiabatic combustion process[6] at constant ambient conditions the maximum flame temperature occurs slightly rich of stoichiometry. If λ >1 the excess of oxygen must be heated and the product temperature drops. For rich combustion, i.e. λ <1, the system is under-oxidized and the fuel conversion is reduced, this results in a temperature drop as well [38]. This circumstance is illustrated in Figure 3.7. Also important is the amount of nitrogen present in the mixture, which increases together with the air/fuel ratio, and it is characterized by a large heat storing capacity. The H/C ratio of the fuel determines the formation of water vapor, CO_2, and their formed dissociation products during the combustion process. For given chemical enthalpy content of reactants, the larger the H/C ratio, the higher the flame temperature. Finally, because of the higher final pressure and the lower rate of dissociation, adiabatic combustion at constant volume is usually characterized by higher maximum temperatures compared to constant pressure combustion [3].

[6]The adiabatic flame temperate is the temperature reached when al the energy released during the combustion process is solely employed to raise the products temperature.

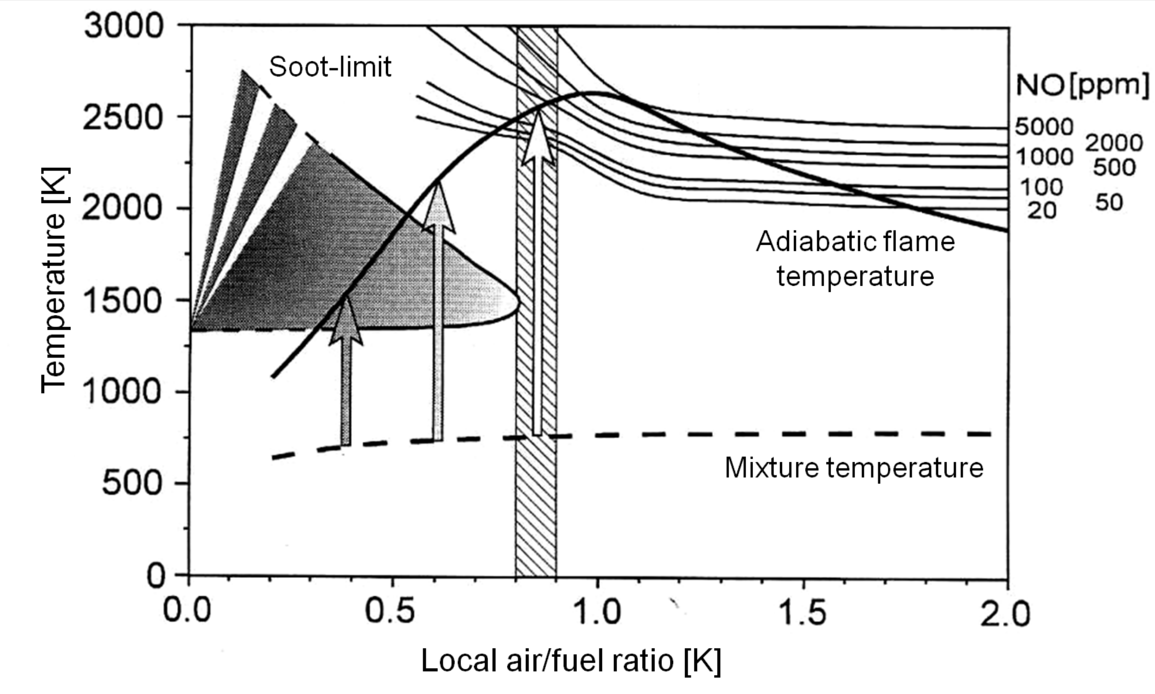

Figure 3.7: Adiabatic flame temperature over a variation of the local air/fuel ratio.

3.4 Emissions

The oxidation process of a hydrocarbon fuel should ideally result in the formation of carbon dioxide and water. Despite of that, in a ICE other pollutant are produced during combustion. Principally it can be differentiated between pollutant resulting from a chemical reaction, such as nitrogen oxides, and pollutant resulting from incomplete combustion, i.e. carbon monoxide and soot. The formation of these molecules does not only depend from the course of the chemical process but it is sensible to a large variety of parameters such as mixture properties, ambient conditions and charge flow.

Conventional DI diesel engines are know for the high combustion efficiency and the very low UHC emissions but, on the other hand, DI combustion is also affected by the trade-off between NOx and PM emissions. As explained in the previous chapter, promoting fuel homogenization and reducing the combustion temperature has been demonstrated to be a feasible solution to mitigate this problem. In this way, the formation of NOx and PM during the combustion process can be avoided. However, even if the combustion of the air/fuel mixture does not occur by flame propagation, the combustion of alternative processes shows conformities to that of gasoline engines with indirect injection, which are known to suffer from higher UHC and CO emissions resulting from a poor fuel conversion.

The following sections present the mechanisms which lead to the formation of the most relevant pollutant. Particular attention is given to those mechanisms present in low-temperature premixed combustion processes.

3.4.1 Nitrogen oxide

Diatomic nitrogen N_2 is the natural component of the Earth's atmosphere wiht the largest volume fraction, i.e. 78 %. Avoiding it to take part by the combustion process is nearly impossible. During the complex chemical reactions which take place in a ICE, nitrogen reacts with the oxygen forming NO, NO_2 or N_2O molecules. These compounds are usually identified as NOx. NOx emissions are responsible for the formation of atmospheric smog and acid rains but can also be the cause of respiratory diseases. Moreover, N_2O is known to play a role in the greenhouse effect.

Two chemical reaction pathways are considered to be responsible for the most part of the NOx emissions formation, the thermal (Zeldovich) and the prompt (Fenimore) mechanisms. Investigations on the conventional DI combustion have shown a preponderance of the thermal mechanism (90 to 95 %) over the prompt one.

To break a N_2 molecule a large amount of energy in form of temperature is required. A combustion temperature above 1900-2000 K must be reached to start molecule dissociation. When it happens, atomic nitrogen reacts very fast with the oxygen present in the charge forming NO molecules. The higher the flame temperature and the oxygen concentration, the more the chemical equilibrium of the reactions favors the formation of NO. For an adiabatic constant-pressure combustion the NO formation peaks at the stoichiometry composition and decreases rapidly as the mixture becomes leaner or richer [3]. Due to the fast temperature drop at the end of the combustion, which is promoted by the volume expansion, the NO chemistry is frozen and the decomposition of the NO molecules is interrupted.

The prompt pathway can be observed in low-temperature fuel-rich mixture areas. In this areas, reactive intermediate species HCN or NH_x are formed and, depending on the ambient conditions, they are either oxidized to form NO or re-formed to molecular nitrogen. This mechanisms has a lower relevance in ICE systems [52].

A third formation mechanism occurs by the high-pressure low-temperature combustion of fuel-lean mixtures. Under these conditions, NO is builted by the oxidation of N_2O, formed in a previous reaction step. The energy needed to overcome the activation energy of the chemical reaction is provided by the high pressure. This process is usually negligible in diffusive combustion where temperatures are high, but becomes important as the combustion temperature remains under $2200\,K$ and the thermal formation mechanism is suppressed. This is the case of alternative combustion processes, which are characterized by low temperatures and high pressure peaks [44].

Reduction of NOx emissions is usually done by means of exhaust gas recirculation. Main effects are the reduction of the flame temperature and the oxygen concentration in the charge, which are the main causes of NO formation. Aim of alternative combustion systems is to avoid the formation of NOx emissions making large use of EGR and limiting the combustion temperature below $2000\,K$. The mechanisms behind this circumstance are explained at the end of this chapter.

3.4.2 Particulate matter

Particulate matter (PM) mainly consists of carbonaceus material (soot) and aerosols, such as ash particulates, metallic abrasion particles, sulfates, and silicates, and mainly results from an incomplete combustion of the hydrocarbon fuel. The main particulate fraction of diesel exhaust PM consists of fine particles. Because of their small dimensions, usually between 10 and $100\,nm$ [3], particulates emissions can be absorbed in the human body during respiration. A long-term exposure has been demonstrated to increase the risk of vascular disfunctions, heart diseases and lung cancer.

Nowadays, the formation mechanisms of combustion related particulate matter are well known. Different types of species and combustion intermediates formed during the fuel pyrolysis and the oxidation processes are ranged as potential precursors, leading to soot particle inception, i.e. polyacetylenes or polyynes, ions, and polycyclic aromatic hydrocarbons (also known as PAH). Recently, PAH has been identified as the most probable soot particles precursor. A detailed description of the particulate formation mechanisms can be found in [53]. Summarized in few words, the fuel molecules are first broken into smaller hydrocarbon molecules and free radicals either by pyrolysis or oxidation reactions. In a second phase, the first aromatic rings are formed, usually benzene or phenyl. These serve as nucleus for the formation and growth of PAHs by the H-abstraction-C_2H_2-addition pathway ($HACA$) in forming larger soot particles.

More generally, fuel-rich areas (λ <0.65) together with combustion temperatures between 1600 and 1900 K are considered to be ideal conditions for soot formation [44]. The rich mixture has less oxygen to completely burn the fuel which turns into a carbon deposit. This is the case of fuel-rich areas of the fuel spray resulting by insufficient mixture formation. The types and quantities of soot particles can vary according to the operating conditions, such as temperatures and pressures, fuel and charge properties.

Soot formation can be avoided improving mixture formation by means of prolonged ignition delay and charge flow or limiting the combustion temperature below 1600 K. These conditions are provided in low-temperature premixed combustion systems. In processes with thorough mixture homogenization, i.e. $HCCI$ and $HCLI$, soot formation is completely avoided, whereas in late injection processes, such as $HPLI$ or MK, soot formation can be observed during combustion but a strong soot oxidation takes place in the second part of the combustion [8]. Finally, up to 90 % of the produced soot can be reduced in the late combustion phase. Required conditions are sufficient oxygen concentration, e.g. by lean combustion processes, and gas temperatures above 1300 K.

3.4.3 Unburned hydrocarbons

Under the acronym UHC, unburned hydrocarbons, a large variety of chemical compounds composed by carbon and hydrogen atoms is meant. As the name reveals, these pollutants result from incomplete combustion of the injected fuel. This means that in some part of the mixture the combustion process do not take place or it is interrupted before completeness. Even if UHC can be partially reduced by post-oxidation late in the combustion process or in the exhaust pipe by means of a diesel Oxidation Catalist (DOC), high enigne-out UHC emissions are a sign for bad combustion efficiency. Once in the atmosphere, hydrocarbons may react with nitrogen oxides by the action of the sunlight generating smog. Moreover, a more stable hydrocarbon compound, i.e. methane (CH_4) is kwon to be one of the pollutants responsible for the greenhouse effect.

Five mechanisms are known to be a possible source for UHC emissions in ICE, these are:

- flame-quenching, due to cold regions of the combustion chamber (wall quenching)
- under-mixing and presence of over-rich zones
- over-leaning and excessively low local air/fuel ratio and low temperatures (bulk quenching)
- fuel deposit on the walls, adsorption and desorption in engine oil (fuel film)
- combustion chamber crevices

An experimental analysis of these mechanisms in LTC combustion is presented in [54]. Using different cylinder bowl geometries and different injectors on a single cylinder engine the authors demonstrate bulk quenching to be the main source of UHC emissions. The long mixing process leads to over-leaning of the mixture. In these regions local temperature, equivalence ratio and charge dilution are inadequate for completing the oxidation process. EGR rate and intake temperature has a direct correlation to the formation of UHC emissions since they influence the mixture temperature. On the other hand, the injection pressure and the swirl are mainly responsible for the local mixture quality but has less impact on the UCH emissions. A correlation between unburned hydrocarbons and the duration of the mixing time is confirmed in [55]. Finally, the global equivalence ratio plays a secondary role in the formation of UHC emission.

Another important aspect that results from these investigations, but that can be also found in many others works, e.g. [36] [49] [14] [10], is the importance of avoiding fuel spray to be injected on the top surface of the piston crown. Otherwise wall impingement occurs with a rapid increase of the UHC emissions. Early fuel injection timing must be limited so that the fuel spray lands into the piston bowl. This must be done considering engine speed and injection pressure.

Finally, results of mass spectrometry measurements of the exhaust gas published in [36] and [50] show low-temperature premixed combustion to be a potential source of methane emissions. CH_4 molecules may results from the low-temperature oxidation of fuel-rich areas and, because of their high stability, survive during the combustion. Since the activation temperature of methane is larger than $900\,K$ the reduction of this molecules in the DOC may be problematic even at part-load engine operation. For this reason, it is suggested to avoid the formation of this pollutant during the combustion process increasing the air/fuel ratio of the charge either reducing the EGR or increasing the boost pressure.

3.4.4 Carbon monoxide

Carbon monoxide (CO) is an intermediate product of the oxidation reaction of hydrocarbon fuels. High concentrations of CO are toxic. In the atmosphere it reacts spontaneously with the oxygen creating carbon dioxide (CO_2) or ozone. CO emissions in ICE are directly correlated to the air/fuel ratio of the mixture and to the fuel homogenization with the charge. For an instance, significant CO emissions can be measured in DI engines at full-load where there is only short time for thorough mixture formation or in the fuel-rich areas of the fuel spray.

In fuel-rich areas, thus by insufficient oxygen concentration, the reaction of carbon monoxide in forming carbon dioxide cannot take place. In this case CO emissions are dominated by the mixing process and concentrations increases steadily with decreasing λ. High injection pressure and an extended ignition delay are suggested for avoiding this problem. Post oxidation processes by means of charge motion (swirl) are also useful in reducing engine-out CO emissions.

By lean combustion CO emissions are usually negligible, as long as the air excess does not result in excessively low combustion temperatures, which also interrupt the reaction mechanism in the formation of carbon dioxide. In fact, below 1500 K the oxidation rate of CO decreases rapidly. This may be the case of low-temperature premixed combustion processes where long ingnition delay and low combustion temperatures are promoted recirculating large rates of EGR. In the premixed burn of a DI process or generally in premixed combustion processes CO emissions are dominated by the chemical kinetics [55].

3.5 Exhaust gas recirculation

The use of exhaust gas recirculation in ICEs has been introduced in the late 1960s as a response to the increasing problems related to the pollutant emissions of the road traffic. Investigations showed NOx emissions to be a critical component of smog, acid rain and to be even responsible for respiratory diseases. The potential of reducing NOx using EGR was already known since longer time but has always been related to an undesired efficiency reduction. During the last decades, legislative regulations has become more and more selective and EGR has been a main topic of a variety of investigations. Nowadays, EGR is widely used in many ICE applications as a common way to control NOx emissions by mantaining comparable combustion efficiency.

The most important effect of EGR is the possibility of reducing the flame temperature and thus the formation of thermal NOx during the combustion process. Increasing charge dilution by means of EGR the air content (and especially O_2) is replaced with inert gas (by non-stoichiometric combustion, i.e. diesel combustion, the reactivity of the exhaust gas depends on the equivalence ratio) and the reactivity of the charge is lowered. The partial pressure reduction of the oxygen in the charge reduces the combustion kinetics. In addition the burned gas has a larger specific thermal capacity compared to fesh air (due to the high CO_2 and H_2O fraction) and it is capable of storing more energy during combustion lowering the temperature [39].

At constant boost pressure increasing EGR reduces the equivalence ratio of the charge and NOx emissions are mainly reduced by means of lowered charge reactivity. In this case a negative effect on the specific fuel consumption can be measured. Increasing EGR but maintaining a constant λ (same amount of fresh air) the thermal effect becomes more important. The NOx reduction is less significant compared to the first strategy but there is less or no combustion efficiency deterioration.

Other secondary effects can be related to use of EGR [39]. The composition of the charge is altered by the presence of exhaust gas components. This has an effect on the chemical reaction process, in particular due because of the endothermic dissociation of the water vapor which leads to a flame temperature reduction. Moreover, as a consequence of the charge dilution, the ignition delay increases giving more time for the mixture formation process and promoting premix combustion.

The main negative aspect related to the use of EGR is the so called "thermal throttling". Exhaust gases are characterized by temperatures in a range between 350 and 1000 K depending on the engine load. Thus, the fresh air temperature, which is around 300 K, increases substantially after mixing with the recirculated gases. This results in a reduction of the charge density and a reduction of the charge mass trapped in the combustion chamber. Moreover, the increase of the charge inlet temperature also leads to increased combustion temperature and wall heat transfer, although this effect is usually compensated by the mitigation of high temperatures during the combustion. Thermal throttling can be limited by cooling the EGR before it mixes with the fresh air. This is usually done cooling the recirculated gases using the water from the engine cooling system. Another negative aspect is related to the increased combustion instability which results from a large use of EGR. In this case, an increase in the covariance of the *imep* can be observed. A massive use of EGR may results in misfire [62].

Finally, the use of EGR becomes fundamental in the application of alternative combustion processes, as illustrated in the previous chapter. The thermal advantage of large amounts of EGR and the reduced combustion reactivity can be exploited for avoiding the formation of NOx emissions. Moreover, the mixture preparation process is promoted, by the longer ignition delay, and the formation of soot is suppressed. In the case of homogenous combustion processes, which are solely kinetically controlled, a precise control of the EGR rate plays an even larger role. Typical EGR rates for low-temperature premixed combustion mode varies in a range between 30 and 60 %.

4

Experimental Setup and Engine Management System

The experimental activity has been performed at the Institute of Internal Combustion Engines and Automotive Engineering at the University of Stuttgart. In this chapter, a brief description of the test facilities is presented.

4.1 Engine and measurement equipment

4.1.1 Engine and test bench specifications

Measurements have been conducted on a Daimler 6-cylinder 3.0 l turbocharged diesel engine of type $OM642$ [56][57]. The engine is equipped with a Robert Bosch GmbH third generation common rail injection system, capable of a maximum injection pressure of 1600 bar and up to five injections per cycle. In order to provide a long ignition delay, the engine configuration with a compression ratio of 15.5:1 has been chosen for this investigation. Moreover, the engine disposes of a variable-geometry turbocharger (VGT), a high-pressure cooled EGR loop, a throttle valve to support high EGR routing and inlet manifold flaps to increase intake gas swirl. Further engine specifications are summarized in Table 4.1

Engine load is set using an asynchronous machine dynamometer controlled by a D2T Morphee 2 automation system. This configuration is capable of operating transient load conditions and test cycles. Exhaust gases are extracted down-

Table 4.1: Engine specifications.

Construction type	$V6\ V72$
Valve system	$4V\ DOHC$
Displacement	$2987\,cm^3$
Bore	$83\,mm$
Stroke	$92\,mm$
Connection rod length	$168\,mm$
Compression ratio	$15.5:1$
Max power at 3800 RPM	$165\,kW$
Max torque at 1600-2800 RPM	$510\,Nm$
Injector type	8 holes, 157°
Injector hydraulic flow	887...923 cm^3/min

stream the Turbocharger and measured using a Horiba MEXA 7100 DEGR gas analyzer and an AVL Smokemeter. At steady state conditions, EGR can be collected making a $CO2$-balance between the intake and the exhaust manifolds. Moreover, the engine has been instrumented to provide pressure and temperature measurements in the most relevant states. A digital AVL 733 fuel balance is used to measure the fuel consumption and a AVL 753 for preconditioning the fuel temperature. The air mass flow rate is measured using a Sensi Flow hot film mass flowmeter. Additionally, engine cooling water temperature and intake manifold temperature can be set independently in a wide rage.

Each engine cylinder is equipped with a piezoelectric transducer of type AVL GU23D. Piezo-resistive pressure sensors of type Kistler 4005BA5FV200S and Kistler 4075A10V39 are installed in the intake and the exhaust manifolds respectively. For protecting the sensors from hot gases and soot accumulation a water-cooled switching adapter is used. In addition, a Fluke current clamp is used for detecting the injection current timing. Sensor signals are synchronized with an optical shaft encoder with a resolution of 1 crank angle (CA) degree and collected using an ADwin system. Values are analyzed and stored using INDIGO, a dedicated software developed at the Institute of Internal Combustion Engines and Automotive Engineering at the University of Stuttgart. More details about the real-time system can be found in the following sections.

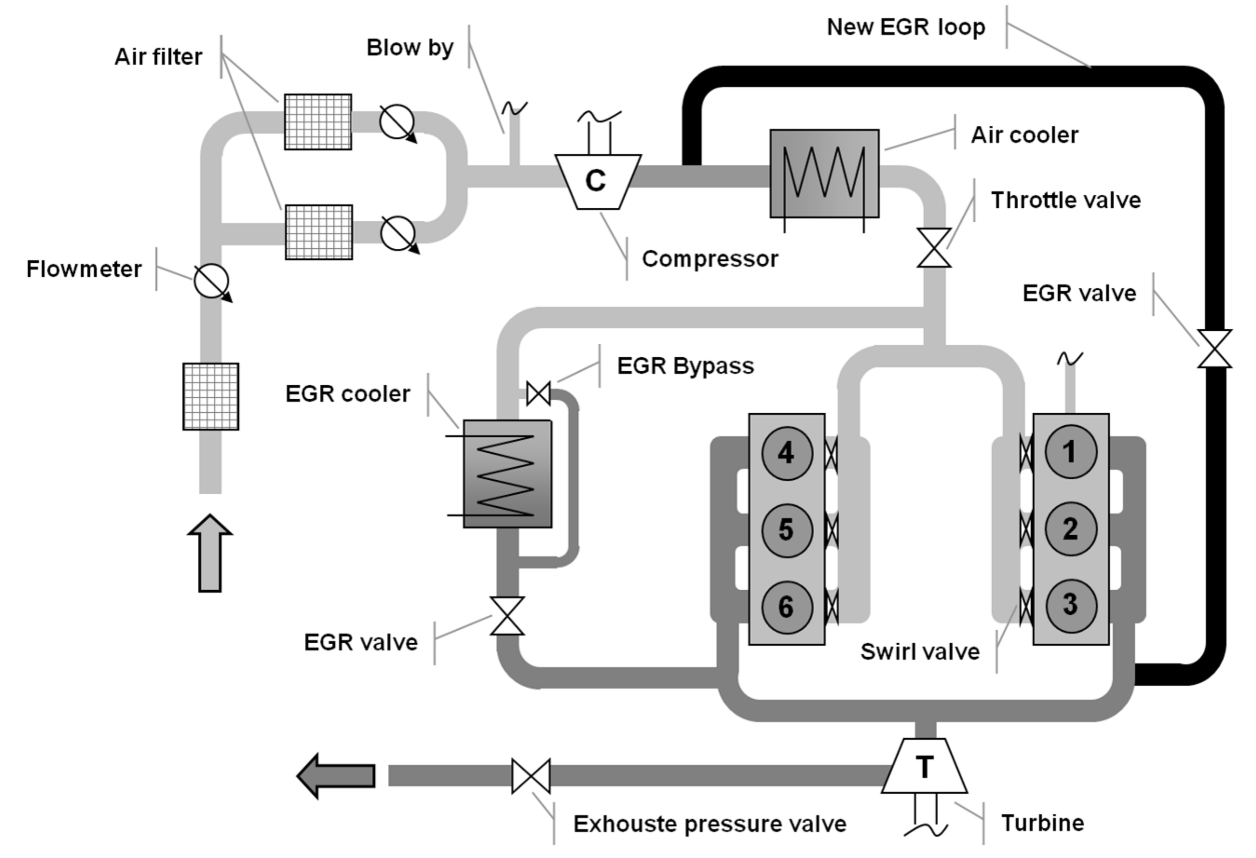

Figure 4.1: Engine hardware configuration.

4.1.2 EGR loop modification

Low-temperature premixed combustion requires high rates of EGR. Preliminary investigations have shown that the original air path configuration can provide a maximum rate of only 45%. Moreover, to attain this concentration the use of the throttle valve is needed. In this way an increased pressure drop between exhaust and intake manifolds is provided but an undesired reduction of the cylinder charge can be measured, yet. Additionally, in the original EGR loop configuration, the exhaust gases are cooled using the water from the engine cooling system, which means that EGR-cooler-out temperatures usually ramain above 350 K. Consequently, by the large ammounts of EGR needed for the low-temperature premixed combustion process, an undesired increase of the intake charge temperatures can be measured. For the reasons mentioned, a new EGR loop has been designed. Following criteria has been taken into account:

- Avoiding of throttling
- Increased gas cooling
- Good mixing with intake fresh air
- Compatibility of the valve with the engine management system
- Robustness and accessibility
- Maintain the original configuration as backup

By the state of the art two feasible solutions can be considered. The first one is a secondary high-pressure EGR loop maintaining the actual connection between exhaust manifold and the intake pipe down-stream the throttle valve. Because of the compact geometry of the engine this solution can not be realized. Moreover, this solution requires the use of an additional cooler and thus additional resistance to the gas flow. On the other hand, a low-pressure EGR, which means an extraction of the exhaust gas after the turbine and recirculation into the intake pipe up-stream the compressor, has also some drawbacks. In fact, since the engine is operated without using any particulate filter in the exhaust pipe, the physical exposure of the compressor to the unfiltered gases could lead to sever damages. Moreover, only using a complex system for mixing the exhaust gas with the fresh air the thermal stress for the compressor can be reduced to sustainable level [59][60]. Finally, a mixed solution of high and low-pressure EGR would increment the effort for a reliable control system considerably [61].

The realized solution is a high-pressure EGR loop with extraction of the gases from the exhaust manifold upstream the turbine and recirculation before the charge-air intercooler, as shown in Figure 4.1. The advantage of this solution is the strong cooling power provided by the intercooler which can be exploited without additional pressure losses. Moreover, since the intercooler is water cooled, arbitrary output temperatures can be set. Because in the drivable load range of LTC combustion the engine is operated at low boost pressure, a pressure drop between exhaust and intake is always provided even without the use of the throttle valve.

4.1.3 Lambda sensor in the intake manifold

To the aim of an engine closed-loop control precise information about the cylinder charge composition are necessary. In modern diesel engines, these information are usually obtained using complex physical models of the air path, which are known to be reliable only to a certain extent. A direct and thus better determination of the charge composition can be done using a broadband Lambda sensor in the intake manifold, e.g. a Bosch LSU 4.2. Connected to a ETAS LA4 Lambda Meter this sensor allows determining the oxygen content in the charge[1]. More informations about the calibration of the sensor can be found in [1]. The oxygen content is of particular interest for the rate of reactivity of the air/fuel mixture. Later in this work it will be shown how to make use of it in controlling the combustion process. Furthermore, since the combustion process takes place under lean conditions, the oxygen balance between intake and exhaust gases can be exploited for a direct and continuous determination of the EGR rate.

4.2 Real-time combustion analysis system

For the closed-loop combustion control a real-time in-cylinder pressure analysis system has been used. Pressure traces are collected using an ADwin-PRO-system which disposes of one ADSP 21162 80 MHz processor for each single signal source. The algorithms for the evaluation of the thermodynamic parameters are implemented in an ADbasic code and have been optimized for real-time use at the Institute of Internal Combustion Engines and Automotive Engineering at the University of Stuttgart during many years of development. The combustion process is characterized using the equation of Bargende [58], which is based on a inverse first-law thermodynamic model and is capable of calculating the gross heat release rate (HRR) in real-time. This simplified model is based on

[1] A broadband Lambda sensor delivers a signal which is proportional to the oxygen partial pressure of the analyzed gas. This signal can be converted either by means of an analytical method that considers ambient conditions or by characteristic curves.

the assumption that the biggest portion of the combustion process results in a in-cylinder pressure change and thus the gross heat release rate is a good approximation of the fuel energy released during the combustion. For the closed-loop control, the parameters shown in Table 4.2 are collected.

Table 4.2: Real-time combustion parameters.

