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TESI DI LAUREA
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**EXPERIMENTAL STUDY ON THE EFFECT OF STOICHIOMETRIC
AIR/FUEL RATIO OF THREE BINARY OXYGENATED FUEL
BLENDS ON COMBUSTION AND EMISSIONS OF A HEAVY DUTY
DIESEL ENGINE**

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THESIS LAYOUT STRUCTURE

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NOMENCLATURE

(A/F)	Air-fuel ratio
(A/F) _{st}	Stoichiometric air-fuel ratio
BEV	Battery electric vehicles
BMEP	Brake mean effective pressure
BD	Burn duration
CA10	Crank angle where 10 % of the heat has been released
CA50	Crank angle where 50 % of the heat has been released
CA90	Crank angle where 90 % of the heat has been released
CAD	Crank angle degree
CN	Cetane number
CO	Carbon monoxides
CO ₂	Carbon dioxide
COV	Coefficient of variability
CVS	Constant volume system
DOC	Diesel oxidation catalyst
DPF	Diesel particulate filter
ECU	Engine control unit
EEPS	Engine exhaust particle sizer
EGR	Exhaust gas recirculation
EVC	Exhaust valve closing
EVO	Exhaust valve opening
IMEP	Indicated mean effective pressure
IMEP _g	Indicated mean effective pressure (gross)
FSN	Filter smoke number
HC	Hydrocarbons
HCCI	Homogeneous charge compression ignition
ID	Ignition delay
ISCO	Indicated specific carbon monoxide

ISFC	Indicated specific fuel consumption
ISHC	Indicated specific hydrocarbons
ISNO _x	Indicated specific nitrogen oxide
ISPM	Indicated specific particulate matter
LHV	Lower heating value
LTC	Low temperature combustion
NO _x	Nitrogen oxides
PM	Particulate matter
PPC	Partially premixed combustion
PRF	Primary reference fuel
PRR	Pressure rise rate
RON	Research octane number
SCR	Selective catalytic reduction
SI	Spark ignition
SOA	Start of actuation
SOC	Start of combustion
SOI	Start of injection
SOOT	Particulate matter
TDC	Top dead centre
V _d	Volume displacement
UHC	Unburned hydrocarbons
ϕ	Equivalence ratio
α	Air-fuel ratio
λ	Air excess ratio

ABSTRACT

Low Temperature Combustion (LTC) is believed to be the best solution to the diesel NO_x and soot emissions trade-off: the latest development in LTC methods is PPC (Partially Premixed Combustion) which can be considered as a combination of Homogeneous Charge Compression Ignition (HCCI) and conventional diesel combustion.

The increasing interest in high-octane alternative fuels, such as alcohols and aromatic compounds, leads to investigate their combustion behaviour, in order to develop cleaner fuels, making another small step towards new cleaner engines.

In this thesis, three binary oxygenated fuel blends are selected, with the aim of studying how their molecular structures and stoichiometric air-fuel ratio influenced the combustion performances and emissions of a truck heavy-duty diesel engine.

The first part describes the fundamental elements of LTC and gives a background on the typical diesel combustion emissions, with the description of the alternative test fuels of this research.

The second part is about the engine setup, with particular attention towards the emission measurement and data acquisition systems.

The third part gives a background on the important parameters of this research, mostly combustion control parameters: it is fundamental to understand the test results, which are presented in the final part of the thesis.

This master thesis project has been developed in collaboration with the “Multiphase and reactive flows” research group at TU/e, Eindhoven University of Technology, in the Netherlands.

CHAPTER 1 - INTRODUCTION

BACKGROUND

Nowadays and in the latest period, pollutants emitted by diesel engines has become an heavy problem for the planet environment, making the society increasingly concerned. Greenhouse effect, acid rain and ozone depletion are just some of the biggest side effects which are directly connected to the worldwide use of combustion engines. The large variety of fuels used in this sector have led to the emission of several harmful pollutants, such as soot, NO_x, CO and HC.

The main focus, in order to enhance cleaner engines, is to control and alter the combustion to reduce the emissions, without losing performances and keeping costs affordable: emissions from vehicles, especially soot emission, can be reduced using renewable fuels from biomass.

Actually the increasing price and decreasing supply of petroleum are leading to an important changing: future transportation fuels are expected to be modified, adding larger amount of alternative fuel components.

Internal combustion engine has been used for transportation for several years, becoming a key factor in the economy: in particular, Diesel engine is currently widely used in many application due to its high efficiency, strong stability, high flexibility for a variety of operating conditions.

The active research on cleaner engines and fuels is really strong, enhanced by the strictly constrains referred to the latest EURO regulations, made to follow the European regulatory pathway for vehicle emissions control.

The EURO track was established in 1992 and progressing through to EURO VI in 2013.

In the table below, standard emissions of NOx and soot for Diesel engines are shown.

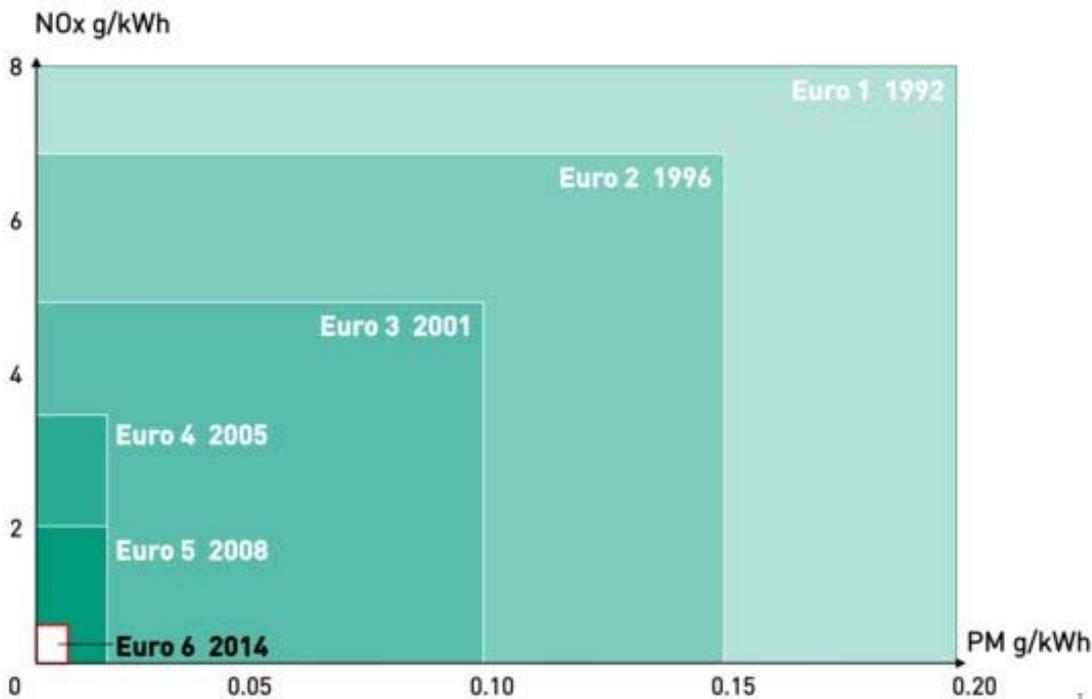


Figure A - EU NOx and soot Emission standards for Diesel

These regulations led to significant revolution in engine-related or fuel-related technologies, as well as exhaust aftertreatment technologies are required. For example, Diesel engines have been provided with Diesel Oxidation Catalyst (DOC), Diesel Particulate Filter (DPF) and Selective Catalytic Reduction (SCR). This aftertreatment system allow to reduce the two main Diesel emissions such as NOx and particulate matter, commonly called soot.

These systems are both expensive and complicate to maintain or be replaced in case of bad conditions or malfunctioning: therefore the need for more advanced combustion concepts that should concern renewable fuels to achieve lower tailpipe emissions to reduce the dependency on aftertreatment.

Latest studies are moving fast towards the use of electric vehicles (BEVs-Battery Electric Vehicles), because it's common belief that BEVs are so much cleaner than Diesel engines. Actually this statement is neither completely right or wrong: the production of electric batteries is not exactly the cleanest one and electric systems are not so comfortable for vehicles. The dimension of the batteries is a crucial point in BEVs continuous development, strictly connected to their energy density which is still far from the suitable duration for a comfortable driving journey. Moreover the most countries are not yet equipped with charging stations and the biggest vehicle batteries could maybe last for 300 km and then they need to be recharged.

That's another reason why it's still worth it to concentrate on renewable fuels and cleaner combustion engines.

The same reasoning can be applied to heavy duty engines developed for trucks: they would need more powerful, durable, long-lasting and high volume power density batteries.

Therefore, even if electric truck do exist (mostly medium duty hybrid), they will still rely in internal combustion engines in the foreseeable future.

Improving fuel efficiency and looking for alternative fuels are the two main strategies to reduce emissions of heavy-duty diesel engines.

Direttiva	Euro	Tipo test	CO (g/kWh)	HC (g/kWh)	NO _x (g/kWh)	PM (g/kWh)	NH ₃ (ppm)	Fumo (m ³)	Step 1	Step 2	
88/77/CEE	/	/	11,2	2,4	14,4		/	/	1/7/88	1/10/90	
91/542/CEE	I	/	4,5	1,1	8,0	0,36	/	/	1/7/92	1/10/93	
	II	/	4,0	1,1	7,0	0,15	/	/	1/10/95	1/10/96	
99/96/CE:	III	ESC, ELR	2,1	0,66	5,0	0,10	/	0,8	1/10/00	1/10/01	
		ETC	5,45	0,78*	1,6**	5,0	0,16	/			/
	IV	ESC, ELR	1,5	0,46	3,5	0,02	/	0,5	1/10/05	1/10/06	
		ETC	4,0	0,55*	1,1**	3,5	0,03	/			/
	V	ESC, ELR	1,5	0,46	2,0	0,02	/	0,5	1/10/08	1/10/09	
		ETC	4,0	0,55*	1,1**	2,0	0,03	/			/
	EEV	ESC, ELR	1,5	0,25	2,0	0,02	/	0,15			
		ETC	3,0	0,40*	0,65**	2,0	0,02	/			/
	595/05/CE	VI	ESC (CI)	1,5	0,13	0,40	0,01	10	/	1/1/13	1/1/14
			ETC (CI)	4,0	0,16	0,40	0,01	10	/		
ETC (PI)			4,0	0,16*	0,5**	0,40	0,01	10	/		
582/05/CE	VI	WHSC (CI)	1,5	0,13	0,40	0,01	10	8x10 ¹¹ #	1/1/13	1/1/14	
		WHTC (CI)	4,0	0,16	0,46	0,01	10	6x10 ¹¹ #			
		WHSC (PI)	4,0	0,16*	0,5**	0,46	0,01	10			/

* = idrocarburi non metanici (NMHC), ** = metano (CH₄), # = misurato in numero di particelle
 PI = accensione comandata (Positive Ignition), CI = accensione spontanea (Compression Ignition)

Figure B - EU Emission standards for HD Diesel engines

1.1 COMBUSTION ENGINE COCEPTS

In order to face emission standards, the research on Diesel combustion engines has led to increasingly innovative combustion concepts. The most important features of these different kinds of combustion, are based on injection position and timing, in order to have more mixed fuel or lower temperature. These concepts are commonly called LTC which stands for Low-Temperature Combustion.

In all LTC strategies, the combustion temperatures are reduced by dilution of the in-cylinder combustible mixtures, either with excess charge gas to create mixtures that are more fuel-lean than stoichiometric, or with moderate to high levels of EGR.

In either case, the diluent gases increase the fuel-specific heat capacity of the combusting mixtures, thereby reducing the combustion temperatures.

In the case of dilution by EGR, the oxygen concentration is also decreased, which further slows NO_x formation kinetics.

In the next chapters the focus will be on some of these prevailing concepts, which modify standard Diesel combustion, improving efficiency and maintaining combustion performances.

The development of these technologies with intrinsically lower soot and nitrogen monoxide could minimize the after-treatment system requirements, leading to lower complexity and costs.

1.1.1 DIESEL COMBUSTION

Diesel combustion is a complex, turbulent, multiphase process that occurs in high-temperature and high-pressure environment. In this type of engine, at the very beginning, only air is compressed: in the meanwhile, when the temperature reaches a certain level and the piston is near TDC (Top Dead Centre) fuel is injected into the combustion chamber. The injection timing is fundamental, it provides the right efficiencies and emissions values. The combustion starts without any spark: a highly reactive fuel is used to enable auto-ignition, after a short ignition delay period, under high pressure conditions. The engine uses a high compression ratio, high temperature and lean fuel mixtures, which lead to high NO_x emissions: in addition, the combustion passes through a temperature window that provides heavy soot emissions in the exhaust gas. That's the so called NO_x-soot trade-off which is a particular feature of Diesel Engines,

because lowering the temperature enhance soot production, while increasing temperature lead to high NOx values.

That's the reason why new combustion concepts have been developed: it is established that a lower temperature inside combustion chamber and a better air-fuel mixture represent the best conditions to improve the NOx- soot tradeoff: the following innovative combustion concepts are looking for this purpose.

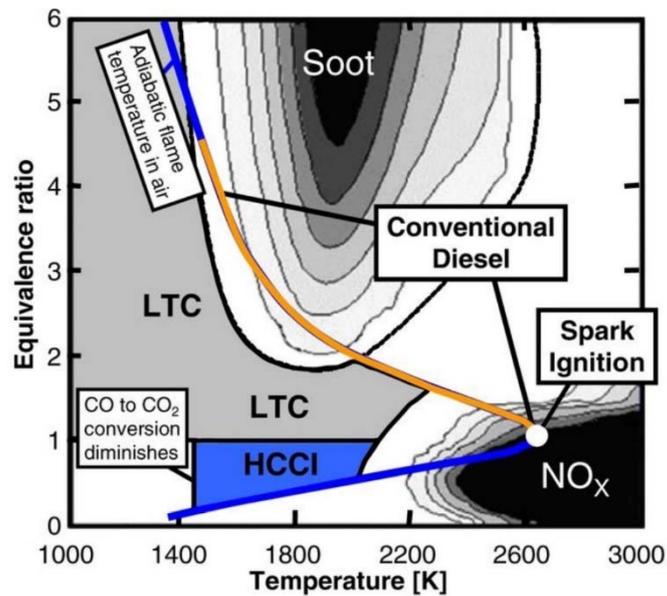


Figure 1.1: Equivalence ratio vs Temperature related to emissions

1.1.2 HOMOGENEOUS CHARGE COMPRESSION IGNITION

The first combustion concept has been introduced to apply such a premixed mixture in a high-efficient compression ignition engine and it is called Homogeneous-Charge Compression Ignition (HCCI).

HCCI is a combustion process controlled by the history of temperature, pressure and composition of the in-cylinder charge. It uses very early port fuel injection to provide the most homogeneous mixture as possible, by the end of the compression stroke, like SI engine. Good results depend on cylinder conditions, because this type of combustion is strongly relied to chemical kinetics, that's why auto-ignition time is highly dependent on cylinder pressure and temperature. The aim is to cause auto-ignition near TDC.

Clearly some draw-backs can show up: HCCI is hard to control, due to the strong dependence on in-cylinder conditions and sometimes knock occurs. It's not so common to see the auto-ignition process moving constant everywhere and that means unstable combustion. Moreover it's hard to minimize fuel condensing in the intake manifold, which can be reduce with high intake temperature.

However, this innovative combustion concept, working with leaner mixture and lower temperature than the Diesel combustion, provides some improvements: it represents a cleaner combustion mode, reducing a lot NO_x emissions (well below EURO VI limit of 0.4 g/kWh) and allowing to avoid expensive after-treatment systems.

The HCCI concept shows some limitations if compared to Diesel combustion: for example the very fast burn-rate leads to poor combustion phasing control and can cause high noise level; moreover the operating range has a short time window and the combustion efficiency is lower than Diesel engine.

In order to move forward and improve this kind of combustion, trying to maintain the primary benefits of Diesel combustion such as higher power density and better phasing combustion control, Partially-Premixed combustion concept has been developed through the years.

1.1.3 PARTIALLY PREMIXED COMBUSTION

PPC is a combination of HCCI and traditional Diesel combustion: the biggest improvement is the possibility to create a homogeneous stratified mixture before ignition, due to a longer mixing time provided by a longer ignition delay.

This important feature allow to strongly reduce the flame temperature and it reduces local fuel-rich regions, dangerous for unstable combustion and soot production.

Usually in PPC, high-octane fuel is injected closer to TDC and combustion is controlled by EGR rate: due to the lower reactivity of high-octane fuels compared to diesel, fuel and air are partially premixed before the combustion starts. Lower temperature make the fuel behave differently from Diesel combustion, in which ignition usually occurs before the end of injection.

Therefore ignition delay, which is defined as the crank angle difference between start of injection and start of combustion, is a key parameter for PPC.

As already said, temperature can be reduced using a high percentage of cooled exhaust gas recirculation (EGR), which brings to reduce NO_x emissions and to a longer ignition delay slowing down the kinetic reactions.

Knocking down the NO_x-soot trade-off is actually possible with PPC, but significant loss in terms of combustion efficiencies occurs and other emissions such as UHC (Uncombusted Hydro Carbon) increase, due to the wall wetting from early injection.

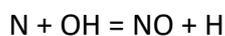
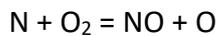
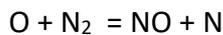
However, PPC represents the latest and more successful combustion concept among the LTC family, showing a wide operating range for an extremely balanced combustion. In the next chapters, the investigation of a huge amount of parameters for PPC combustion on a heavy duty diesel engine is described: the best ways of looking for the right NO_x-soot trade off and higher efficiencies, controlling a whole set up from EGR rate to SOI (Start of Injection) at different loads and for different fuel blends are presented.

1.1.4 EMISSIONS

In this paragraph, a quick review of the most common exhaust gas pollutants will be given, referring mostly to Diesel combustion.

Nitrogen Oxides (NO_x) and Particulate Matter (Soot) are usually detected at higher percentage after Diesel combustion, while Unburned Hydrocarbon (UHC) and Carbon Oxide (CO) have lower percentage. This trend is the opposite for gasoline spark ignition (SI) engines.

- Nitrogen Oxides (NO_x) → nitric oxide (NO) and nitrogen dioxide (NO₂) are usually grouped together as NO_x and NO is the predominant oxide of nitrogen produced inside the engine cylinder. The mechanism of NO formation from atmospheric nitrogen is well explained by Zeldovich mechanism, which states that in combustion of near-stoichiometric fuel-air mixtures the principal reactions governing the formation of NO from molecular nitrogen are:



The in-cylinder environment is prosperous for NO_x formation: Diesel engines work with air excess and lean mixtures, related to high pressure which lead to high temperature that enhance the chemical reaction of air nitrogen with oxygen.

Diesel engines presents extremely complex processes such as fuel-air mixing and combustion. During the premixed and uncontrolled combustion phase, immediately following the ignition delay, the composition near stoichiometric ratio burns due to spontaneous ignition flame propagation. After that, during the mixing controlled combustion phase, the burning mixture is closer to stoichiometric (turbulent, unsteady, diffusion flame). The critical equivalence ratio (ϕ) for NO formation in high-temperature, high pressure burned gases is close to stoichiometric ($0.85 \leq \phi \leq 1.1$). High temperature provides higher NO_x formation, mostly between start of combustion and shortly after the peak cylinder pressure, where burned gas temperatures reach their maximum value.

Subsequently the decreasing temperature due to expansion freezes the NO chemistry.

In order to avoid NO_x formation, the combustion should be more homogeneous as possible, avoiding peak temperature and slowing down the combustion, which is basically upstream the common Diesel combustion. That's the reason why Diesel engines present more NO_x than SI engines.

- Particulate Matter (soot) → Diesel particulate mostly consists in carbon agglomerates, generated by carbonaceous material (soot) on which some organic compounds have become absorbed.

Soot formation takes place in Diesel combustion environment, pressures between 50 to 100 atmospheres and temperatures from 1000 to 2800 K. Very short time is required for soot formation, in the order of milliseconds.

The first stage in soot formation involve the first condensed phase material which comes from oxidation of fuel molecules: this products are typically made of unsaturated hydrocarbon (such as acetylene and polycyclic aromatic hydrocarbons) and they are considered as the most likely precursor of soot in flames. The first particles have a very little diameter, smaller than 2 nm.

The second stage involve particle growth, both coagulation and aggregation: gas-phase species contribute to form the bulk of the solid-phase material. Surface growth reactions lead to increase the amount of soot, the number of the particle remain the same.