Parameter	**Symbol**	**Unit**
10% of integral heat release, combustion beginning	$CA10$	$°CA$
50% of integral heat release, center of combustion	$CA50$	$°CA$
Indicated mean effective pressure	$imep$	bar
Cylinder pressure at defined piston position	p_θ	bar
Maximum cylinder pressure	p_{max}	bar
Maximum rate of cylinder pressure rise	$dp/d\theta_{max}$	$bar/°CA$

The location of the 10% of the integral heat release has been found to provide a more stable detection of the combustion beginning compared to the 5% value. Especially for low-temperature premixed combustion, which is characterized by a two stage ignition process followed by a vary fast energy release, this can be a convenient solution. Differently to other real-time systems, the $imep$ is calculated over an entire combustion cycle. In order to obtain an $imep$ value between two consecutive combustion cycles the algorithm starts to integrate at 500°CA before TDC and ends at 220°CA after TDC, for each cylinder respectively. The CAN message, containing all the combustion information, is than sent to the control unit at 240°CA after TDC. Further and more detailed information about the real-time combustion analysis system can be found in [1].

4.3 Engine management system

The standard Bosch EDC 17 ECU has been replaced by the rapid prototyping AFT PROtroniC ECU. This ECU works on a MPC5554 120 MHz processor and can be easily programmed using Matlab Simulink. The PROtroniC is connected to the engine using a breakout box which makes all the engine functionality easily accessible. Moreover, a switch with the Bosch *EDC* 17 can be realized in very short time without any further modification of the engine hardware. The high voltage *TTL* signals for operating the piezoelectric injection system are delivered by a Continental Piezo Driver, which is programmed to deliver the injectors with the suitable voltage and current curves and gets the trigger signals from the *ECU* over an optocoupler. For the calibration and diagnostic of the *ECU* functionalities the software AFT MARC is used.

For this project, a completely new engine control software architecture has been developed. The structure is divided in "input", "function" and "output" components which makes the information flux particularly easy to understand and to administrate. In the first one, signals coming form the engine sensors and the real-time combustion analysis system are metered and converted into physical values. In the "function" part, engine maps and control algorithms are implemented. This part is also responsible for the combustion closed-loop control. In the latter, new control target values are first converted in current signals and than sent to the engine actuators.

Engine load requirements are set using the gas pedal which signal is directly commutated into a desired *imep*. In this way, the *imep* becomes a main control parameter that characterize the engine load together with the engine speed. In fact, using cylinder pressure transducers which provide real-time *imep*, fuel injection quantity, which is commonly used in commercial *ECU*s and is rather based on injection models, becomes obsolete. Detailed information about the software architecture can be found in [1].

5

System analysis and definition of the combustion control strategy

Despite the large amount of investigations about the influence of operating parameters on the application of low-temperature premixed combustion, most part of the closed-loop control systems published in the last years does not rely on the thermodynamic mechanisms and the physical correlations between actuating and target variables. To the point of view of the author, the definition of a solid control strategy can only result from a deeper understanding of the combustion process and its mechanisms. This approach is illustrated in the following chapter. After the identification of the most relevant mechanisms and the correlations between actuating and target variables, a new control strategy is proposed.

A modern diesel engine is a very complex system in which the variation of a single parameter usually affects a large variety of different mechanisms. This strong interaction between the actuating and the measuring parameter, or target parameters, makes the investigation and the control of a diesel engine a particular difficult task. As illustrated in Chapter 2 many different solutions have been proposed in the last years for the control of low-temperature premixed combustion systems. In order to propose a new closed-loop combustion control strategy an experimental analysis of the combustion system is carried out on the 6-cylinder engine presented in the previous chapter. Main goal is the identification of clear and direct physical correlations between the actuating and the target variables, which definition is also part of the present chapter.

The right choice of the actuating variables depends mainly from the system properties and the definition of the control targets. The first step is the identification of the actuators and their functionalities. In this work, the focus has been set on the realization of a closed-loop control strategy for a multi-cylinder series engine and not, differently to other approaches, for a modified engine or an experimental apparatus. Therefore, except for the modification of the EGR loop, which has been amplified as explained in the previous chapter, it is made use of the standard engine equipment. This means that any modification of the engine hardware, such as compression ratio or valves timing, injection system or fuel properties are investigated in this work.

Figure 5.1 shows the actuating and the target parameters selected for this investigation. Actuating variables have been divided in two main groups, the fuel path, which includes the variables of the injection system, and the air path, which includes the VGT, the throttle valve, the swirl flaps and the EGR valve. The target variables, which has been arbitrary selected, are here defined in three groups. The engine load, which can be calculated using the indicated mean effective pressure, the engine protection and noise, including the maximum pressure rise $dp/d\theta_{max}$ and the standard deviation of the *imep* (as a measure for the combustion stability), and finally the emission and fuel consumption group.

The first step by applying a new combustion process is to identify system constrains and application limits for the actuating and measuring variables. In this way the operation limits of the low-temperature premixed combustion can be identified as well as the most relevant mechanisms of the combustion process. This information are fundamental for the definition of a closed-loop control.

The sensitivity analysis of alternative combustion processes has been the focus of many investigations of the past years, so that almost every sort of parameter variation can be found in the literature. A list of some publications is shown in Table 5.1. However, every engine type has own characteristics and its behavior may differ substantially from what has been demonstrated in other investigations. Moreover, a sensitivity analysis is always suggested in order to get a feeling about the reactivity and limitations of the system under investigation.

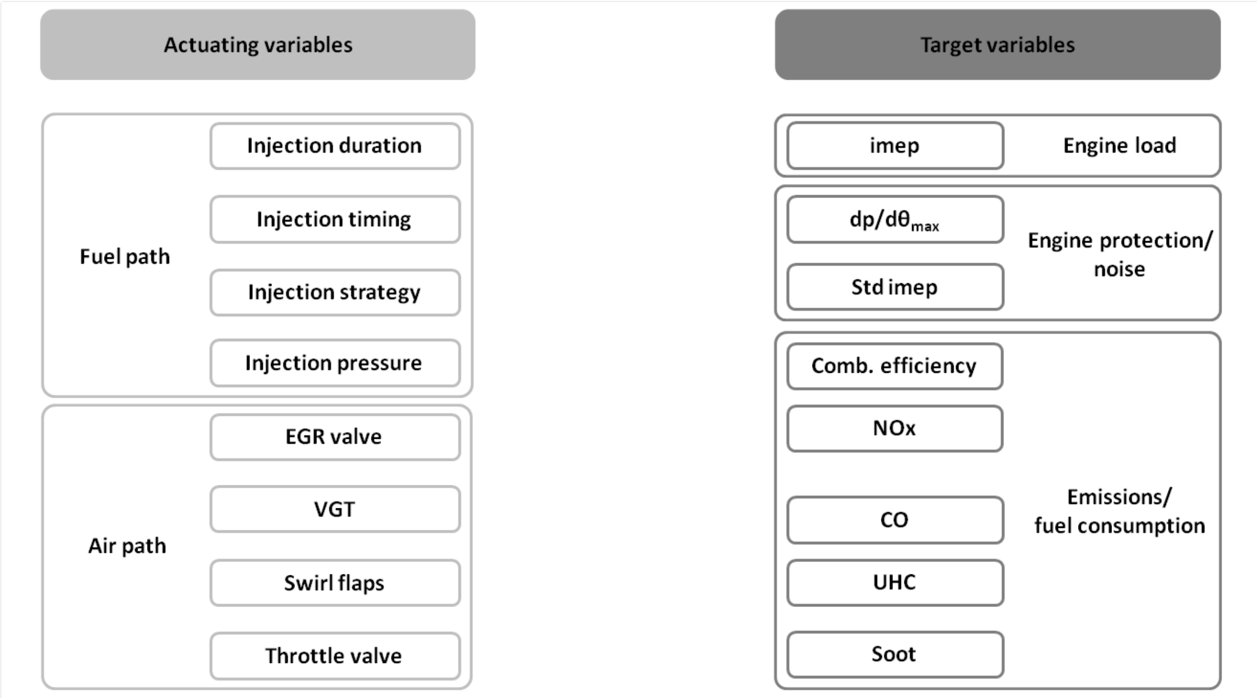

Figure 5.1: Definition of actuating and target variables in a modern diesel engine.

5.1 Engine constrains

Engine constructive constrains has to be taken as "hard" system limitations which can not be arbitrary adjusted during engine operation. These are the geometry of the combustion chamber, the characteristics of the air path and the turbocharger, the valves timing, the characteristics of the injection system and the fuel quality[1].

5.1.1 Combustion chamber geometry

The most relevant constrain related to the combustion chamber geometry is the shape of the piston bowl. Since the combustion chamber of the $OM642$ has been designed and optimized for the application of a conventional DI combustion, the piston bowl has the typical ω-shape which guarantee optimal mixture formation

[1]The influence of fuel properties on the low-temperature premixed combustion has been demonstrated in many works, e.g. [20] [10] [49]. For practicability reasons in this work the use of different fuel blends is not considered as a feasible actuating parameters.

Table 5.1: List of publications regarding the sensitivity analysis of some combustion variables in low-temperature premixed combustion

Reference	[10]	[11]	[64]	[65]	[66]	[49]	[14]	[39]	[67]	[54]	[55]
Engine cylinders	1	1	1	1	4	1	1	4	1	4	4
Fuel mass	●						●		●		
Inj. timing	●		●		●	●	●		●	●	
Inj. strategy						●					
Inj. pressure	●	●			●	●	●		●		●
EGR rate	●	●	●		●	●	●	●	●	●	●
EGR temperature				●	●	●					
Swirl	●	●		●						●	
Air/fuel ratio	●	●									
Nozzle geometry	●					●			●	●	
Bowl geometry	●								●	●	
Compression ratio		●			●		●				
Fuel properties	●				●	●					●
p_{Int} and T_{Int}	●			●		●		●		●	●
p_{Exh}				●							

at high load conditions, where a large amount of fuel is injected at TDC.

In Figure 5.2 the interaction between the fuel spray and the piston bowl is illustrated. It can be seen that by early injection timing the fuel spray misses the piston bowl landing on the top surface of the piston crown. As already mentioned in the previous chapter, this leads to a deterioration of the combustion with immediately measurable consequences on the engine power output and the UHC emissions. To overcome the out of bowl injection problem a narrow spray cone angle injector in combination with a deeper piston bowl geometry should

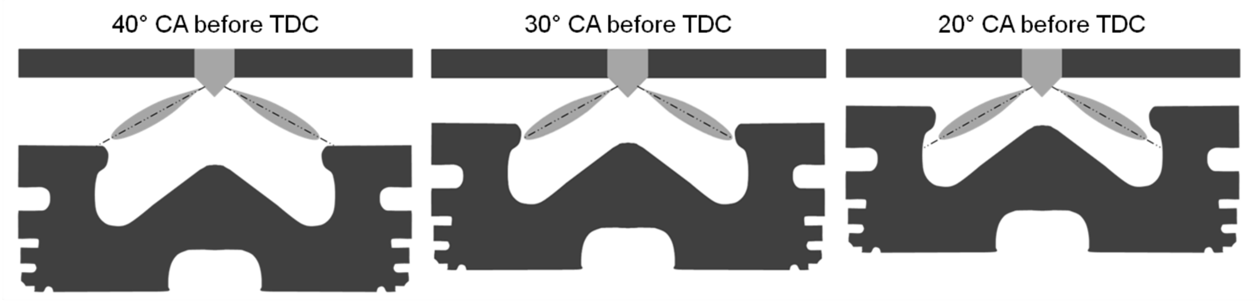

Figure 5.2: Schematic diagram of the interaction of the fuel spray with the piston bowl.

be used [63]. Otherwise, prolonged mixture formation will be limited by this constrain.

This limit depends from many operation parameters, i.e. injection pressure, engine speed, charge composition and intake pressure and temperature. In other words, all the parameters which affect the spray penetration in the combustion chamber or the piston speed, sicne the injection process is characterized by a physical time delay. Measurements on the $OM642$ has shown injection events earlier than $25°CA$ at low engine speed and earlier $35°CA$ at higher speed to suffer from this constrain. In this work a limitation of $30°CA$ before TDC is suggested. This limit has been identified measuring the engine-out UHC emissions by varying the injection timing for different engine operation conditions as shown in Figure 5.3. The increase in the UHC emissions by early injection events can be easily recognized. As expected, an increased injection pressure leads to a reduction of the injection timing range.

5.1.2 Injection system

The wide injection angle, which, as explained above, limits the application of early injection events, and the maximum injection pressure, limited at 1600 bar, represent the most relevant injection system constrains. However, in the system under investigation, the injection pressure does not represent an application limit since the maximum pressure needed for the low-temperature premixed combustion is of only 1100 bar.

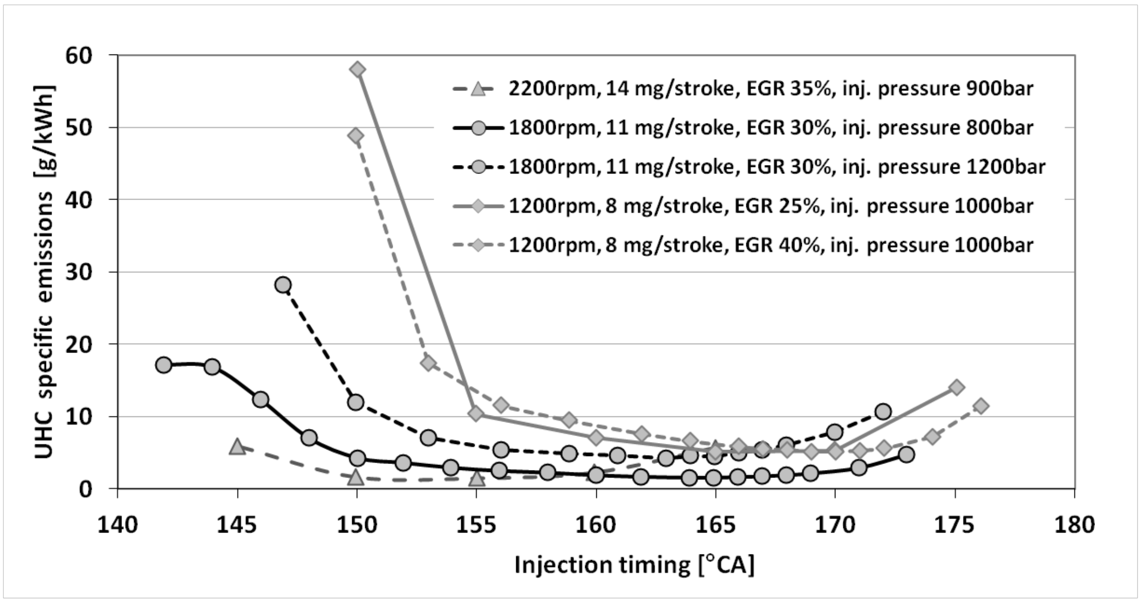

Figure 5.3: UHC emissions by a variation of the injection timing at different engine loads showing the interaction between the fuel spray and the piston bowl.

5.1.3 Air path

The air path is mainly characterized by its volume, the pressure losses along the pipes, and the possibility of varying boost pressure, intake temperature and EGR rate during engine operation. The volume of the pipes, the intercooler and the intake manifolds have a large influence on the system response of the air path. This is of particular relevance under transient load opertation, where a closed-loop control may become difficult. Detailed investigations about the air path system response of the $OM642$ can be found in [1].

Another constrain of relevance by the application of alternative combustion processes is the availability of recirculated exhaust gas. EGR can be recirculated only when there is a positive pressure drop between the exhaust and the intake side of the engine. Since the enthalpy of the exhaust gases is increased by the combustion process this pressure difference is usually given in a ICE, especially using a turbocharger. Exploiting the pressure drop, exhaust gas can be recirculated and mixed with the fresh air. However, the maximum amount of EGR is additionally limited by the pressure losses in the EGR loop. In order to avoid a degradation of the volumetric efficiency by means of reduced charge density, the EGR must be cooled before being mixed with the fresh air. Due to the flow resistance in the EGR cooler, the maximum available rate is further reduced. In the previous chapter a different solution to the conventional EGR loop has been discussed. In Figure 5.4 the improvements of this modification are illustrated. Thanks to the reduced pressure losses in the pipes the maximum EGR rate has been increased of about 5 %. Moreover, recirculating the exhaust gases upstream the intercooler, the charge temperature entering the cylinder remains constant. Positive effects can be measured by a slightly larger equivalence ratio and by a NOx emissions reduction. Depending on the engine load, a maximum rate up to 55 % can be recirculated using this solution. This limit is achieved closing the VGT by 92 %.

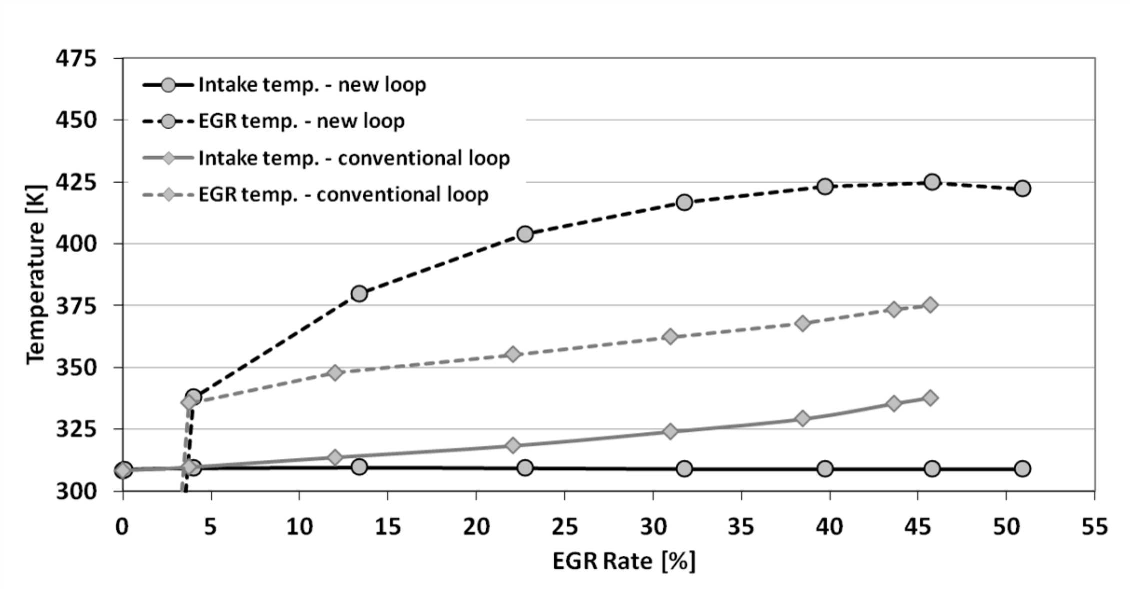

Figure 5.4: Comparison between new and conventional EGR loop. EGR valve variation at constant speed 1800 rpm and injection quantity 10.7 $mg/stroke$.

A main limiting factor of the air path is the boost pressure provided by the turbocharger. In fact, since the turbocharger of the $OM642$ has been designed for supplying the engine with maximum amount of fresh air at full load operation, there is no or little changes of varying the boost pressure in the low load range, where low-temperature premixed combustion is applied. Investigations have shown a VGT position of 92 % to be the best compromise for achieving maximum EGR by sufficient boost pressure. This position was kept constant for the entire low-temperature premixed mode. However, high boost pressure at low load would have a greater potential in extending the operating limits of alternative combustion processes and improving the trade-off between emissions and combustion efficiency [73] [74]. Moreover, increasing the boost pressure there would be the possibility of varying EGR rate and fresh air mass independently, thus increasing the degrees of freedom of the closed-loop control.

5.2 System limitations

Differently to the engine constrains, system limitations, or "soft" limitations, are related to the interaction between engine actuating variables and system behavior. These can be identified by means of system sensitivity analysis.

5.2.1 Injection strategy

The potential of using multiple injection strategies in low-temperature premixed combustion has been reported in many publications. An improved fuel homogenization resulting form a multiple injection strategy is shown in [49]. However the author reports over-mixing problems related to fast heat release and high rates of pressure rise as the mixture approaches stoichiometry. In this case, multiple injection has any remarkable emission improvement compared to the single injection. On the contrary, fuel wall impingement and degradation of the oil film may result. In [69] the pilot injection, which is separated by the main injection event, takes place at the begin of the expansion stroke. In this way the fuel burns for the two injections in apremixed split-combustion process. Thanks to low-temperature resulting from the late combustion phasing both NOx and PM formation are suppressed. Also in [70], problematics related to the multiple injection strategy are highlighted. At low load, early injection events may result in fuel over-mixing. The distribution of the fuel quantity between the first and the second injection plays an important role. Moreover, the fuel injected during the second event may burn diffusive with significant increase of soot emissions. On the other hand, at part and medium load a split combustion results in reduced engine noise, lower UHC and CO emissions. Of particular interest is in this case the increased sensitivity of the combustion from engine operating conditions, i.e. speed and load, using multiple injection strategy. In [74] and

[68] any significant advantage of multiple injection at low load can be identified. Using multiple injections at higher engine load the leanness and homogeneity of the mixture is improved with benefits for CO, UHC and PM emissions with little expense of NOx emissions. Furthermore, there is the possibility to extend the operation range of the low-temperature premixed combustion to higher loads.

Preliminary investigations on the $OM642$ have shown very little potential in improving mixture formation using multiple injection, by significantly increasing the risk of diffusive combustion. In the short range of possible injection timing, limited from the one side by the necessity of injecting the fuel in the piston bowl (as explained in the previous section) and from the other side by the combustion deterioration resulting from a late combustion phasing, there is only short time for multiple injections. As a result of this investigation and in consideration of the increased complexity highlighted in the publications cited above, it was decided to use a block injection strategy. This means, the in the entire application range of the low-temperature premixed combustion mode the fuel is completely injected during a single event.

5.2.2 Injection duration

Injection duration is an extent for the fuel quantity injected during an engine cycle. The correlation between these two parameters mainly depends from the injection pressure and the injection timing. In the first case, increasing injection pressure, the fuel injection rate per time unit increases and a shorter injection timing is needed in order to maintain a constant injected fuel mass. By large variations of the injection timing a correction over the injection duration is also needed. In fact, departing from the TDC the in-cylinder pressure and charge density decrease and the fuel jet encounters less resistance. Finally, the fuel quantity is directed related to the indicated piston work. Even if the rate of the fuel conversion varies with most part of the engine operation parameters, this correlation can be assumed to be monotonically increasing.

5.2.3 Injection timing

The combustion strategy employed in this work is characterized by a late single injection event. This strategy is particularly suited for the use of a conventional diesel engine equipped with standard piston ω-bowl geometry and standard injection system with a wide injection angle. As explained in the previous section, it is necessary to match the injection spray with the piston bowl, by means of correct application of injection timing and injection pressure, in order to avoid fuel wall impingement on the piston crown. For this reason, early injection timing is limited by $30°CA$ before TDC. Moreover, this investigation focuses on those combustion processes in which long mixture formation is promoted but there still exists the possibility of controlling the combustion phasing over the injection timing. Figure 5.5 shows the correlation between these two parameters. At low load conditions, i.e. $1600\,rpm$ λ 6 and $1800\,rpm$ λ 4, for very early injection timing the correlation runs into an asymptote, this means that the combustion beginning can not be further anticipated varyind the injection timing. Generally, however, a strictly monotonic increasing correlation can be racognized. This circumstance is fundamental for the application of a robust closed-loop control strategy.

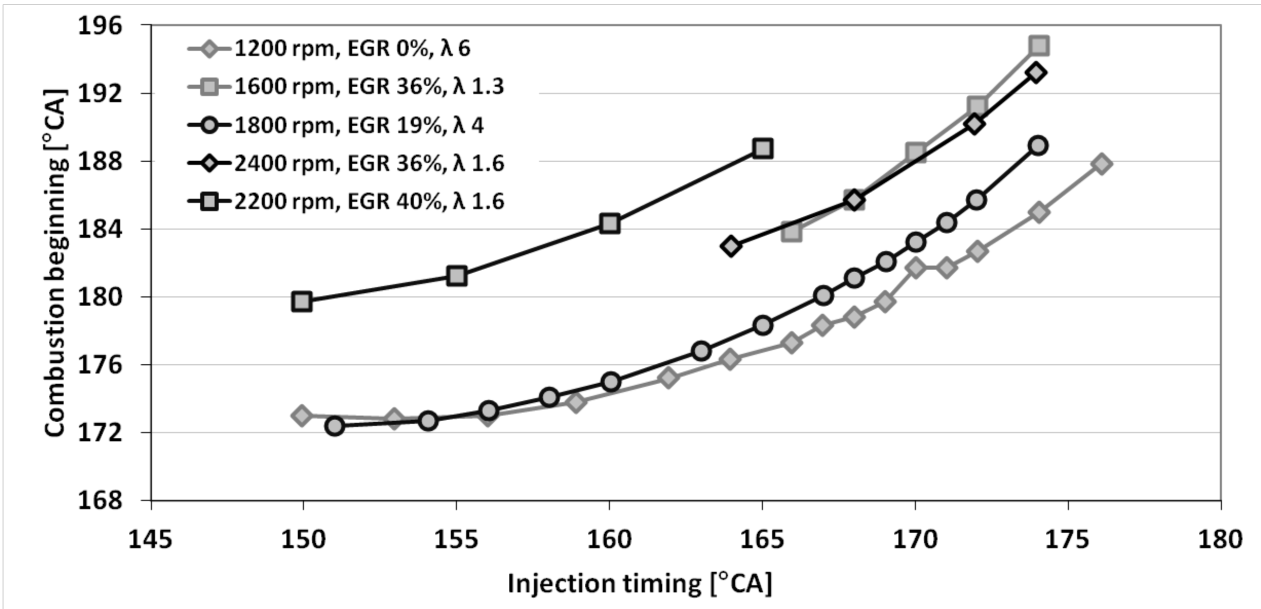

Figure 5.5: Injection timing variation at different engine loads and its correlation to the combustion beginning.

The measurements illustrated in Figure 5.5 and in the following two figures show the results of injection timing variations at different engine loads and speed. Each variation has been carried out at constant EGR rate and constant λ. In fact, as explained above, keeping the injection pressure constant, it is necessary to compensate the variation of in-cylinder pressure and charge density at the different injection timing in order to avoid fluctuations of the injected fuel mass. To overcome accuracy problems related to the measurement of the fuel quantity using the fuel meter (not adequate for this kind of investigation), the injection duration is set by means of constant λ. Constant intake pressure and temperature are also provided.

Because of the strictly monotonic correlation between injection timing and combustion beginning illustrated above, the next two diagrams are plotted over the combustion beginning, which is defined as the 10 % of the integral heat release. In this way the mechanisms taking place before and during combustion can be better highlighted.

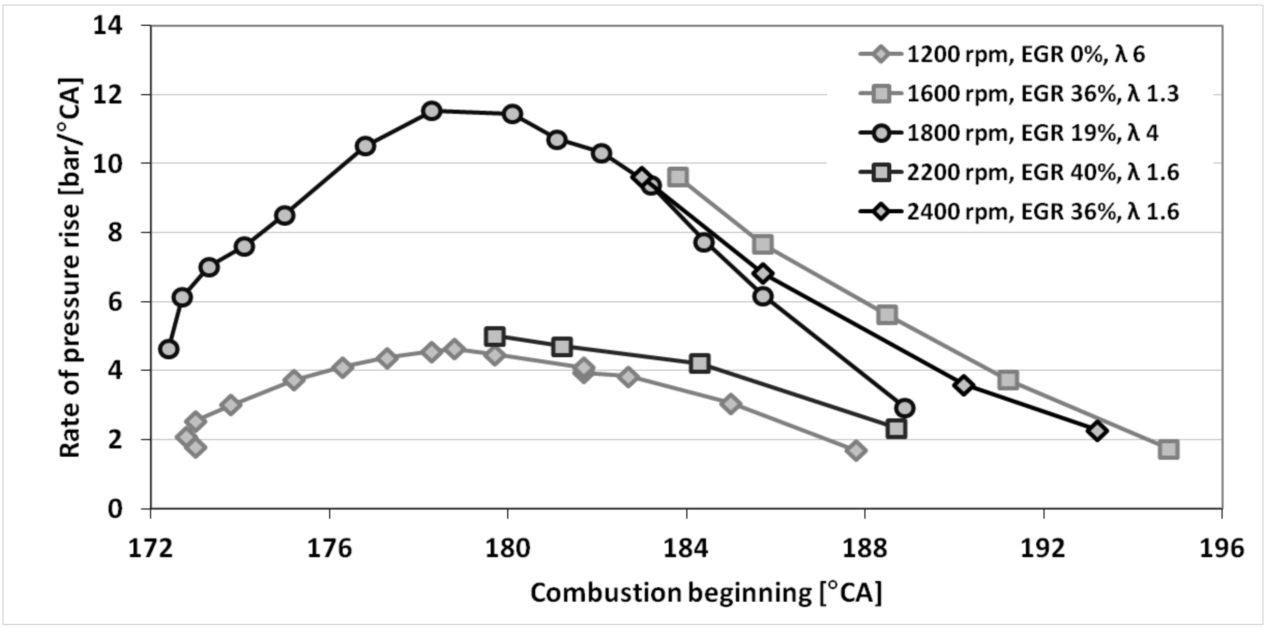

Figure 5.6: Maximum rate of pressure rise for different engine loads plotted over the combustion beginning.