In the exhaust, an additional mass fraction comes from dilution: the exhaust gases are cooled and diluted with air. Adsorption and condensation occur in the dilution tunnel and in the exhaust system.

The measurement of particulate emissions is conducted either by diluting the whole exhaust in a dilution tunnel with constant volume sampling or by extracting a small proportional part of the exhaust gas and diluting it. Particulate emissions are measured by passing part of the diluted exhaust aerosol through a filter paper. The increase of the weight of the filter is used to calculate the particulate matter mass (PM) emissions.

Particulate matter comes easy from incomplete combustion conditions which come from a non-homogenous fuel mixture inside the cylinder that provides fuel-rich regions, especially at high loads. The fuel need more time as possible for a complete combustion.

Some of the main causes of soot formation are the following:

- . Regions of high fuel concentration (especially at high loads)
- . Heterogeneous combustion process
- . High equivalence ratio (ϕ)
- . SOI retarded
- . CA50 retarded

LTC concepts provide a solution to decrease soot formation, lowering temperature and equivalence ratio and creating a more homogeneous fuel combustion, compared with Diesel.

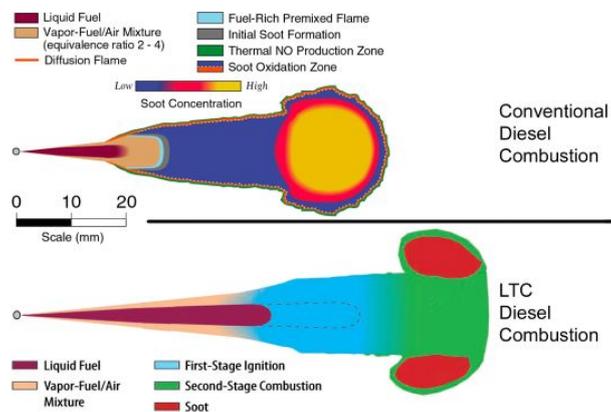


Figure 1.2: Fuel droplet from conventional Diesel combustion and LTC combustion

- Hydrocarbons (UHC) → Diesel fuel shows a heterogeneous combustion, which lead to a more complex composition if unburned and partially oxidized hydrocarbons in the exhaust gas and extends over a larger molecular size range.

As a result of complexity, diesel combustion has been divided into four stages:

- 1) Ignition Delay → the time between the start of injection and start of ignition
- 2) Premixed combustion phase → during which the fuel that has mixed to within combustible limits during the delay period burns
- 3) Mixing controlled combustion phase → the rate of burning depends on the rate of fuel-air mixing
- 4) Late combustion phase → mixing of residual combustibles, excess of oxygen and low rate of heat release that provides kinetics of oxidation

Fuel unburned can occur if fuel-air mixture is too lean to auto-ignite or maintaining the flame propagation, or too rich to ignite during the first part of the combustion.

Another important cause is flame quenching, which comes from fuel remaining on the cylinder walls and get scratched away by the piston during cycles. Flame quenching is mostly caused by wrong injection timing, especially too advanced SOI. Instead, if SOI is retarded, under-mixing is enhanced and fuel can't burn properly because the combustion is already in a late phase.

Under-mixing (fuel that leaves fuel injector at low velocity, late during the combustion), over-leaning (fuel mixed to leaner conditions than the limits) and flame quenching cause hydrocarbons presence in the exhaust gas.

Over-leaning zones are usually located at the boundaries of the real schematic behaviour of fuel spray, as it is shown in the figure above. In a swirling flow, ignition occurs first in the spray core: the fuel at a certain distance from the core has already mixed beyond the lean limit and in these regions there will be unburned fuel. The amount of HC in the exhaust gas depends on the amount of fuel injected during ignition delay time and the mixing rate in this period. Certainly there is a strong correlation between ignition delay and HC values.

It's important to keep the ignition delay as shorter as possible and control the injection timing, not too late (unburned fuel) and not too advanced (flame quenching). The equivalence ratio doesn't affect HC emissions, until a critical value of 0.9 is reached: from that value, an increasing equivalence ratio will bring critical HC levels.

- Carbon Monoxide (CO) → CO emissions from Diesel are low enough to be unimportant. That is because carbon monoxide production comes mostly from high equivalence ratios that never occur inside a Diesel engine which works with lean fuel mixture.

1.2 BIOFUELS

1.2.1 GENERAL CHARACTERISTICS

Research on biofuel systems covers sustainability aspects in the supply chain from biomass cultivation, harvesting and transport via conversion processes to distribution and end use.

A great number of fuel additives are currently produced based on fossil feedstock, but could be replaced by renewables.

As biofuels are of high interest for reasons such as climate change, security of supply and economy the demand for the different fuels is steadily increasing from both governmental and users as passengers and heavy duty vehicles.

Systems analysis of biofuel production basically covers the whole chain from biomass resource extraction via processing and distribution to end-use.

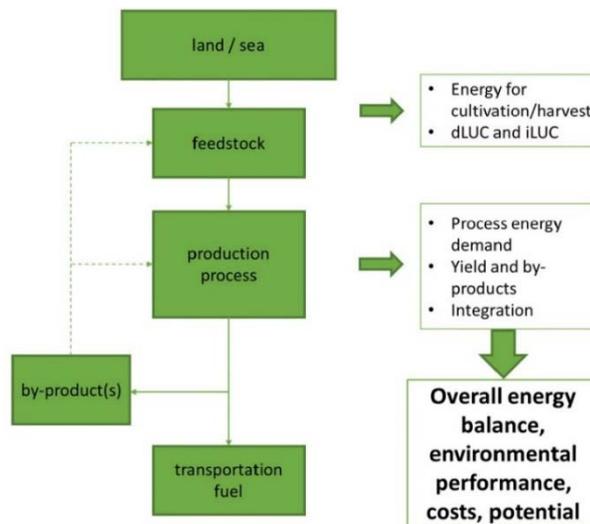


Figure 1.3: Major steps accounted for biofuel production systems analysis.

dLUC= direct land use change iLUC= indirect land use change

Basically four major types of biofuel feedstock exist:

- Sugar-based feedstock (e.g. sugar cane or sugar beet)
- Starch-based feedstock (e.g. corn and other cereals)
- Oil-based feedstock (e.g. palm oil, animal fats and others types of oils)
- Lignocellulosic feedstock (e.g. wood, forest residues)

In addition, related to the feedstock, three principal process platforms for biofuel processing exist:

- Thermochemical conversion (suitable for lignocellulosic material)
- Biochemical conversion (e.g. fermentation, anaerobic digestion. Suitable for sugar and starch-based feedstock)
- Hydrogenation (conversion of oil-based materials, such as vegetable oils or animal fats to fuel)

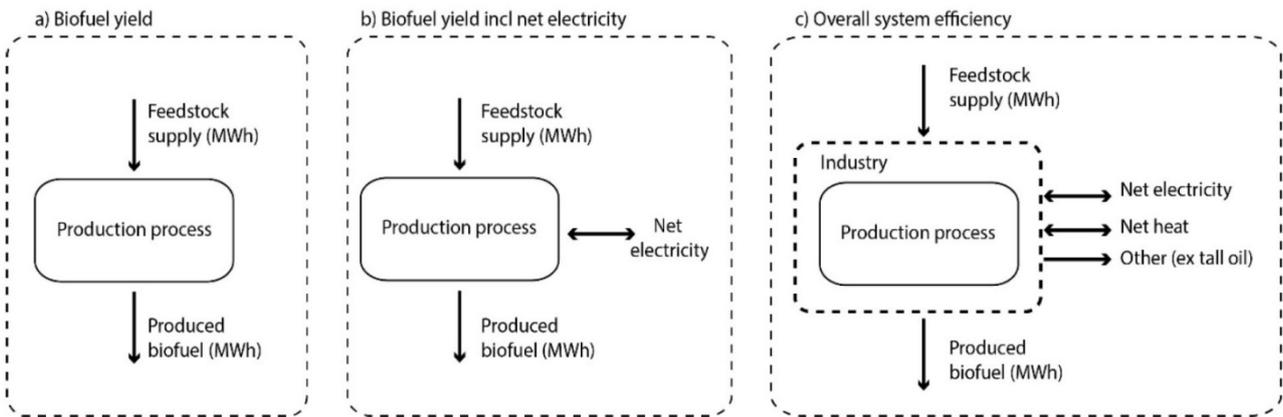


Figure 1.4: Schematic representation of biofuel production

The simpler and schematic representation of the biofuel production is shown above: the production process are also designed to produce net electricity and heat besides the main objective of biofuels.

It is possible to have an extended vision of the production processes representing a scheme in which feedstock and processes are completely shown as a block diagram.

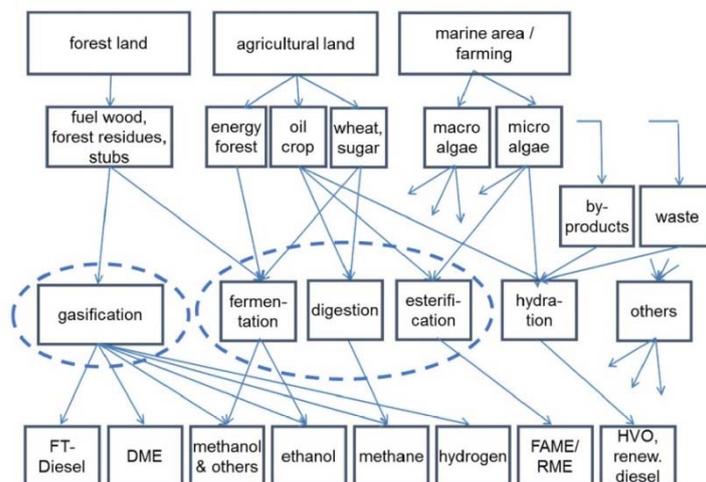


Figure 1.5: Biofuels feedstock and process block diagram

The constant research on biofuels is propelled by the knowledge about fossil fuels closing end, reinforced by environment care: it's well known that great emissions come from all types of fossil fuels and petroleum precursor and the common idea is to move towards a solution that provides cleaner engines related to cleaner fuels.

The variations and usage of alternative fuels is vast for the liquid ones: lots of different fuel configurations have been tested and used over the years.

Biodiesel is the most widely used as fuel, starting from low grade blend (7%) in standard diesel, up to higher percentages and also to 100% neat fuel. The most common biodiesel is called FAME (Fatty Acid Methyl Esters), typically produced by an alkali-catalyzed reaction between fats and methanol.

Nowadays there is plenty of renewable sources that could be used as fuels base, include alcohols (methanol, ethanol, propanol, butanol), derivatives of alcohols and different ethers. These kind of fuels can't run an engine by themselves, they need to be blended to fossil fuels, mostly for a matter of reactivity in Diesel engines (there is no spark ignition, more reactivity of the fuel is required).

1.2.2 COMBUSTION REACTIVITY

The most important quality for the fuels is how good and how fast they burn inside the cylinder. A fuel that doesn't burn properly is useless, that is why combustion reactivity is the first property to be investigated.

In order to see how a fuel would be prone to auto ignition, two parameters need to be investigated:

- RON (Research Octane Number) → it represents the measurement of combustion reactivity for gasolines. The higher the RON, the lower the reactivity (the gasoline is more resistant to auto ignition)
- CN (Cetane Number) → combustion reactivity parameter for Diesels. The higher the CN, the higher the combustion reactivity. A fuel with high CN burns easily than another one with lower CN value.

So there is an inverse correlation between RON and CN. Usually gasolines need to be resistant to auto ignition to avoid knock, which represents an usual phenomenon due to too much powerful combustion, with unstable flame that causes literally an explosion above the piston.

The tests on RON number are based on the so called PRF (Primary Reference Fuel), which are n-heptane and iso-octane. They have got respectively 0 and 100 RON values.

The CN of a fuel is measured by comparing its ignition characteristics with reference fuels in a single-cylinder diesel engine. The reference scale is based on normal cetane (n-hexadecane) that shows a CN value of 100.

It is also possible to conduct some tests looking at the ignition delay, which is basically directly related to the properties of auto ignition of a fuel, in terms of time.

Normally is defined that a fuel falls in the Diesel auto ignition range if $CN > 30$ and belongs to the gasoline auto ignition range if $RON > 60$.

Diesel fuels have a CN range from 40 to 60 and gasolines have RON range between 90 and 100.

The emergence of low temperature combustion strategies, particularly the ones that concern different ways of chemical reactivity, controlled ignition timing and combustion like PPC (Partially Premixed Combustion), offers the potential to significantly improve operating efficiency and

reduce emissions. Combustion reactivity is one of the most important property to choose which fuels are better to be tested and what type of combustion control they would need.

The possibility of blending alternative fuels with petroleum-derived fuels, depends on their combustion reactivity.

Diesel RON is in a range between 15 and 25, and the CN is about from 45 to 55. For example, butanol CN is about 25. That means a higher reactivity compare to these alternative fuel alcohols: these fuels can't burn as neat fuel for a Diesel engine but they could be blended with Diesel or others petroleum-based fuels. Researchers are looking into the real possibility in blending fuels in order to get lower emissions at the same performances: cleaner fuels for cleaner engines.

In the next chapters there will be some results shown, which come from several tests on one if these alcohols blended with n-heptane as a high reactivity based fuel in a certain percentage. The results involve emissions properties if this blend under different combustion control parameters.

1.2.3 CHARACTERISTICS OF TEST FUELS

In the next part, test fuels used for the tests on the heavy duty diesel engine will be described.

The focus is on creating different fuel blends that could provide relevant combustion results in terms of control, emissions and efficiency: in particular the effect of chemical structure and stoichiometric air/fuel ratio on emissions and combustion control are investigated for three different fuel blends. The blends need to be suitable for a heavy duty diesel engine for PPC application strategy.

The straight-chain alkane n-heptane chemical formula is C_7H_{16} and has the important property of being a PRF as a 100% n-heptane fuel shows RON=0.

Due to its high reactivity, n-heptane presents a really good base to be blended with alternative fuels which are way less reactive: n-heptane burns really fast and it's dangerous for gasoline engines because knock can occur. It is perfectly suitable, instead, for diesel engines.

Combustion strategy concerned the LTC concept called PPC: the main benefit is to concurrently reduce flame temperature and allow more fuel-air premixing to produce both low NO_x and soot emissions.

Some binary oxygenated fuels such as alcohols and other organic compounds are investigated on their chances to be blended with a higher reactivity fuel base, such as n-heptane.

Alcohols chemical structure presents carbon, hydrogen and oxygen with one particular functional group –OH. They are organic compound too, but with a particular structure. Organic compounds present carbon, hydrogen and oxygen in their structures as well, but without the –OH group. In particular, the table below shows a list of binary oxygenated fuels (except for toluene and iso-octane which are aromatic hydrocarbons), that will be investigate and analysed in terms of combustion properties such as (A/F)_{st}, RON, LHV (Lower Heating Value) and density.

Fuels	(A/F) _{st}	RON	LHV [<i>MJ/Kg</i>]	DENSITY [<i>Kg/m³</i>]
N-HEPTANE	15,2	0	44,6	684
BUTANOL	11,1	96	34,4	810
ETHANOL	9,0	108	27,0	794
METHANOL	6,5	136	20,1	792
TOLUENE	11,5	121	40,6	867
ANISOLE	10,8	112	33,5	995
BENZYL-ALCOHOL	10,8	108	33,1	1044
ISO-BUTANOL	11,2	113	33,0	802
TERT-BUTANOL	11,1	103	33,0	791
BENZALDEHEYDE	10,4	120	33,2	1040
ISO-OCTANE	15,1	100	44,8	690
DIESEL	14,4	20	42,6	832

Table 1A: Alternative fuels characteristics

The base fuel of the different blends is always n-heptane, because of its higher reactivity: without it, the other fuels listed in the table above are very difficult to ignite. That's why n-heptane represents the starting point for the fuel blends: the analysis is conducted calculating the different values of (A/F)_{st}, RON and LHV for different blending ratios for each fuels related to n-heptane, using a developed Matlab code.

1.3 AIM OF THE THESIS PROJECT

The aim of the research is to investigate the influence of stoichiometric air/fuel ratio and chemical structure of the fuels related to NO_x and soot emissions.

PPC has great potential to achieve high engine efficiency and very low emissions of NO_x and soot: the need of specific fuel properties for PPC cannot be fulfilled by using a single component fuel. That's why test fuels use PRF blends as primary fuels and organic compounds (including alcohols) to have specific RON and $(A/F)_{st}$ but different structures. It is also investigated the case of very similar molecular structures of the blends but different $(A/F)_{st}$, in order to analyse both chemical structure and $(A/F)_{st}$ influence on emissions.

Alcohols have shown good prospects for application in PPC because of their higher latent heat of vaporization and lower reactivity: even at high loads, alcohols provide short ignition delay which is comparable to diesel but they still help to reduce soot emissions.

The first step is to identify test fuel blends, using PRF n-heptane base blended with organic compounds and alcohols, calculating $(A/F)_{st}$ values and RON. The calculation is done for different blend percentages for all the fuels.

The aim is to find three binary oxygenated fuel blends, with two of them showing the same $(A/F)_{st}$ and different molecular structure and other two showing very similar molecular structure but different $(A/F)_{st}$.

The target RON is set around 70, because higher octane number fuels are considered problematic with low load and idle conditions. The main focus will be on $(A/F)_{st}$ and structure between the blends, RON from 70 to 80 will be accepted as well: it comes from an approximate calculation, based on a weighted average between RON of n-heptane and RON of the organic compound depending on blending ratio (chemical tests in the lab would be more precise but it would have been much more time consuming).

As an example, in the table below, n-heptane/butanol blend characteristics are shown, from 10% to 50% n-heptane and from 90% to 50% butanol. It's easy to notice that, due to the lower reactivity of butanol compared to n-heptane, if butanol percentage decreases, RON decreases as well, while $(A/F)_{st}$ increases.

Heptane (%)	1-butanol (%)	A/F st	LHV	RON
10	90	11,5008	34,0607	86
15	85	11,6753	34,5653	82
20	80	11,8526	35,0780	77
25	75	12,0329	35,5990	72
30	70	12,2161	36,1286	67
35	65	12,4024	36,6668	62
40	60	12,5918	37,2139	58
45	55	12,7844	37,7702	53
50	50	12,9803	38,3359	48

Table 2B: Butanol-Heptane blend

Organic compounds percentage must not exceed a certain percentage in the blend, otherwise the fuel would have a too much high RON and would be too difficult to ignite and control during combustion.

The results don't show three blends with the same (A/F)st, but two of them present very similar values with reasonable blending ratio: the first one is indicated by the red line in the table above, referred to butanol blend. The second one involves benzaldehyde blended with n-heptane, shown in the table below.

Heptane (%)	Benzaldehyde (%)	(A/F)st	LHV	RON
10	90	10,7113	34,0253	108
15	85	10,8817	34,4313	102
20	80	11,0584	34,8522	96
25	75	11,2418	35,2888	90
30	70	11,4322	35,7422	84
35	65	11,6299	36,2131	78
40	60	11,8356	36,7027	72
45	55	12,0496	37,2122	66
50	50	12,2724	37,7427	60

Table 1C: Benzaldehyde-Heptane blend

As it is presented, the (A/F)st is around 11.8 for both the blends and RON is slightly higher than 70: these are suitable conditions for PPC application in a diesel engine.

Butanol blending percentage is quite high in the first blend: that fact could lead to unstable combustion at low loads but it's interesting to investigate also this possibility to test limit of organic compounds before misfire.

The other solutions show lower or higher values of (A/F)_{st} compared with these first two: it might be worth it to investigate chemical structure, looking for a similar blending structure with butanol or benzaldehyde.

The blending ratio is also important: there is already a high organic compound ratio provides by butanol blend (80%) so now it's required to remain around 60% like in the benzaldehyde case.

The table below shows anisole properties: its chemical structure presents a benzene ring and oxygen just like benzaldehyde, so their structures are quite similar (C₇H₆O for benzaldehyde and C₇H₈O for anisole).

Heptane (%)	Anisole (%)	A/F st	LHV	RON
10	90	10,8449	34,2853	100
15	85	10,8520	34,6972	95
20	80	10,8600	35,1229	90
25	75	10,8690	35,5630	84
30	70	10,8792	36,0183	78
35	65	10,8909	36,4896	73
40	60	10,9045	36,9777	67
45	55	10,9205	37,4835	62
50	50	10,9395	38,0081	56

Table 1D: Anisole-Heptane blend

The three fuel blends used for the tests on the heavy duty diesel engine are the following:

- 1) 20% heptane – 80% butanol
- 2) 40% heptane – 60% benzaldehyde
- 3) 40% heptane – 60% anisole

CHAPTER 2 – EXPERIMENTAL SETUP AND MEASUREMENTS

2.1 ENGINE AND SETUP

The test setup used, referred to as CYCLOPS, based on a 12.6L six-cylinder DAF diesel engine, model XE355c. Cylinder 1 is isolated as test cylinder, cylinder 4, 5 and 6 are operated by the stock engine control unit and used to propel the engine, even when the test cylinder is not fired. Cylinders 2 and 3 are empty, without pistons.



Figure 2.1: CYCLOPS setup



Figure 2.2: Engine



Figure 2.3: CYCLOPS setup, hydraulic dynamometer and smoke meter

The figure below shows the schematic overview of the test engine.

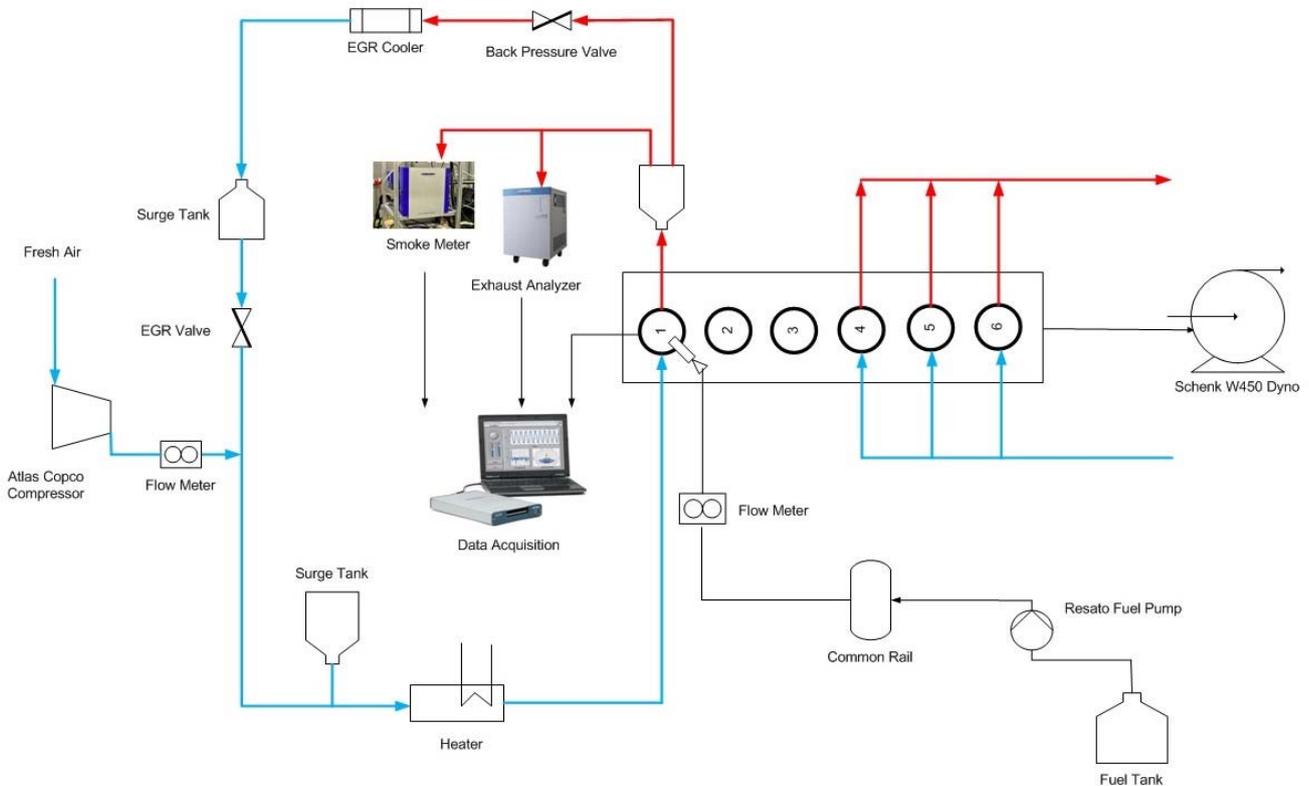


Figure 2.4: Schematic overview of the CYCLOPS setup

A standalone fuel injection system propels the test cylinder, which is composed of a Resato double acting air-driven high-pressure fuel pump, a pressure regulator and a prototype Delphi injector. The Atlas Copco air compressor provides up to 5 bar of intake air pressure, while exhaust pressure is set to 0.3 bar higher than intake pressure (the reason stands on a correct use of EGR). EGR flow is cooled by an EGR cooler using a variable flow of process water as a coolant medium, temperature can be lowered to 30°C. After the EGR cooler there is a cooled EGR tank, which acts like a intermediate storage for the cooled flow, before the flow pressure drops in the EGR valve. The cooled EGR flow is mixed with the fresh intake air from the air compressor, inside a second tank located before a heater that heats the intake charge to the desired temperature. The air-circuit and fuel-circuit of cylinder 1 is separated from the other cylinders. The fuel is directly injected inside the test cylinder using a single ncv (nozzle check valve) 2mm injector.

Engine geometrical characteristics are listed in the table below.

Bore [mm]	130
Stroke [mm]	158
Connecting rod [mm]	266,7
Compression ratio [-]	15,7
Bowl shape	M-shaped
Bowl diameter [mm]	100
Number of valves	4
Exhaust valve close (EVC) [°CA]	-346
Intake valve close (IVC) [°CA]	-153
Exhaust valve open (EVO) [°CA]	128
Intake valve open (IVO) [°CA]	344

Table 2A: Geometrical properties of the engine

The compression ratio of the test cylinder has been altered from the original 17:1 to 15,7:1 to allow longer fuel-air mixing.

The engine is connected to a Schenk W450 hydraulic dynamometer: it can be used to measure force, torque and power (for CYCLOPS it is not useful for that purpose because there are the three cylinder propelled by diesel connected to it and not only the test cylinder). It works as a hydraulic brake connected to the crankshaft.



Figure 2.5: Resato fuel pump



Figure 2.6: Schenk W450 hydraulic dynamometer

2.2 EMISSION MEASUREMENT SYSTEM

The exhaust system of the test cylinder is separated from the other diesel cylinders.

Exhaust gas from the first cylinder goes directly into a tank and then it is recirculated into the EGR cooler: part of it is sent to the emission measurement systems if the setup. These systems include a Horiba measurement system, a smoke meter and the engine exhaust particle sizer (EEPS).

These devices detect HC, CO, NO_x and soot emissions, analysing exhaust gas samples after test cylinder combustion and they are directly connected to the data acquisition system inside the engine control room.

2.2.1 HORIBA MEASUREMENT SYSTEM

In order to fight exhaust emissions, research and development into the engine periphery is being vigorously pursued. To facilitate engine tests, there is a strong need for a measurement system capable of adapting to changing test requirements. The Horiba Mexa 7100 DEGR exhaust analysis system can measure HC, CO and NO_x with three different methods for each emission respectively:

Flame ionization method (FID) → The sample gas is introduced into a hydrogen flame inside the FID. Any hydrocarbons in the sample will produce ions when they are burnt which are detected using a metal collector which is biased with a high DC voltage. The current across this collector is thus proportional to the rate of ionisation which in turn depends upon the concentration of HC in the sample gas.

Non-dispersive infra-red method (NDIR) → Each constituent gas in a sample will absorb some infra red at a particular frequency. By shining an infra-red beam through a sample cell (containing CO or CO₂) and measuring the amount of infra-red absorbed by the sample at the necessary wavelength, a NDIR detector is able to measure the volumetric concentration of CO or CO₂ in the sample.

Cheluminescent method → The reaction between NO and O₃ (ozone) emits light. This reaction is the basis for the CLD in which the photons produced are detected by a photo multiplier tube (PMT). The CLD output voltage is proportional to NO concentration.

2.2.2 SMOKE METER

Increasingly stringent emission regulations for combustion engines are the main drivers for high-performance emission measurement instruments. The measurement challenges of engines equipped with exhaust after-treatment systems are high pressures and temperatures at the exhaust sampling point combined with high emission rates.



Figure 2.7: AVL smoke meter

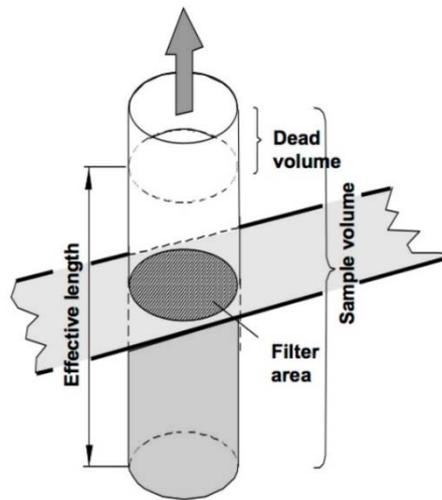
The AVL Smoke Meter uses the filter paper method to determine the Filter Smoke Number (FSN defined according to ISO 10054) and the soot concentration in mg/m^3 .

This smoke meter has a measurement range of 0 to 10, in which 0 corresponds to the original reflection of the white paper and 10 corresponds to no light reflection.

The sampled engine exhaust gas passes through clean filter paper: soot causes the blackening of the filter paper. The paper blackening is measured by a photoelectric measuring head and the result is analysed by a microprocessor.

By default the AVL Smoke Meter is equipped with a remaining filter paper indicator on the front of the housing. The led is clearly visible and indicates when the paper supply is low, making it easy to tell whether the paper roll should be replaced.

The exhaust sampling scheme of the smoke meter is shown in the figure below.



$$\text{Effective length} = \frac{\text{Sample volume} - \text{Dead volume} - \text{Leak volume}}{\text{Filter area}}$$

Figure 2.8: Exhaust sampling

Some technical data are also presented in the following table.

Measurement principle	Measurement of filter paper blackening
Measured value output	FSN (filter smoke number), mg/m ³ (soot concentration)
Measurement range	0 to 10 FSN
Detection limit	0.002 FSN or 0.02 mg/m ³
Resolution	0.001 FSN or 0.01 mg/m ³
Exhaust pressure ranges	(-300*) -100 to 400 mbar (-500*) -200 to 750 mbar with the special sampling option 0 to 3000 mbar with the high-pressure option (* with activated altitude simulation)
Maximum exhaust temperature	600 °C with standard 340 mm sample probe (800 °C with 780 mm long sample probe)

Figure 2.9: Technical data

2.2.3 ENGINE EXHAUST PARTICLE SIZER

The engine exhaust particle sizer (TSI EEPS 3090) spectrometer is a fast-response, high resolution instrument that measures very low particle number concentrations in diluted exhaust. It is used in the experiments to obtain detailed information about particle concentration and size distribution. It is well-suited for dynamic and transient test cycles, with real-time data collection: it measures the size distribution and number concentration of engine exhaust particle emissions in a range from 5.6 to 560 nanometers.



Figure 2.10: EEPS

The EEPS works charging positively the particles: these are then introduced to the measurement region near the centre of a high-voltage electrode column. A positive voltage is applied to the electrode and creates an electric field that repels the positively charged particles outward according to their electrical mobility. Charged particles strike the respective electrometers and transfer their charge: a particle with higher electrical mobility strikes an electrometer near the top, whereas a particle with lower electrical mobility strikes an electrometer lower in the stack. This multiple-detector arrangement using highly sensitive electrometers allows for simultaneous measurements of multiple particle size.

A data rate of 1 Hz (electrometers currents are read 1 times per second) enables to identify and correlate particle emissions with specific engine events during the cycle.

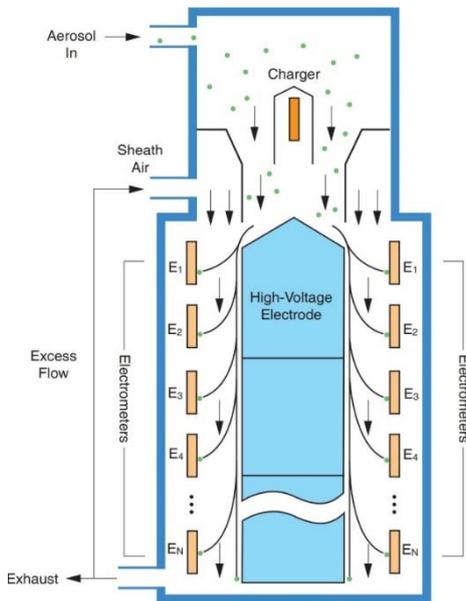


Figure 2.11: EEPS operation process

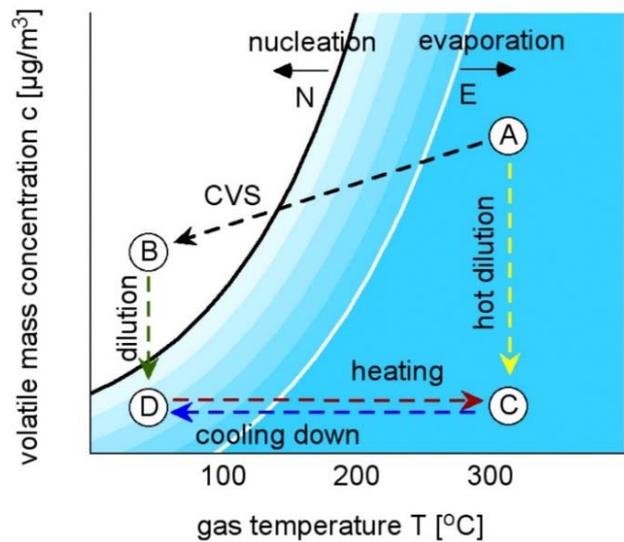


Figure 2.12: Dilution effect

Hot engine exhaust contains both solid particles (for example carbonaceous soot and ash) and volatile vapours (such as water, sulfate and hydrocarbons). In these experiments, a constant volume system (CVS) tunnel is used for emission testing: that means volatile substances have a chance to condense into droplets, which are detected, as particle together with non-volatile solid particles. In order to measure only the solid fraction, dilution and thermal condition is required. Dilution can decrease the number concentration of droplets formed in the CVS but it is unable to evaporate them. (passing through the CVS lead to decrease concentration and temperature of the volatile compounds, it is already a first dilution). The second dilution keeps the temperature constant but decreases the volatile mass concentration further. However, the sample needs to be heated above the evaporation point of the volatile compound, after the second dilution, using a thermal conditioner. Before entering the EEPS, the volatile compounds remain in the vapour phase during the cooling process and don't cross the nucleation curve.

In these experiments, real-time sampling rate and a sampling period of EEPS is set at 1 Hz and 1 minute respectively. The overall dilution ratio is 355.

2.3 DATA ACQUISITION SYSTEM

The engine is equipped with all the required sensors for engine tests, such as intake and exhaust pressures, intake and exhaust temperatures, cooling water and oil. These quasi steady-state engine data, together with air and fuel flows and emission levels are recorded at 20 Hz for a period of 40 seconds by means of an in house data acquisition system called TUEDACs. The average of these measurements is taken as the value for the operating point under investigation.

Recording and processing the so-called fast changing parameters is on the system SMETEC Combi combustion analyser: these fast data, such as in-cylinder pressure (measured by an AVL GU21C piezo-electric pressure transducer), common rail fuel pressure and injector current, are fundamental to understand the cyclic variations on each experimental point.

All of the channels are logged at 0,1°CA increments for 50 consecutive cycles. From these data, the average and standard deviation of important combustion parameters, such as CA10, CA50 and IMEP (Indicated Mean Effective Pressure) are calculated online by SMETEC software.

For the final analysis, the data re post-process with a custom Matlab script.

CHAPTER 3 – PARAMETERS BACKGROUND

3.1 FEATURED COMBUSTION PARAMETERS

The aim of the tests on the heavy duty diesel engine is to control the combustion parameters in order to acquire target information, which allows to study and analyse the engine performances in terms of combustion and emissions.

Therefore, it is important, before the tests, to define which combustion parameters will be controlled and acquired real-time (while the engine is running) and which ones will be investigated with the post-processing analysis.

In the next part, these different parameters will be described: the results of the research are based on the three different fuel blends (designed in the first chapter) behaviour related to the controlled combustion parameters.

3.1.1 COMBUSTION PHASING – CA50

Advanced model-based combustion control strategies are typically used to maintain optimal combustion timings, regulate cycle-to-cycle variation (CA50 and IMEP) and adjust combustion rate to control peak pressure and PRR (Pressure Rise Rate).

Optimum combustion timing can decrease the emission levels (HC and CO emissions), reduce the cyclic variability and extend the PPC operating range. Crank angle position corresponding to 50% heat release (CA50) is the most commonly used combustion phasing parameter. In PPC combustion strategy, combustion phasing is also a control variable for engine combustion control: the optimum CA50 indicates the best combustion conditions in terms of stability and efficiency.

The control of the CA50 is directly related to the SOA (Start Of Actuation) parameter, which allows to moving the combustion advanced or delayed, changing temperature and pressure inside the cylinder.

The faster calculation of CA50 can use signals from sensors inside the cylinder, such as ion current, knock and microphone sensors. It is possible to calculate CA50 from cylinder pressure: CYCLOPS setup has an in-cylinder piezo-electric pressure transducer and the fast data acquisition system (SMETEC Combi) can calculate real-time the CA50 position during the combustion.

The need to underline combustion phasing as an important combustion control parameters, comes from tests strategy: before testing the three alternative fuels at different loads (8, 12 and 16 bar IMEPg), the optimum CA50 needs to be find. Once the optimum CA50 is set, during the test it will not change: it will remain constant while the other parameters will change to analyse the combustion.

The research of the optimum CA50 and the importance of phase combustion changing will be investigated closer in the next chapters.

3.1.2 GROSS INDICATED MEAN EFFECTIVE PRESSURE - IMEPg

Normally, if torque and displacement of an engine are known, it is a very practical way to calculate the Brake Mean Effective Pressure (BMEP), for studying the performance of an engine. The CYCLOPS layout does not allow the use of BMEP because the measured torque from the dynamometer is not directly related to the test cylinder but it refers to the whole setup (including the other three cylinder). That is the reason why, in this research, the IMEP value is used, which does not consider the mechanical efficiency.

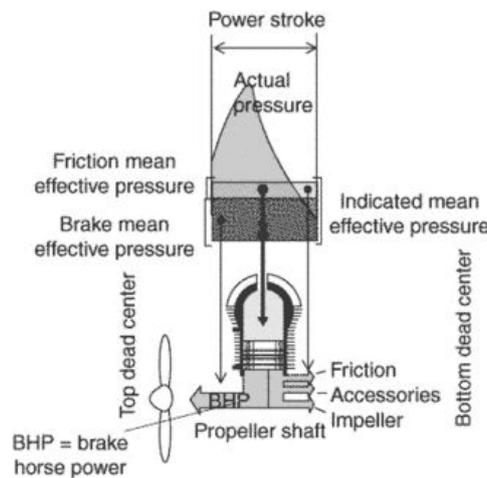


Figure 3.1: IMEP representation

The net indicated mean effective pressure (IMEP_n) is calculated from in-cylinder pressure over the complete engine cycle (720° in a four-stroke engine), where P is the in-cylinder pressure and V_d is the displacement volume.

$$IMEP_n = \frac{\int_{-360}^{360} P \cdot dV}{V_d}$$

In this research, to evaluate the combustion performance at different intake pressures and exhaust pressures, the gross indicated mean effective pressure (IMEPg), which excludes the gas exchange stroke (it refers to compression and expansion stroke only, which is 360° in a four-stroke engine), is used as the engine reference load and calculate the indicated emissions.

$$IMEPg = \frac{\int_{-180}^{180} P \cdot dV}{V_d}$$

3.1.3 AIR EXCESS RATIO - λ

Engine air-fuel ratio is defined as the mass flow rate of fresh air to fuel and it is called α .

$$\left(\frac{A}{F}\right) = \frac{\dot{m}_a}{\dot{m}_f} = \alpha$$

The air excess ratio is defined as:

$$\frac{(A/F)}{(A/F)_{st}} = \lambda$$

Diesel engines normally work with lean combustion, which means $\lambda > 1$: compression ignition engines need an air excess to work properly and this leads to high NOx and soot, with low HC and CO emissions.