In Figure 5.6 it can be seen how, independently from the engine load (here represented by λ) and speed, the maximum rate of pressure rise grows to a maximum, which corresponds to a combustion beginning at TDC. These means, that phasing the combustion from the compression to the expansion, the conditions for the fastest heat release are given at TDC. Two mechanisms are supposed to be responsible for the illustrated behavior. The change of the in-cylinder conditions (pressure and temperature) due to piston motion, which are in first order responsible for the rate of pressure rise, and the change of the ignition delay and thus the mixing time, which also depends from the in-cylinder conditions and thus has a second order effect on the rate of pressure rise[48]. The ignition delay is a very important parameter in alternative combustion processes, since it is an extent for the fuel homogenization process. In other words, at constant boundary conditions, a longer ignition delay is a clear sign for a better mixture formation and a global mixture leaning, and vice versa. Optical investigation demonstrating this circumstance can be found in [10]. Moreover, the mixture quality is, together with pressure and temperature, directly responsible for the devolution of the combustion process and its characteristics [2] [71]. The effects of these mechanisms work simultaneously and lead to the behavior illustrated in Figure 5.6.

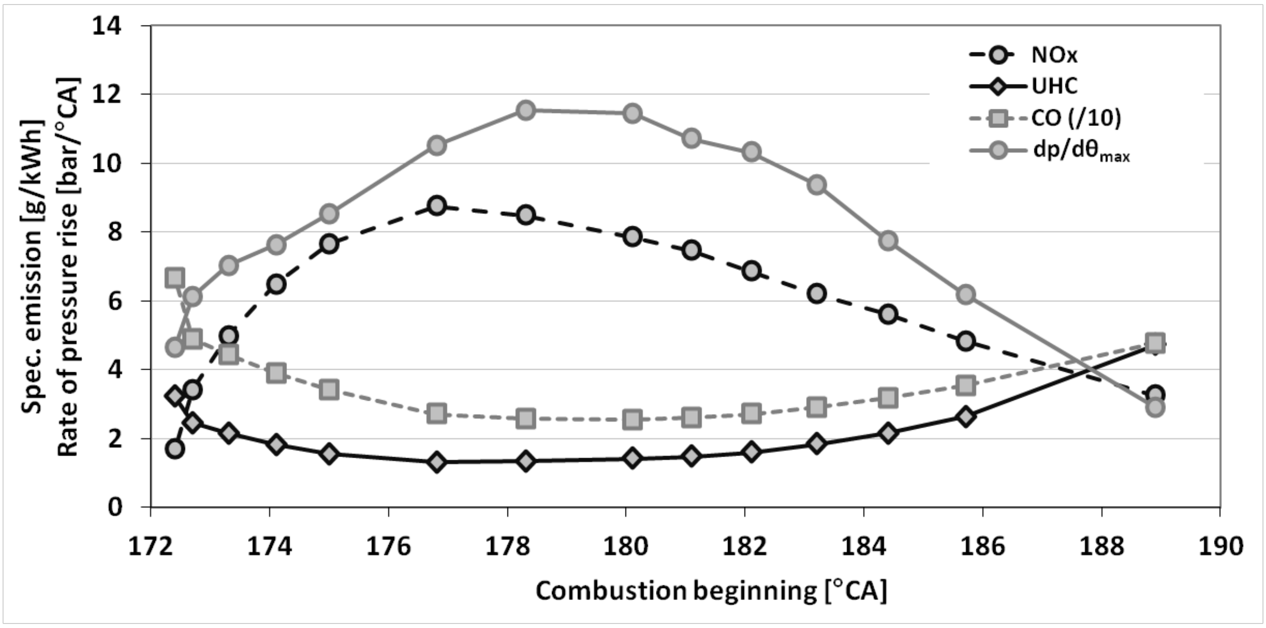

Figure 5.7: Specific emissions and maximum rate of pressure rise plotted over the combustion beginning for the selected engine load at 1800 rpm and λ 4.

The analysis of the engine-out emissions confirms the mechanisms identified above. These are plotted together with the maximum rate of pressure rise over the combustion beginning in Figure 5.7, for the representative load point at $1800\,rpm$ λ 4. Easy to recognize is the good correlation between NOx emissions and the $dp/d\theta_{max}$. This is because the conditions which lead to the highest rate of pressure rise, i.e. high pressure, high temperature and shorter time for fuel homogenization, also promote the formation of nitrogen oxides. The reason behind the different peak position of the traces has probably to be found in the effects of the combustion process on the emissions, whereas the maximum pressure rise is mainly related to the conditions at combustion beginning. UHC and CO emissions show an opposite trend. Where the highest rates of pressure rise are measured, high temperatures and short time for mixture formation inhibit their formation [54].

The results of this investigation show a strict correlation between the maximum rate of pressure rise and the combustion beginning, and thus the injection timing. This correlation is strictly monotonic increasing as long as the combustion beginning is phased in the compression stroke and becomes strictly monotonic decreasing when the combustion beginning moves in the expansion. A peak of the maximum rate of pressure rise is reached approximatively at TDC. NOx emissions shows a very similar behavior in contrast to UHC and CO emissions.

The realization of a closed-loop control system that take care of this discontinuity is a difficult task. Since combustion during compression is not possible under every load condition, as shown in Figure 5.6, and the injection beginning range is limited at $30°CA$ before TDC for early injections, for a robust closed-loop control system it is suggested to limit the combustion in the expansion. In this phase is the correlation indipendent from the engine load and there is the possibility of controling of the maximum rate of pressure rise over the injection beginning (with consideration of the combustion beginning). This solution would provide a constant engine protection, which is one of the target variables defined at the beginning of this chapter. Similar results has been found in [36], where a closed-loop control of the maximum rate of pressure rise over the injection timing has been implemented.

5.2.4 Injection pressure

The injection pressure usually increases with the engine load in order to provide short injection duration by increasing fuel quantity. In the DI combustion process, however, the injection pressure is limited in order to avoid over-mixing of the pilot injections. Because of the small fuel quantity injected during these short events, at high injection pressures the fuel would not ignite properly and the main combustion would burn with an increased premixed character increasing the combustion noise. This is not the case of low-temperature premixed processes where it is made use of larger injection pressures. In this way, the entire fuel can be injected prior to combustion onset and fuel homogenization is promoted.

A variation of the injection pressure has been carried out. In order to better isolate the effect of the injection pressure on the premixed combustion, a constant beginning of the combustion at TDC during the entire parameter variation has been chosen. In this way, the effect of the piston motion and the change in the combustion chamber volume are reduced at minimum. Other boundary conditions such as charge composition ($34\,\%\ EGR$), temperature ($780\,K$) and pressure ($28\,bar$) at injection beginning as well as the injected fuel mass ($7.7\,mg/stroke$) have been also kept constant.

The results illustrated in Figure 5.8 show a clear trade-off between NOx and UHC-CO emissions. In accordance to the emission formation mechanisms, explained in Chapter 3, this circumstance can be explained with a variation of the local mixture quality at combustion onset. Increasing the injection pressure two mechanisms take place. The first one is an increased vaporization and homogeneization of the fuel, resulting from the larger exchange momentum between the fuel spray and the charge. The second, as shown on the left side of the figure, is an increase of the mixing time (defined as the time gap between end of the injection and the combustion beginning) due to the shorter injection duration by constant ignition delay. Both this effects promote fuel homogenization leading to an increased local air/fuel equivalence ratio. Over-mixing takes place and combustion temperature decreases, going from 2600 to $2500\,K$. As a consequence NOx formation is reduced whereas engine-out UHC and CO emissions increases.

A validation of these results can be found in the literature listed in Table 5.1.

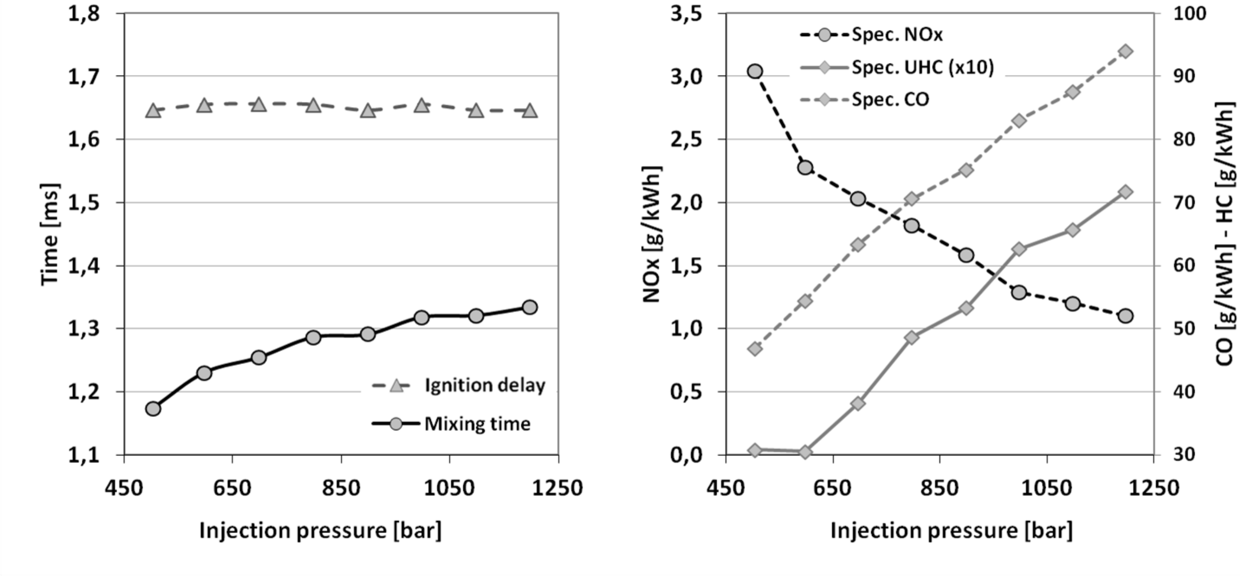

Figure 5.8: Ignition delay and mixing time on the left and specific engine-out emissions on the right by an injection pressure variation at constant combustion beginning.

In [10] optical investigations of the OH-chemiluminescence show an increased mixture homegenization as the cause of increased UHC-CO emissions. In [66] and [14] a reduction of soot formation for $HPLI$ combustion by means of increased injection pressure is reported. The same effect of the injection pressure on the trade-off between the emissions is shown in [67]. Also in this case any overall engine performance improvement can be achieved by means of injection pressure variation.

The trade-off in the emissions shows the importance of the right choice of an adequate injection pressure. High injection pressure promote fuel mixture formation but a limit must be set in other to avoid combustion deterioration due to over-mixing. This must be done matching other engine operating parameters such as injection timing and charge dilution. In this work a concentration limit of 500 ppm for UHC and 1000 ppm for CO engine-out emissions has been applied. In compliance with this limits, an injection pressure map for the entire application range of the low-temperature premixed combustion has been defined. As illustrated in the contour maps showed in the Appendix, the injection pressure increases together with the load ($imep$) from 900 to 1100 bar but remain constant over the speed.

In conclusion, the injection pressure can be utilized for the fine tuning of the combustion process but can not be considered as a feasible parameter for controlling the combustion process.

5.2.5 Throttle valve

In the application of DI combustion processes, throttling is usually exploited in only few engine operating conditions, such as during the regeneration of the DPF, where combustion takes place under fuel-rich conditions. In this case, the pressure drop behind the valve does not only decrease the total air mass entering the engine, but it increases the recirculation of the exhaust gas reducing the fresh air content of the charge. An increase of the EGR rate by means of throttling is not considered to be a feasible solution in low-temperature premixed combustion, since maximum cylinder filling is desired. As explained in Chapter 4, with the proposed extension of the EGR loop, where the exhaust gases are mixed with the fresh air upstream the intercooler, the use of the throttle valve is no more necessary for the recirculation of high rates of EGR.

5.2.6 Swirl flaps

Through actuation of the swirl flaps (also known as duct cutoffs) one of the two inlet channels of one cylinder can be partially or completely closed creating a swirling of the fresh air. This solution has been developed for DI combustion processes to overcome the poor mixture formation problematic at low engine speed, in particular because of the large formation of PM affecting this load reagion. Swirl helps primarely in increasing the fuel homogenization prior to combustion onset, avoiding so fuel-rich areas, or in promoting fuel oxidation in the late combustion phase. Moreover, as explained in Chapter 3, swirl may reduce fuel wall impingement by deviating the spray cone.

Investigations of the swirl effect on the low-temperature premixed combustion has been carried out at two different load points. The swirl has been varied closing the flaps progressively. Since the actual swirl number in the combustion chamber is not known, the data are plotted over the flaps opening position, where 100 % means the valve is completely closed and maximum swirl is achieved whereas 0 % means that both inlet channels are fully opened.

The results for the $HCLI$ mode are illustrated in Figure 5.9. The load point at 1800 rpm is characterized by an early injection timing of 25°CA before TDC and an EGR rate of 50 %. The entire fuel mass of 9 $mg/stroke$ has large time for mixture formation, which is also promoted by the high rate of EGR. Also in this case, a constant combustion beginning by TDC has been kept constant over the entire variation in order to better highlight the effect of charge swirling.

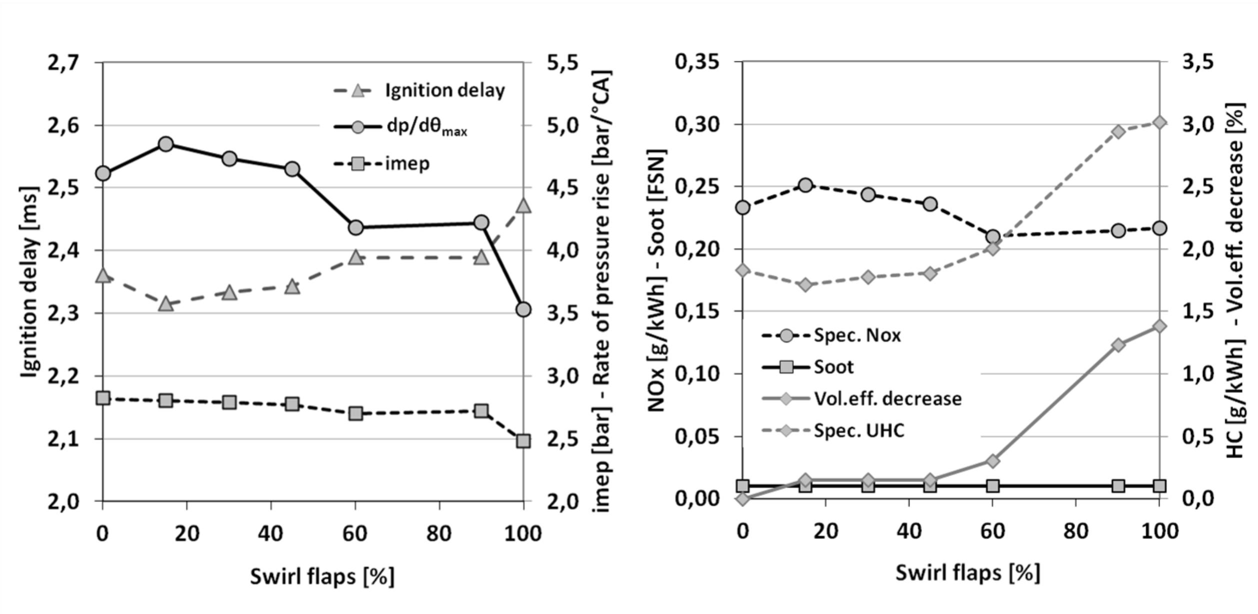

Figure 5.9: Ignition delay, $imep$ and rate of pressure rise on the left, specific engine-out NOx and UHC emissions, smoke number and decrease of the volumetric efficiency on the right, by a swirl variation at constant combustion beginning in $HCLI$ mode.

All the parameters plotted point to an increased fuel homogenization by higher swirl. Similar to the results seen by the variation of the injection pressure, a trade-off between UHC (and CO, which is not plotted) and NOx can be recognized on the right side. The slightly increase in the ignition delay and the reduction of the maximum rates of pressure rise on the left side confirm the prolonged mixing time. Because of the high charge dilution, NOx emissions are globally low and soot formation does not take place. Generally, it can be said that the use of the swirl valve in $HCLI$ mode has little or no positive effects but leads to a slightly increase of the pumping losses and deterioration of the volumetric efficiency, together with an excessive fuel homogenization.

Figure 5.10 shows the measurement at higher load. Engine speed is again $1800\,rpm$ but injection quantity is increased at $21\,mg/stroke$. In this case, the exhaust gas recirculation can be increased at a maximum rate of only 27 % because of the λ limit of 1.2. Due to the large fuel quantity and the low EGR rate, the combustion is characterized by high rates of heat release and must be phased later in the expansion. For this reason, a constant combustion beginning of $12°CA$ after TDC has been kept constant during the swirl variation, which corresponds to a rate of pressure rise of $9\,bar/°CA$. This is realized with an injection timing of $2°CA$ before TDC. At the injection pressure of $1100\,bar$ the injection delay measures approximatively $1.8°CA$. These operating conditions are typical for $HPLI$ combustion.

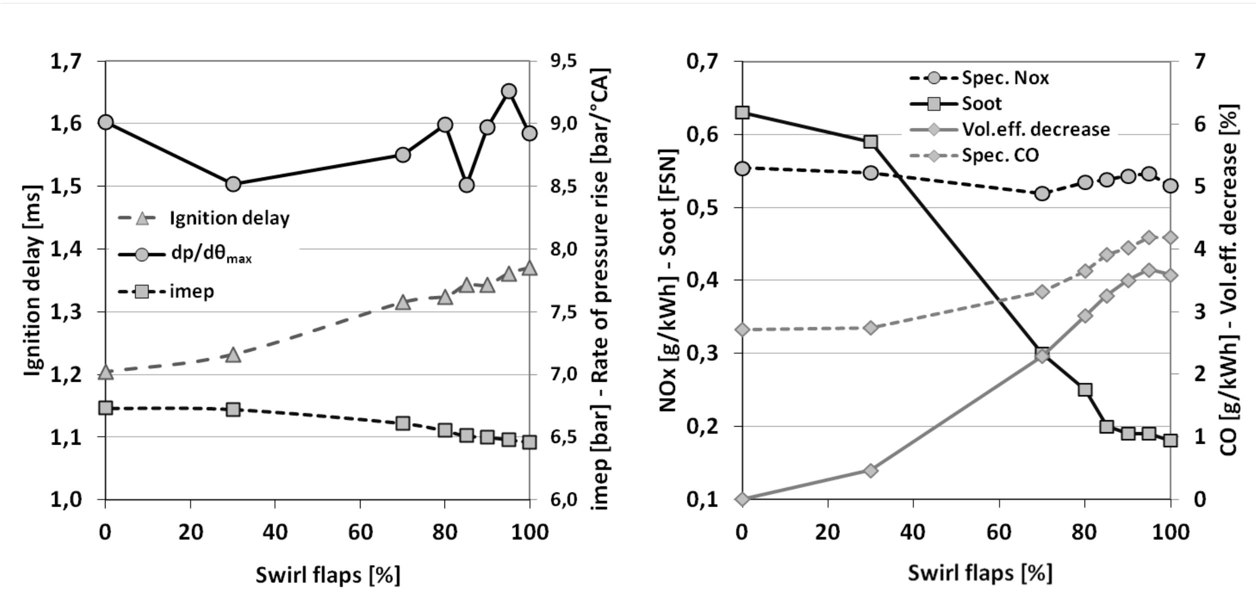

Figure 5.10: Ignition delay, $imep$ and rate of pressure rise on the left, specific engine-out NOx and CO emissions, smoke number and decrease of the volumetric efficiency on the right, by a swirl variation at constant combustion beginning in $HPLI$ mode.

Due to the high reactivity of the charge (short mixing time and high combustion temperatures) and the low global equivalent ratio, soot formation can not be avoided during combustion. Increasing the swirl however, a significant reduction is possible. In this case, looking at the engine performances and the engine-out emissions, it cannot be clearly said if this reduction results either from a better fuel homogenization prior to combustion or from an improved oxidation of the soot oxidation in the late combustion phase. In [8], optical investigation of late injection homogeneous combustion processes has pointed out late oxidation mechanisms to be the reason behind the soot reduction. However, the soot emissions improvement has to be weighted with the increase in pumping losses (4 %) and CO emissions (in this case UHC emissions remained constant over the entire parameter variation).

The trade-off in the emissions related to the magnitude of the fuel homogenization and the decrease in volumetric efficiency resulting by the throttling effect of the swirl flaps has been also indentified in [11] and [65]. The use of the swirl flaps can effectively increase the mixture homogenization at higher loads but a compromise between premixed level and efficiency losses must be found. Moreover, due to the reduced volumetric efficiency, the use of the swirl flaps may reduce the application range of low-temperature premixed combustion processes. Therefore, even if there exists the possibility of fine tuning of the combustion process, in this work the integration of the swirl flaps in a closed-loop control system is not suggested.

5.2.7 Rate of exhaust gas recirculation

Because of its large influence on the combustion, which mechanisms are explained in Chapter 3, the EGR rate is considered to be, together with the injection timing, the most important engine parameter in low-temperature premixed processes. Especially at low load, i.e. in $HCLI$ mode, the EGR rate can be varied in a very wide range. In Figure 5.11, at constant engine speed, injection timing and quantity, the EGR has been increased from 0 to 50 %. The intake temperature has been kept constant cooling the charge (EGR is mixed with the fresh air upstream the intercooler) at 313 K. The results show a significant effect of the EGR on the entire combustion process. Differently to the injection beginning, the effects of EGR begin already with the reduction of compression-end pressure and temperature, due to the increased heat capacity of the charge, and ends with lower exhaust temperatures. During the combustion process the reduction of the flame temperature and the mixture reactivity results in a longer ignition delay, lower rates of heat release and a marked combustion phasing. A detailed analysis of these processes on a ICE is not practicable and it is not the aim of this investigation. However, it is possible to recognize the most relevant mechanisms.

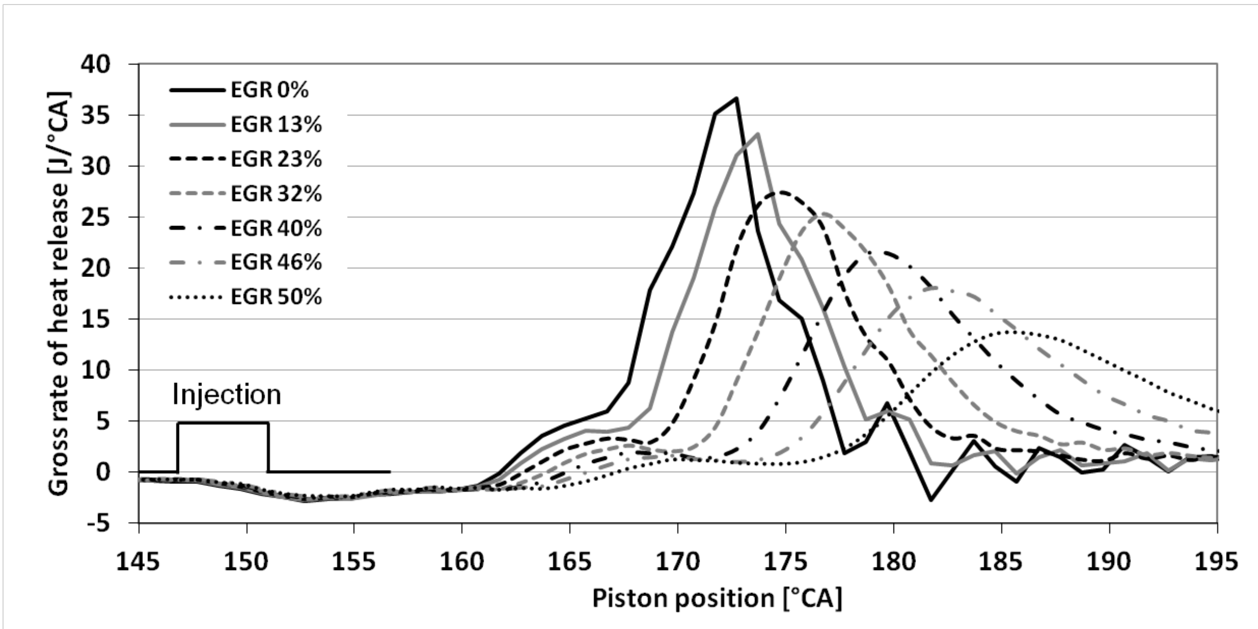

Figure 5.11: Gross rate of heat release by an EGR variation at constant injection beginning at $33\,°CA$ before TDC and constant injection quantity in $HCLI$ mode.

The injection event is clearly separated from the combustion and there is long time for the formation of an homogenous mixture. After the injection event the rate of heat release drops due to the evaporation enthalpy of the fuel. The ignition delay of the low temperature reaction, which starts at ca. $160\,°CA$, seems to remain relatively constant, as demonstrated in [72]. The small differences can be attributed to the change in compression-end in-cylinder conditions. A significant reduction of the mixture reactivity can be observed both for the low-temperature and the high-temperature reaction. The combustion delay between these two phases is stretched at higher EGR rates showing a lower reaction rate of the low-temperature combustion. Due to the reduced oxygen concentration, the reactive radicals needs longer time to find a partner and the rate of reaction is reduced. This leads to a reduced temperature increase, additionally suppressed by the larger heat storing capacity of the exhaust gases, and the chemical equilibrium is shifted in the production of metastable intermediates, which are less reactive. These two mechanisms, i.e. the reduction of the flame temperature and the mixture reactivity, take place simultaneously and their effect is prolonged over the entire combustion. Similar results are illustrated and discussed in [36] [10] [64].

In order to investigate the effect of the exhaust gas recirculation under more realistic engine operating conditions, measurements at constant maximum rate of pressure rise and constant *imep* has been carried out. In consideration of the results illustrated in the previous sections, a constant maximum rate of pressure rise has been provided by means of injection timing variation. The *imep* has been kept constant adjusting the injection duration. Intake charge temperature was kept constant at $313\,K$ but the pressure varied slightly, as well as the charge quality and its quantity, depending on the rate of EGR.

In Figure 5.12 the results for a representative $HCLI$ load point at $1800\,rpm$ and $4\,bar$ *imep* are plotted over the combustion beginning. On the right side of the figure the charge is characterized by an air/fuel ratio of 2.3 and a low rate of EGR of about $27\,\%$. The injection event takes place at ca. $2\,°CA$ before TDC. Starting from this point, the EGR rate is systematically increased and the charge subsequently diluted. As shown in Figure 5.11, this results in a longer ignition delay and a lower mixture reactivity, thus a later combustion beginning and a lower peak of pressure rise. In order to keep this last parameter constant, the combustion beginning must be anticipated injecting the fuel earlier. Approaching the TDC, in fact, the higher pressures and temperatures resulting from the piston motion can be exploited for increasing the mixture reactivity and overcome the dilution effect, which results both form the higher EGR rate and from the increase in ignition delay. This can be done until the combustion beginning has reached the TDC, since after that point in-cylinder temperature and pressure decrease again. Shifting the combustion beginning in the compression, the EGR can not be further increased but on the countrary, in order to maintain the same level of pressure rise, it must be reduced. It can be seen that the investigated range with combustion beginning in the compression phase is significantly shorter if compared with the range with a combustion beginning in the expansion. This is because the very left point in the diagram corresponds to an injection timing of $30\,°CA$ before TDC, which is considered to be the limit for piston bowl-in combustion.