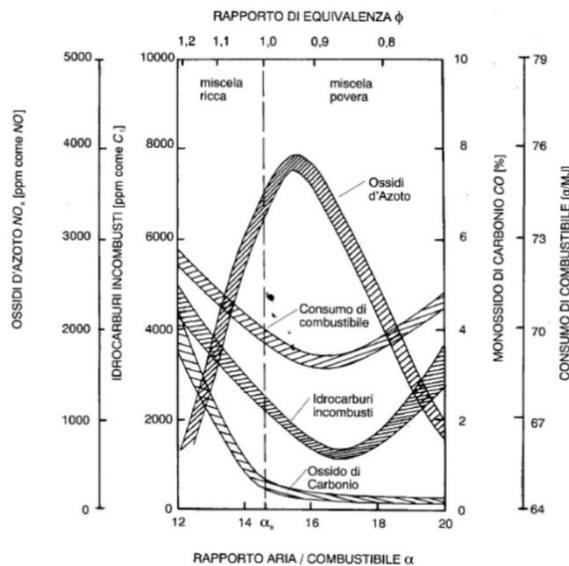


Figure 3.2: Emissions trend with AFR

The air-fuel ratio is a key parameter in LTC methods, which are designed to improve the NO_x-soot trade off lowering the combustion temperature and burning cleaner fuels.

The λ value is directly connected to the EGR (Exhaust Gas Recirculation) ratio, defined as the mass flow rate ratio of EGR to the sum of fresh air and EGR. It is also experimentally determined from the ratio of the CO₂ volume concentration in the intake gas to the CO₂ volume concentration in the exhaust gas.

If EGR rate increases, λ increases too and temperature decrease fast. During the tests, EGR rate will be changed between certain value to investigate how the emissions behave when the combustion is richer or leaner at the same load.

The calculation of λ in this research is based on Brettschneider equation, which provides the λ -value based on exhaust emissions and fuel composition.

$$\lambda = \frac{[CO_2] + \left[\frac{CO}{2}\right] + [O_2] + [NO_2] + \left(\frac{H_{cv}}{4} \times \frac{3.5}{3.5 + \frac{[CO]}{[CO_2]}} - \frac{O_{cv}}{2}\right) \times ([CO_2] + [CO])}{\left(1 + \frac{H_{cv}}{4} - \frac{O_{cv}}{2}\right) \times ([CO_2] + [CO] + (C_f \times [HC]))}$$

H_{cv} is the atomic ratio of hydrogen to carbon in the fuel, O_{cv} is the atomic ratio of oxygen to carbon in the fuel and C_f is the number of carbon atoms in each of the hydrocarbons molecules being measured by Horiba system. The λ obtained is accurate within 1% and it describes different mixtures: stoichiometric ($\lambda = 1$), lean combustion ($\lambda > 1$) and rich combustion ($\lambda < 1$).

3.1.4 INTAKE TEMPERATURE AND PRESSURE

Managing the supply of the intake charge up to the start of combustion position is a critical aspect in modern engines and can impact on emissions, performance and fuel economy. The intake charge supplied to the combustion chamber at all operating conditions meets a number of requirements:

- Sufficient quantity of Oxygen is available to ensure complete combustion
- Sufficient amount of diluent (e.g., EGR) is present to control the combustion temperature
- The temperature and pressure (density) of the charge air is controlled

- Suitable bulk motion and kinetic energy is given to the charge air in the cylinder to support the mixing of air, fuel and intermediate combustion products

In older diesel engines, that did not have to meet stringent exhaust emissions requirements, intake charge systems were relatively straightforward. It was sufficient to ensure that air was clean and that the flow capacity of the intake system was adequate to ensure peak torque and power objectives.

After a while an increasing sensibility towards lower emissions required better controlled intake air properties and matched to suit the engine operating conditions.

Turbocharger, EGR rate were introduced to have higher intake pressure and to lower cylinder temperature: moreover, higher and higher intake pressures required that the intake temperatures, resulting from compression, have to be limited.

There are also two main aspects of intake temperature:

- Limiting the maximum temperature
- Managing low charge temperatures to facilitate engine start-up, warm-up and emission control

A too much high intake temperature leads to lower energy density of the intake charge and combustion temperature can become too high, which means higher exhaust emissions.

A too low temperature instead can be problematic in firing the engine and the emissions can be too high during the engine warm-up. That is the reason why intake air coolers are installed in boosted engines.

In conclusion, higher intake air pressure allows a better mixing inside the cylinder, but higher pressure means higher temperature which is not suitable for a good combustion. Intake air temperature must be controlled, if it is too high in a Diesel engine a lot of NO_x emissions will occur.

In this research, the intake pressure value is set higher than the exhaust pressure (0.3 bar difference) to prevent a return of emission gases from the exhaust manifold back to the cylinder.

3.1.5 INJECTION DURATION

It is the period of time during which fuel enters the combustion chamber from the injector.

In order to meet increasingly stringent emissions regulations and to satisfy growing demands in relation to economy and engine performance, precise injection timing and exact fuel quantity dosing have become key elements over the entire operational life of the engine.

The injection duration can regulate the injection quantity of fuel: with increasing injection quantity, the injection pressure increases, but injection duration is shorter and injection timing can be retarded.

3.1.6 INJECTION PRESSURE

It is a real key factor in LTC combustion strategy and in combustion engines in general. The injection pressure is given by the ECU control and it also depends on the injector characteristics. High injection pressure contributes reduced sized droplet with very fine atomization and improved combustion resulting in reduction of smoke emission. In addition, atomizing the fuel into very fine droplets increases the surface area of the fuel droplets resulting in better mixing and subsequent combustion.

Atomization is done by forcing the fuel through a small orifice (injector nozzles) under high pressure (even more than 2000 bar). Fuel penetration distance become longer and the mixture formation of fuel and air is improved when the combustion duration become shorter ad the injection pressure become higher.

The effect of high injection pressure benefits are:

- Smaller fuel particles due to higher atomization
- Fuel may be injected closer to TDC in hotter air giving shorter ignition delay, resulting in emissions control
- Higher spray penetration
- Smaller droplets evaporate faster leading to rapid fuel-air mixing
- Shorter injection duration which leads to retarded injection timing

In this research the injection pressure factor will be investigated at different values during the tests in order to study its effect on performances and emissions of the heavy duty diesel engine, particularly related to soot emissions.

3.1.7 START OF ACTUATION - SOA

This parameter expresses the time at which the injection is actuated: it is important to define that the start of actuation is not the start of injection. The difference between these two parameters is the time that lays between the actuation and the injection and it is given by the injector reaction to the electric trigger command from the ECU (injector lag).

The tests are conducted acquiring the SOA value because the injection control system provides the SOA set and not the SOI (start of injection). The SOA is not the SOI, the difference between the two parameters is the injection delay, which can be calculated as crank angle degrees.

The start of injection indicates the exact moment in which the fuel leaves the injector nozzles and enters the combustion chamber. It is usually expressed in crank angle degrees (CAD) relative to TDC (Top Dead Centre) of the compression stroke.

The SOA (and directly the SOI) affects the whole combustion timing: moving the SOA between different crank angles degrees directly moves back and forth the combustion, changing sensibly the combustion parameters and the exhaust emissions. Specially it moves the CA50 position: if the CA50 is advanced (even before TDC) the combustion will be stronger with higher peak pressure and consequently more unstable and explosive (the compression stroke is not finished yet and the cylinder pressure reach its maximum value). Otherwise, if the CA50 is retarded, the combustion is moved during the expansion stroke with consequent lower temperature and higher HC and soot, due to incomplete combustion of the fuel.

These are just few reasons why the SOA and directly the SOI have to be accurately investigated: moreover, during the test, SOA will change to adapt to different load conditions and EGR rate, in order to keep the combustion more stable as possible.

3.2 POSTPROCESSING PARAMETERS

These kind of parameters will be studied as results of the tests for the three fuel blends, after the data post-processing with Matlab. The following will be investigated:

- Efficiencies (Combustion efficiency, thermal efficiency, gross indicated efficiency)
- Burn Duration
- Ignition Delay
- COV
- aRoHR (apparent Rate of Heat Release)
- Exhaust Temperature
- Cylinder Temperature
- Maximum Pressure Rise Rate (Max PRR)
- Emissions (HC, CO, NOx, Soot)

Combustion efficiency parameter describes the completeness of combustion: smoke and other incomplete combustion products such as unburned hydrocarbons and carbon monoxide represent a combustion inefficiency.

The magnitude of this inefficiency is small, due to the low emissions of both the pollutants (at full load conditions, if only 0.5 % of the fuel supplied is present in the exhaust as black smoke, the result would be unacceptable). Under lean operating conditions, the amounts of incomplete combustion products are small while under fuel-rich conditions these amounts become more substantial: that is due to the insufficient oxygen to complete the combustion.

Combustion efficiency is normally higher than 98 %. It is useful to define it because a fraction of fuel's chemical energy is not fully released inside the engine during the combustion.

The equation used to calculate the combustion efficiency is written below:

$$\eta_c = 1 - \frac{\sum_i(x_i \cdot LHV_i)}{\left[\frac{\dot{m}_f}{(\dot{m}_a + \dot{m}_f)}\right] \cdot LHV_f}$$

In this equation, the x_i are the mass fractions of CO, H₂ HC and particulates, respectively, the Q_{HVi} are the lower heating value of these species (subscripts f and a denote fuel and air).

The heating values for CO and H₂ are defined:

$$LHV_{(CO)} = 10.1 \text{ MJ/Kg}$$

$$LHV_{(H_2)} = 120 \text{ MJ/Kg}$$

The heating value of HC is not usually known: so the HC heating value is assumed equal to the fuel LHV (42 – 44 MJ/Kg).

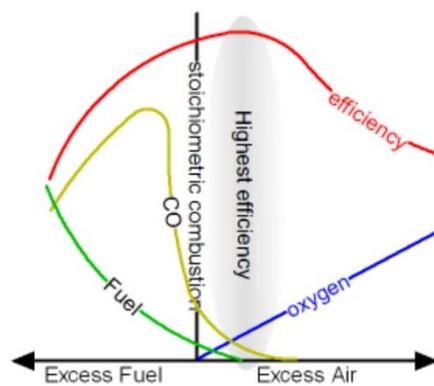


Figure 3.3: Highest efficiency zone

In this calculation in the Matlab script, soot mass fraction is low enough for their contribution to be small and negligible in the equation. The term in square brackets indicates the fuel fraction which is used and it can be express with indicated fuel consumption.

The gross indicated efficiency is related to the compression and expansion stroke, as the IMEP_g (indicate mean pressure gross).

The equation used to describe it, concerns the difference between the total output power (P) and the output power related to the two strokes of intake and exhaust (P_p): this difference is then divided by the multiplication between the optimum fuel flow rate ($\dot{m}_{f(opt)}$) and the fuel heating value (LHV_f).

$$\eta_g = \frac{(P - P_p)}{\dot{m}_{f(opt)} \cdot LHV_f}$$

As a result from the two previous calculation, the thermal efficiency is the ratio between the gross indicated efficiency and the combustion efficiency.

It is important to analyse the efficiencies trend during different loads combustion in the heavy duty engine: these efficiencies are strictly related to the emissions and they give an expression of the quality of combustion, which characterize the engine performances.

The burn duration is also commonly called combustion duration, because it completely refers to the combustion phase.

$$BD = CA_{90} - CA_{10}$$

Its definition is the difference between the crank angle related to 90 % of heat released during the combustion and the crank angle related to 10 % of heat released.

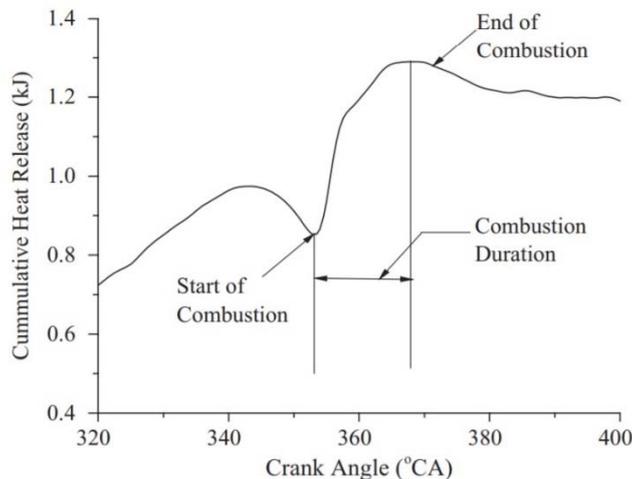


Figure 3.4: Combustion duration

This parameter is affected from the equivalence ratio: usually the burn duration is higher if the combustion is lean ($\lambda > 1$). This behaviour will be investigated in relation with different EGR ratio that will change the air-fuel ratio inside the cylinder, lowering the combustion temperature.

It is lower if there is sufficient turbulence in the combustion chamber and it depends on the generation of air swirl during the induction process.

The duration of the early stage and main stage of the burning process decrease as the turbulent velocity at the start of combustion is increased.

The burn duration also decreases with an increasing compression ratio and this is because of the increase in the end of compression temperature and pressure and decrease in fractional residual gases (favourable condition for the reduction of ignition lag and increase in flame speed).

Besides, the burn duration describes the burning rate of the air-fuel mixture, giving an expression of the type of combustion, a slow one or an explosive one with high peak pressure.

The ignition delay in a diesel engine is defined as the crank angle interval between the start of injection and the start of combustion.

$$ID = CA_{10} - SOI$$

The start of combustion is not well defined, but it is set to the respective crank angle of 10 % of heat release in the combustion. It refers to the change in slope of the heat-release rate, determined from the cylinder pressure.

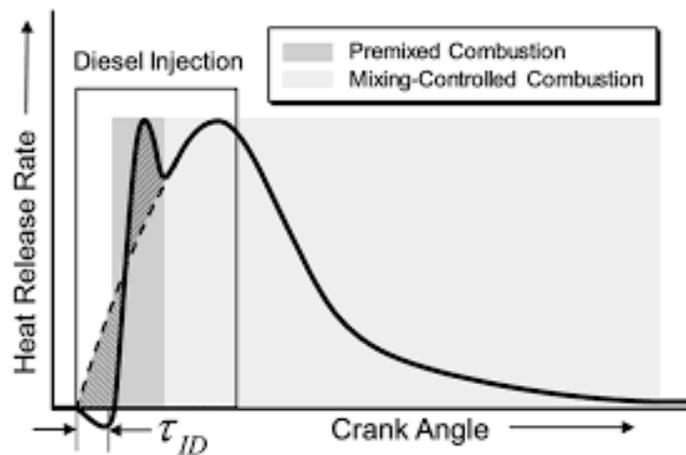


Figure 3.5: Diesel combustion rate of heat release

Clearly the start of combustion is harder to define in IDI conditions (indirect injection), because the ignition time is not perfectly predictable, but it is easier for DI engines (direct injection).

In some application, also flame luminosity detectors are also used to determine the first start of the flame.

Due to vaporization effects, the apparent rate of heat release (aRoHR) curve will typically exhibit negative values before combustion for DI diesel engines: the start of combustion is taken at the angle value at which the heat release curve change from negative to positive.

The ignition delay has strong bases on physical and chemical characteristics of the fuel and injection system: it depend on the atomization of the liquid fuel, the vaporization of the fuel droplets and the mixing of the fuel vapour with air. The chemical part includes pre-combustion reactions of the fuel, air and residual gas mixture that provide the autoignition.

Some engine designs use a pre-chamber or swirl chamber to create the vigorous air motion necessary for a rapid air-fuel mixing.

Furthermore, the penetration of the spray depends on the size of the fuel droplets, the injection pressure, the air density and the air flow characteristics.

The ignition quality of a fuel is defined by its cetane number: if this parameter is low and the fuel has long ignition delay, most of the fuel is injected before the ignition and that results in rapid burning rates with high pressure rise rate (PRR).

The Arrhenius type of equation described the ignition delay.

$$\tau_{id} = a \cdot \Phi^{-k} \cdot P^{-n} \left[\exp \left(\frac{E_a}{R_u \cdot T_{cyl}} \right) \right]$$

In this equation τ_{id} is the ignition delay, ϕ is the equivalence ratio, E_a is the activation energy, T_{cyl} is the cylinder temperature and R_u is the gas constant. Then a , k and n are empirical constants.

Moreover, talking shortly about stringent emissions, to reduce NOx a lower ignition delay is suitable: that because the combustion is more explosive with higher peak pressures but it is also shorter in terms of burn duration. This means that cylinder temperature increases really fast but then, with the beginning of the expansion stroke, it drops, preventing high temperature forward in the combustion, with the result of low NOx emissions.

The coefficient of variability (COV) is used to evaluate the intensity of cycle-to-cycle variation of a parameter, which is the gross indicated mean pressure (IMEPg) in this research.

$$(COV)_{a_i} = \frac{\sqrt{\frac{\sum_{i=1}^n \left(a_i - \frac{\sum_{i=1}^n a_i}{n} \right)^2}{n-1}}}{\frac{\sum_{i=1}^n a_i}{n}} \cdot 100$$

In this equation, n is the number of cycles, a is the parameter of which variability is studied and it is the IMEPg.

The $(COV)_{IMEPg}$ shows how the engine responds to combustion variability.

The heat release analysis in diesel engine provides an efficient way to acquire the combustion information: the RoHR is defined as the rate at which the chemical energy of fuel is released by the combustion process. The heat release is a key factor for the understanding of the in-cylinder process, which can be evaluated using basis on the thermodynamic laws.

The trend of heat release (instantaneous RoHR and maximum RoHR) can be obtained by processing in-cylinder pressure data: the calculation is based on the application of the first law of thermodynamic for an closed system.

It is assumed that the cylinder contents are a homogeneous mixture of air and combustion products and it is at uniform temperature and pressure at each instant in time during the combustion process.

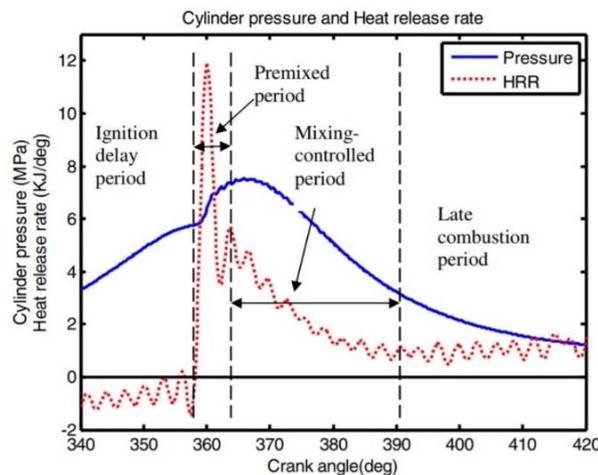


Figure 3.6: Rate of heat release and pressure

The rate of heat release describes the combustion behaviour: the first heat released peak comes from the premixed combustion phase which is usually stronger and more unstable, with the highest pressures of the cycles because it corresponds to the end of the compression stroke. After

TDC (indicated at 360 CAD in the graph above) there is another smaller peak and then the heat decreases.

The RoHR trend acts as a description of the combustion phase, because it underlines the exact moment in which the combustion starts and how long it lasts, pointing out the highest in-cylinder pressure.

In the graph, four different periods are shown: ignition delay, pre-mixed, mixing-controlled and latest combustion period. The combustion analysis is based on RoHR and pressure trend, because changing the start of injection these trends will be shifted forward or backward, defining the combustion type related to the position of the CA50.

Normally about 80 % of the total fuel energy is released in the pre-mixed and mixing controlled combustion: the cylinder pressure and the heat release shown, are typical of a direct injection engine.

In this research, the calculation of heat release is based on the first law of thermodynamic: heat release analysis is most commonly used for diesel engine combustion studies and produces absolute energy with units of Joules or Joules/degree.

$$\frac{dQ}{d\alpha} = \frac{\gamma}{\gamma - 1} p \frac{dV}{d\alpha} + \frac{1}{\gamma - 1} V \frac{dp}{d\alpha}$$

In this equation, α is the crank angle, Q is the net heat release, p and V are the cylinder pressure and volume respectively and γ is the ratio of specific heats (correlated to the in-cylinder temperature).

The γ value is known to be a function of charge temperature and charge composition (AFR, EGR, fuel specification and ratio of reactants to combustion products) and its value has to be around 1.35. In this analysis, the value is given by the following equation, with temperature dependence:

$$\gamma = 1.38 - 6.0 \cdot 10^{-5} \cdot T + 1.0 \cdot 10^{-8} \cdot T^2$$

In the exhaust stroke the mass flow rate and the properties of exhaust gas vary significantly: the thermodynamic state (pressure, temperature, etc.) of the gas in the cylinder varies continually during the exhaust blowdown phase, until the cylinder pressure closely approaches the exhaust

manifold pressure. The blowdown phase refers to a drop in pressure inside the combustion chamber, due to the anticipated EVO (Exhaust Valve Opening).

The exhaust gas temperature varies substantially through the exhaust process and decreases due to heat loss as the gas flows through the exhaust valve and into the exhaust manifold.

The gas temperature near the exhaust valve leaves the cylinder and meets cooler gas inside the exhaust manifold, that comes from the previous exhaust process and is stationary in the exhaust manifold from the previous EVC.

The best indicator of exhaust thermal energy is the enthalpy-averaged temperature:

$$\bar{T}_h = \frac{\int_{EVO}^{EVC} \dot{m} \cdot c_p \cdot T_g \cdot d\vartheta}{\int_{EVO}^{EVC} \dot{m} \cdot c_p \cdot d\vartheta}$$

The average exhaust gas temperatures are usually measured with a thermocouple, it is an important quantity for determining the performance of turbochargers, catalytic converters and particulate traps.

It is assumed that in the intake stroke the cylinder temperature is constant as inlet temperature and in the exhaust stroke the cylinder temperature is constant as exhaust temperature (both measured by sensors and thermal couples). That is not possible for the compression and expansion stroke, because there is no sensors that can measure that high temperature inside the cylinder without burning: for that reason the cylinder temperature is calculated based in the ideal gas law. This calculation represents just a global cylinder temperature.

The cylinder temperature is useful to indicate the start of combustion and the highest temperature point in which the combustion is at its maximum power.

Obviously, during the late combustion phase this temperature will decrease (late expansion and exhaust strokes).

The cylinder temperature also gives important information about the NOx emissions and the temperature behaviour allows checking the correct combustion phase.

The rate of heat release presents, in its equation, a key parameter for the combustion:

$$PRR = \frac{dp}{d\alpha}$$

This value is usually called PRR (Pressure Rise Rate) and it describes the pressure gradient related to the crank angle degrees: the representation of the cylinder pressure rise rate is useful as an indicator of the energy release rate of the combustion process. The fast release of energy at the onset of combustion results in a near linear cylinder pressure increase from start of combustion (SOC) to the resulting peak cylinder pressure, especially when an engine is operating at high load. Therefore, the slope of this near linear increase is chosen as the value to quantify the pressure rise rate and as an indicator of the energy release rate of the combustion process (the slope refers to the average PRR).

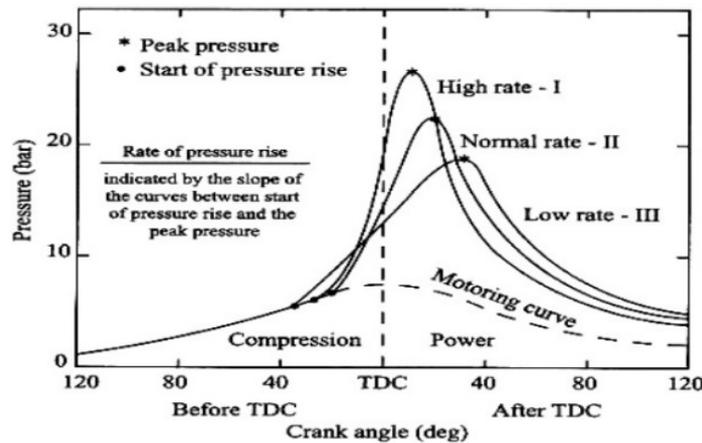


Figure 3.7: Pressure slope

The pressure rise rate shows the combustion behaviour in terms of pressure: a high rate condition is referred to an explosive and faster combustion which can damage the piston and the cylinder, while a low rate is typical of low reactivity fuels (high RON).

The pressure rise rate signal always comes from a filtered pressure signal acquisition: a piezo-electric sensor inside the cylinder measures the in-cylinder pressure and then the signal is filtered using a Savitzky-Golay filter. This is a method of data smoothing based on local least-squares polynomial approximation: the lowpass filters obtained by this method are widely known as S-G

filters, that fit a polynomial to a set of input samples and then evaluate the resulting polynomial at a single point within the approximation interval. This method of computing the S-G filters is implemented using Matlab polynomial functions.

`sgolayfilt (X,ORDER,FRAMELEN)`

It smoothes the signal X using the polynomial low-pass filter: the polynomial order (ORDER) must be less than the frame length (FRAMELEN).

The length of the input X must be \geq FRAMELEN and if the order equals FRAMELEN-1 no smoothing will occur.

Subsequently, with the filtered pressure signal, the filtered pressure rise rate is known and it is possible to graph the correct rate of heat release.

The most important parameter that is worth analysing is the maximum value of the pressure rise rate (PRRmax) because it describes the combustion behaviour and explosiveness.

CHAPTER 4 - TESTS AND RESULTS

4.1 TEST STRATEGY

The tests are conducted on a modified single-cylinder, direct-injection, DAF heavy-duty diesel engine, running at constant speed (1200 rpm), with three different alternative fuel blends, following a well-defined strategy.

The main objective is to investigate the influence of air/fuel ratio and molecular structure of the fuels on emissions, in particular NO_x and soot, and combustion performances.

The first part of the strategy involves the definition of the three fuel blends to test (see chapter 1): the blends, mixture of n-heptane and an organic compound (even alcohols), need to have a RON in a range between 70 and 80 to ensure a correct combustion.

After the analysis (reported in chapter 1), the choice falls on these three blends:

- 1) 20% heptane – 80% butanol
- 2) 40% heptane – 60% benzaldehyde
- 3) 40% heptane – 60% anisole

The second part of the strategy is to define the different test loads, in order to study different operating conditions of the engine. The loads are controlled by the variation of the gross indicated mean effective pressure (IMEP_g): the aim is to analyse the combustion at low, medium and high load, to reproduce the daily stress of a combustion engine, driving in the city or on the highway. Three different loads are chosen to represent the operating conditions: 8 bar, 12 bar, 16 bar IMEP_g.

The combustion control at low loads is not easy for this type of blends, because they are not as reactive as the diesel, so their auto-ignition needs more pressure than common fuels (RON is higher than diesel). At the beginning some trials have been done, trying to control the combustion with 8 bar IMEP_g but it was not possible to maintain a stable flame inside the cylinder.

For that reason, it was decided to continue the tests with only two different loads, 12 bar and 16 bar IMEP_g.

The next step of the strategy is to decide the parameters directly related to combustion properties and emissions: these parameters have been varied through certain values (sweep) to analyse the combustion behaviour and the emissions records.

That is in order to reproduce extreme conditions for the engine, leading to an unbalanced combustion at non-conventional regime, which enhance the soot emissions in order to acquire many data on soot behaviour.

The tests have been conducted with EGR rate sweep at 12 bar and 16 bar load and with only injection pressure sweep at 16 bar load: increasing or decreasing the EGR rate the combustion temperature will change, while increasing or decreasing the injection pressure, the atomization of the fuel particles will change.

The injection pressure sweep is not necessary at 12 bar load: the density inside the cylinder is not relevant and it would only be an excessive fuel consumption. Moreover, high injection pressure at low load would enhance the possibility to have fuel on the cylinder walls, increasing HC emissions. Besides, increasing the atomization of the fuel is useful to reduce soot emission, which is already low at 12 bar load.

The EGR rate concerns NO_x emissions, while the atomization affects the particles dimensions, which are directly related to the particulate matter emissions.

So far the strategy concerns the fuels, the loads and the sweep controlled parameters (because the other parameters will change consequently, for example the λ value will change during the EGR sweep). Apart from the starting conditions, there is a last point in the strategy to be defined: the most important control parameter for the combustion phasing is CA50 and it is necessary to find the optimum CA50 value to conduct the tests properly.

The only way to find it, is running the engine, acquire data for each fuel, and investigate the CA50 value that provides the best efficiencies and combustion balance.

So at the beginning of each fuel test, the first investigated parameter will be the optimum CA50, that will be set as a fixed parameter for the subsequent tests.

Therefore, the strategy is defined with the following points:

- Fuel
- Starting conditions
- Different loads
- Investigation of the optimum CA50 at three different loads for each fuel
- EGR sweep
- Injection Pressure sweep
- Data analysis

4.1.1 STARTING CONDITIONS

Fuel	Bu80-He20		Benz60-He40		An60-He40	
IMEPg load	12 bar	16 bar	12 bar	16 bar	12 bar	16 bar
Intake Pressure [bar]	1.97	2.5	1.87	2.5	1.87	2.5
Intake Temperature [°C]	40	40	40	40	40	40
Engine speed [rpm]	1200	1200	1200	1200	1200	1200
Oil Temperature [°C]	> 80	> 80	> 80	> 80	> 80	> 80
Water Temperature [°C]	> 80	> 80	> 80	> 80	> 80	> 80
CA50 (cad)	9	9	6	9	9	9
Injection Pressure [bar]	1500	2000	1500	2000	1500	2000
EGR rate [%]	25	25	25	25	25	25
Lambda λ (+/- 0.05)	1.5	1.5	1.5	1.5	1.5	1.5

Table 4A: Starting test conditions

The table above shows the starting parameters at each load for each fuel: as already said in the previous paragraph, the combustion is not stable for an 8 bar IMEPg load, because the reactivity of the fuels is not enough to sustain the auto-ignition and the combustion flame.

The last two parameters are varied, depending on the load: for the EGR sweep the rate is varied from 25% to 45% with steps of 5%, while the injection pressure is varied from 2000 bar to 1500 bar with steps of 100 bar.

The EGR sweep is done for both loads, 12 bar and 16 bar, while the injection pressure sweep is only done for 16 bar.

The SOA (start of actuation) is always adapted to the set conditions, in order to maintain a stable combustion and a stable flame inside the cylinder: it is important to notice that the SOA is always before TDC because, with these fuels, it is important to have the CA50 near TDC.

If the SOA is near the end of the compression stroke, the cylinder pressure is higher at the start of combustion and the fuel manage to auto-ignite: otherwise, with a retarded SOA (after TDC), at the start of combustion the cylinder pressure and temperature is already decreasing and misfire can occur, due to the low reactivity of these fuels.

4.1.2 DIFFERENT VALUES OF IMEPg

The experiments were carried out over different loads: 12 bar and 16 bar IMEPg respectively, to cover both premixed dominated combustion and diffusion dominated combustion. At the very beginning of this research, also a lower load has been tested but with no good results. In fact, at the lower load of 8 bar IMEPg is completely unstable in terms of combustion, due to the lower reactivity of the fuels.

Misfire can also occur if the EGR rate is too high, because the pressure and the temperature are not high enough to sustain the combustion.

The IMEP is defined as the average pressure value that, multiplied for the cylinder displacement, produce the same work obtained during the operating cycle. It is an expression of the theoretical, frictionless power known as indicated horsepower.

The aim of this study is to investigate the potential of high-octane fuels operating at different loads: the main focus is to achieve reasonable emissions, efficiency and stable combustion. Due to the layout of the test setup, the IMEP is calculated from the in-cylinder pressure signal.

In order to evaluate the combustion performance, the IMEPg is used (excluding the gas exchange stroke).

In the case of 12 bar IMEPg load, the intake pressure is set at 1.97 bar, the intake temperature is 40°C, the engine speed is 1200 rpm. These parameters are fixed during the tests: the EGR rate is set at 25% initially and it is changed during the EGR sweep, with values of 25%, 30%, 35%, 40%, 45%, trying to maintain a stable combustion with advanced SOA values, which are manually controlled during the whole test, to control the combustion. The λ value changes, related to the EGR rate: precisely, λ decreases if EGR rate increases, but lambda starting value is always around 1.5 (± 0.05).

Besides, at higher load, the injection pressure sweep concerns the value from 2000 bar to 1500 bar, decreasing in steps of 100 bar: in this case, the correct combustion is controlled by the injection duration, which can be changed as a manual input on the ECU from the control room. Naturally, if the injection pressure decreases, it is necessary to increase the injection duration, in order to maintain the fixed fuel mass flow rate.

At 16 bar IMEPg load, the intake pressure is set at 2.5 bar, the intake temperature is 40°C, the engine speed is 1200 rpm, with an injection duration which is different from the previous case of 12 bar load. That is because a higher fuel mass flow rate is required, to face the higher cylinder pressure and maintaining a stable combustion. At high load, the only injection pressure sweep is investigated, with the same values, from 2000 bar to 1500 bar.

4.1.3 EFFECT ON COMBUSTION PHASE CHANGING

In order to characterize the combustion phasing, CA50 is used, which is defined as the crank angle where 50 % of the heat has been released. The test strategy considers the use of a fixed CA50 for each fuel and each load, which comes from a study of the optimum CA50.

It is difficult to control the CA50 in a specific position while the engine is running, because during the combustion it can easily change its value: so the first tests are conducted to investigate an optimum value, related to the efficiency and combustion stability (COV). Once the optimum value is known (in the best conditions it is between 0 and 10 CA/ATDC, it is 9 CA/ATDC in this research, except for the benzaldehyde blend at 12 bar IMEPg which show 6 CA/ATDC), the other tests will be all done with a fixed CA50 which corresponds to the optimum value.

The ECU control allows to manually change the SOA, injection duration and injection pressure parameters: the strategy to maintain a fixed value of CA50 is to phase the combustion. The easier way is to regulate the SOA, trying to obtain a stable combustion with a fixed CA50 value.

The main effects of combustion phasing involve pressure and temperature inside the cylinder: also, the ignition delay is a key factor strictly related to the combustion phasing. In fact, fixing a certain value of CA50, the fuels with short ignition delay are injected near TDC, while longer ignition delay needs an advanced SOA.

In the next paragraph, the data analysis from the first tests on the investigation of the optimum CA50 is shown: the post-processing main parameters are plotted in relation with CA50 different acquired values.

4.1.4 OPTIMUM CA50

The following paragraph shows the result for the first tests, observing the starting conditions (keeping the lambda constant), conducted with different CA50 values. The aim of these tests is to individuate the best combustion phasing, related to the efficiency and combustion parameters.

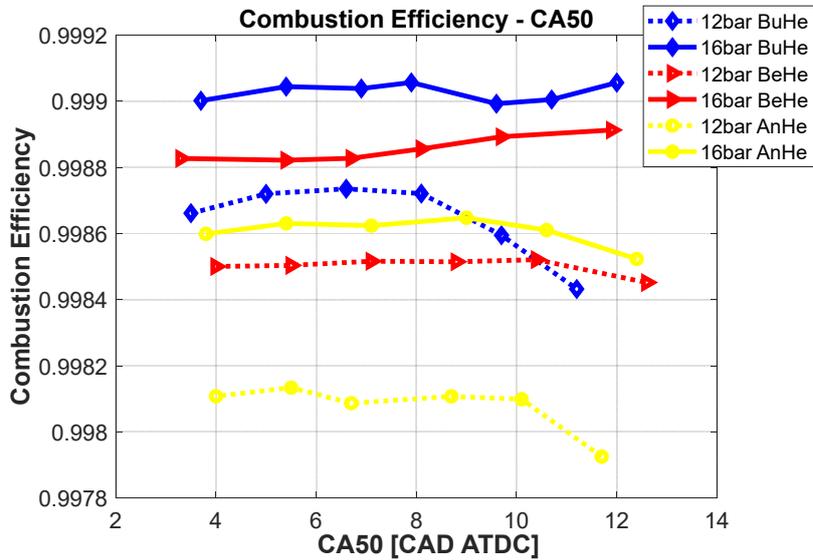


Figure 4.1: Combustion Efficiency – CA50

Figure (4.1) shows the combustion efficiency of the three fuel blends, at each load: the trend is quite stable, but it mostly decreases above 10 CAD ATDC, due to incomplete combustion.

Below, figure (4.2) represents the combustion stability through the COV value. The COV trend is chaotic but with a tendency to increase.

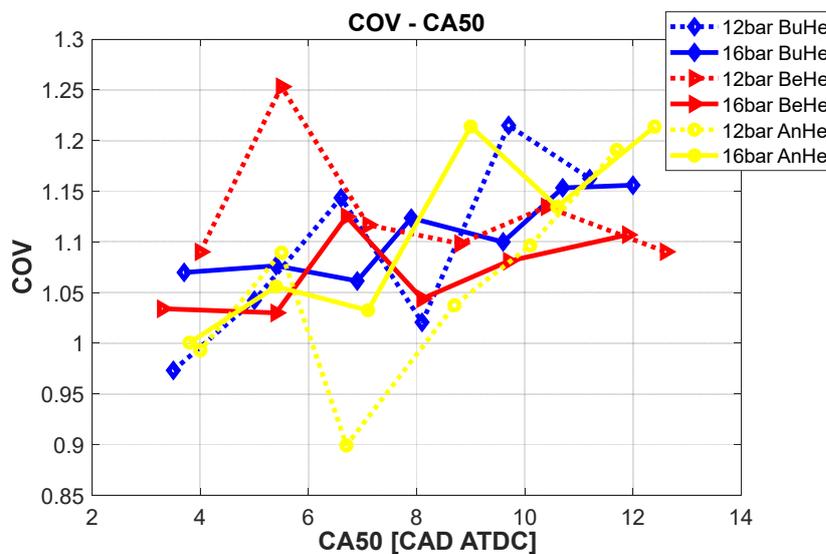


Figure 4.2: COV – CA50

The thermal efficiency value remains between 47 % and 50 % with several fluctuations, as it is shown in figure (4.3). Generally, the best range of CA50 values for all the parameters is between 0 and 10 CA ATDC. That is because before TDC the combustion would be too advanced with a high presence of fuel between the cylinder walls and the piston (flame quenching), with consequent higher HC emissions, while after 10 CAD ATDC the combustion would be too retarded, enhancing misfire. However, once an engine is considered as non-adiabatic system and has heat transfer, the excessively intensified combustion process (with quite short combustion duration) results in a strong increase of the cylinder pressure and temperature peak (with high PRRmax, figure (4.4)). These are the conditions for a too advanced CA50 (near TDC), which enhance the heat transfer losses as a disadvantage on the improvement of efficiency. On the contrary, a retarded CA50 and relatively long combustion duration lead to a decreasing heat losses: on the other hand it could lead to incomplete combustion.

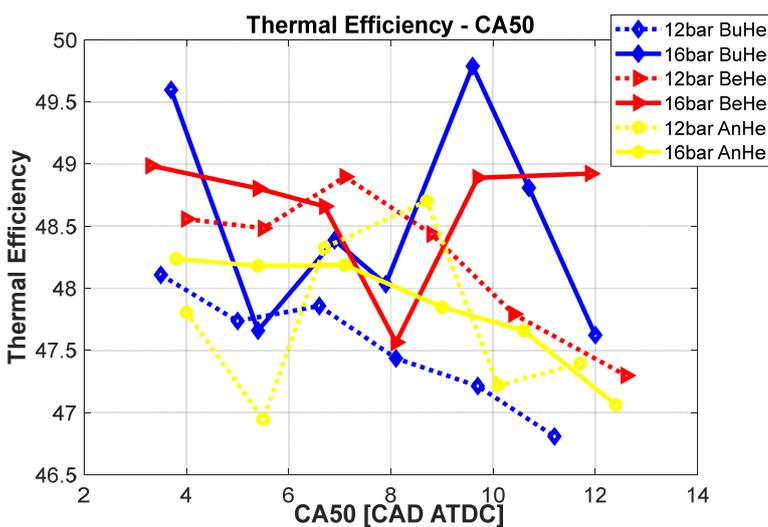
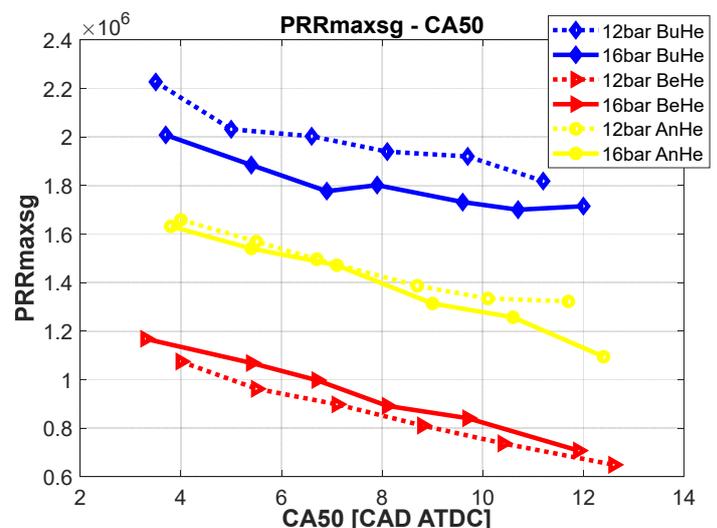


Figure 4.3: Thermal efficiency – CA50

Figure 4.4: Maximum Pressure Rise Rate – CA50



The ignition delay is strongly related to the CA50 and to the combustion phasing:

$$ID = CA10 - SOI$$

When the CA50 is retarded, the ignition delay slightly increase, as it is shown in figure (4.5).