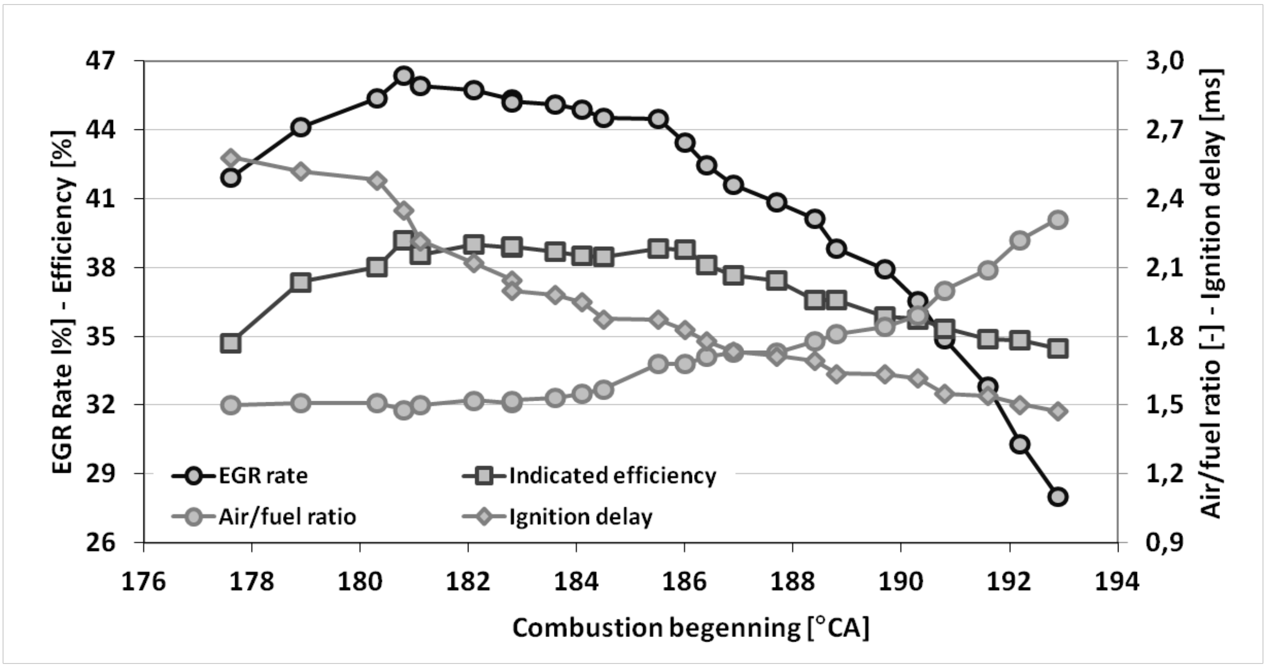

Figure 5.12: Indicated efficiency, charge quality and ignition delay by an EGR variation at constant maximum rate of pressure rise $6\,bar/°CA$ and constant $imep$ $4\,bar$ in $HCLI$ mode.

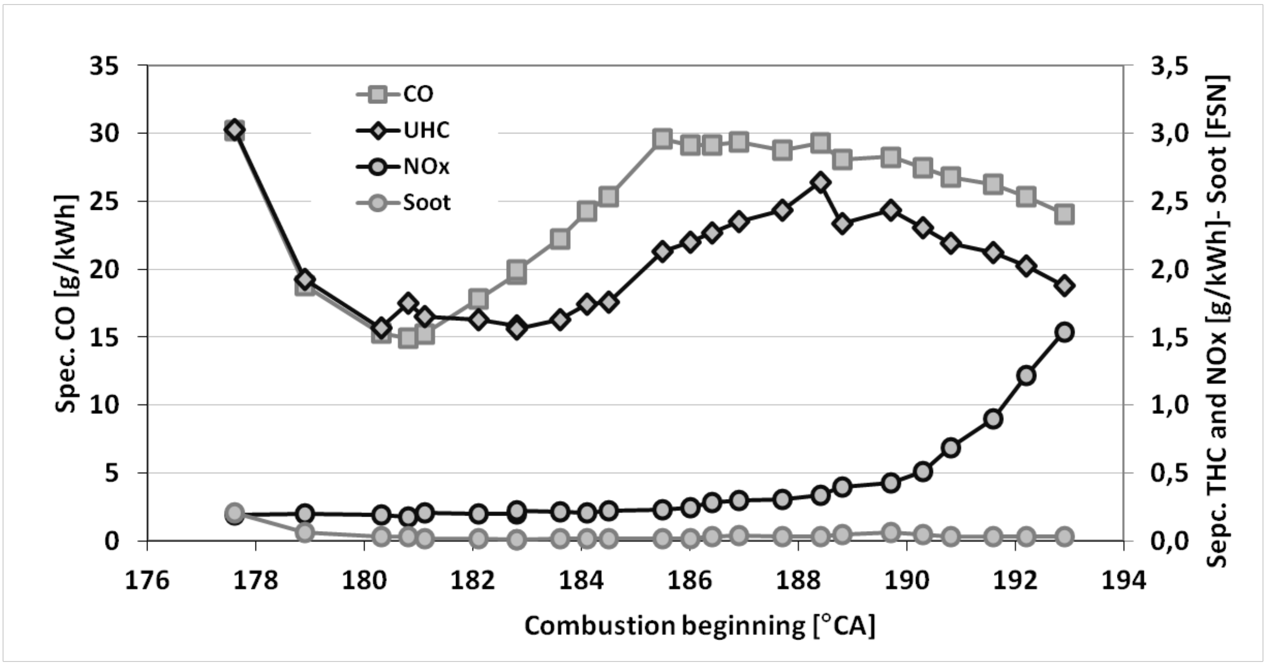

Figure 5.13: Specific NOx, UHC and CO emissions and smoke number by an EGR variation at constant maximum rate of pressure rise $6\,bar/°CA$ and constant $imep$ $4\,bar$ in $HCLI$ mode.

The increased ignition delay induces to suppose that the mixture becomes more and more homogenous approaching the TDC. As demonstrated in the previous sections, this leads to an increase in UHC and CO emissions in favor of lower NOx. However, since the maximum rate of pressure rise is kept constant, the mixture reactivity is not significantly deteriorated. In fact, the combustion duration (not showed in the diagram) reaches its minimum for combustion beginning at TDC. This explains the plateau in the efficiency located between 180 and 186 $°CA$, where constant volume combustion is approached. The engine-out emissions, shown in Figure 5.13, helps in the interpretation of the results. As expected, increasing the charge dilution and the ignition delay the NOx formation is significantly reduced to a minimum of $0.2\,g/kWh$. Because of the long mixing time the soot formation is suppressed over the entire EGR variation, excepted for the very left point where fuel is injected on the piston crown is supposed. Looking at the UHC and CO emissions trends two phases can be recognized. In the range between 178 and 186 $°CA$ there is direct effect of the piston motion and the emissions reach a minimum for combustion beginning at TDC, where pressure and temperature peak. The mixture reactivity reaches here its maximum and the combustion is very short. This effect seems to overlay the over-mixing problem. In the second phase, i.e. for combustion beginning after 186 $°CA$, the combustion is slower and it suffers more from the volume expansion. An interruption of the combustion process resulting from charge cooling is supposed to be reason behind the higher UHC and CO emissions.

Figure 5.14 shows the engine performances by a variation of the EGR rate at constant maximum rate of pressure rise of $6\,bar/°CA$ for a higher load point, i.e. $1800\,rpm$ and $8\,bar\ imep$. The injection timing rage goes from 182 $°CA$ for the very right point to 170 $°CA$ for the very left one and it is typical for $HPLI$ combustion. Phasing the combustion earlier in the expansion stroke, thus moving from the right to the left in the diagram, the indicated efficiency increases continuously. This is done increasing the rate of EGR. However, due to the high amount of injected fuel, the air/fuel ratio set a limit to the charge dilution. This means, that at higher loads charge dilution for a defined maximum rate of pressure rise is not limited by the combustion beginning at TDC (as it was for the $HCLI$ mode) but by the global air/fuel ratio of the mixture.

Looking at the measured engine-out emissions this limitation becomes more relevant. As reported in [55], the CO emissions are strictly related to the

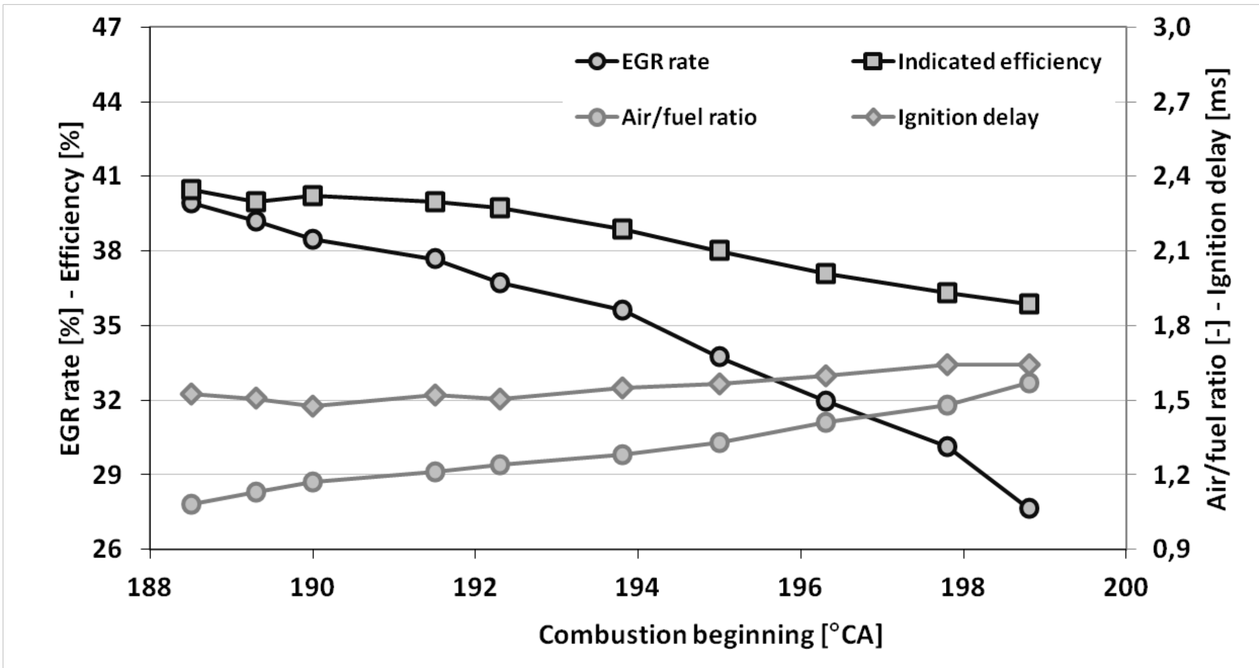

Figure 5.14: Indicated efficiency, charge quality and ignition delay by an EGR variation at constant maximum rate of pressure rise $6\,bar/°CA$ and constant $imep$ $8\,bar$ in $HPLI$ mode.

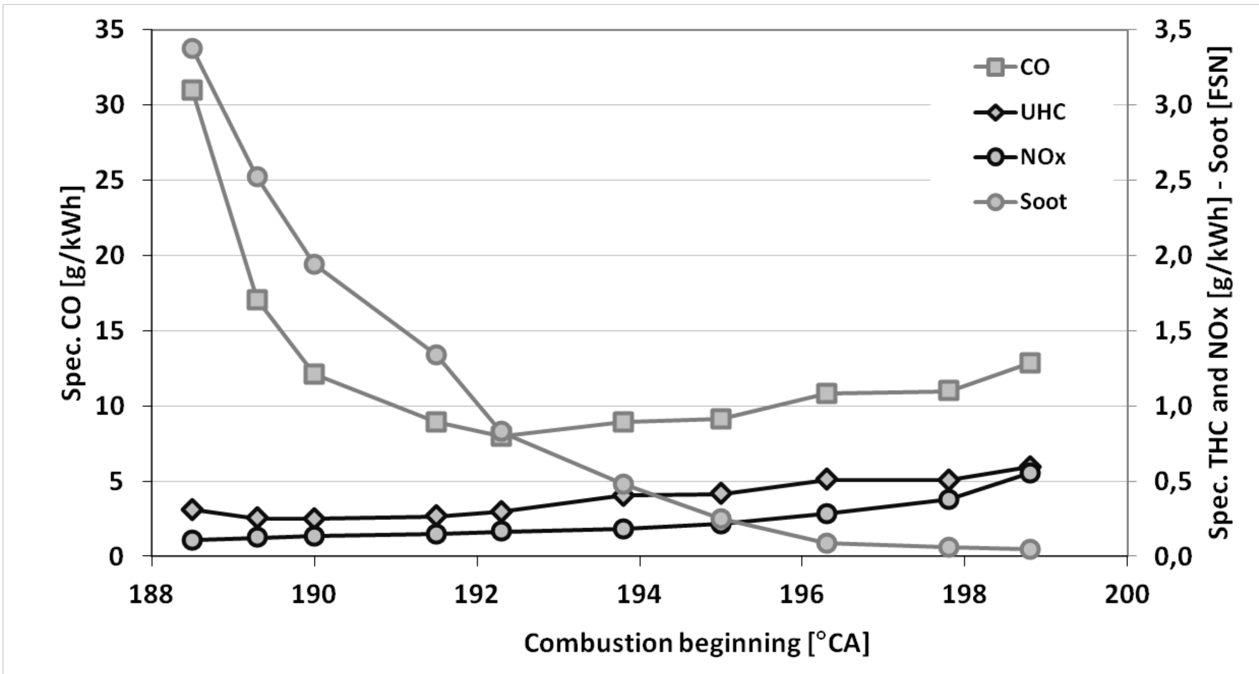

Figure 5.15: Specific NOx, UHC and CO emissions and smoke number by an EGR variation at constant maximum rate of pressure rise $6\,bar/°CA$ and constant $imep$ $8\,bar$ in $HPLI$ mode.

global equivalence ratio of the mixture and, as soon as it falls below 1.2 (at ca. 192 $°CA$), CO emissions increase rapidly together with the PM. This limit is known as the soot-limit and it characterizes the maximum charge dilution. Since the ignition delay does not signifacantly vary, specific NOx and UHC emissions remains relative constant. The UHC can be slightly reduced anticipating the combustion thanks to an increased mixture reactivity. Also in this case, at the highest EGR rate NOx emissions are limited below $0.2\,g/kWh$.

Similar EGR rate variations at constant maximum rate of pressure rise has been carried out at three different load points both for $HCLI$ and $HPLI$ combustion mode. The results confirm the tendencies illustrated above. Generally it can be said, that an increase of the charge dilution by means of EGR has positive effects on both the engine efficiency and the emissions, since the combustion beginning can be phased closer to the TDC, where the conditions for a fast combustion process are given. Moreover, charge dilution promotes long mixture formation and low combustion temperatures. Two limits has been identified. In the low load range, starting from a combustion beginning located in the expansion, the combustion can be anticipated until the TDC is reached. After this point the strictly monotonic correlation between injection timing and maximum rate of pressure rise is no more available. Increasing the load, thus the fuel quantity, the global air/fuel ratio of the mixture decreases. In this case, in order to avoid high CO and PM emissions, the EGR rate must be limited as soon the soot-limit of λ 1.2 is reached.

5.3 Combined parameter variation

In the previous section, the injection timing and the rate of EGR have been identified as the most relevant parameters for controlling the combustion process. The injection timing has a direct correlation to the combustion beginning and following to the maximum rate of pressure rise. This correlation can be exploited in combination with high rates of EGR for increasing combustion efficiency and reducing engine-out emissions by phasing the combustion beginning early in the expansion stroke. However, the variation of one parameter at a time by keeping the others constant has a limited range of investigation.

In order to identify overall tendencies, the injection timing and the EGR rate has been varied in a 50 measurement points matrix at constant engine speed of 1800 rpm and constant load of 4 bar $imep$. Measurement of similar contour maps at different engine loads are carried out in [67]. In this case the injection quantity constant is kept constant and not the $imep$.

The results of this investigation, illustrated in Figure 5.16, confirm the tendencies found in the previous investigations. There is a similarity between the NOx emissions and the maximum rate of pressure rise which is opposite to those of UHC and CO. Combining some of the contour maps shown in Figure 5.16 in a single diagram it is possible to identify the mayor mechanisms of the low-temperature premixed combustion investigated in the previous section. This is illustrated in Figure 5.17. The actuating variables are on the diagram axis. The target variables maximum rate of pressure rise, combustion beginning and λ are plotted as contour lines whereas the indicated efficiency as contour surfaces. The correlations between actuating and target variables as well as the defined limit for the combustion beginning at TDC can be easily identified. Since the presented measurement has been carried out at low-load, where larger parameters variations are possible, charge dilution is not critical ($\lambda > 1.2$). However, it can be seen how the air/fuel ratio represent a sort of horizontal limit in the map. The combination of this limit with the area of maximum combustion efficiency defines the target area for the application of the combustion process. This can be achieved by means of EGR rate variation, which would result in a vertical shift, or injection timing variation, which would result in a horizontal shift. This sort of representation is very useful in defining the combustion control strategy. In fact, the correlations illustrated here can be extrapolated in the whole application range of the low-temperature premixed combustion.

5.4 Definition of a new closed-loop control strategy

In Figure 5.1, shown a the beginning of this chapter, fuel and air path are clearly separated. Near the differences shown till now, these systems are also characterized by a very different response behaviour, which consideration is fundamental

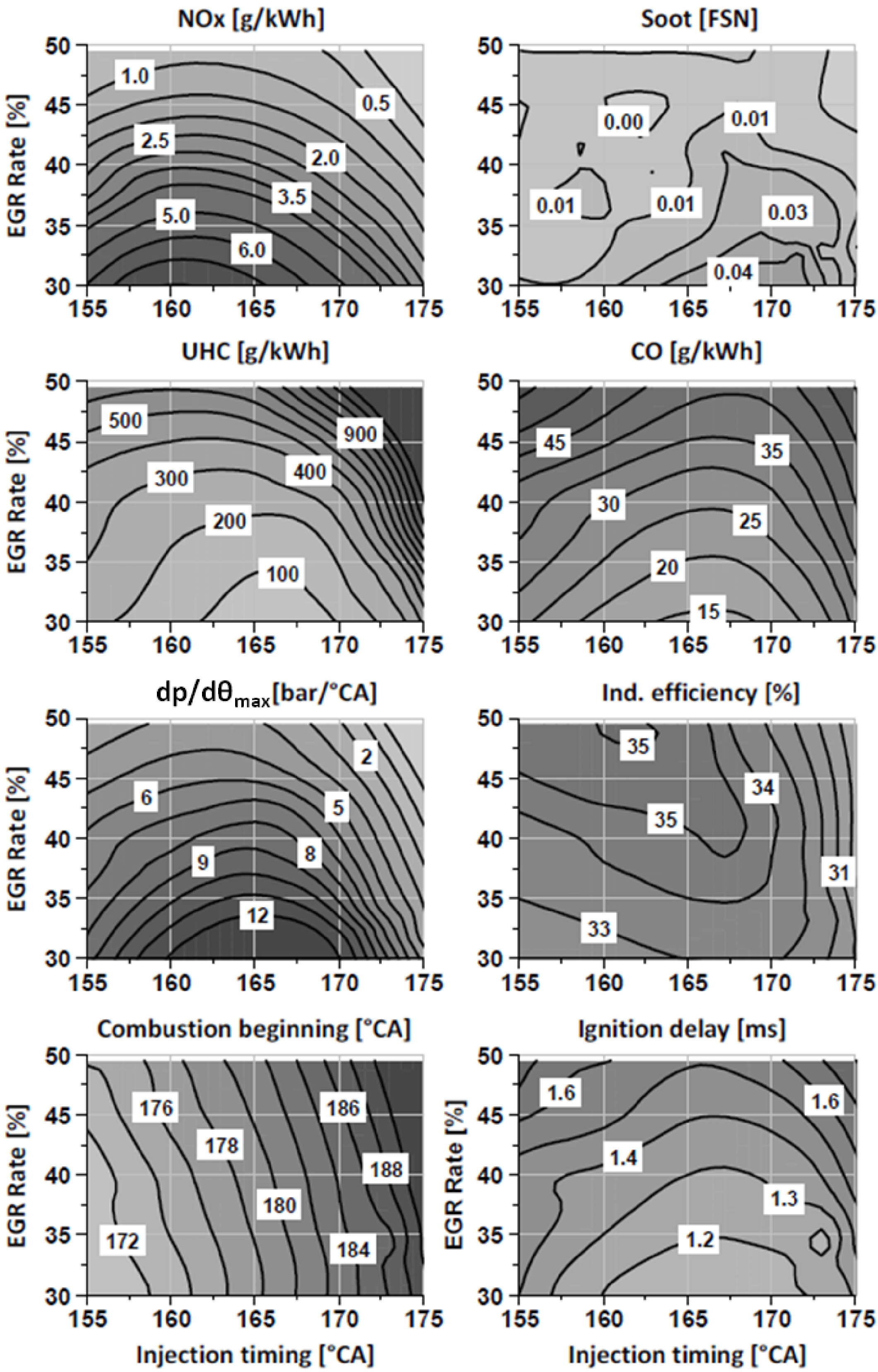

Figure 5.16: Contour maps of engine-out emissions and performances on the injection timing-EGR rate plane at constant engine speed of $1800\,rpm$ and constant load of $4\,bar\ imep$.

in the definition of the control strategy. In fact, only the fuel path (injection system) is capable of very rapid changes. If the injection pressure can be varied very rapidly, injection duration, injection timing and also the number of injection events can be changed by every new combustion cycle. Moreover, these last three parameters can be adjusted individually for every single cylinder. This makes the fuel path a very effective device for controlling the combustion process. On the contrary, parameters of the air path have only a global effect on all the cylinders. Additionally, because of the large volume of the intake and exhaust pipes, it usually takes several engines revolutions until a change in the combustion process is measurable and the desired target value is reached.

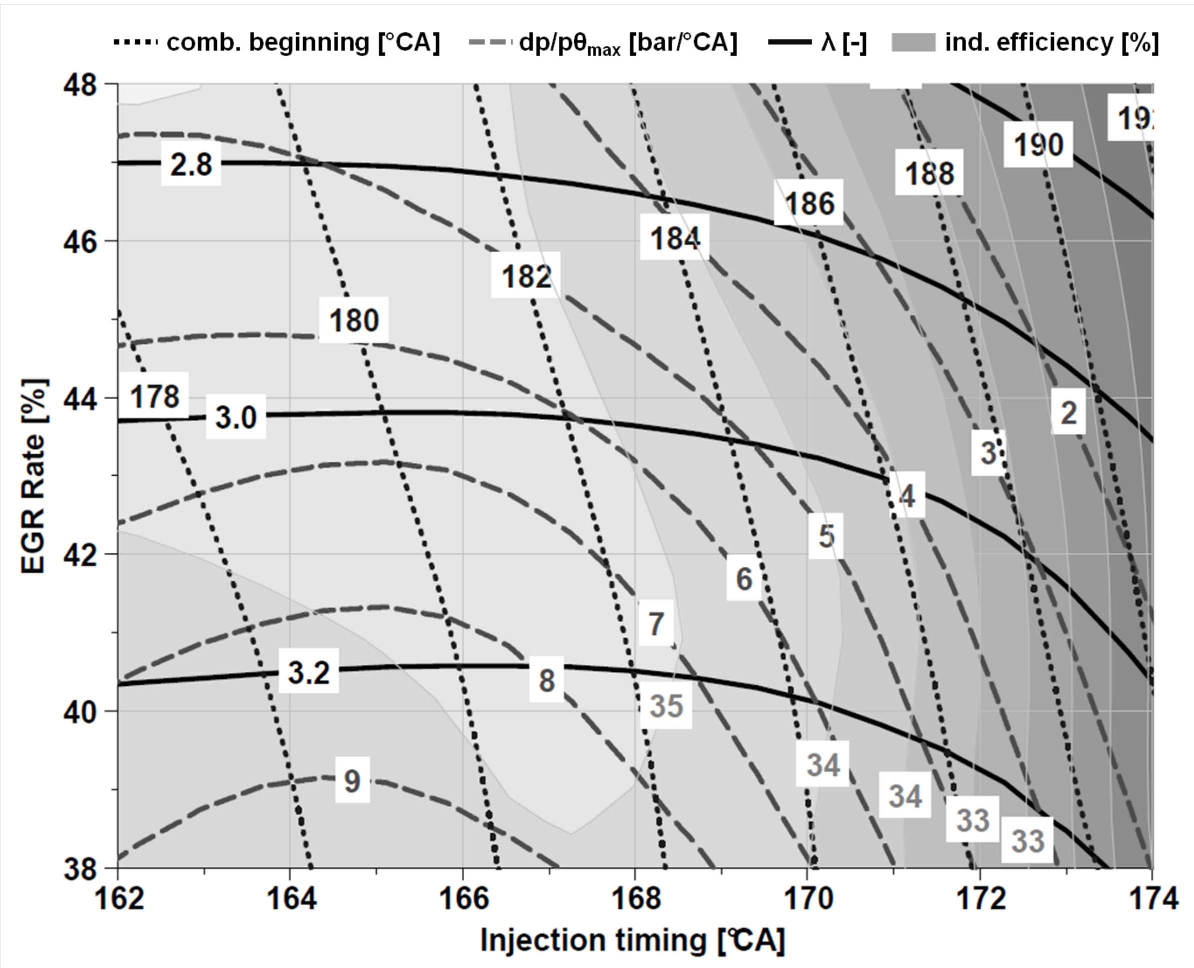

Figure 5.17: Combination of contour maps on the injection timing-EGR rate plane.

A further important differentiation can be done between measureable and non-measureable target variables. Using in-cylinder pressure sensors the variables of the first two target groups can be calculated immediately by means of real-time pressure traces analysis. On the other hand, the determination of the variables of the last target group, i.e. emissions and efficiency, is usually done using complex empirical correlations which calculation effort is very time costing for the *ECU* and thus not suitable for real-time calculation.

The goals set by the definition of a new control system for the low-temperature premixed combustion are

- ensure a long time for fuel homogenization
- but still providing a direct control of the combustion beginning,
- high combustion efficiency
- and no misfire,
- NOx and smoke-less combustion
- by maintaining low UHC-CO emissions,
- low engine noise (low rates of pressure rise)
- and good transient behavior.

The system sensitivity analysis proposed in the first part of this chapter has been useful for identifying mayor constrains ans system limitations, and for highlighting the most relevant mechanisms of the low-temperature premixed combustion. Deriving from these information a combustion control strategy can be finally defined. This is schematic illustrated in Figure 5.18.