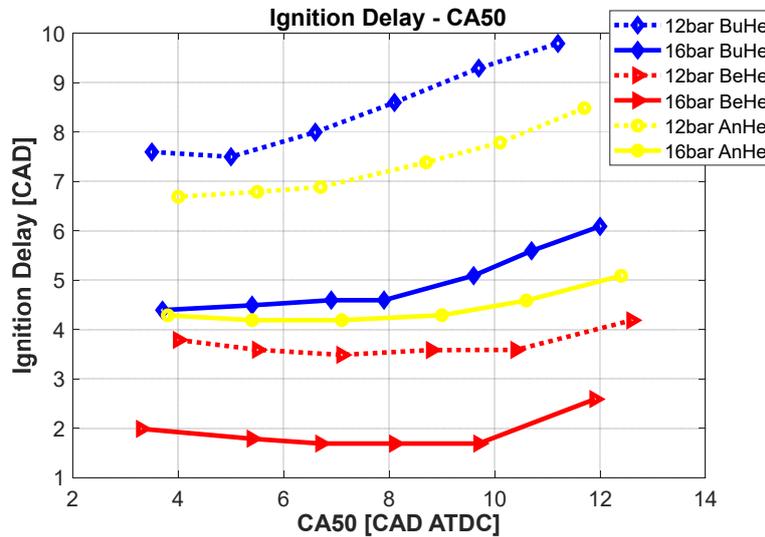


Figure 4.5: Ignition delay – CA50

The following listed graphs represents the emissions behaviour, depending on the CA50. The first one, in figure (4.6), shows the ISHC (Indicated Specific Hydrocarbons) emissions, with unit of measure indicated with [g/kWh].

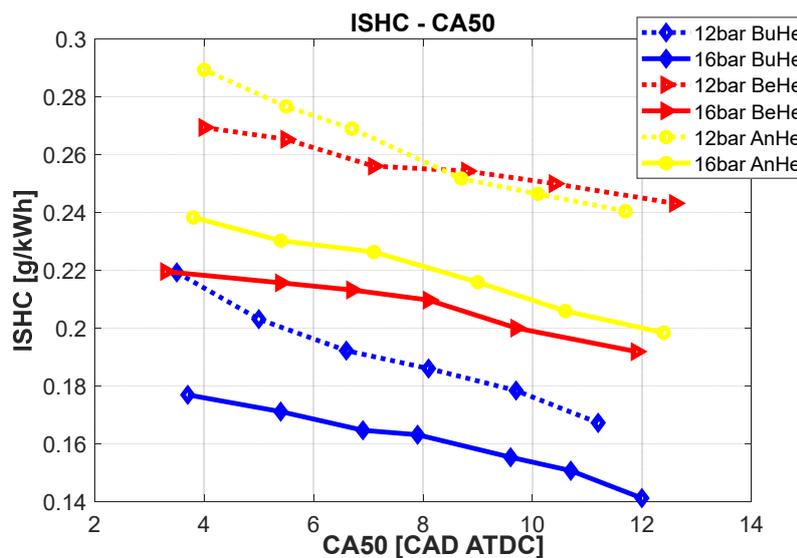


Figure 4.6: Hydrocarbons – CA50

It can be noticed that the earlier the CA50 the higher the ISHC: that is strictly related to the SOA, which is much more advanced the more the CA50 is advanced.

The ISHC suffer the too earlier combustion phase because the fuel is injected before TDC and it is trapped between the cylinder walls and the piston, leading to flame quenching and producing a lot of unburned fuel.

The next graph, figure (4.7), shows the SOA behaviour, depending on which value corresponds to the CA50: the SOA is not the SOI, the difference between the two parameters is the injection delay.

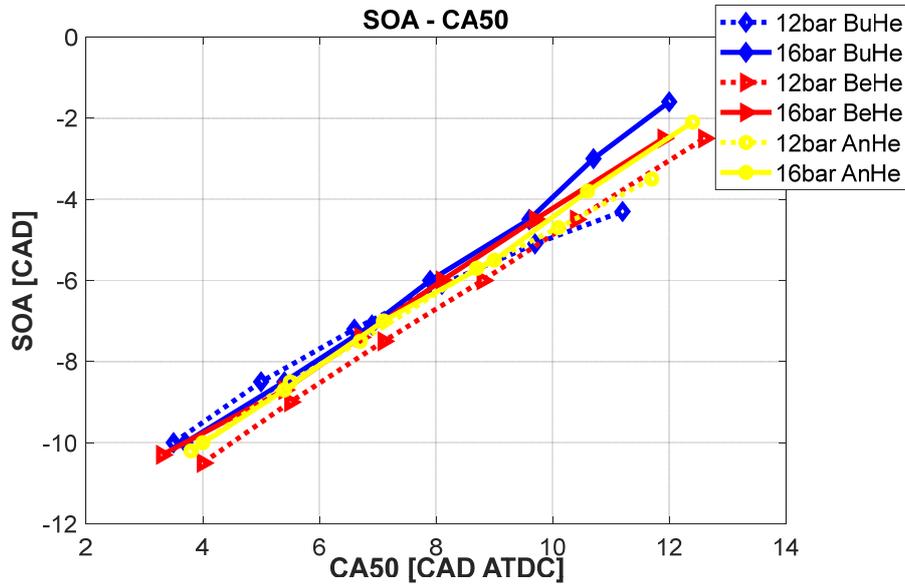


Figure 4.7: Start of actuation – CA50

The combustion behaviour of three fuel blends is quite similar and the SOA trend, related to the CA50, is the description of the same reactivity of the fuels.

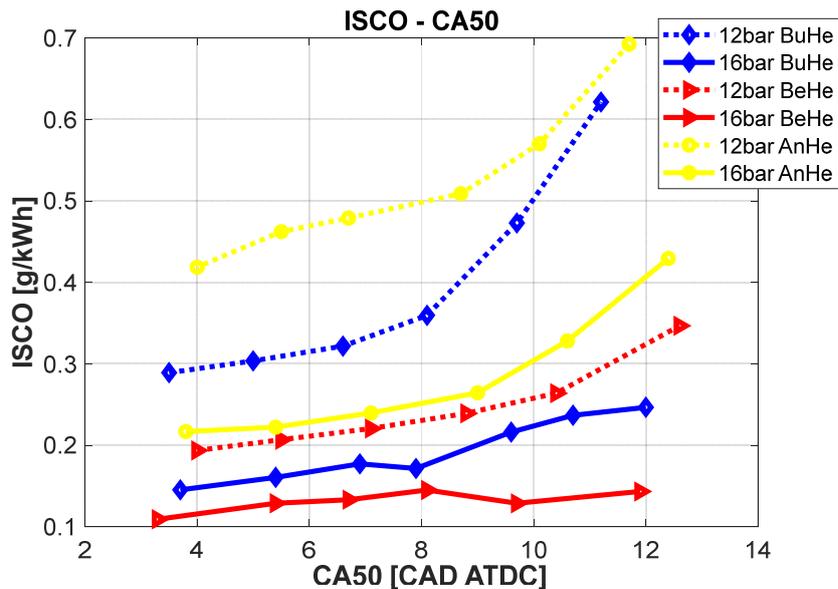


Figure 4.8: Carbon monoxide – CA50

Whether the combustion phase is retarded, the CO is higher because of an incomplete combustion with a temperature dropping during the exhaust stroke. The case of an incomplete combustion of an over-lean fuel-air mixture enhances the CO emissions.

The NOx emissions (figure (4.9)) increase with higher cylinder temperature: and advanced CA50 generated a strong combustion with high pressure peaks and high PRR, which leads to high temperature (explosive combustion near the TDC). On the other hand, with a retarded CA50 the combustion slides away from the TDC, largely inside the expansion stroke: the temperature drops with the pressure and the combustion could be incomplete. Lower temperature allows to reduce NOx, the challenge is to reduce the temperature maintaining a good combustion efficiency, with the more suitable CA50.

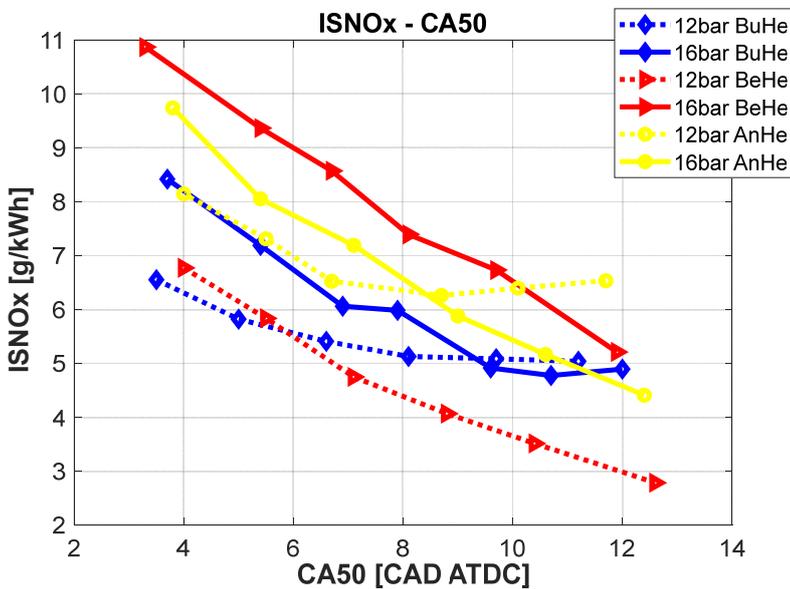
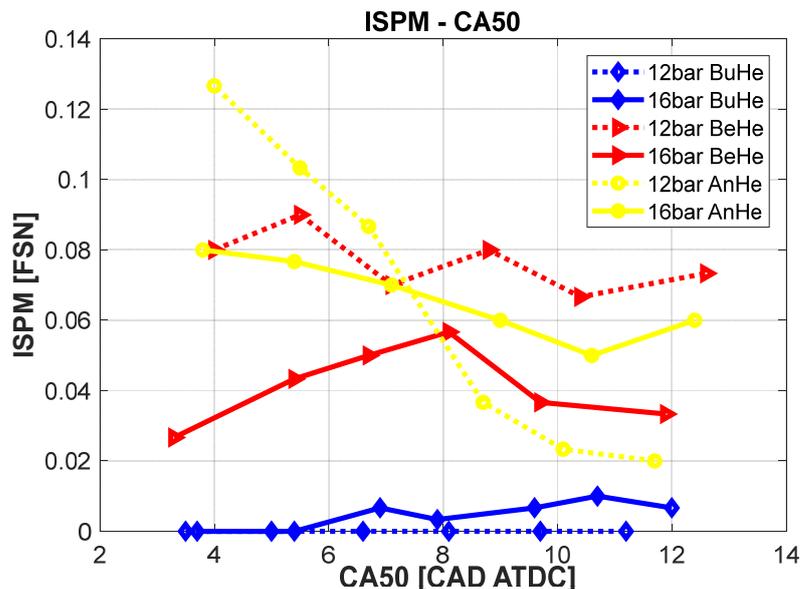


Figure 4.9: NOx – CA50

Figure 4.10: Particulate matter – CA50



The particular matter emission is shown in figure (4.10), expressed in FSN (Filter Smoke Number) instead PPM (Parts Per Million). This is due to the smoke meter measurement: the system is set for Diesel combustion acquisition and the conversion from FSN to PPM is not accurate for these alternative fuels.

This analysis of the combustion phasing influence on combustion parameters allows to choose the optimum value of CA50: it is useful to refer to the efficiencies and to the maximum pressure value which is reached inside the cylinder (it is always better to avoid too high pressure).

The efficiencies are based on the trade-off between heat losses and combustion completeness and the choice of the optimum CA50 has to respect this trade-off.

Therefore, for advanced CA50 (near TDC) the heat losses are high because of the high pressure and explosive combustion due to the proximity with the end of the compression stroke. On the other hand, instead, for retarded CA50 (expansion stroke) an incomplete combustion occurs, due to the expansion stroke which cool down the cylinder.

The optimum value lays somewhere in the middle of these two conditions: the aim of the CA50 analysis is to locate the optimum value, respecting the trade-off.

In this research, accordingly to the efficiency and COV (combustion stability parameter), the following CA50 value will be taken as fixed parameters for the different tests:

- Bu80 – He20 , 12 bar IMEPg , CA50 = 9 CAD/ATDC
- Bu80 – He20 , 16 bar IMEPg , CA50 = 9 CAD/ATDC
- Benz60 – He40 , 12 bar IMEPg , CA50 = 6 CAD/ATDC
- Benz60 – He40 , 16 bar IMEPg , CA50 = 9 CAD/ATDC
- An60 – He40 , 12 bar IMEPg , CA50 = 9 CAD/ATDC
- An60 – He40 , 16 bar IMEPg , CA50 = 9 CAD/ATDC

4.2 RESULTS

All the tests have been conducted on CYCLOPS setup: in the following paragraph all the results for the three fuel blends will be presented, showing the fuels behaviour, under the same test conditions, related to different combustion parameters and emissions.

4.2.1 EGR SWEEP

The exhaust gas recirculation, which returns a portion of the engine exhaust gas to the combustion chamber via the intake system, shows a great potential to reduce NO_x emissions. The possibility of reducing NO_x emissions results from thermal, chemical and dilution effects.

In particular, the thermal effect refers to the high specific heat of the exhaust carbon dioxide (CO₂) and water (H₂O), compared to oxygen (O₂) and nitrogen (N₂) in the intake air, reduces combustion temperature. In the chemical effect, recirculated H₂O and CO₂ are dissociated during combustion and they modify the NO_x formation process. In the dilution effect, the reduction of Oxygen mass fraction, in the fresh intake air charge, by inert gases, causes a reduction in the local flame temperature.

The use of EGR is limited by increased soot emissions and low efficiencies: when the EGR rate exceeds a certain percentage, the combustion become unstable and even misfire can occur, mostly in LTC techniques with low reactive fuels. LTC and alternative fuels, such as the tested blends, aim to separate the event of diesel injection from combustion, to reduce NO_x and soot emissions simultaneously, which has always been impossible with normal Diesel engines (the well-known NO_x/soot trade-off).

In this paragraph, the results based on EGR sweep will be presented: these results describe the reactions of the fuels to a varying EGR rate, presenting their combustion parameters and the engine emissions. The EGR quantity is regulated by an electric valve position, which is controlled by an electronic system: the EGR is defined as the ratio between the recirculated mass from the exhaust manifold and the mass of intake mixture. It can be also defined by the recirculated CO₂ fraction: this is due to the strong difference between the amount of CO₂ in the fresh intake air (approximately zero) and the amount in the exhaust gases which is much higher, as the CO₂ is a combustion product.

The results analysis is conducted on the efficiencies, on the emissions (HC, CO, NO_x and particulate matter) and combustion parameters, such as COV, ignition delay, SOA, PRRmax, rate of heat release and cylinder temperature. All these parameters are investigated under variations of the EGR rate, increasing from 25 % (with steps of 5 %) until the combustion remain controllable (normally up to 45% EGR).

Combustion efficiency

Figure (4.11) shows the combustion efficiency trend, for an EGR rate sweep from 25% to 45% with an interval of 5% for each step: the combustion efficiency usually presents a value around 1, but in this case over 35% EGR it shows a sharp decrease. The graph indicates the decline from 0.999 to 0.930 if the EGR rate is increased from 35% to 45%.

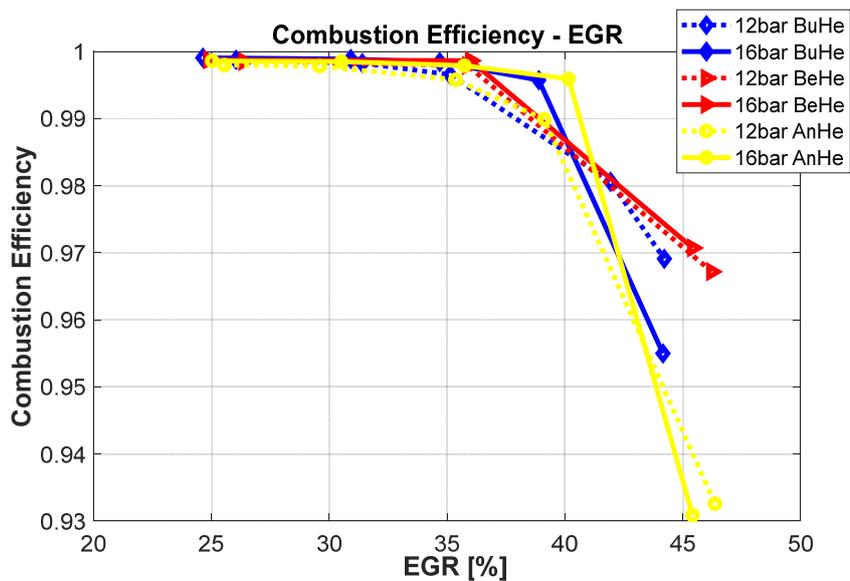


Figure 4.11: Combustion Efficiency – EGR percentage

The An60 - He40 fuel blend (60% anisole, 40% heptane) is the most resistant at high load (16 bar IMEPg) but it is also the one that reaches the lowest efficiency value for an EGR rate of 45%. The other two blends behave similarly, but the Bu80 – He20 reaches lowest efficiency values at high load than Be60 – He40. However, if the EGR rate is below 35%, the three fuel blends maintain a high combustion efficiency value: the diesel combustion works normally with an EGR rate of 15%.

Unburned hydrocarbons (HC)

The unburned hydrocarbons emissions are usually low in the traditional Diesel combustion: besides, aromatics compounds (such as anisole and benzaldehyde) combustion produce a higher percentage of HC.

The analysis of the results shows that the acquired values are still low (average lower than 0.25 g/kWh), but there is an actual difference between Bu80 – He20 and the other aromatic blends, which present higher values as expected.

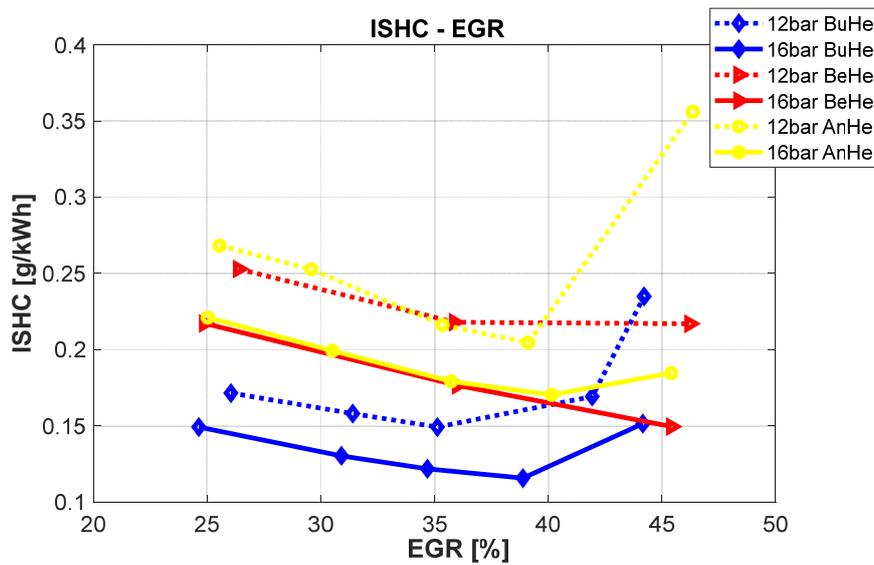


Figure 4.12: HC – EGR percentage

The main cause of HC emissions is an incomplete combustion: HC can be formed by flame quenching in the cylinder walls or by over-rich fuel mixture that doesn't complete the combustion due to a too retarded combustion phase. As shown in figure (4.12), the higher load presents lower HC emissions than the lower load.

The graph also shows that HC emissions decrease with an increase load, for all the fuels and for every EGR rate: that is due to a higher pressure condition inside the cylinder, which provides a better auto-ignition of the fuel, suppressing quenching effect in the leaner regions, leading to a decrease in HC emissions.

Bu80 – He20 blend is apparently characterized by a more clean and complete combustion, compared to the aromatic compounds.

Carbon monoxide emission (CO)

The CO emissions are mainly controlled by the λ value and tend to increase with insufficient oxygen. The increase in the CO levels is a result of incomplete combustion of the air-fuel mixture or any factor that can deteriorate the combustion.

In figure (4.13) the CO emissions trend is shown: CO emissions strongly increase with higher EGR rate: the emissions are stable until the EGR rate is below 35%, then the production of CO rise very fast.