5.4.1 Engine load

In commercial *ECU*s the engine load is usually characterized as a "desired" torque which is derived from the actuation of the gas pedal. In the standard vehicle equipment there are no sensors capable of measuring the effective torque delivered by the engine. Introducing in-cylinder pressure sensors however, the engine load can be directly measured in form of indicated piston work, i.e. *imep*. With good approximation, the *imep* is directly correlated to the amount of fuel injected per engine stroke, which results from the injection duration and

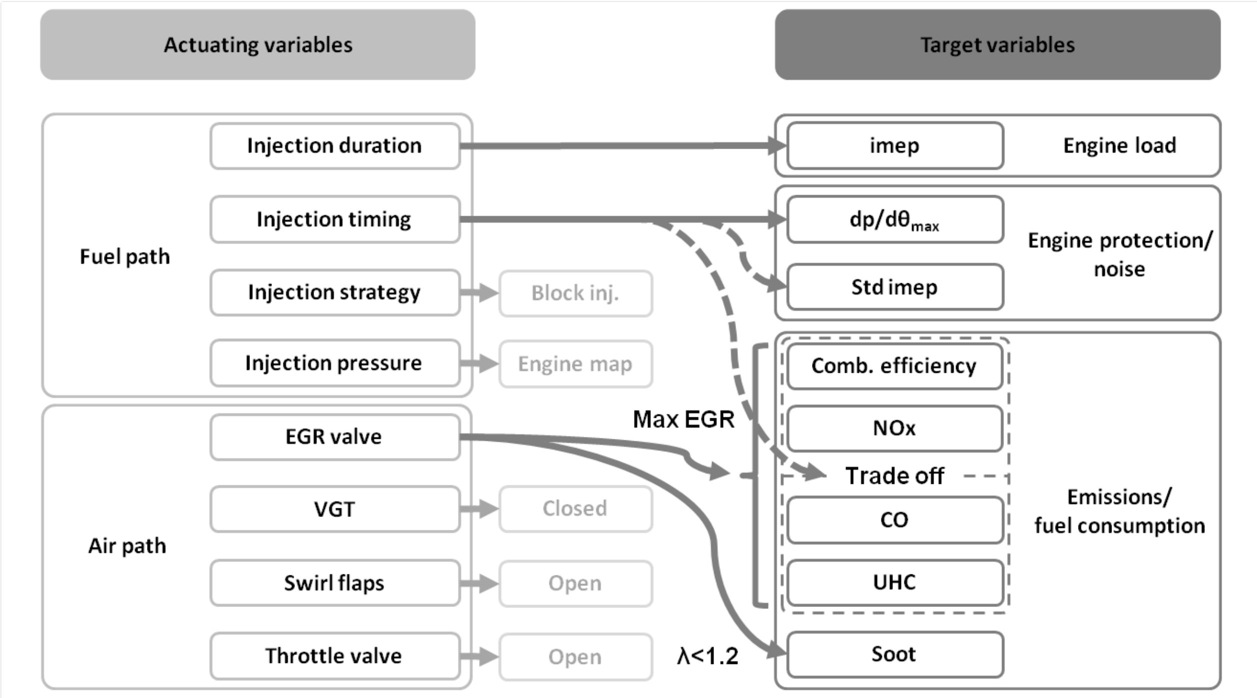

Figure 5.18: Proposed strategy for the closed-loop control of the low-temperture premixed combustion.

the injection pressure. Therefore, a closed-loop control of the indicated engine torque can be easily realized over the injection duration.

The ability of controlling the eingine load in closed-loop structure offers many andvantages. One of these is the possibility of balancing the cylinder torque output with a significant reduction of the engine vibrations. Moreover, this solution provides a continuous compensation of engine aging processes such as drifts in the injector hydraulic flow or increased blowby, for an istance.

5.4.2 Engine protection and noise

As resulted from the sensitivity analysis performed during this investigation there is a direct correlation between the injection beginning and the maximum rate of pressure rise. Even if this correlation is not linear, a closed-loop control between this two parameters can be realized. In fact, as long as the combustion beginning takes place in the expansion stroke, the correlation shows a strictly monotonic decreasing character. This means that shifting the injection timing back from the TDC (later in the expansion) the maximum rate of pressure rise decreases and inversely.

This solution offers many advantages. The first, and most direct one, is a constant monitoring of the combustion reactivity and the avoidance of high rates of pressure rise which would increase the engine noise and could damage the engine. The second benefit is the possibility of assuring good combustion stability, especially under transient engine operation avoiding excessive combustion phasing in the expansion. In fact, differently to the center of combustion, which is largely used in other closed-loop control systems, the maximum rate of pressure rise is directly related to the reactivity characteristics of the mixture. Therefore setting a target $dp/d\theta_{max}$ helps in providing a stable and balanced combustion for every cylinder.

5.4.3 Emissions and fuel consumption

Measurements performed during this investigation have pointed out a target area in the proximity of the TDC in which both low emissions and highest combustion efficiency can be obtained. Even if a direct measurment of these target parameters is not possible, because of the short calculation time of a real-time system, this target area can be reached setting the right oxigen concentration of the charge, which is monitored using the labda meter placed in the intake manifold. This can be done controlling the EGR valve position in closed-loop structure.

If the desired $dp/d\theta_{max}$ value is controlled by means of injection timing variation, for a combustion beginnng placed in the expansion the target area can be reached increasing the EGR rate, shifting so the combustion beginning towards the TDC. Once the combustion beginning has reached the TDC there is no need of further charge dilution. This would result in an unnecessary reduction of the maximum rate of pressure rise. At this point, an anticipation of the injection timing must be limited in order to avoid a shift of the combustion beginning in the compression stroke, where the correlation between injection timing and maximum rate of pressure rise is inverted.

If the charge is over-diluted and the desired $dp/d\theta_{max}$ can not be achieved (this situation cuold results for an instance from a fast load reduction), the controller must reduce the EGR rate, closing the valve, as long as the target oxygen concentration and the desired maximum rate of pressure rise can be provided. Also in this case, a solution for limiting the combustion beginning at TDC must be provided.

At part load, soot formation can be avoided monitoring the λ value of the exhaust gases. Increasing the load, the EGR can be increased until the soot-limit of λ 1.2 is reached. In this case the combustion beginning can not be phased at TDC but, for the given configuration of target maximum rate of pressure rise and maximum EGR rate for λ at soot-limit, the best possible combustion efficiency is provided.

Virtual emission sensor

The choice of the desired maximum rate of pressure rise mostly depends on the desired engine noise level and the allowed engine-out emissions concentrations. As shown in Figure 5.17, the combustion efficiency seems to be less dependent from this parameter, when the combustion beginning is kept constant. The emissions, on the contrary, are directly related to it. As shown in Figure 5.19 a clear trade-off in the emissions is obtained setting different target maximum rates of pressure rise. The reason behind this has been discussed many times in the system sensitivity analysis. The in-cylinder conditions which lead to high rates of pressure rise, i.e. near-stoichiometric mixture (short ignition delay), in which the oxidation reactions have high speed and the flame temperature is high, also promote the formation of NOx emissions. Inversely, high EGR rates combined with long ignition delay reduces the mixture reactivity and the flame temperature resulting in combustion deterioration and higher UHC and CO emissions. Therefore, the maximum rate of pressure rise can be exploited as a virtual emissions sensor. Implementing a weighting function in the application of the combustion process an optimum position in the emission/noise trade-off can be calculated in order to set the target $dp/d\theta_{max}$ value.

The proposed closed-loop combustion control strategy can be summarized as follows. Starting from arbitrary in-cylinder conditions the maximum rate of pressure rise is rapidly set, by means of injection timing variation, under the

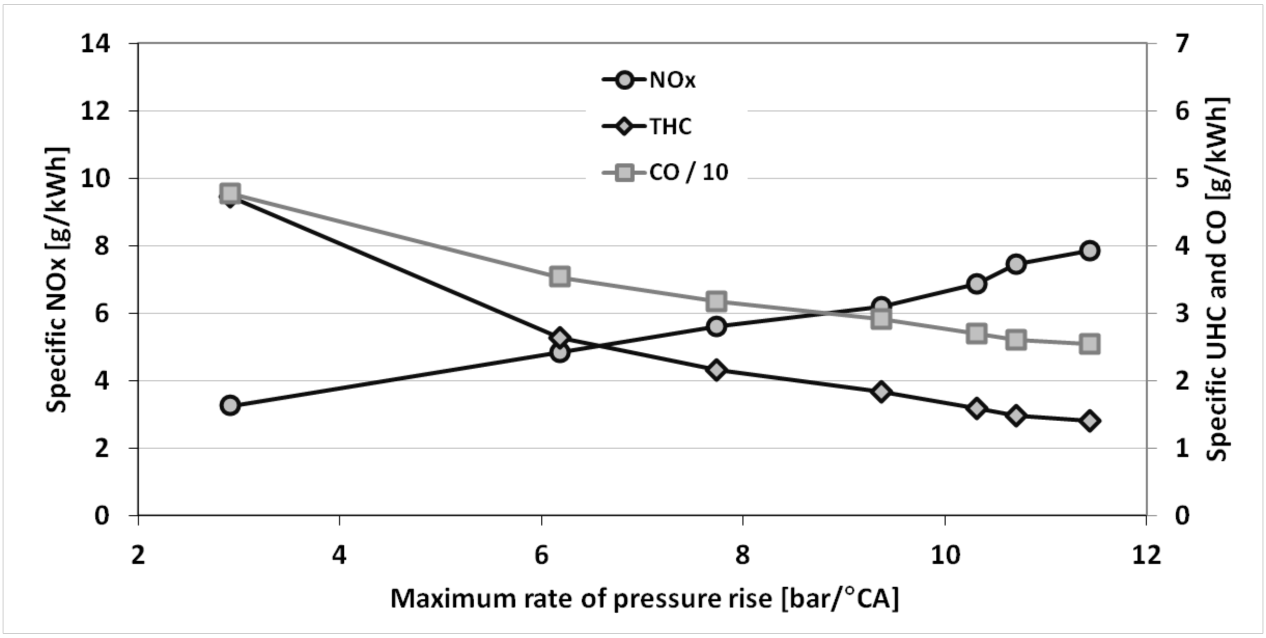

Figure 5.19: Variation of the target maximum rates of pressure rise at constant engine load conditions, 1800 *rpm* and 4 *bar imep*.

critical level and to a given target value. The quality of the charge rate is than varied using the *EGR* valve. The goal is to phase the combustion beginning towards the *TDC*. This is done under consideration of the *lambda* limit of 1.2.

Choosing the injection duration, the injection timing and the *EGR* valve as main actuating parameters it is possible to uncouple the functionalities of the injection path from those of the air path. Since the engine load is controlled using the injection duration and the maximum rate of pressure rise using the injection timing, the properties of the charge can be set independetly controlling the *EGR* rate. Moreover, using this strategy a clear target range for the combustion can be defined, where high combustion stability and low fuel consumption are provided.

5.4.4 Implementation of the closed-loop control strategy in the ECU software architecture

Detailed information about the implementation of the proposed closed-loop control strategy in the *ECU* software architecture can be found in [1]. For reasons of completeness a brief description is given here.

The control strategy of the low-temperature premixed combustion is implemented using three independent $SISO$ closed-loop circuits. The desired $imep$ is set controlling the fuel injection duration. Targeted maximum rates of pressure rise are achieved by means of injection timing control and the EGR valve has been chosen as actuating variable for setting the desired oxygen concentration in charge, which is measured using an additional lambda sensor in the intake manifold. The circuits are based on simple proportional-integral-derivative (PID) controllers supported by feed-forward maps for the actuating variables. These maps have been fitted on the base of stationary measurements. For the load and engine protection controllers it is made use of a single PID parameter set for the entire range of engine operation, whereas for the charge quality controller, because of the strong dipendecy of the air path response from the engine torque and speed, maps are used for the P and I factors.

Investigations in transient engine operation mode have shown the need of improving the robustness of the control strategy for avoiding a deterioration of the combustion process. This may results from a strong deviation of the charge quality as a consequence of the low system response of the air path. Two effective solution has been realized.

Inverted PID controller

In transient engine operation fluctuations of the charge quality occur. By insufficient charge dilution the engine protection controller phases the combustion later in the expansion maintaining constant maximum rates of pressure rise. In the other case, i.e. by large rates of EGR, the mixture reactivity is reduced and the controller reacts anticipating the injection timing. As shown in Figure 5.17, this strategy works until the combustion beginning has reached TDC. In order to avoid the combustion beginning to be phased in the compression stroke, the desired maximum rate of pressure rise must be reduced. This has been realized using the actual values of the PID controller factors in an inverted PID controller structure. This structure is capable of calculating a new, lower, desired $dp/d\theta_{max}$ value. The reduction of the $dp/d\theta_{max}$ target value is proportional to the needed injection timing correction for shifting the combustion beginning at or after TDC. In this way a dynamic injection timing limitation is obtained without disturbing the controller functionality.

Injection timing feed-forward dynamic adaptation over the oxygen concentration

An additional side-effect related to the poor transient response of the air path is the occurrence of undesired oscillations in the $dp/d\theta_{max}$ signal. These could be reduced increasing the PID factors value of the engine protection controller, which however would also result in an increased system instability. A solution has been found comparing the $dp/d\theta_{max}$ signal with the measured oxygen concentration of the charge. Not the actual oxygen concentration value but the difference to the desired one is in fact proportional to the increase in the mixture reactivity and thus to the maximum rate of pressure rise. Exploiting this mechanism, a dynamic correction of the feed-forward value of the injection timing has been realized. This solution provides an effective reduction of the $dp/d\theta_{max}$ signal oscillation and the avoidance of high peaks of pressure rise.

6

Combustion Mode Switch

The ability to switch the combustion mode between alternative and conventional diffusive combustion is necessary to overcome the operation limit of the low-temperature premixed combustion and represent a fundamental task of a the combustion control strategy. An effective mode switch procedure, based on targeted engine measurements, is proposed in this chapter.

The application range of the low-temperature premixed combustion is mainly limited by the trade-off between high mixture reactivity and charge dilution. In order to avoid fuel-rich mixture and the formation of soot, increasing the engine load and thus the required fuel mass the EGR rate must be reduced. The fuel/EGR mass ratio increases and the mixture becomes more reactive. In order to limit the desired maximum rate of pressure rise the combustion muss be phased later in the expansion stroke with a deterioration of the combustion efficiency. For this reason, switching to diffusive combustion mode once the soot-limit is achieved is suggested. The limit can be defined using the λ value of the exhaust gases or setting an upper load limit, which in this case is assumed at $4.5\,bar\ imep$.

The goals by the definition of the mode switch strategy are:

- short duration of the switching procedure
- maintain the engine noise unvaried
- ensure a steady monotonic increase of the engine load under dynamic engine operation

In this chapter an analysis of the problematic related to the combustion process by switching between alternative and conventional mode are discussed. Further information about the realisation of the proposed switching procedure in the software architecture as well as a detailed description of the behavior of the closed-loop control strategy during this process are given in [1].

6.1 Diffusive combustion mode

Table 6.1: Differences in the parameters application between combustion modes at switching limit.

Low-temp. premixed		**Diffusive**	
		Fuel path	
Strategy	single injection	Strategy	multiple injection
Inj. timing	15-30 $°CA\ BTDC$	Inj. timing	0-5 $°CA\ BTDC$
Inj. pressure	900-1000 bar	Inj. pressure	350-450 bar
		Air path	
EGR valve	48-56 %	EGR Valve	24-32 %
(Oxygen rate	13-16 %mass)	(Oxygen rate	20-22 %mass)
Throttle valve	5 %	Throttle valve	5 %
VGT	92 %	VGT	92 %
Swirl flaps	5 %	Swirl flaps	5 %

As explained in Chapter 4, a new software structure has been designed for the aims of this projects. In order to investigate the problematic of the combustion mode switch it is first necessary to extend the software structure in order to run the engine with multiple injections. In the diffusive mode only the *imep* closed-loop control is implemented. This is realized varying the injection duration of the main injection but keeping it constant for the two pilot injections. The injection timing and the injection pressure are open-loop controlled using engine maps. The actuating variables of the air path, i.e. the VGT and the EGR valve, are also controlled in open-loop. The use of the EGR rate closed-loop control based on the measurement of the oxygen concentration using the lambda meter in the intake manifold is not suggested for the diffusive combustion mode, since the high oxygen concentration in the exhaust gases makes the measurements inaccurate. For seek of simplicity also in this case it is not made use either of the throttle valve or the swirl flaps.

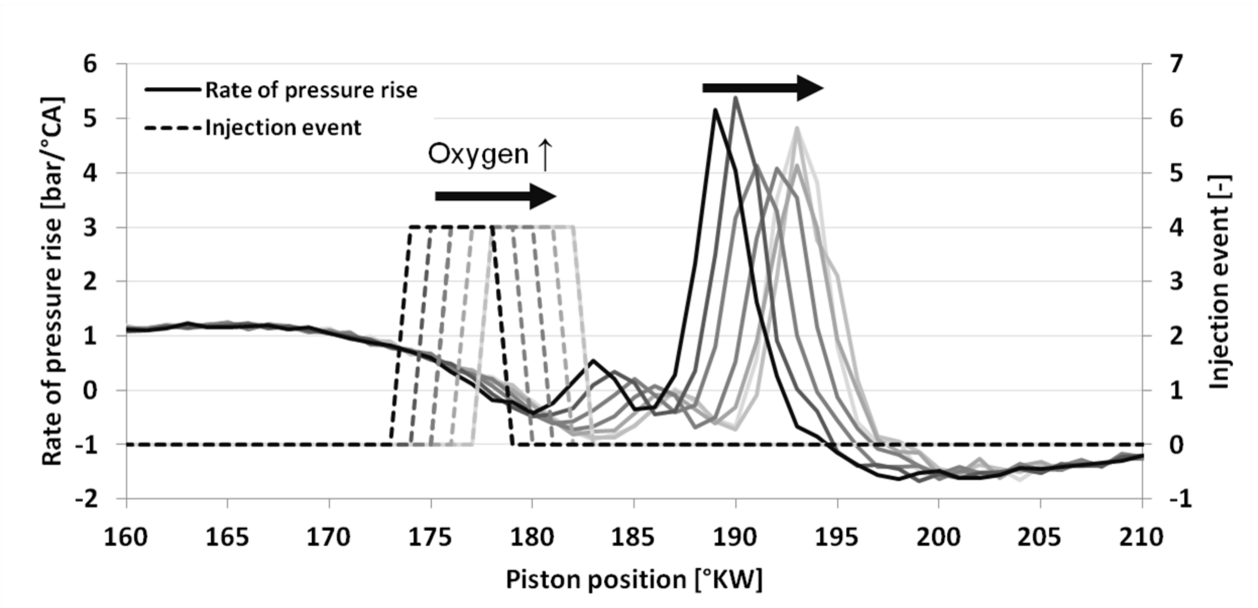

Figure 6.1: Consecutive combustion cycles by an oxygen concentration increase in low-temperature premixed combustion mode.

For ensuring diffusive combustion two pilot injections are used. In the application of the injection duration and the injection pressure, particular attention is given to the occurrence of a clear premixed combustion prior to the main injection event, in order to provide the in-cylinder conditions needed for a fast diffusive combustion and low combustion noise. A constant time dwell of 1000 ms between the first pilot and the main injection, and a time gap of 900 ms between the second and the fist pilot are applied. The center of combustion is set between 8 and 10 $°CA$ after TDC. The application range of the diffusive combustion extends from 1 to 20 $bar\ imep$ and from 800 to 2400 rpm.

The above described application of the diffusive combustion must be intended as a solution for reproducing the basic functionalities of a standard series software architecture, but it has any pretention of achieving the high complexity standard of the series production.

6.2 Principles of the combustion mode switch

As shown in Table 6.1, in the proximity of the switching limit, which is set at 4.5 *bar imep*, there is a large difference in the parameters application. the most significant are the differences in injection pressure and oxygen concentration of the fresh charge. In low-temperature premixed combustion mode it is made use of high charge dilution and high injection pressure whereas in diffusion combustion mode this parameters have significant lower values.

Investigations have shown a strong incompatibility between the two applications. In other words, it is not possible to change combustion mode by simply switching the fuel path between the two applications. The consequences are in one case an inadmissible rates of pressure rise and, in the other, an instable combustion up to misfire. This means, that a structured switching strategy which takes in consideration the complexity of the thermodynamic mechanisms is needed.

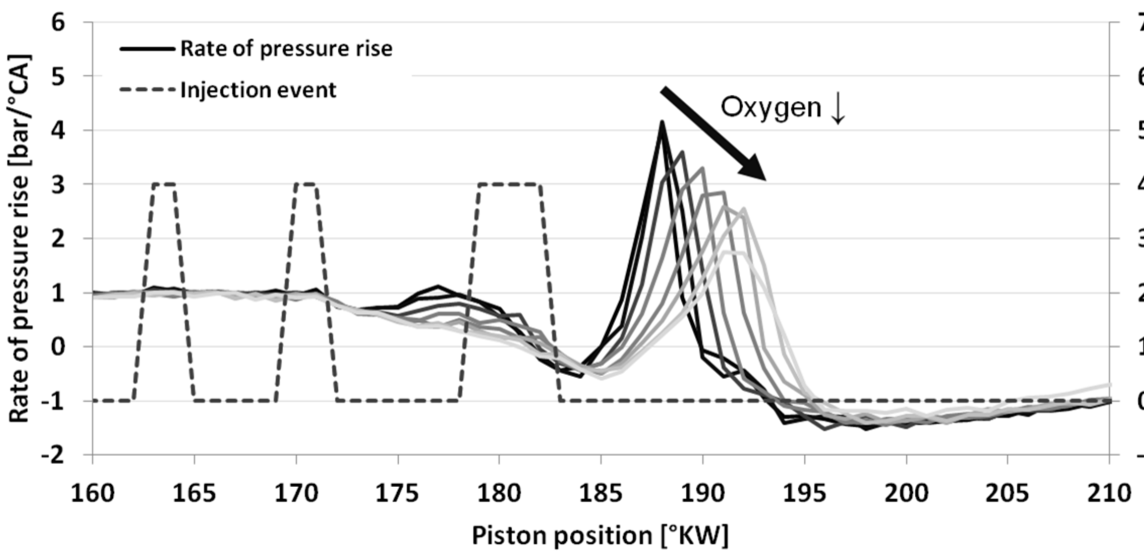

Figure 6.2: Consecutive combustion cycles by an oxygen concentration reduction in diffusive combustion mode.

As already mentioned, parameter variations of the fuel path, such as injection strategy and injection pressure, are much faster than those of the air path, e.g. the oxygen concentration in the fresh charge. Even if the EGR valve position can be changed rapidly, the EGR rate, because of the large volumes of the pipes, takes more than one second to reach its target value. This circumstance is analyzed in [1]. For this reason, it suggested to start the procedure switching the air path actuators at first. An investigation has been carried out in order to highlight the oxygen concentration tolerance of the two combustion modes. Starting from the stationary engine application the oxygen concentration has been increased in the case of the low-temperature combustion and reduced in the case of diffusive combustion by means of EGR valve opening variation.

Figure 6.1 shows some consecutive combustion cycles by a rapid oxygen concentration variation in the low-temperature combustion mode. In the rate of pressure rise traces the cool flame combustion followed by the main combustion can be recognized. The injection event is represented by the dashed lines. Closing the EGR valve the oxygen concentration increases and thus the mixture reactivity becomes faster. In order to provide a constant maximum rate of pressure rise the closed-loop control shifts the injection timing towards the TDC.

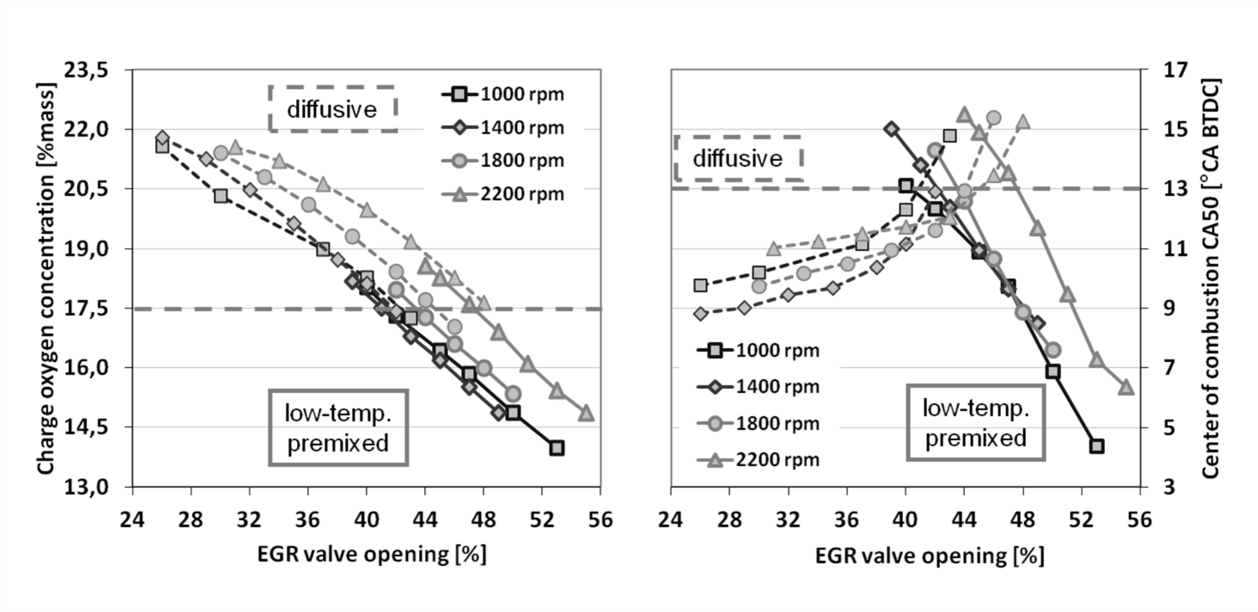

Figure 6.3: Investigation of the oxygen tolerance. Comparison between low-temperature premixed and diffusive combustion.

In diffusive combustion mode, the injection timing of the main and pilot injections is open-loop controlled and does not vary by reducing the oxygen concentration. This is illustrated in Figure 6.2. A significant reduction of the mixture reactivity can be recognized for both the pilot and the main injection. The premixed ratio becomes smaller and the ignition delay of the main combustion longer. The main combustion shows a lowered heat release and a slower fuel conversion. A further decrease of the oxygen concentration would lead to misfire.

In both cases, the combustion is phased later in the expansion until quenching occurs. In the first case as a consequence of the defined combustion control strategy, in the latter due to the reduced reactivity of the charge. Before quencing takes place, it is necessary to switch the fuel path.

In Figure 6.3 the oxygen tolerance of the two combustion modes at diffent engine loads is illustrated. Also in this case, starting from the default parameters configuration of Table 6.1, the oxygen mass rate in the fresh charge and the center of combustion have been measured varying the EGR valve opening position at $4.5\,bar\ imep$ and different engine speed. Interestingly, for all the traces there exist an overlap region between alternative and conventional combustion. The oxygen concetration of $17.5\,\%$ can be adopted as switching criterion for the fuel path. This limit is independent from the engine speed and corresponds to a center of combustion of approximatevely $13\,°CA$ after TDC.

6.3 Switching from premixed to diffusive combustion

A schematic step by step description of the proposed mode switching strategy from alternative to conventional combustion is shown in Figure 6.4. In the diagram a combustion mode switching procedure as a result of a load increase is illustrated.

Once the $imep$ value achieves $4.5\,bar$ the switching process is triggered. The EGR valve is rapidly closed to the value stored in the diffusive combustion application map. After a short delay, the oxygen concentration in the charge increases. From the rapid EGR valve actuation it results a sudden increase of the exhaust gas pressure upstream the turbine with consequences on the cylinder filling, which causes a drop in the $imep$ signal.

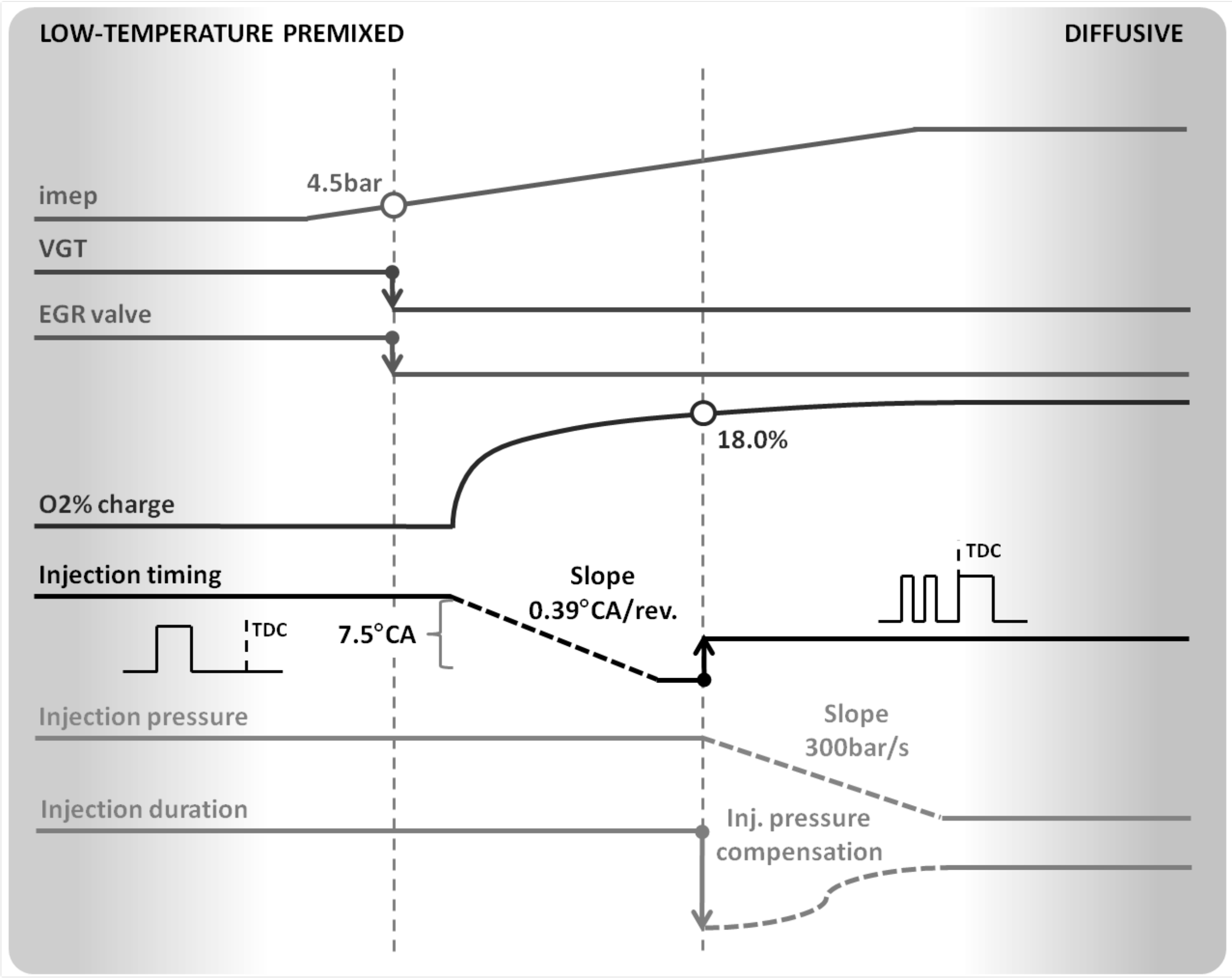

Figure 6.4: Schematic representation of the switching procedure from alternative to conventional combustion mode as a result of a load increase.

In Figure 6.8, this circumstance is illustrated. It must be considered that in the default parameters application the same VGT position is set for both alternative and conventional combustion modes. However, beacause of the different position of the EGR valve, the pressure level upstream the turbine is not equal between the two modes and once the valve is operated the pressure literally jumps from one to the other level. In order to provide a steady monotonic $imep$ signal, a compensation of this effect is needed. A pressure-neutral increase of the oxygen concentration can be realized applying a different VGT position for the diffusive mode, lowering the exhaust gas pressure. Switching the combustion mode the VGT is operated simultaneously with the EGR valve providing a constant pressure. For higher engine loads outside the mode switching region the usual VGT position can be used.

The measurements illustrated in Figure 6.5 and in the following four diagrams have been all carried out increasing the load from 3.5 to 5.5 $bar\ imep$, switching from alternative to conventional, or decreasing it from 6 to 4 $bar\ imep$, switching in the opposite direction, using a constant slope duration of 10 seconds. Data has been collected with a 100 Hz rate. The engine speed has been kept constant at 1400 rpm.

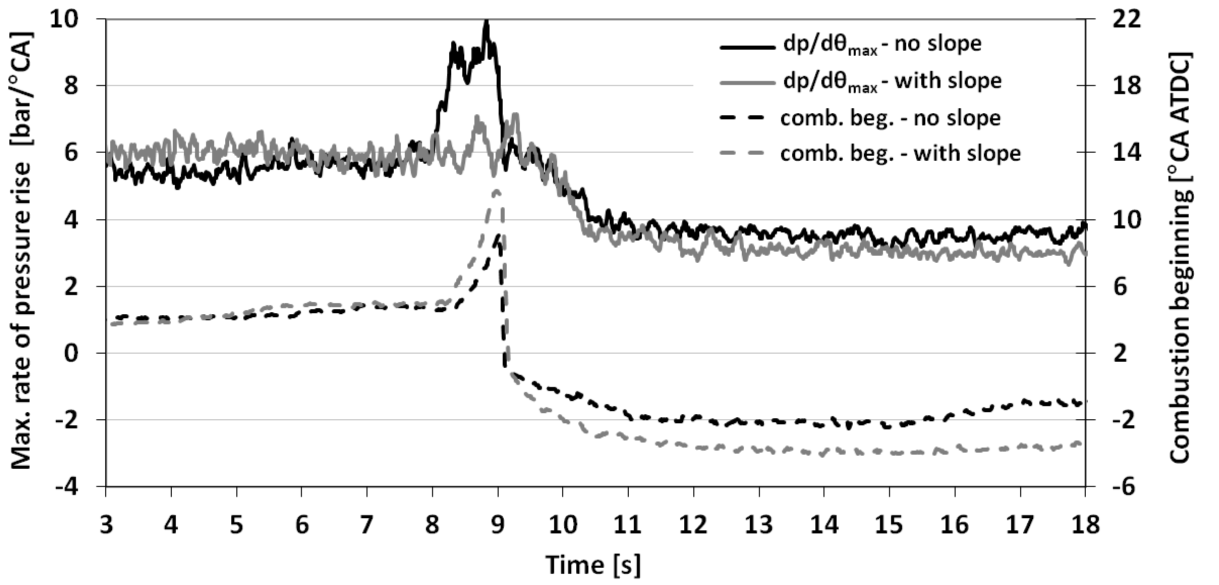

Figure 6.5: Consequences of a rapid oxygen concentration increase on the low-temperature premixed combustion and the improvement by using a compensation of the feed-forward injection timing value.

After a short delay, the oxygen concentration increases very rapidly and the mixture becomes more reactive. As shown in Figure 6.5, the $dp/d\theta_{max}$ controller is to slow for avoiding an increase of the maximum rate of pressure rise. To avoid an undesired increase of the combustion noise it is suggested to apply a correction on the feed-forward value for shifting the injection timing towards the TDC using a slope function. The beginning and the rate of the slope must be set in consideration of the air path characteristics (system response), whereas the magnitude of the correction must be limited in order to avoid combustion quenching. This solution supports the engine protection controller effectively

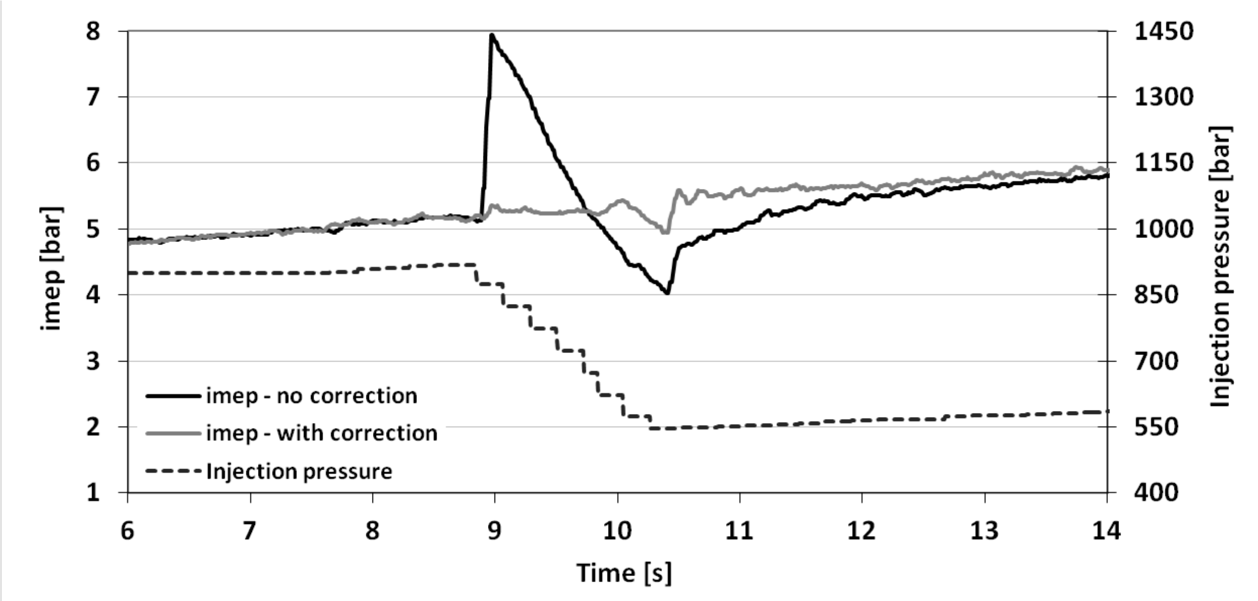

Figure 6.6: *imep* oscillations as a consequence of the injection pressure difference switching from single to multiple injection and the improvement by using a dynamic correction of the feed-forward value of the main injection duration.

in maintaining a constant maximum rate of pressure rise during the switching procedure.

The fuel path is first switched when the defined oxygen concentration is achieved. At this point, the injection timing is changed and two pilot injections are added to the combustion. This is done between two consecutive combustion cycles. Even if effects such as pressure waves in the common-rail are less influent on the single injection event, it is suggested to actuate the injection pressure only after having switched the injection strategy to multiple injection. Since the fuel supply is splitted in many injections and because of the late position of the main injection (usually at TDC or after it), the multiple injection pattern is characterized by a larger combustion stability even at higher injection pressures. At this point, the injection pressure must be lowered to the application value of the diffusive combustion. This is done by a controlled decrease limited at $300\,bar$ per second. This means, that for a typical pressure difference between the combustion modes of $450\,bar$, this procedure last about 1.5 seconds. During this time it is necessary to compensate the pressure difference reducing the duration of the main injection in order to avoid abrupt variations of the *imep* signal. Depending on the difference between the actual and the target injection pressure, the feed-forward value of the *imep* controller, i.e. the injection duration applyied in stationary mode, is corrected using a specific shaped function. In Figure 6.6 a comparison between this strategy and a combustion switch without injection pressure based dynamic feed-forward correction is shown. A linear correction of the injection duration over the injection pressure is not sufficient and only applying a S-shaped multi parameter function a steady monotonic *imep* signal is obtained. Otherwise a significant increase of the *imep* can be observed. A detailed description of this function can be found in [1].

6.4 Switching from diffusive to premixed combustion

The combustion mode switching procedure from diffusive to alterantive is similar to the one discussed in the previous section. This is illustrated in Figure 6.7.

Also in this case, when the engine load drops below $4\,bar\ imep$, the air path is switched at first. The EGR valve position is actuated to the feed-forward value of the low-temperature premixed combustion controller, which is than reactivated. The oxygen concentration drops, after a short delay, to a lower value. As previously discussed, the influence of the EGR valve on the exhaust pressure upstream the turbine must be compensated closing the VGT simultaneously. Figure 6.8 shows the advantages of a steady exhaust pressure profile achieved using this solution.

Differently to the mode switching procedure from alternative to conventional, in this case the injection pressure is switched together with the air path. As already mentioned, the diffusive combustion is more stable and thus suitable for an injection pressure variation. With a rate of $300\,bar$ per second the target pressure is reached before the injection strategy is commuted. As shown in Figure 6.9 also in this case a correction of the feed-forward value of the main injection duration, proportional to the actual pressure difference, is needed. A solution has been found designing a G-shaped multi parameter correction function, which is described in [1].

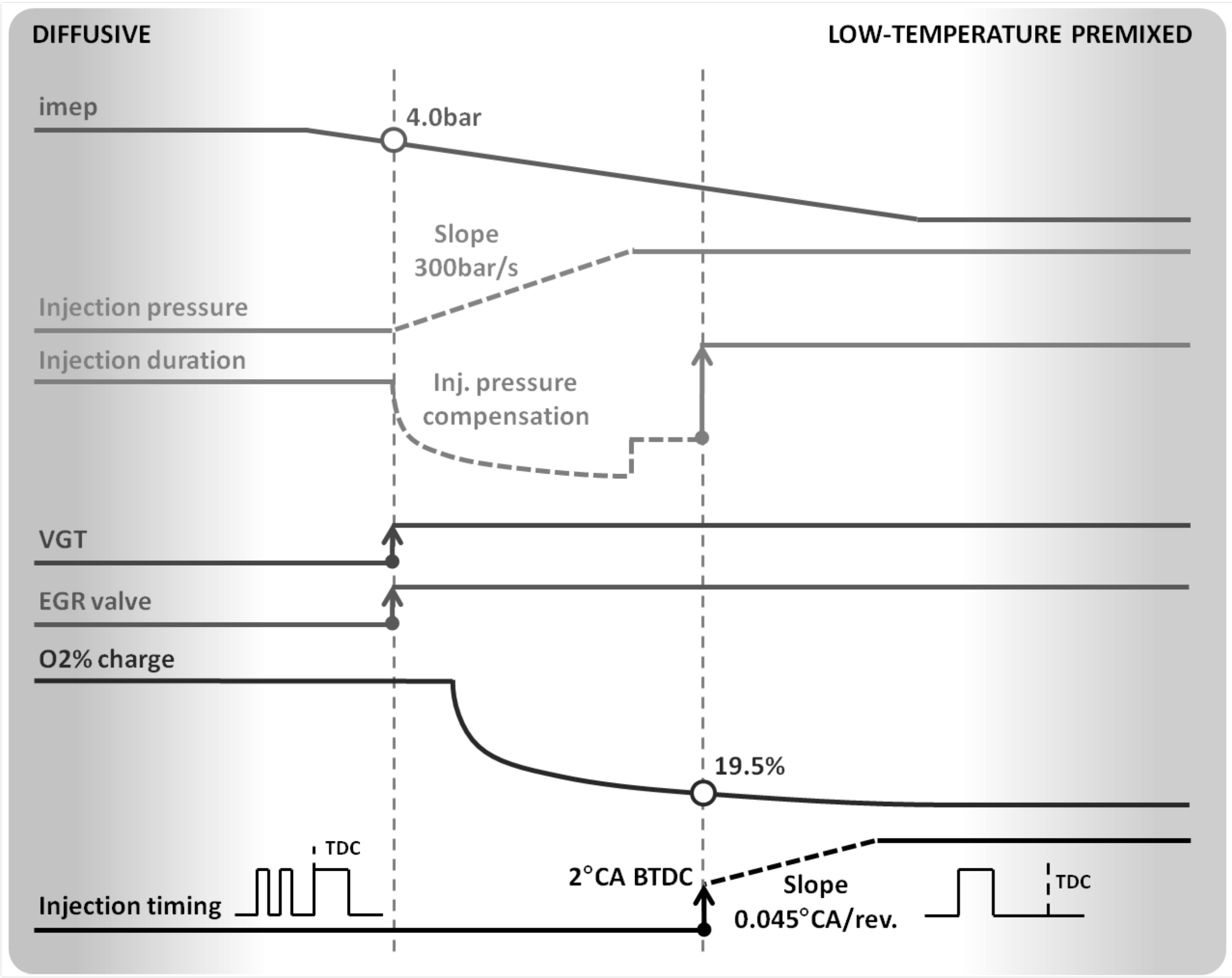

Figure 6.7: Schematic representation of the switching procedure from conventional to alternative combustion mode as a result of a load decrease.

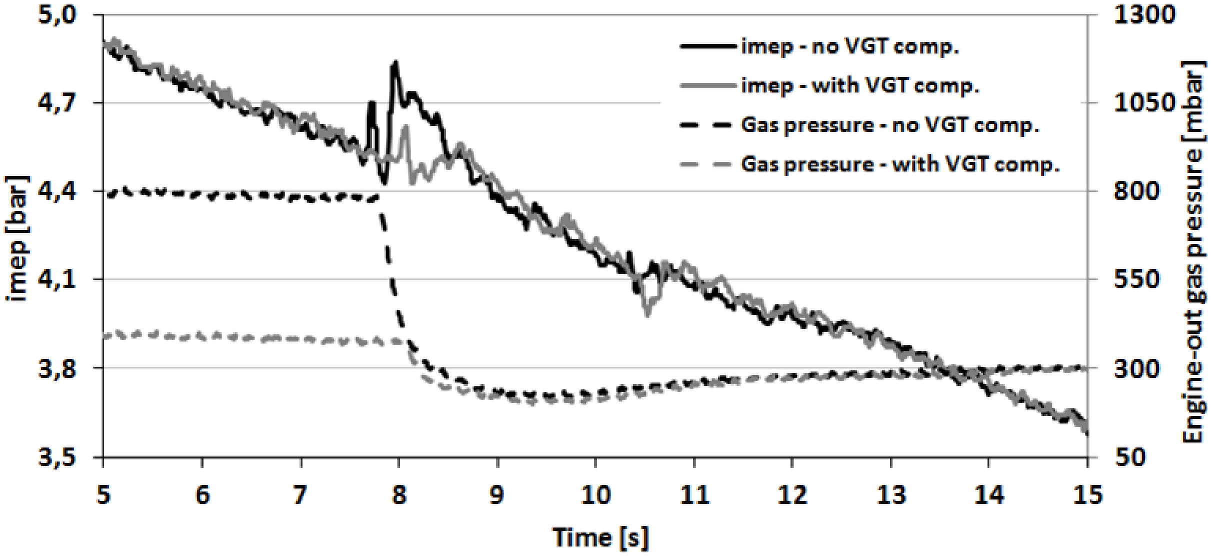

Figure 6.8: Effects of an exhaust gas pressure drop on the *imep* signal by fast EGR valve actuation. A steady pressure profile is achieved compensating it closing the VGT.

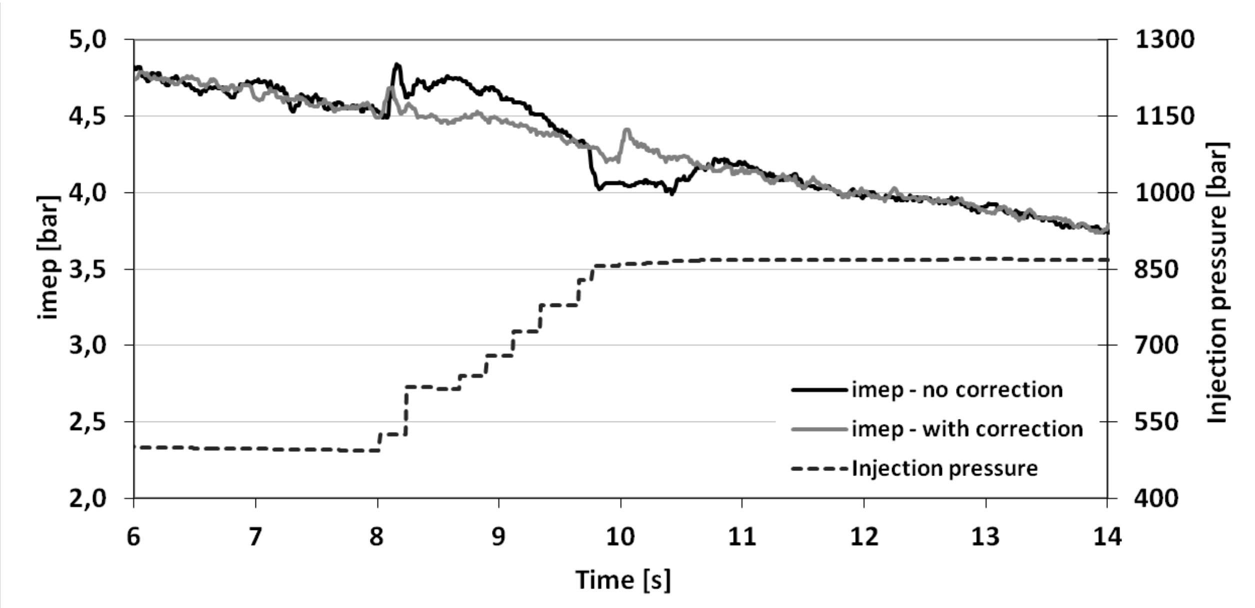

Figure 6.9: Oscillations of the *imep* signal as a consequence of the injection pressure variation difference and improvement using a dynamic correction of the feed-forward value of the main injection duration.

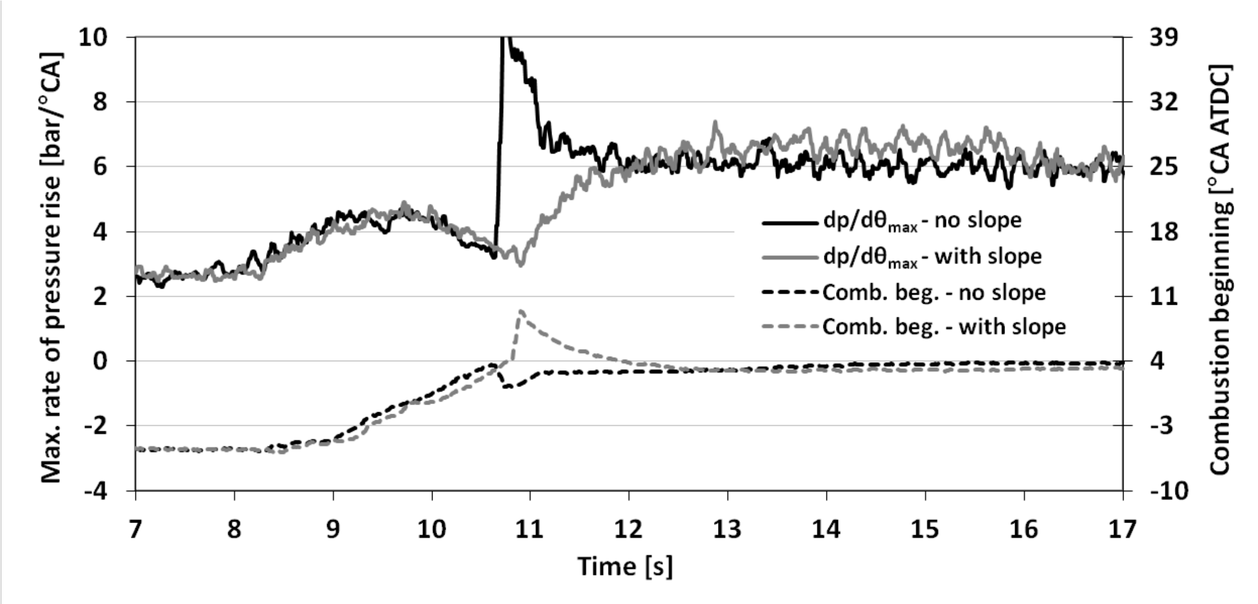

Figure 6.10: High rates of pressure rise resulting from the higher oxygen concentration in the charge in combination with early injection timing and improvement using a correction of the injection timing feed-forward value.

Once the target oxygen concentration is achieved, the injection strategy is commuted to a single injection event. The injection duration value assumes for the first combustion event the feed-forward value of the *imep* controller. This is not the case of the injection timing. At this point, because of the deviated oxygen concentration of the charge, high rates of pressure rise may results if the injection timing would be directly switched to the feed-forward value of the $dp/d\theta_{max}$ controller. Figure 6.10 illustrates this circumstance. Switching directly to the feed-forward value the combustion is anticipated and the maximum rate of pressure rise increases rapidly. Only after some combustion cycle the $dp/d\theta_{max}$ controller is able to reduce the engine noise. A solution has been found applying a correction to the injection timing feed-forward value, switching so the injection timing to a predefined crank angle position closer to TDC. In this way the combustion beginning is phased in the expansion and the mixture reactivity is lowered. Starting from this position the correction of the feed-forward value is slowly reduced. In this way the controller is supported in providing low rates of pressure rise.

7

Results

In this chapter advantages and drawbacks of the proposed closed-loop combustion control system are illustrated and commented. After the definition of the application range for the low-temperature premixed combustion in the speed-torque map, the control strategy is tested under both stationary and transient engine operation mode. For measurements on the engine dynomometer, a simplified $NEDC$ has been designed in accordance with the legislative guidelines. Finally, the results of the combustion mode switch are shown.

7.1 Application range of the low-temperature premixed combustion

The first result of this investigation is the identification of the drivability range of the low temperature premixed combustion mode in the speed-torque map. The engine is run in stationary mode using the proposed closed-loop combustion control strategy. The speed is increased starting from 1000 in $200\,rpm$ steps whereas the load from $1\,bar\ imep$ in $1\,bar$ steps. This is done until the occurrence of combustion instability problems. Engine maps for target parameters and feed-forward values has been previously set operating the engine in a open-loop procedure. The desired maximum rate of pressure rise has been set overall at $6\,bar/°CA$, whereas other actuating parameters has been applyied following the information collected during the system sensitivity analysis, described in Chapter 5. Special attention has been given in providing a good combustion quality and stability and avoiding NOx and PM emissions.

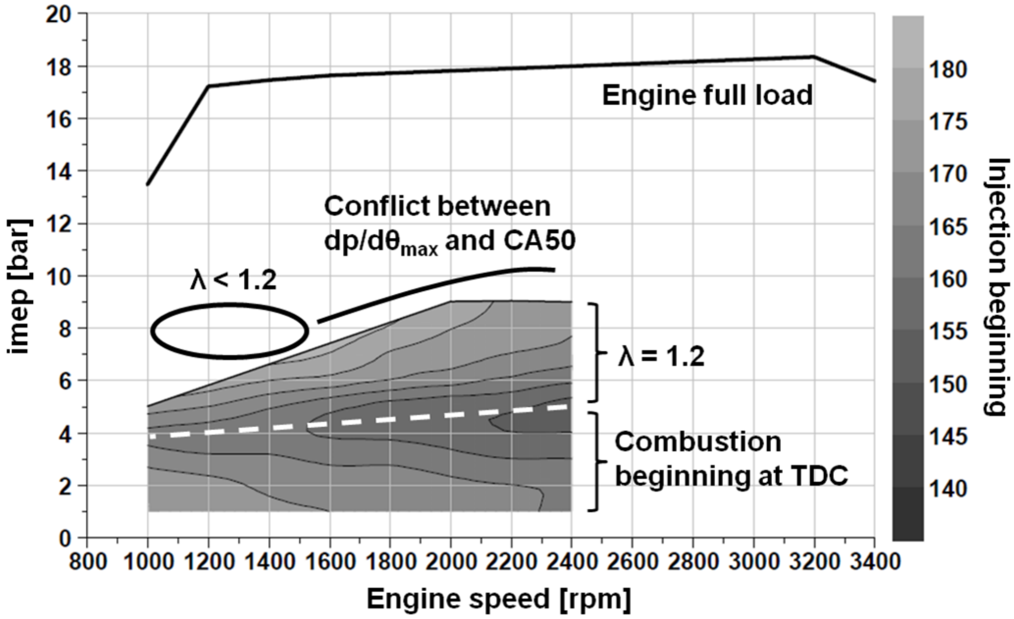

Figure 7.1: Contour map of the injection timing showing the application range for the low-temperature premixed combustion in comparison with the full load capability of the engine.

The resulting drivability range is shown Figure 7.1. In this case the injection timing is plotted. Other actuating and target parameters are illustrated in the Appendix. Increasing the engine speed the controller anticipate sistematically the injection timing in order to compensate the increase in piston velocity. This can be done until 2400 rpm. At this point the time for the mixture formation and for the combustion process becomes very short. Even if the combustion is very fast, the center of combustion is progressively phased in the expansion. A further increase in engine speed would lead to misfire.