The main reasons are related to a retarded combustion, such as the ignition delay, which increases as the increase of EGR, resulting in higher amount of fuel burned in the expansion stroke. This fact leads to a reduced CO oxidation reaction rate due to the expansion cooling effect and consequently to increasing CO emissions.

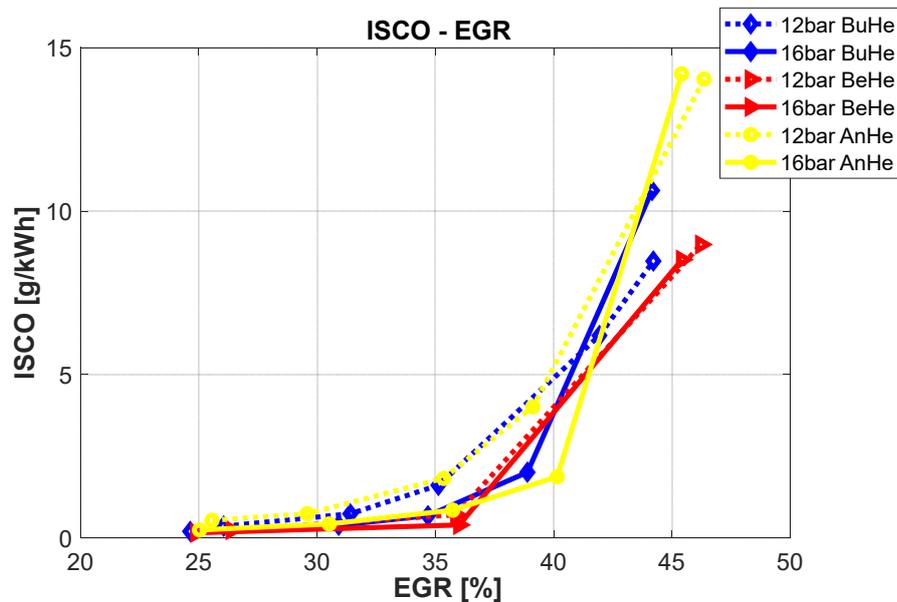


Figure 4.13: CO – EGR percentage

In addition, the exhaust gases dilutes the oxygen in the incoming air stream providing inert gases to act as absorbent of combustion heat: this leads to a decreasing cylinder temperature and higher CO emissions.

When the EGR rate is higher than 35%, the decreasing oxygen concentration exceeds a threshold, which define the start of the combustion deterioration, weakening the oxidation of the CO, with a consequent increase of CO emissions.

It is worth noting that, under 35% EGR rate, CO emissions are higher for lower loads: this is because at lower loads the temperature is lower than higher loads.

Nitrogen oxides (NO_x)

In figure (4.14) the NO_x emissions trend is shown: the in-cylinder gas temperature and the oxygen concentration are two main parameters for NO_x emissions. Nitrogen oxides are formed at high temperatures but, despite this fact, these temperatures are easily reachable from diesel combustion. The residence time of the fuel/gas mixtures in locations where the local temperature and local λ are suitable for the NO_x formation, is another parameter that influences the NO_x production.

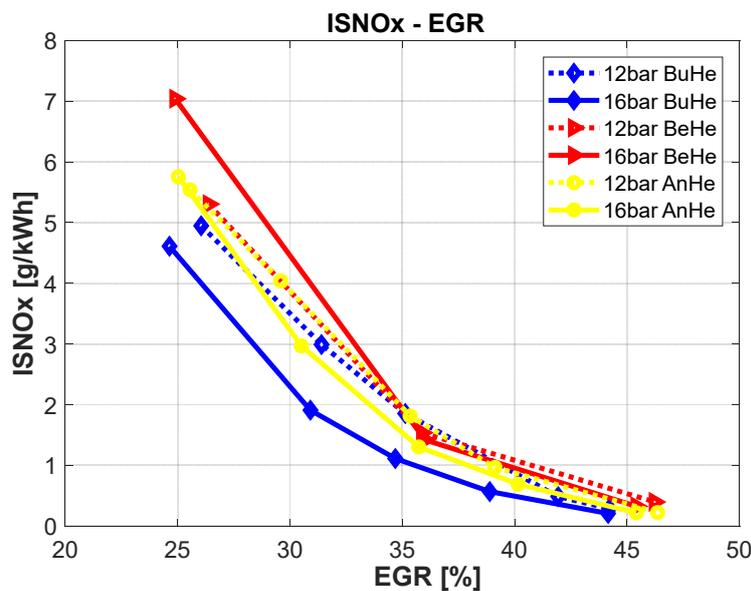


Figure 4.14: NO_x – EGR percentage

The EGR rate strongly reduce NO_x emissions when the 40% threshold is passed: the reason is that the dilution effect of EGR reduced the in-cylinder temperature and the oxygen concentration, reducing the NO_x.

Theoretically, higher loads should present higher NO_x emissions, due to the different amount of fuel needed, compared to the lower loads. The temperature is higher for 16 bar IMEPg than the 12 bar IMEPg load and that should lead to higher NO_x emissions: contrarily, the graph shows that the 12 bar load brings to higher NO_x production, even if the combustion temperature is lower. Probably the reason is the higher oxygen concentration in lower loads that brings to create a bigger amount of nitrogen oxides.

The best achievements in NO_x emissions reduction are found for an EGR rate higher than 40%: low reactive fuels hardly manage to present a stable combustion at that point, but, until 45% EGR rate, the tests has been correctly conducted.

Particulate Matter (Soot)

There is a well-defined gap between the levels of soot production of a common diesel and these fuel blends: diesel combustion produces soot emissions from 10% of EGR rate and reaches a FSN value of one, for an EGR rate value of around 30%. As it is shown in figure (4.15), the soot emissions increase rapidly with EGR also for these alternative fuels, but it starts increasing at 25% for the two blends with similar structure and over 35% for the alcohol based fuel.

Actually, the butanol blends shows remarkable low soot emissions, even at high EGR rate, compared with benzaldehyde and anisole blends.

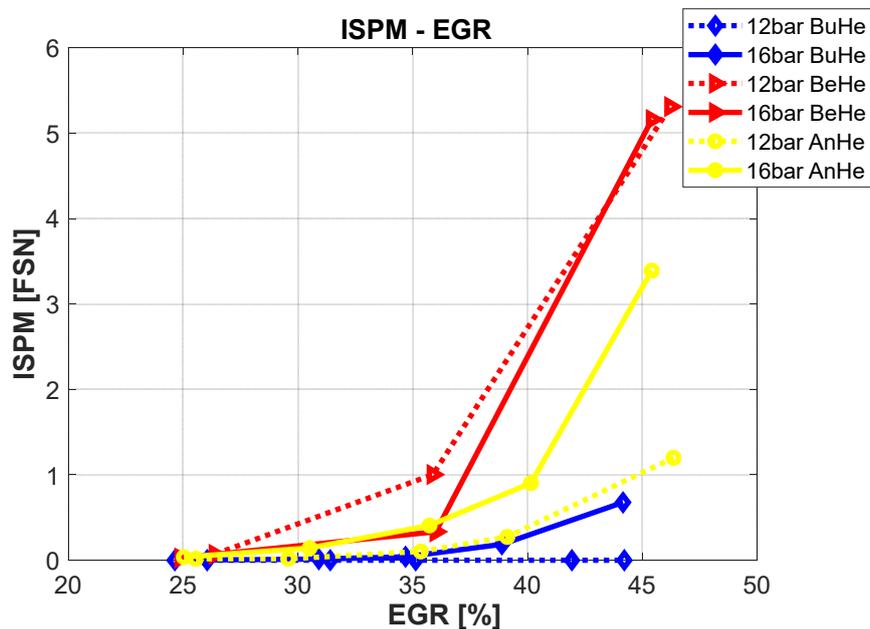


Figure 4.15: Particulate matter – EGR percentage

The big difference with diesel, according to other studies, can be due to longer ignition delay, which provides a reduction of the fuel-rich combustion zones where is easy to generate soot.

Except for one point of Be60 – He40 at 12 bar load, the higher load produces more soot: that is because at low oxygen concentration condition (higher load), relatively short ignition delay time could generate more fuel-rich zones and enhance the soot production.

The Be60 – He40 and An60 – He40 blends have a very similar structure, as aromatic compounds: their soot emission trend is higher than Bu80 – He20 fuel blend. It could be that the fuel molecular structure actually affects the soot emission, showing an almost negligible emission value for the Bu80 – He20 fuel blend.

The oxygenated benzaldehyde is the only case of higher soot production at lower load than higher load.

Ignition delay

It consists in physical delay, wherein atomization, vaporization and mixing of air fuel occur and in chemical delay, due to pre-combustion reactions.

Lower ignition delay provides higher NO_x emissions: it is also strongly dependent on ambient temperature and pressure. The ignition delay is defined as the time interval between the start of injection and the start of combustion.

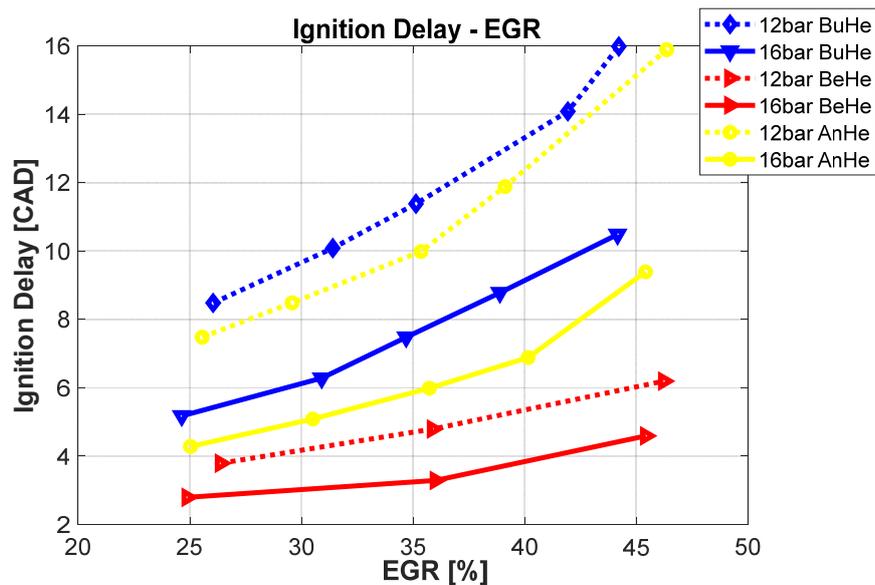


Figure 4.16: Ignition delay – EGR percentage

Short ignition delay enhances soot reduction but NO_x emissions are higher: that happens because the combustion is more violent with shorter ignition delay, due a lack of time to properly mix air and fuel. On the other hand, higher ignition delay leads to lower NO_x and higher soot, because a better pre-mixing of air and fuel in the combustion chamber lowered the combustion temperature. The EGR rate influence on the ignition delay is shown in figure (4.16): the higher the EGR rate, the higher the ignition delay, because of the lowered temperature. The benzaldehyde fuel blend shows the smaller ignition delay and the best reactivity at each load, compared to the other to fuel. The anisole blend is also more reactive than the butanol blend, underlining that the aromatic compounds, with benzene ring structure, have higher reactivity compared to the alcohol blends such as Bu80 – He20.

The higher load shows a shorter ignition delay in comparison with the lower load, because in-cylinder temperature and pressure are higher for a 16 bar IMEPg and the combustion is favourable. Increasing EGR rate leads, keeping the same SOA, to a retarded combustion.

Start of actuation (SOA)

The combustion analysis starts with the SOA parameters, which defines the exact crank angle degree in which the injector is solicited: after the choice of the optimum CA50 (in order to have the same combustion phasing for a better comparison of the parameters), it is a fixed parameter: the SOA value acts consequently, referring to the optimum CA50.

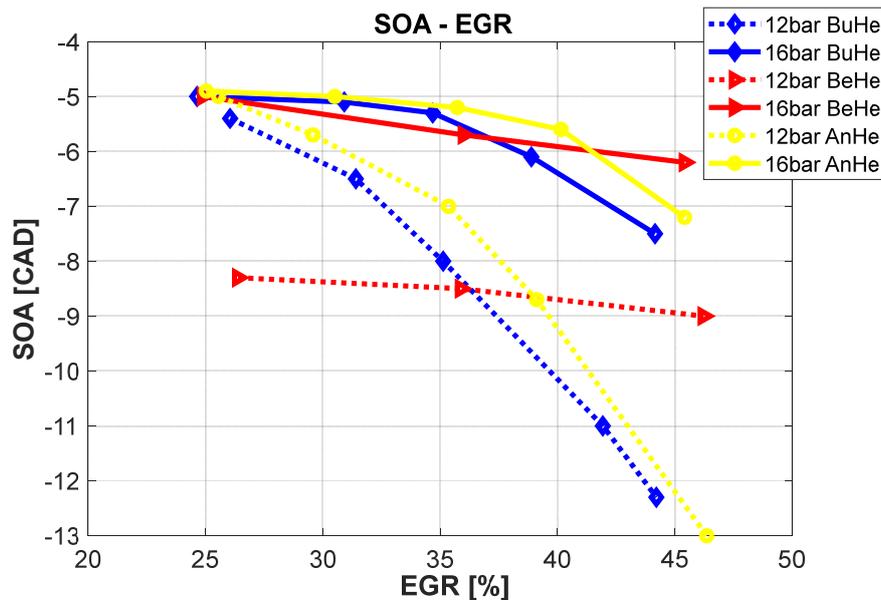


Figure 4.17: Start of actuation – EGR percentage

The EGR rate has a strong effect on anisole and butanol: the SOA for their combustion needs to be sharply anticipated, to respect the fixed CA50. This is because a higher EGR rate is unfavourable to combustion and slows it down: these fuels can even run into misfire for an extreme EGR rate (above 45%), because of the excessive dilution and low temperature.

The Bu80 – He20 and An60 – He40 blends have a similar trend, but Be60 – He40 shows a small variation at both loads: the benzaldehyde is not so affected by the EGR rate, in terms of combustion rate. The difference between the SOA values at different loads for benzaldehyde is bigger than the others because the optimum CA50 for 12 bar IMEPg of Be60 – He40 is set at 6 CAD/ATDC, so it is advanced.

It is worth noting that the butanol blend is the less reactive and need the most advanced SOA, compared to the other fuels conditions with the same optimum CA50 value.

As it is clear from the figure (4.17), the higher load does not need a too much advanced SOA, because combustion conditions are better (pressure and temperature) and allows to reach the CA50 crank angle faster than the lower load.

Maximum pressure rise rate (PRRmax)

Fuels with high RON and long ignition delay are injected, inside the combustion chamber, even before the ignition occurs. This results in very rapid burning rates once combustion starts, with higher values of pressure rise rates compared to more reactive fuels (with lower RON).

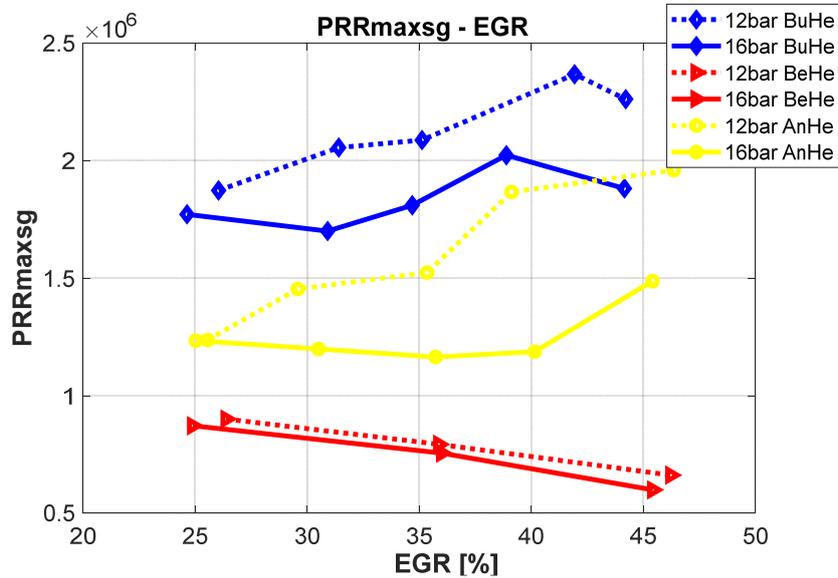


Figure 4.18: Maximum pressure rise rate – EGR percentage

The figure (4.18) shows that the less reactive fuel blend Bu80 – He20 has the highest value of pressure rise rate. In addition, the lower load presents higher value due to a lower cylinder pressure that allows a faster pressure rise rate.

Burn Duration

The burned duration is described as the difference between CA90 and CA10.

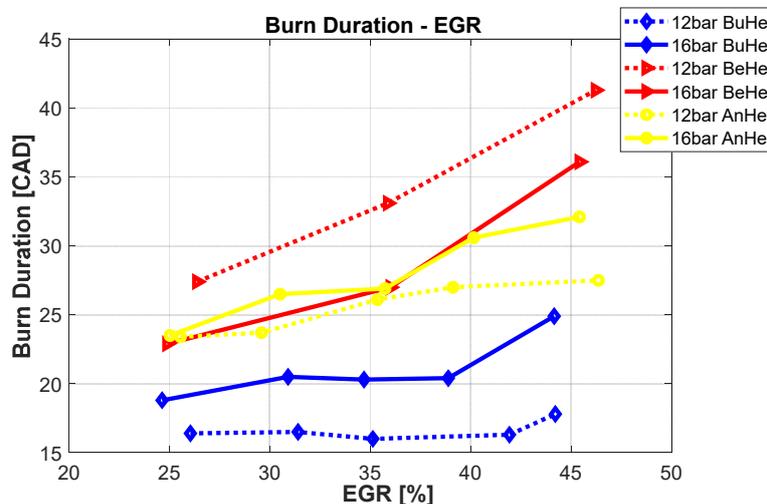


Figure 4.19: Burn duration – EGR percentage

The alcohol fuel blend Bu80 – He20, has a molecular structure that burns faster than the aromatic fuel blends structure.

As it is shown in figure (4.19), the anisole and benzaldehyde benzene rings provides a longer burn duration, stating that the molecular structure has a strong influence on combustion duration.

The Bu80 – He20 behaviour is coherent with the pressure rise rate: the faster the PRR, the shorter the burn duration. The structure of alcohols fuels is burns faster, with high pressure peaks that could damage the piston and the cylinder. The Bu80 – He20 shows a more unstable combustion, probably due to the molecular structure.