Different is the occurrence of combustion istability problems increasing the engine load. In the low-speed range the maximum torque is limited by the lack of fresh air resulting from the lack of boost pressure. The charge motion in the combustion chamber is slow and, even reducing the EGR rate, a further increase in fuel mass results in higher soot emissions. Increasing the engine speed the fresh air content rises and more fuel can be injected. However, this can be done only to a certain extent. Once the soot-limit of λ 1.2 is achieved, the EGR rate must be decreased for allowing more fuel to be injected. The fuel to EGR ratio increases and the mixture becomes more reactive. Additionally, by larger fuel quantity the combustion duration increases shifting the center of combustion late in the expansion. It results a conflict between keeping the maximum rate of pressure rise at the desired level, by retarding the injection event, or preventing combustion deterioration. To overcome these limitations an increased boost pressure would be necessary. This can be realized modifying the turbocharger as demonstrated in [73] and [74].

7.2 Definition of the dynamic driving cycle

In order to investigate the potential of the defined closed-loop control strategy under transient and thus more realistic operating conditions, a dynamic driving cycle is designed. $NEDC$ load/speed roller chassis dynamometer data, provided by Daimeler AG, are simplified for the use on the engine test bench. In accordance with the legislative guidelines [76], following assumptions are taken:

- constant load of $0\,Nm$ and a speed of $800\,rpm$ during idle phases (engine or gearing)
- load and speed drop of the duration of 2 seconds at gear switching
- load drop at the end of an acceleration phase
- constant load and speed during constant speed phases

A better reproducibility of the measurements is achieved changing the load/speed cycle over to a gas pedal/speed cycle. In this way, there is only a little influence of the dynamometer control on the cycle devolution and a larger validity of the results is expected. Moreover, in the software structure the desired *imep* is directly proportional to the gas pedal position, as explained in Chapter 4. The obtained *imep*/speed for the urban part of the cycle is shown in Figure 7.2.

Both the urban (ECE-15) and the extra-urban ($EUDC$) part of the cycle has been simplified for the use on the engine dynamometer as described above. In Figure 7.3 a comparison with the roller chassis dynamometer data on the speed-load map is illustrated. The black diagonal line shows the drivability range limits of the low-temperature premixed combustion. It can be seen that the urban part

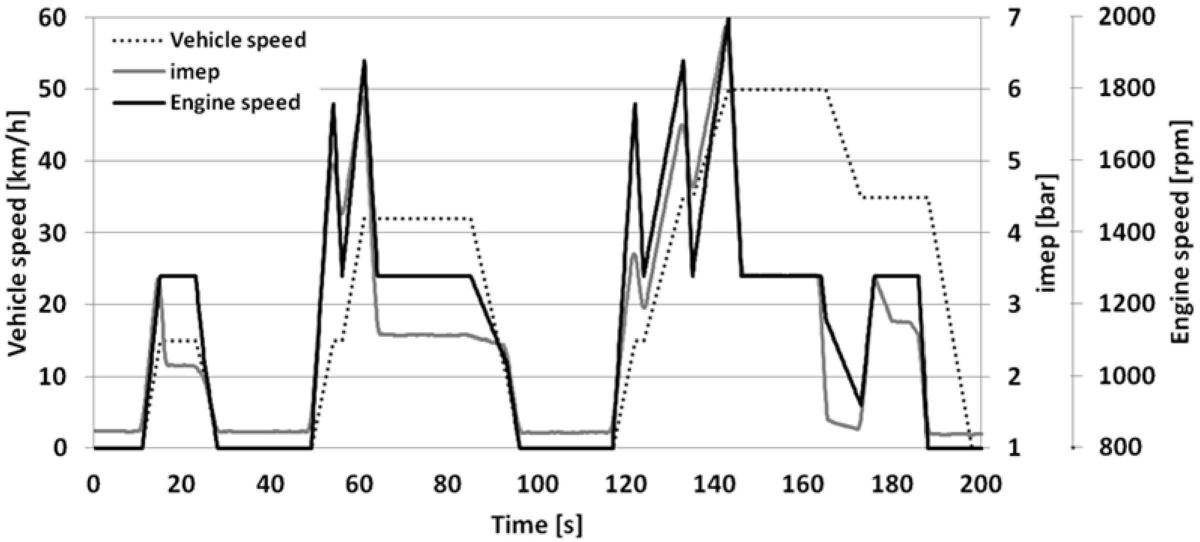

Figure 7.2: *imep*/speed dynamic cycle defined for transient measurements on the engine dynamometer.

of the cycle is completely located below the line whereas the extra-urban can be driven only switching to conventional diffusive combustion. Moreover, the designed cycles reproduces the rapid load and speed changes characteristics of the $NEDC$ very well.

7.3 Measurements in ECE-15

7.3.1 Measurements in low-temperature premixed combustion mode

The feasibility of the defined closed-loop control strategy is confirmed by the measurements in transient mode. As shown in Figure 7.4 the desired engine load (*imep*) is always performed, even by rapid engine speed changes. Optimal system response is achieved also for the highest rates of load increase, which adds up to $0.8\,bar/s$ *imep*. The largest error (defined as the difference between desired and actual parameter value) is about $0.5\,bar$ and the standard deviation over the entire cycle duration measures only $0.105\,bar$. In comparison, the mean

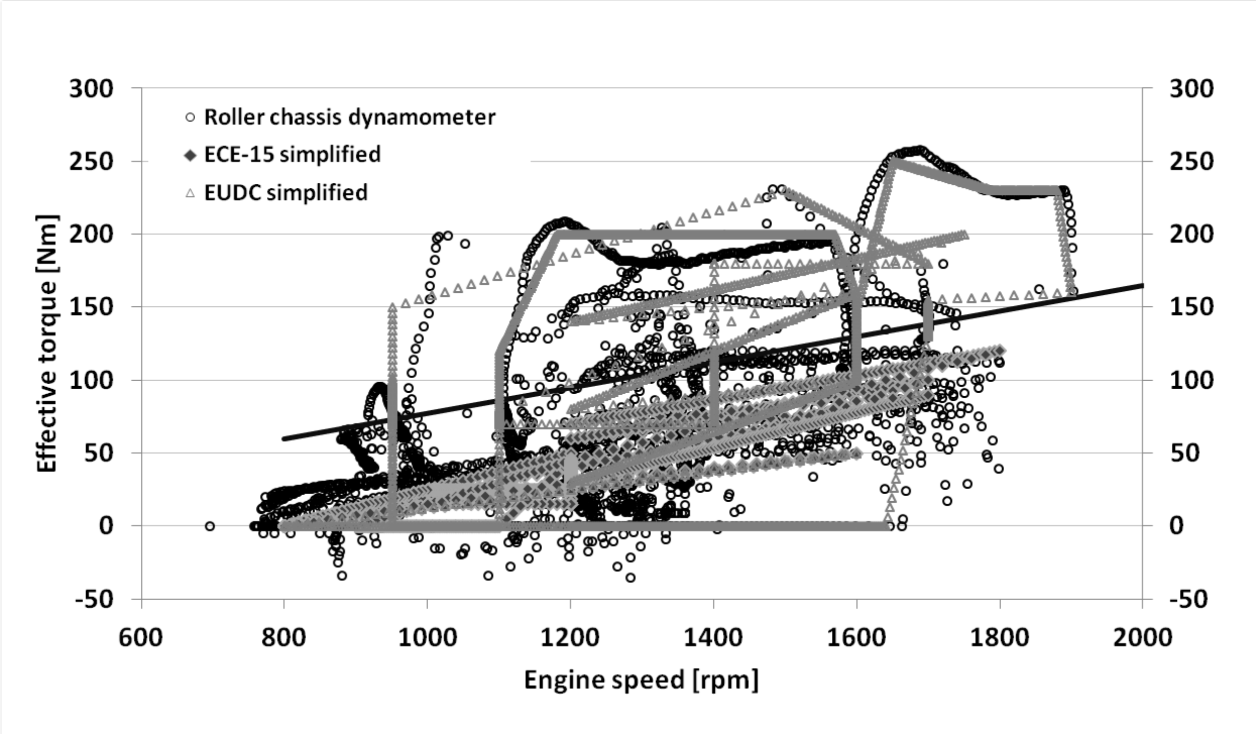

Figure 7.3: Comparison between the roller chassis dynamometer measured data and the simplified dynamic driving cycles.

standard deviation between the six cylinders is about $0.036\,bar$ (with active *impe* controller). At the end of the sharp load reductions, such as at 20 and 130 seconds, oscillations of the *imep* signal can be observed. These are attributed to the slow system response of the air path and the resulting oscillations in the charge oxygen concentration.

More interesting are the results about the maximum rate of pressure rise closed-loop control, which serves as engine protection controller. These are illustrated in Figure 7.5. It can be seen how high rates of pressure rise can be effectively avoided controlling the injection timing. In fact, adopting this strategy, local increases of the $dp/d\theta_{max}$ value are immediately corrected. The largest error is about $3.03\,bar/°CA$ and its standard deviation over the 200 seconds of the ECE-15 duration is $0.534\,bar/°CA$, which is in the same order of magnitude of the statistical $dp/d\theta_{max}$ spread among the six cylinders.

Due to the high cycle dynamic, phases characterized by rates of EGR larger than expected (if compared to the stationary application) occur. In these phases the injection timing limitation, described at the end of Chapter 5, is active.

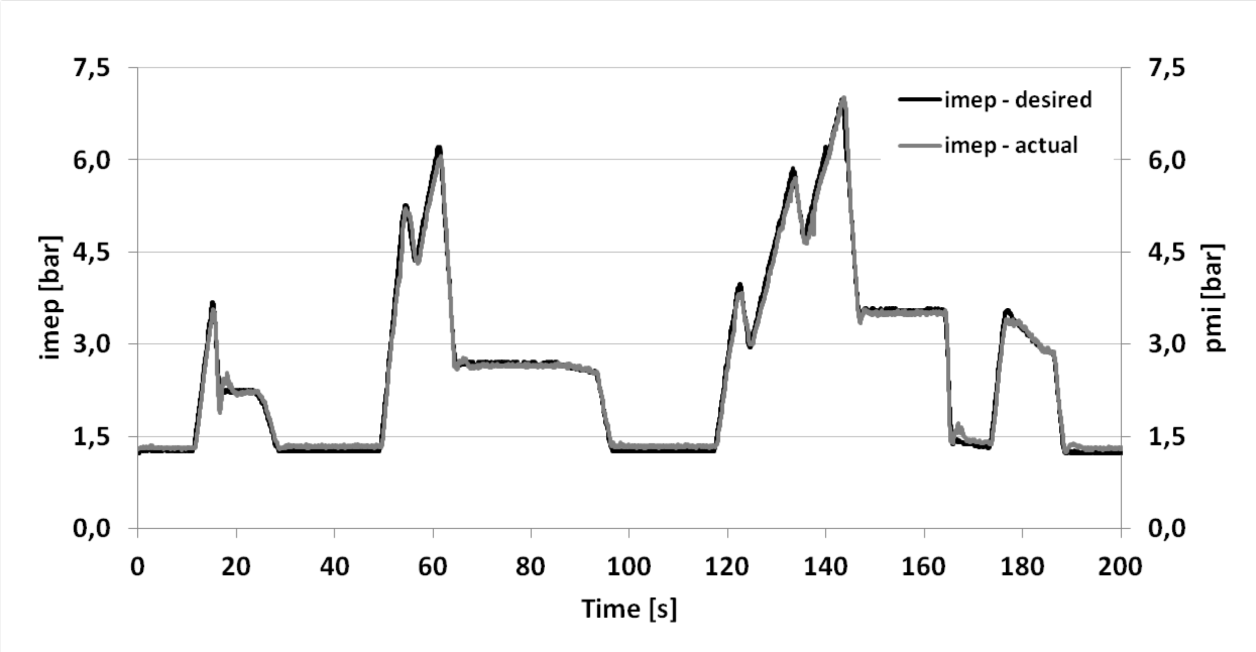

Figure 7.4: Desired and actual *imep* over the simplified ECE-15 in low-temperature premixed combustion mode.

This means that a corrected, lower, desired maximum rate of pressure rise is calculated by the inverted PID structure, avoiding so the combustion beginning to be phased in the compression stroke. The new values are plotted together with the original desired ones and the actual values. At the bottom of the figure the activation of the correction is shown. Because of the different charge filling among the cylinders, this limitation must be activated individually and a spreading of the engine noise may results.

The good quality of the results must be also attributed to the dynamic correction of the injection timing feed-forward value over the deviation of the oxygen concentration in the charge, which has been also presented at the end of Chapter 5. This is an effective solution for improving the controller response. In fact, without this future, the $dp/d\theta_{max}$ signal error increases at $5.13\,bar/°CA$ and the standard deviation at $0.968\,bar/°CA$, which is significantly higher than the values discussed above. This has a direct effect both on the combustion noise and the engine-out emissions. In fact, using the dynamic correction for avoiding high fluctuations of the rate of pressure rise the NOx emissions can be reduced of about 3% over the entire ECE-15. Moreover, due to the smoother $dp/d\theta_{max}$ signal, resulting from the improved system response, there is no need

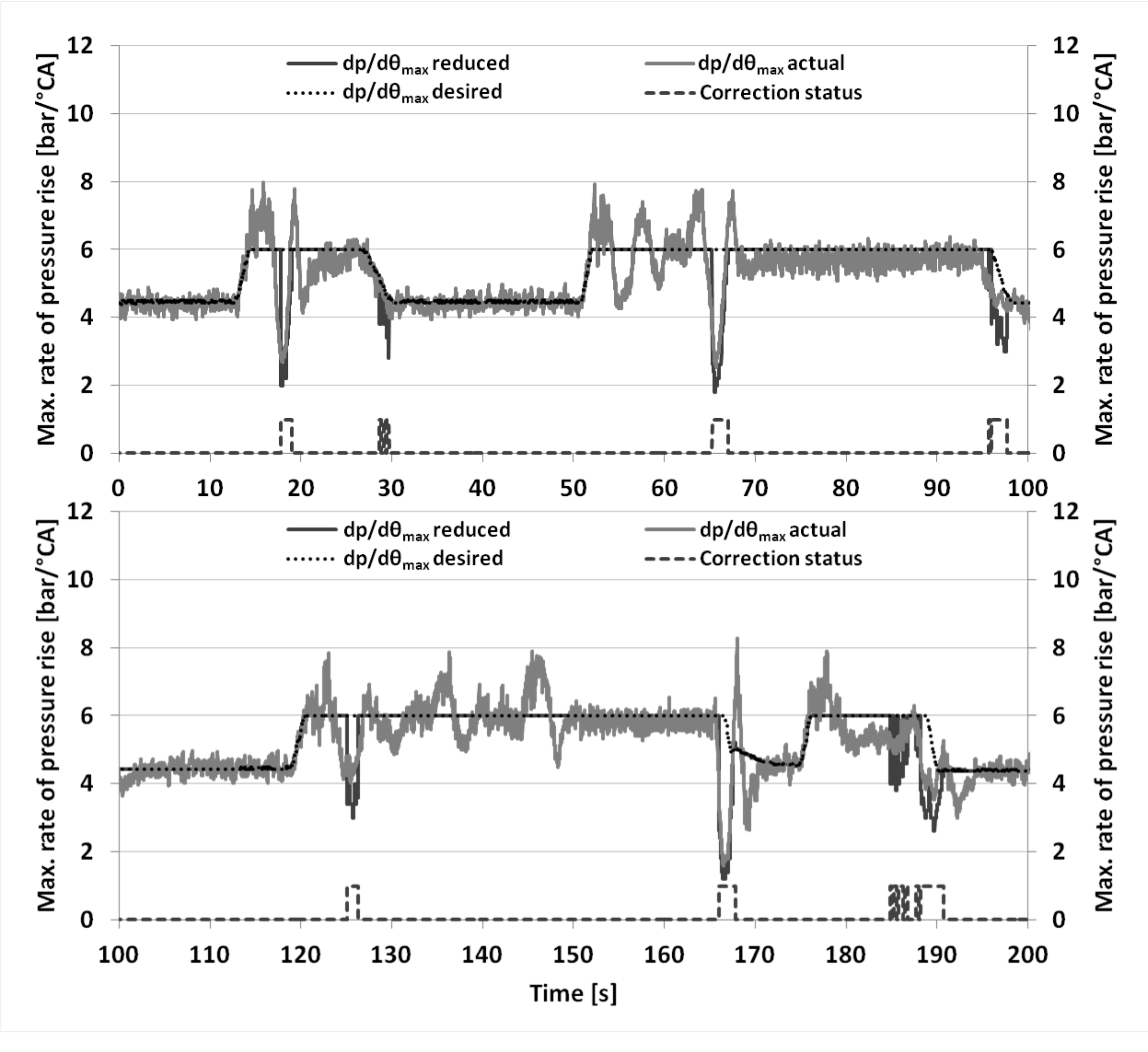

Figure 7.5: Desired, corrected and actual $dp/d\theta_{max}$ over the simplified ECE-15 in low-temperature premixed combustion mode.

of phasing the combustion beginning late in the expansion. This results in an improved combustion efficiency and lower UHC emissions during transients. As shown in Figure 5.17, the best conditions of pressure and temperature for an efficient fuel conversion can be found in the proximity of the TDC. The overall mean combustion beginning is phased $0.4°CA$ earlier using the dynamic feed-forward correction, and an improvement of the UHC emissions of about 4.4% is measured.

Finally, it can be said, that the proposed strategy provides an effective solution for the closed-loop control of the low-temperature premixed combustion. Thanks to a clear separation between the different tasks, engine load, engine protection and charge quality, the controllers can work independently and efficiently. The slow system response of the air path, and following of the charge quality controller, becomes under transient engine operation mode a mayor disturbance variable of the system. However, this can be compensated improving the response of the engine protection controller by monitoring the deviation of the oxygen concentration and avoiding the combustion beginning to be shifted before TDC.

7.3.2 Measurements with combustion mode switch

As already discussed, using the proposed combustion control strategy it is possible to drive the entire ECE-15 only using alternative combustion. However, a switch to conventional diffusive combustion is suggested for avoiding fuel consumption deterioration and high soot and UHC emissions. Moreover, in the middle load range, where the low-temperature premixed combustion is limited by the soot-limit of λ 1.2, high dynamic load changes may lead to a breach of this limit resulting in an additional soot emission increase. Responsible for these fluctuations of the charge quality are the high need of EGR for limiting the charge reactivity and the slow system response of the air path. Switching to diffusive combustion however, this problematic can be avoided. Setting the switch limit at $4.5\,bar\ imep$, in the urban cycle the combustion mode is switched two times from low-temperature premixed to diffusive and inversely. Figure 7.6 shows a comparison of the λ signal between low-temperature premixed and mixed mode combustion. The air/fuel ratio drops below the soot-limit, at 60 and 140 seconds.

This circumstance is avoided switching the combustion mode.

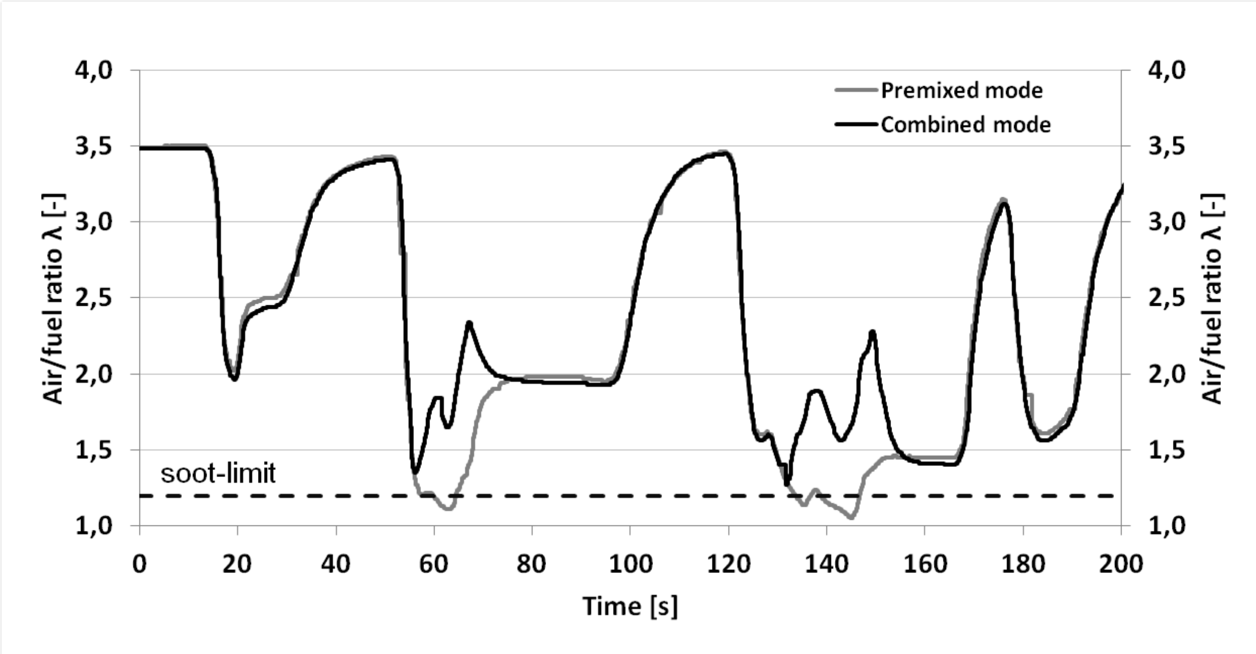

Figure 7.6: Oscillations of the lambda signal in low-temperature premixed combustion mode and possibility of avoiding over-dilution of the charge switching to diffuse combustion.

Switching the combustion mode it results a slightly increase in the overall *imep* signal error. The largest value becomes $0.86\,bar$ and the standard deviation $0.111\,bar$. The larger *imep* oscillations can be recognized at the end of the deceleration phases at 65 and 150 seconds in Figure 7.7. Generally, however, the proposed switching procedure, together with the combustion control strategy, provides a good engine performance over the entire cycle.

An enlargement of the switching phases is shown in Figure 7.8. At the bottom of the diagrams the fuel and air path switching timing are plotted. In order to better understand the switching process dynamic the charge oxygen concentration is also shown. As explained in Chapter 6.3, the air path is switched at first once the *imep* has reached the limit value of $4.5\,bar$. The fuel path, in this case injection strategy and injection pressure, is switched 1.7 seconds later, when the oxygen concentration has reached the target value. During this time the injection timing feed-forward value is shifted toward TDC for avoiding an increase of the combustion noise. At this point the injection pressure is reduced and the pilot injections are added to the combustion. The switching procedure

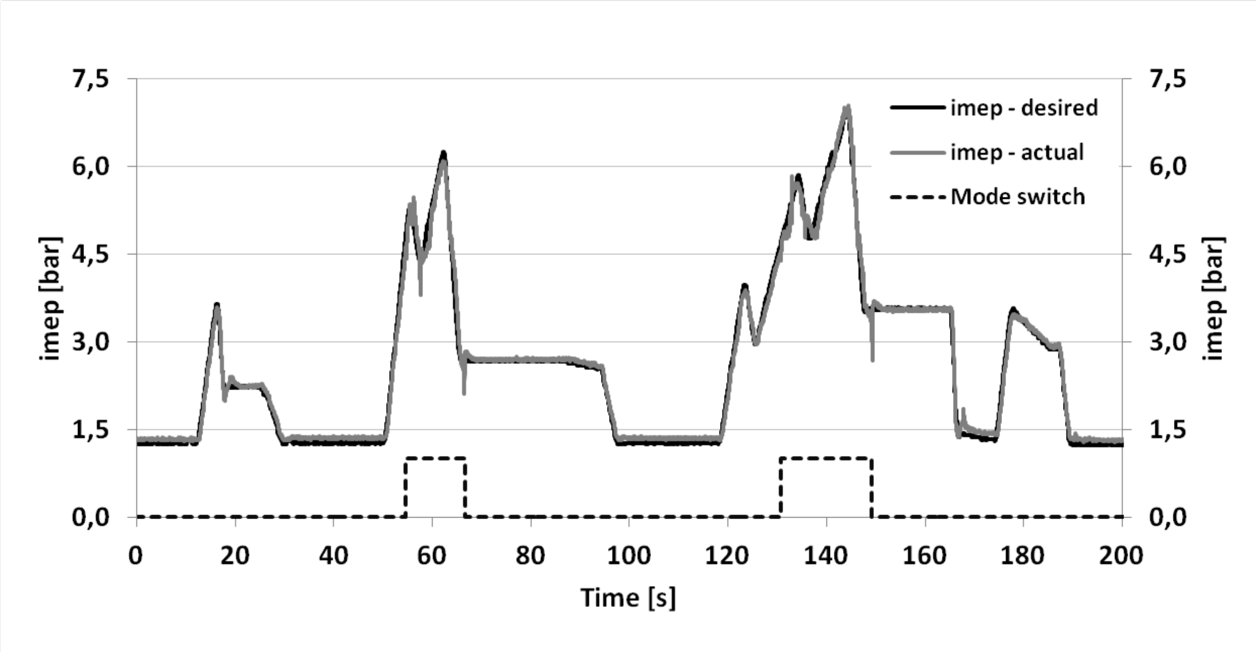

Figure 7.7: Desired and actual *imep* over the simplified *ECE*-15 in combined combustion mode.

ends when the injection pressure has reached the desired value for the diffusive combustion, this means approximatively 3.5 seconds after the beginning of the procedure. At 58 seconds a marked discontinuity in the *imep* signal can be recognized. This is attributed to the lack of compensation of the wave oscillations in the common-rail using multiple injection strategy.

As long as the *imep* signal does not drop below 4 *bar* the engine is operated in diffuve combustion mode. Since in this mode any active control of the charge quality is used, the desired oxygen concentration value is set equal to the actual one. At switching time, it literally jumps to the lower desired value of the premixed combustion and the controller is reactivated. In this case the injection pressure is switched together with the oxygen concentration. A steady *imep* signal is obtained reducing the injection duration, as explained in Chapter 6.4. However, this strategy still show some drawbacks. These can be recognized in the decreasing signal value short before the fuel path is switched, i.e. at 67 and 149 seconds. The switching procedure for diffusive to low-temperature premixed combustions lasts for approximatively 2.5 seconds.

A comparison of the maximum rates of pressure rise during the urban cycle using

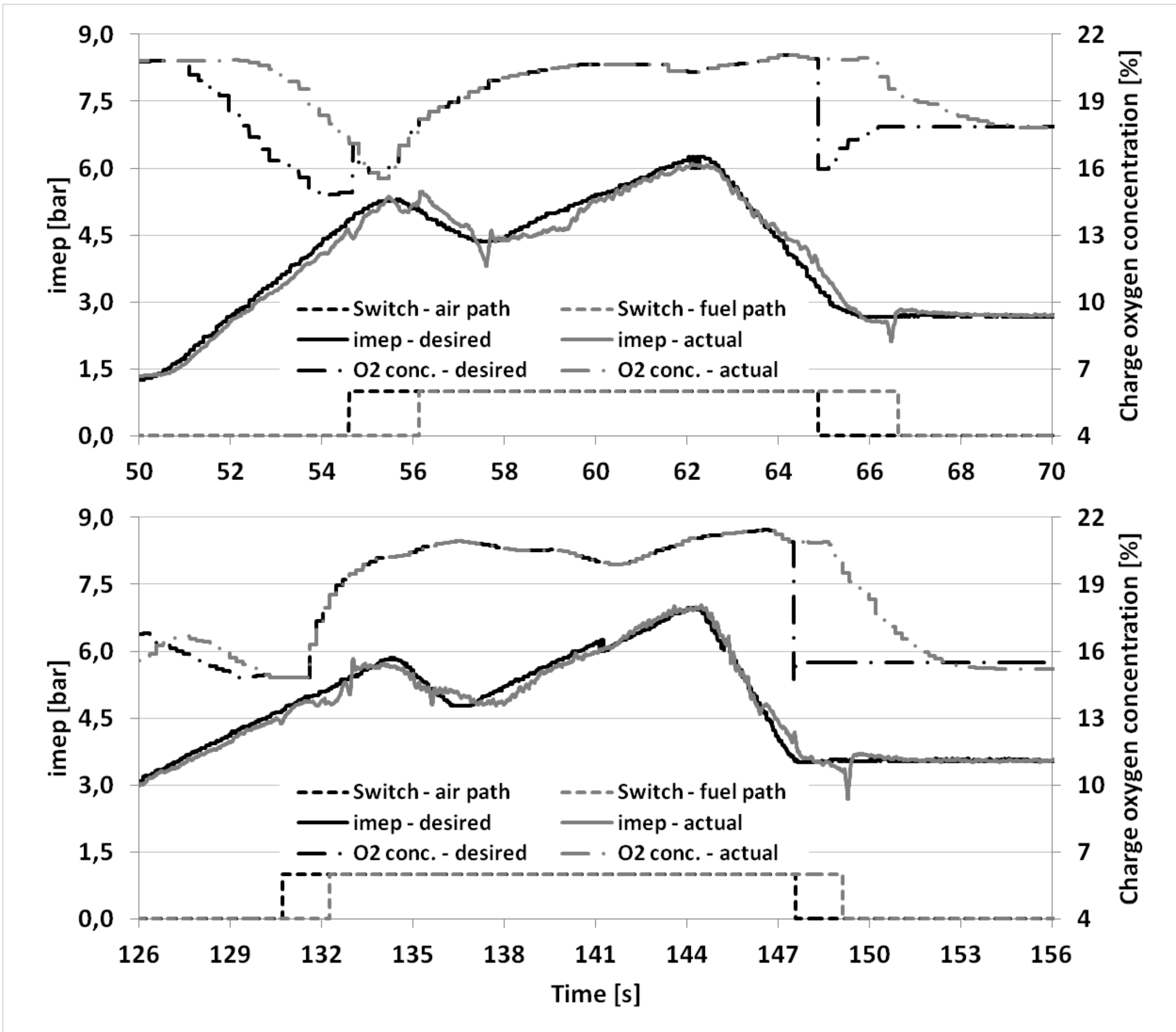

Figure 7.8: Enlarged view of the switching phases.

solely the alternative or the combined combustion mode is shown in Figure 7.9. As expected a reduction of the $dp/d\theta_{max}$ signal is observed when the engine is operated in diffusive mode. During the switching procedure some oscillation occurs. It must be considered that during these phases the oxygen concentration in the charge, as well as the engine load, varies very rapidly and the engine protection and charge dilution controllers must be de- or reactivated depending on the actual combustion mode. Since any significant increase of the rates of pressure rise is measured, it can be said that the goals set at the beginning of Chapter 6 have been successful achieved.

For sake of completeness, a comparison of the engine-out NOx and UHC emis-

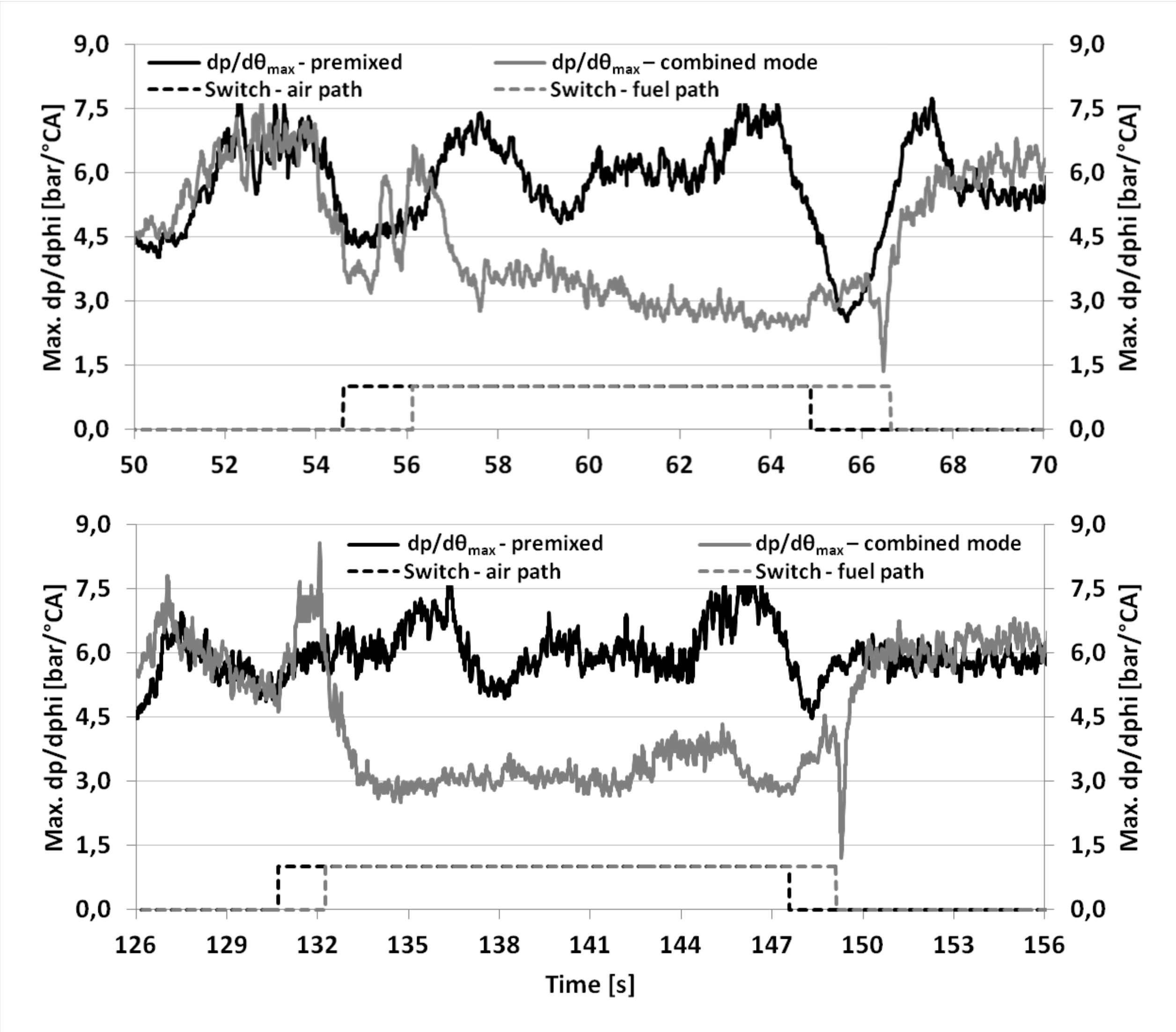

Figure 7.9: Comparison of the maximum rates of pressure rise during the urban cycle using alternative or combined combustion mode.

sions is illustrated in Figure 7.10. Low UHC emissions characterize the phases with multiple injection strategy and diffusive combustion whereas an increase of the NOx emissions is measured. This results from the higher oxygen concentration and higher combustion temperatures. Particularly during the second phase, i.e. between 130 and 150 seconds, the NOx emissions increase becomes significant showing the potential of low-temperature premixed combustion processes.

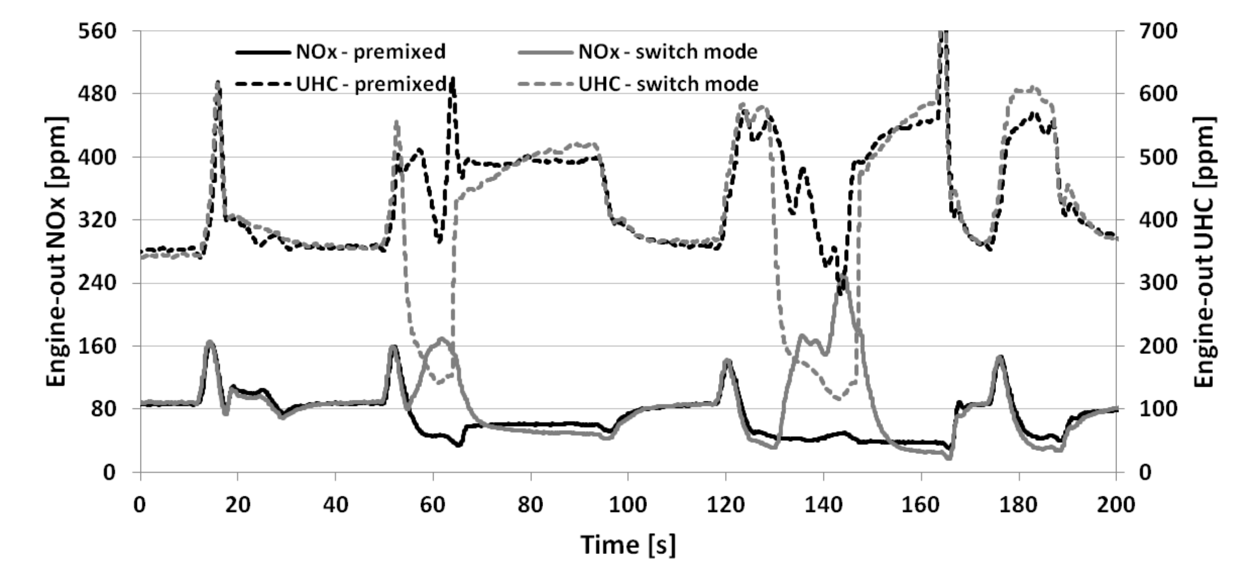

Figure 7.10: NOx and UHC engine-out emissions during the ECE-15. Comparison between alternative and mixed combustion mode.

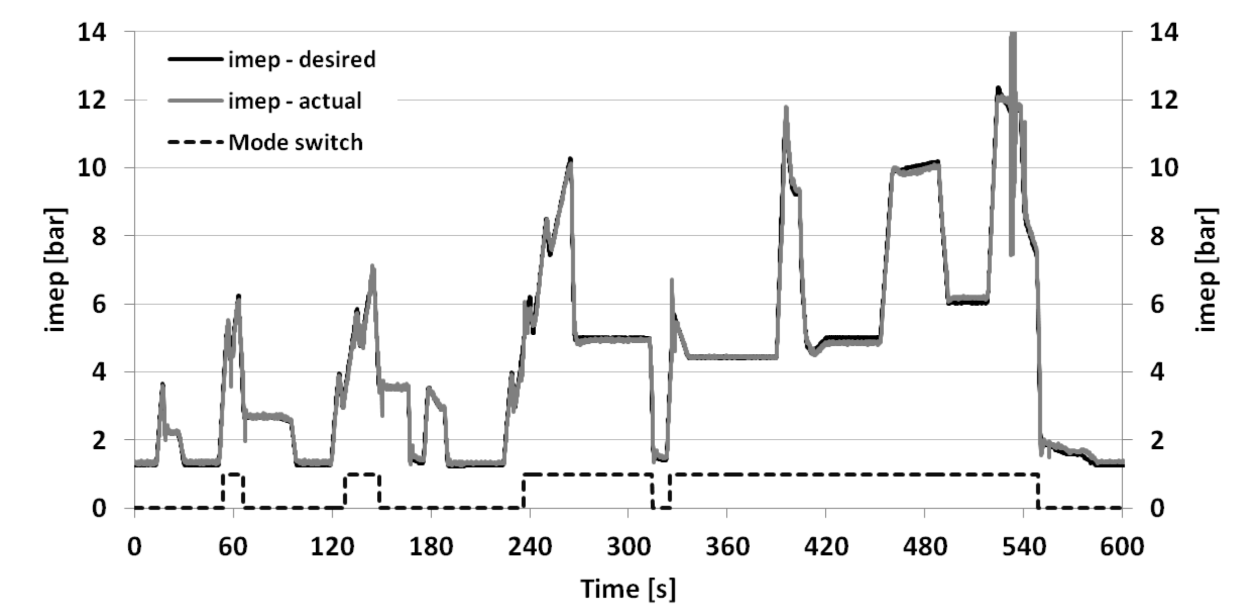

Figure 7.11: Desired and actual $imep$ signal over the entire $NEDC$.

7.4 Measurements in entire NEDC

Switching from low-temperature premixed to diffusive combustion mode the engine can be operated also outside the limited drivability range of the alternative combustion. This is demonstrated driving the entire $NEDC$ at the test bench. The results are shown in Figure 7.11. In the extra-urban part of the cycle the engine is mainly operated using diffusion combustion, however four additional mode switching events take place.

8

Summary and Conclusions

In this chapter a summary of the research activities conducted during this work is given. Following, the success of the presented investigation is discussed.

8.1 Summary

In the present work an investigation of the thermodynamic mechanisms ruling the combustion in a low-temperature premixed process has been carried out. The investigation has been based on an extensive literature search covering the main aspects related to alternative combustion processes for automotive engines applications. A sensitivity analysis of the system under investigation has been than carried out. Dependencies and correlations between actuating and target parameters could be so identified. On the base of these, a new closed-loop control strategy has been proposed.

A strict correlation between the injection timing and the maximum rate of pressure rise has been identified. This has been found to be monotonic increasing as long the combustion beginning is phased before TDC, and becomes monotonic decreasing for combustion beginning in the expansion stroke. This correlation is independent from the engine speed and load. Moreover, it is suggested to phase the combustion process during the expansion stroke, where the system flexibility is larger and the performances are better. Based on this correlation a closed-loop controller of the maximum rate of pressure rise is proposed. This should provide an engine protection function avoiding the maximum rate of pressure rise to exceed mechanical stress limits of the engine hardware. At the same time, driving the engine at constant maximum rate of pressure rise, it also avoids the deterioration of the combustion process resulting by inappropriate combustion phasing.

A closed-loop control of the combustion phasing over the injection timing and of the *imep* over the injection duration has been realized. The charge quality, i.e. the oxygen concentration or the rate of EGR, has been set independently operating the EGR valve. The control loop has been closed using a lambda sensor in the intake manifold which is capable of measuring the oxygen concentration in the charge, consisting of a mixture of fresh air and recirculated exhaust gas. This strategy allowed a clear separation of the controllers tasks.

A target area for the closed-loop control strategy has been found measuring an EGR rate/injection timing map at constant engine load. Combining the contour plots of different parameters on a single diagram, a target area characterized by high combustion efficiency and low emissions has been identified. This area can be achieved increasing the EGR rate maintaining the maximum rate of pressure rise constant, shifting so the combustion beginning towards the TDC.

The control strategy of the low-temperature combustion is mainly limited by two factors. For a desired maximum rate of pressure rise, the EGR rate can be increased until the the combustion beginning has reached the TDC. After this point a further increase of the EGR would result in a reduction of the maximum rate of pressure rise. This is usually the case for low-load einge operation. Increasing the load, an additional limitation must be taken into account. Since the fresh air quantity does not increase significantly, the air/fuel ratio decreses with increasing fuel quantity. The rate of recirculated exhauste gas must be limited for avoiding the charge quality to drop below the soot-limit of λ 1.2. It resulted a clear separation between a low-torque combustion beginning limited operation range and a mid-torque λ-limited range. The border between this two modes measures approximatively $4.5\,bar\ imep$. Looking at the injection timing two different patterns could be identified, i.e. $HCLI$ and $HPLI$ combustion.

The proposed closed-loop control strategy has been first tested under stationary engine operation and laboratory conditions. A speed-load drivability range which extends between 1000 and $2400\,rpm$ and between 1 and $9\,bar\ imep$ has been measured. This range is mainly limited by the low boost pressure and the lack of fresh air at low speed and by combustion instability at higher speed.

In order to overcome the limited application range of the low-temperature premixed combustion, a switching procedure to and from conventional diffusive combustion has been designed. The proposed solution is based on engine measurements which allowed the characterization of the most relevant problematics related to this topic. These mainly result from fast actuations of the EGR valve or the injection pressure. Dynamic corrections for supporting the controllers has been defined. These provide a steady monotonic increase (or decrease) of the engine load under stationary and transient engine operation.

Finally, a high dynamic $NEDC$-based cycle for the engine dynamometer has been defined in order to test the closed-loop control strategy under realistic and repeatable transient conditions. The results show a good response of the controllers and confirm the robustness of the defined strategy.

8.2 Conclusions

This work demonstrates the potential of a thorough investigation of the combustion process as fundament for the definition of a new closed-loop control strategy. Understanding the mechanisms ruling the combustion, system limitations (together with the hardware constrains) and clear dependencies between actuating and target parameters can be identified and implemented in a robust control strategy of the combustion process. The validity of this approach is demonstrated by the good system response under stationary but particularly under transient engine operation. This methodology can be further employed in the definition of a closed-loop control strategy for the diffusive diesel combustion, which, because of the multiple injection strategy, is characterized by an even higher complexity. In this case a deeper understanding of the thermodynamic mechanisms related to the manipulation of the pilot injections is needed.

The proposed pressure-based closed-loop combustion control provides a feasible solution for implementing low-temperature combustion processes in the coming series engine generation. Since the conventional injection system, as well as a standard combustion chamber geometry can be utilized, the need of hardware modifications is very little. However, an appropriate boost pressure strategy supported by a low pressure EGR loop and twin-stage turbo-charging would provide further improvements. Near the possibility of extending the divability range of the low-temperature premixed combustion, the implementation of a further actuating parameter, i.e. the variable boost pressure, would increase the flexibility of the proposed strategy.

The switching procedure to and from diffusive combustion completes the strategy and expand the drivability range up to engine full load. In this case a refinement of the proposed dynamic corrections is suggested.

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Amstblatt der Europäische Union, 70/220/EWG, 2007.

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Appendix

A.1 Contour maps low-temperature premixed combustion

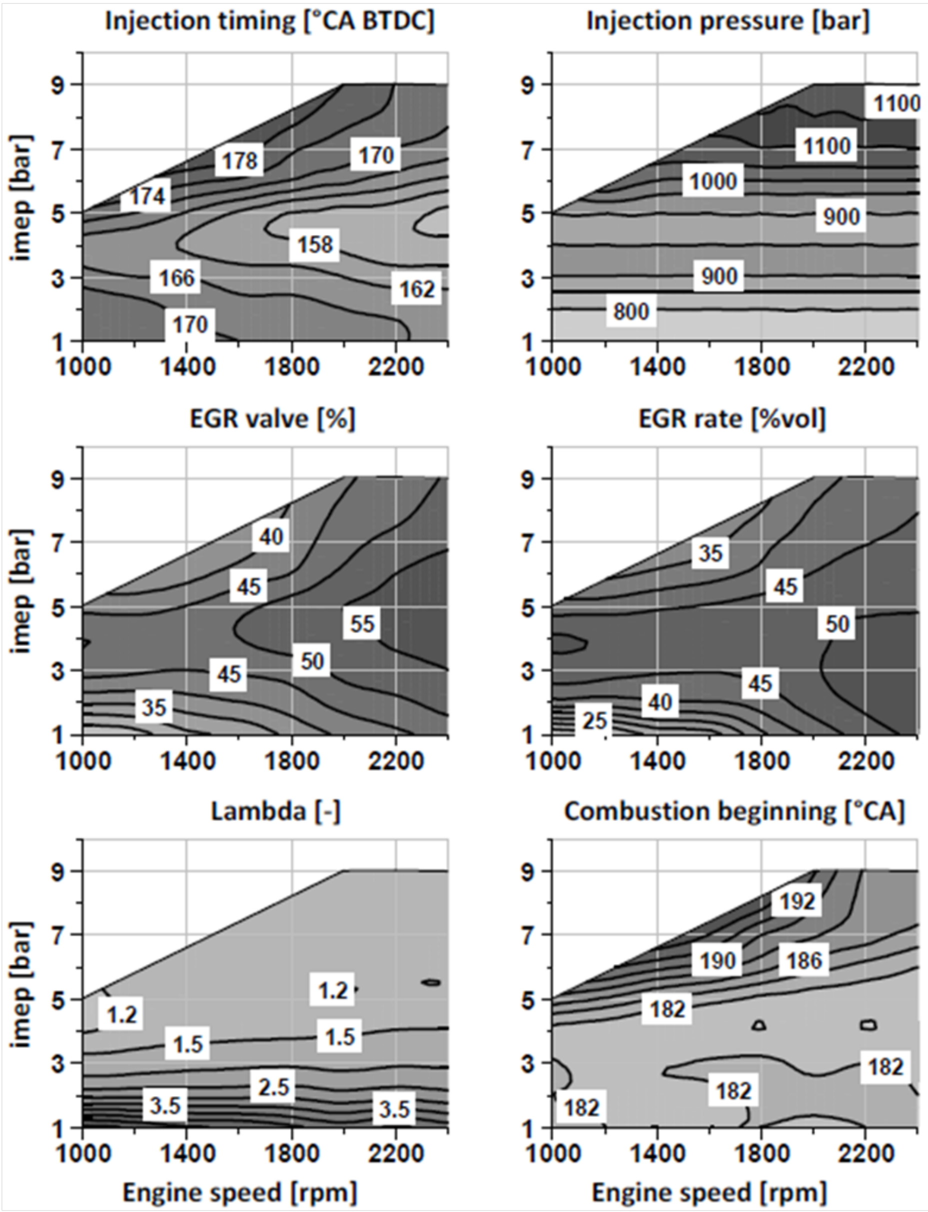

Figure A.1: Actuating parameters values under stationary operation mode.

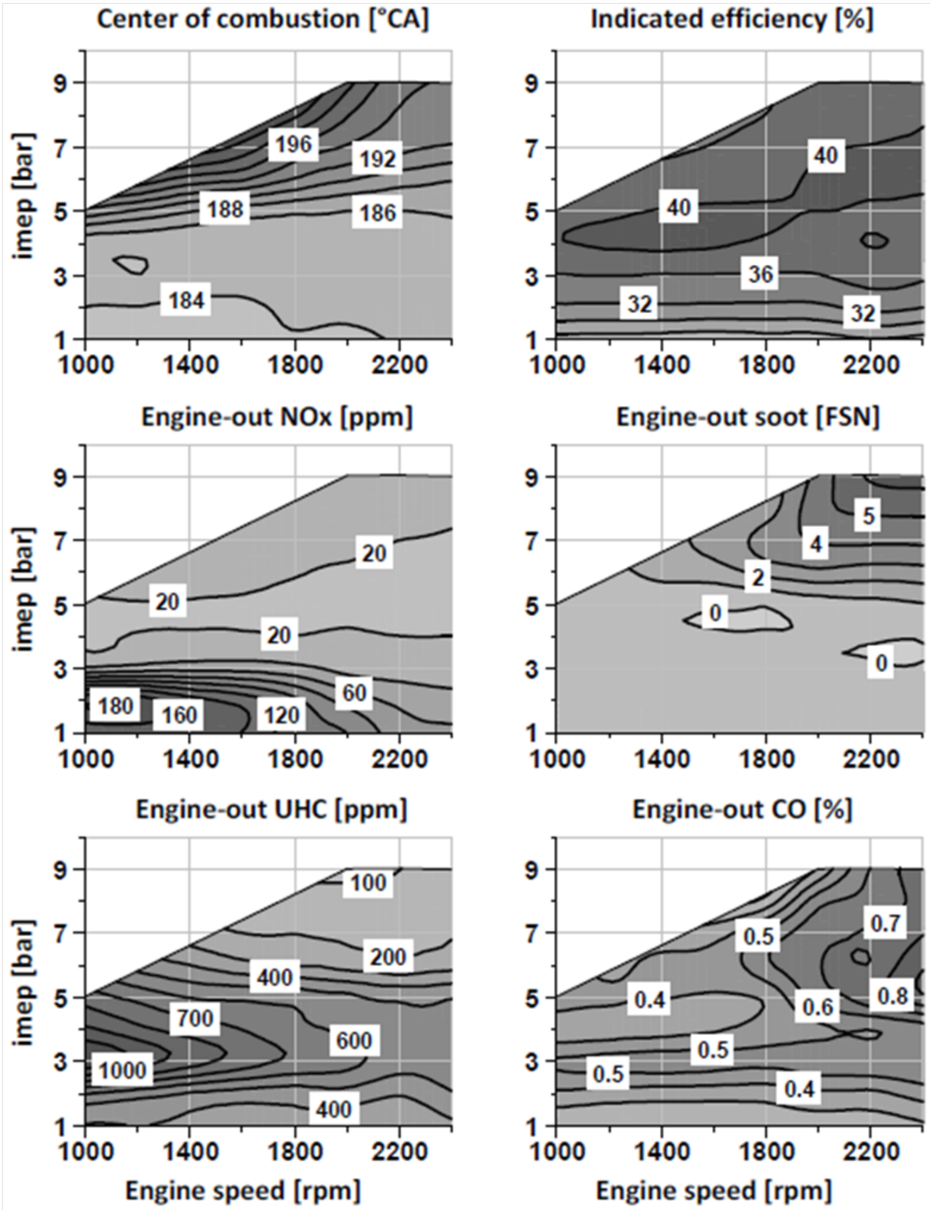

Figure A.2: Target parameters values under stationary operation mode.

A.2 Contour maps diffusive combustion

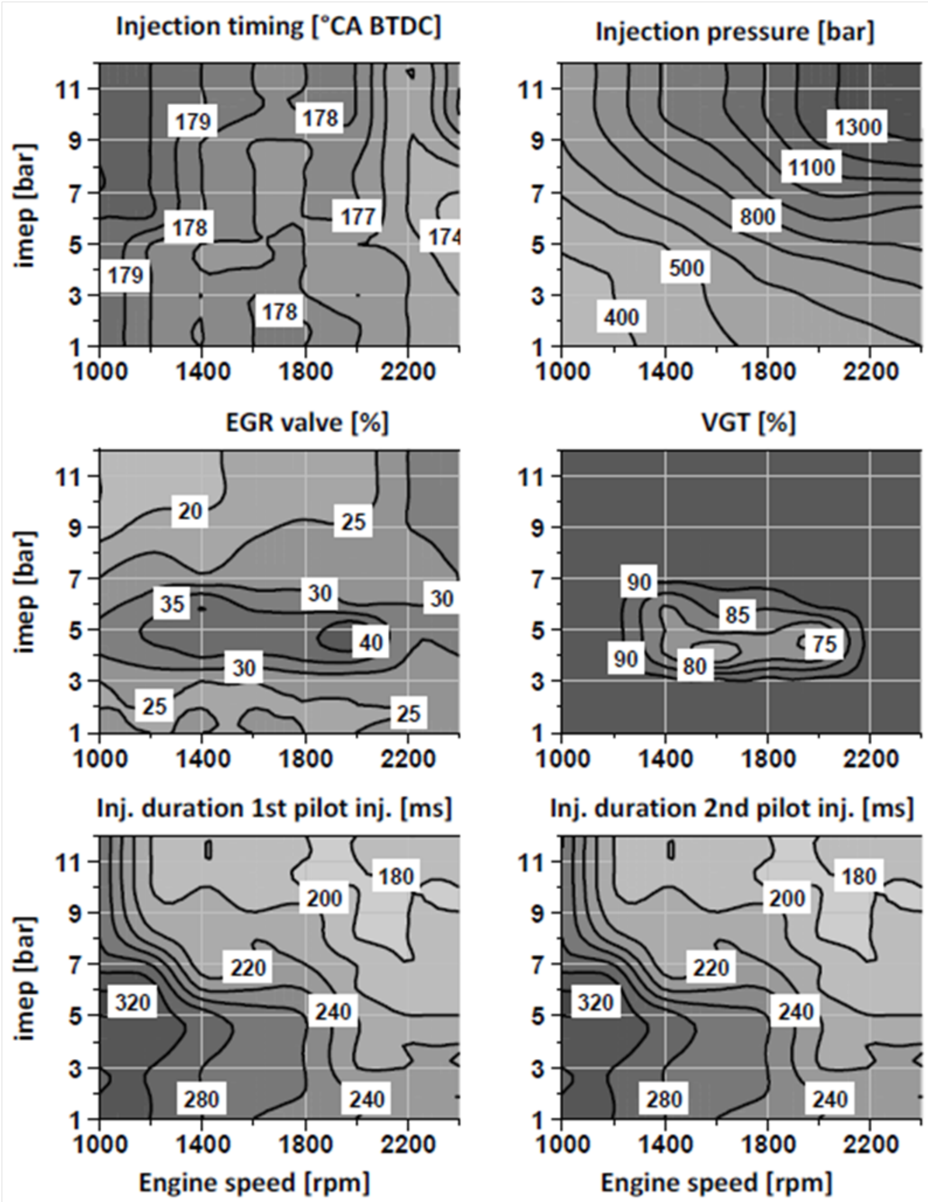

Figure A.3: Actuating parameter values under stationary operation mode.