Rate of heat release (RoHR)

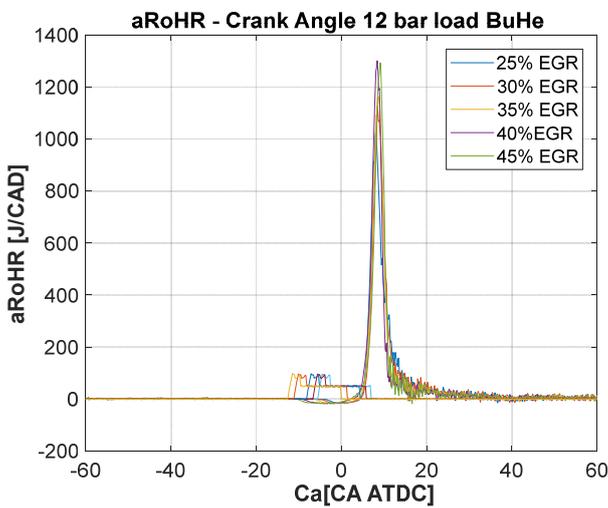


Figure 4.20: Rate of heat release – CA50 for 12-bar load Bu-He

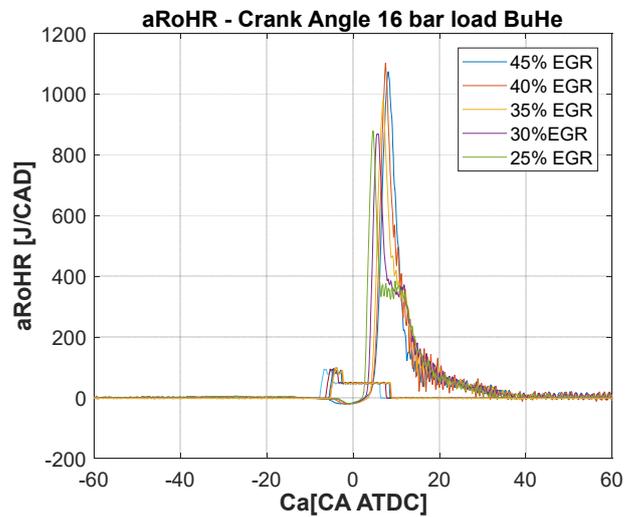


Figure 4.21: Rate of heat release – CA50 for 16-bar load Bu-He

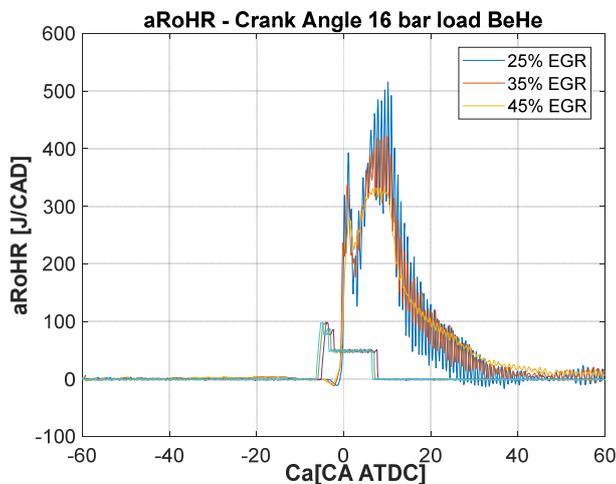


Figure 4.22: Rate of heat release – CA50 for 16-bar load Be-He

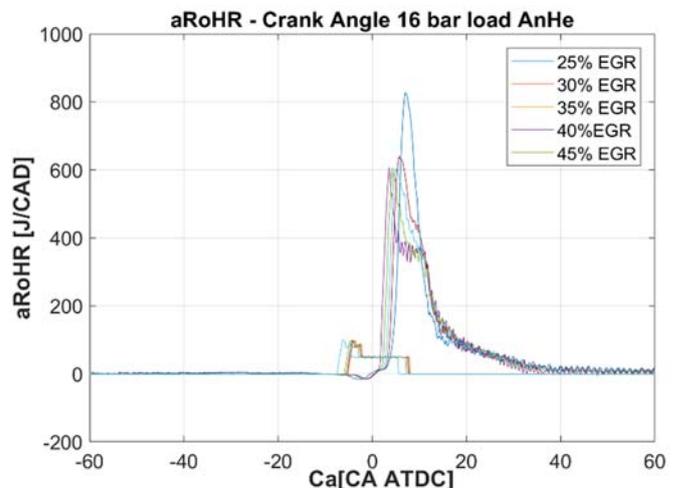


Figure 4.23: Rate of heat release – CA50 for 16-bar load An-He

The plots above show the rate of heat release data and the injector current signal for the three fuels respectively, two loads (12bar and 16bar) for Bu80 – He20, 16bar load for Be60 – He40 and 16 bar for An60 – He40.

The rate of heat release is presented both for medium load (12bar) and for high load (16bar). At 12bar load, the fuels provide almost pure premixed combustion while diesel normally shows both premixed and diffusion combustion (two peaks).

At high load, Bu80 – 20He and An60 – He40 are still controlled by premixed combustion but start to show some diffusion combustion: besides, Be60 – He40 combustion is dominated by diffusion combustion and it burns in a conventional diesel combustion mode.

The EGR rate effect is to slide forward the energy released peak of the premixed combustion, retarding the maximum rate of heat release between 5 CA/ATDC to 10 CA/ATDC.

It seems that the increase of EGR rate overrides any start of diffusion combustion, moving forward the premixed combustion: this effect can be seen in figure (4.20), (4.21) and (4.23). On the other hand, for Be60 – He40, the EGR rate enhanced the diffusion combustion (figure (4.22)), while for lower load of Bu80 – He20 the effect of EGR is only about increasing the rate of heat released, maintaining the premixed combustion.

The Be60 – He40 fuel has the most similar diesel combustion mode because is dominated by diffusion combustion at high loads. The other fuels, at high load and low EGR rate, shows a start of diffusion combustion but they are still mainly controlled by the premixed combustion. The same fuels at high EGR rate shows only premixed combustion, shifted to retarded CA/ATDC.

The three fuels at low load are mainly controlled by premixed combustion.

Cylinder Temperature

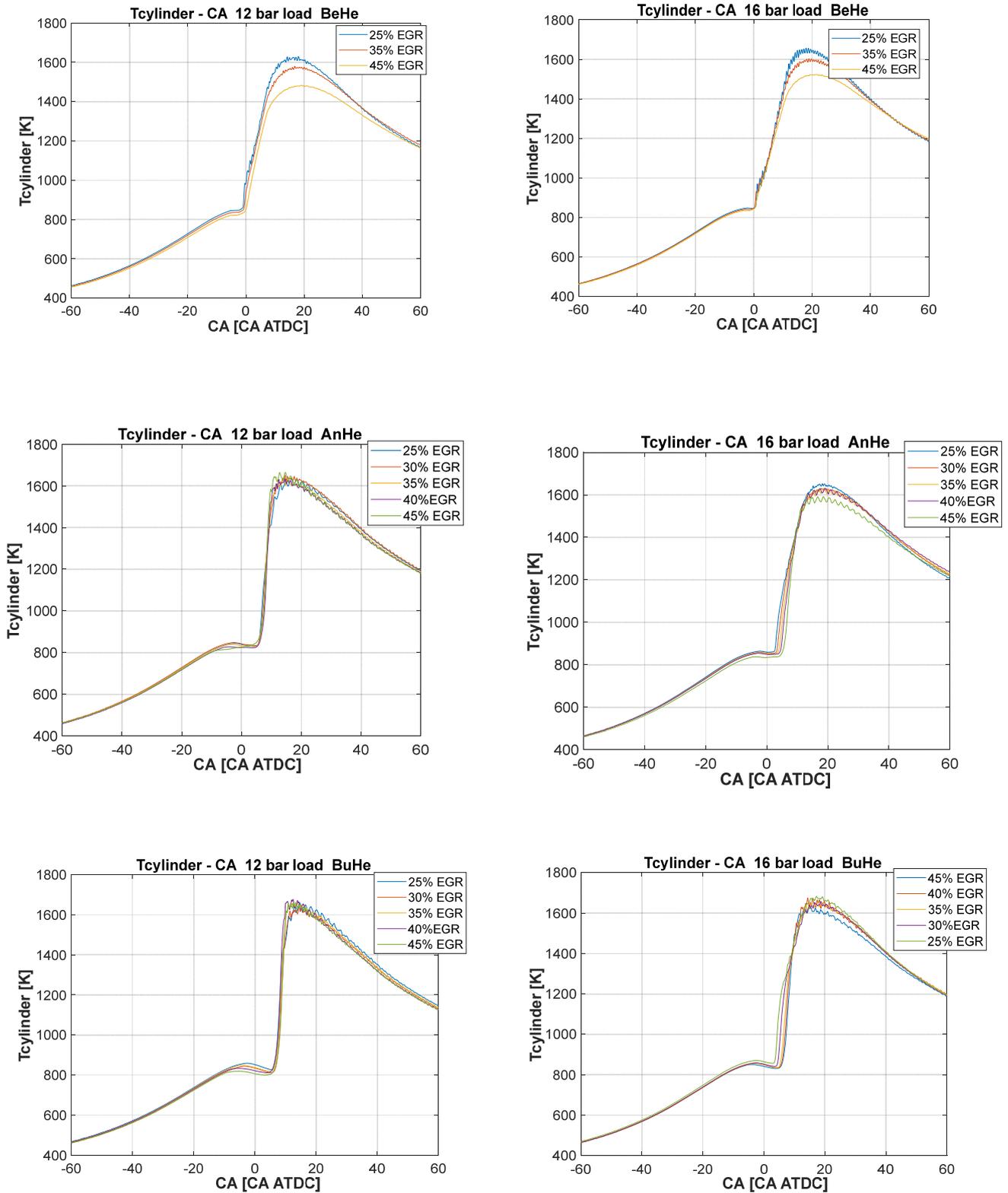


Figure 4.24: Cylinder temperature – CA50 for different EGR percentage and for the three fuel blends at each load

4.2.2 INJECTION PRESSURE SWEEP

The injection pressure variation analysis is important to study the soot emissions behaviour and consequently the NO_x-soot trade-off.

Injection pressure is a key parameter in alternative fuels combustion: it provides higher atomization of fuel particles and easier air-fuel mixing. In order to analyse the effect of stronger fuel atomization on soot emissions, the injection pressure sweep is done only at high load, 16bar IMEP: soot emissions at lower load are already low and there is no need to investigate the possibility of higher injection pressure.

The results are presented for the three fuels, at 16bar load IMEPg, for an injection pressure sweep from 1500 bar to 2000 bar with five steps of 100 bar each.

Emissions and combustion parameters are presented in this paragraph.

Combustion Efficiency

The combustion efficiency is not so much affected by the injection pressure: the efficiency value is near one for each fuel as it is shown in figure (4.25).

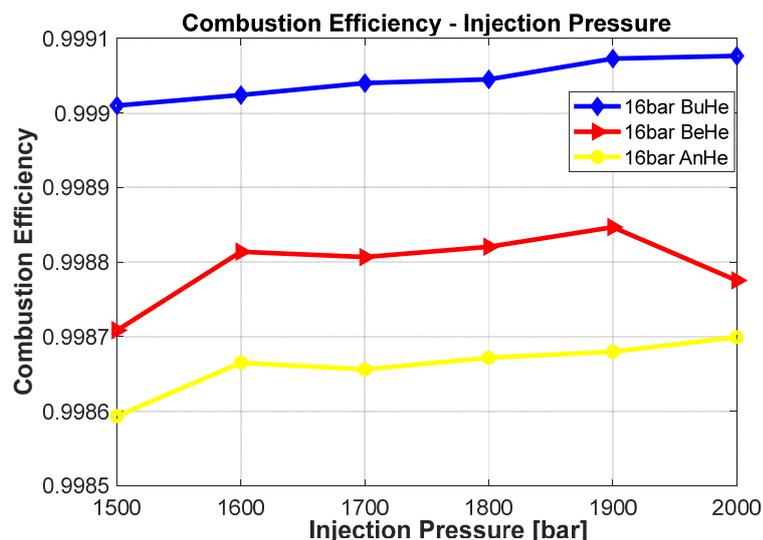


Figure 4.25: Combustion efficiency – injection pressure

Nitrogen Oxides (NOx)

The NOx increases with an increase of injection pressure and high load. The main reason is that an increase of injection pressure leads an increasing peak cylinder pressure and heat release rate, increasing the NOx emissions.

The oxygen concentration, the combustion temperature and a high temperature duration are the main causes for NOx emissions.

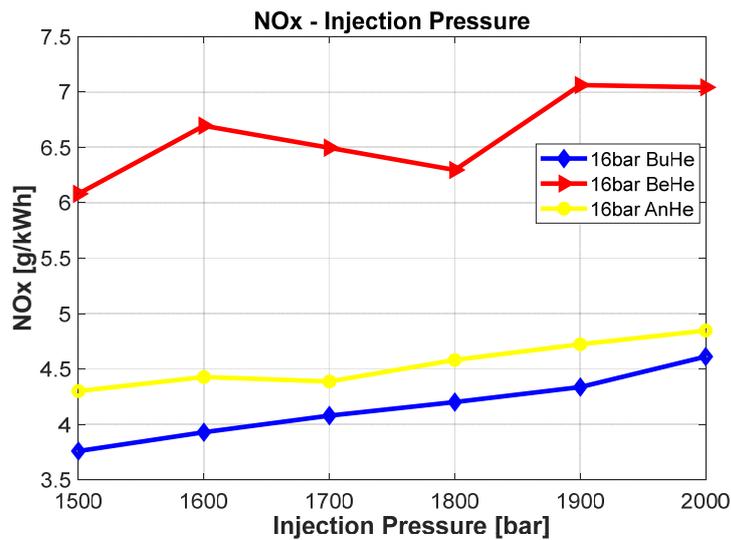


Figure 4.26: NOx – injection pressure

Particulate Matter (soot)

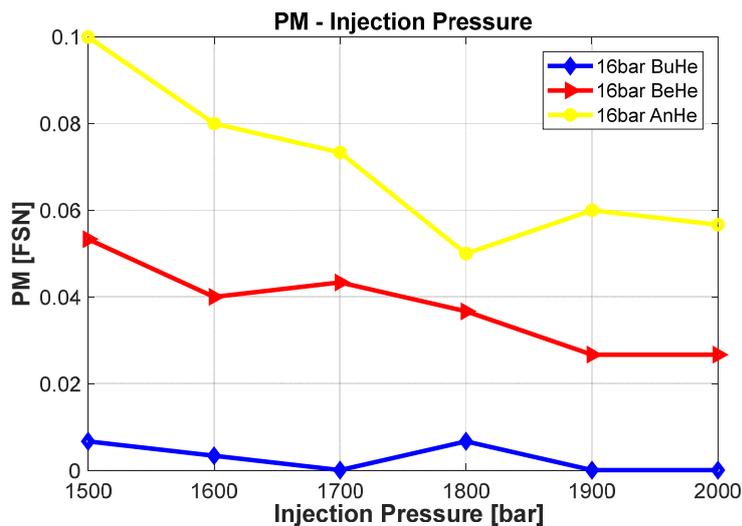


Figure 4.27: Particulate matter – injection pressure

The soot emissions decrease with higher injection pressure, due to the atomization of the fuel particles.

Smoke emissions decreases also because the increase of fuel injection pressure enhances a higher contact face area between the fuel and air and, in addition, increases the speed of mixture formation. Moreover the combustion temperature rises, in order to accelerate oxidation, resulting in a more complete combustion.

The aromatic hydrocarbon is the main component of soot precursor, Be60 – He40 and An60 – He40 shows higher soot emissions value, while the alcohol fuel has almost zero soot emissions. The NOx-soot trade-off is represented by the opposite trends of NOx and soot: comparing the two plots, it is noticed that NOx can decrease, lowering the injection pressure, without increasing smoke emissions sharply.

Ignition delay and SOA

The ignition delay clearly decreases for higher fuel particles atomization, due to a better mixing inside the combustion chamber. The combustion becomes easier, so the injection pressure has a positive effect on combustion.

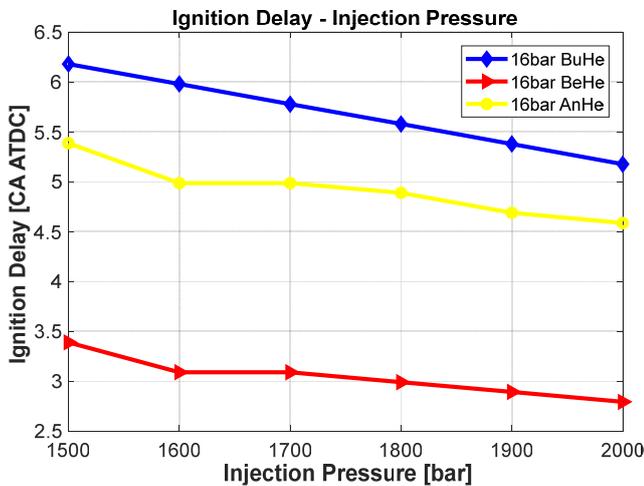


Figure 4.28: Ignition delay – injection pressure

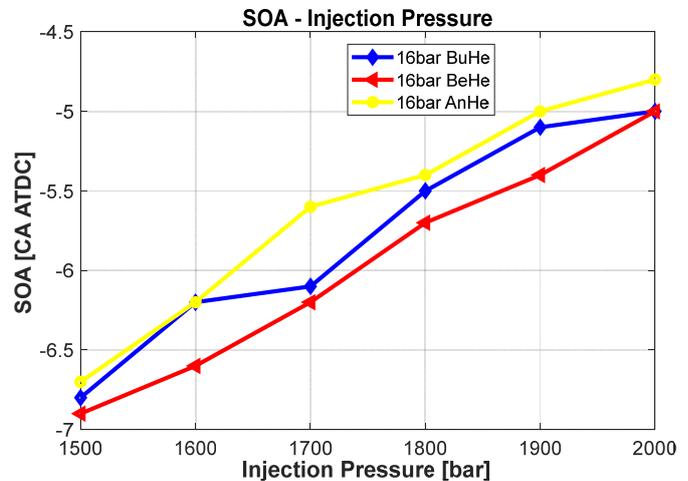


Figure 4.29: Start of actuation – injection pressure

The shorter ignition delay is directly related to the SOA: in order to maintain the same CA50 and the same combustion phasing, it is necessary to retarded the start of actuation, because of the faster ignition of the fuel. The figure (4.29) represents the SOA trends: it is shifted forward as the injection pressure increase, with a similar behaviour for all the fuels.

Figure (4.28) shows the ignition delay trend, again quite similar for all the fuels.

Burn Duration

The injection pressure determines the mixing rate of fuel and air inside the combustion chamber: if higher mixing occurs, the burn duration becomes shorter, due to the better ignition conditions. The fuel injection rate gets faster and the rate of heat release increases with higher injection pressure.

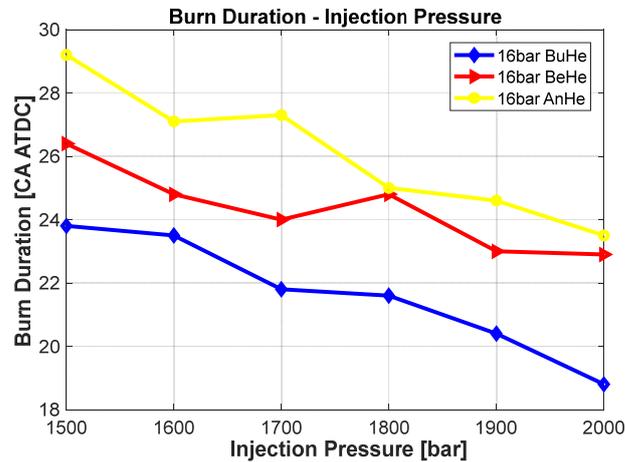


Figure 4.30: Burn duration – injection pressure

Cylinder Temperature

The effect of the injection pressure on the cylinder temperature is shown below.

As expected, a higher burning rate leads to higher temperature: the burning rate increases with the injection pressure, resulting in a shorter combustion duration.

The cylinder temperature increases with injection pressure for all the fuels.

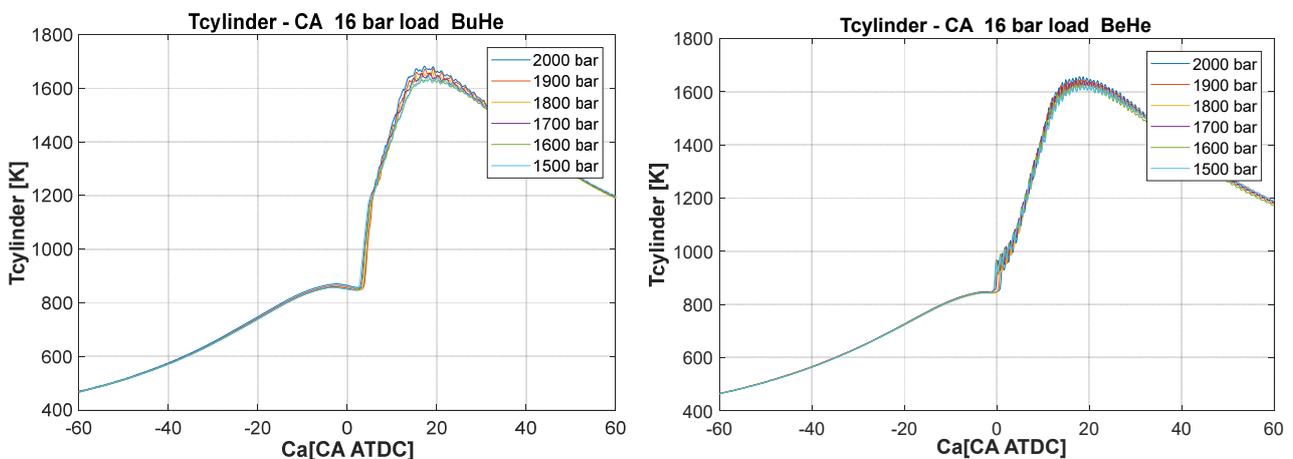


Figure 4.31: Cylinder temperature – CA50 for Bu-He and Be-He at 16-bar load

4.2.3 ENGINE EXHAUST PARTICLE SIZER - EEPS

High-octane fuels have shown significant properties for reducing soot emissions, but the study of the nanoparticles emitted is needed. Load, dilution and fuel structure can be major factors influencing that. It is worth to acquire knowledge about the particulate matter in terms of number concentration and size distribution.

The EEPS is a fast-response, high-resolution instrument that measures very low particle number concentrations in diluted exhaust.

Below, some figures from EEPS are shown: the representations regards the EGR rate influence on the nanoparticles distribution at different load.

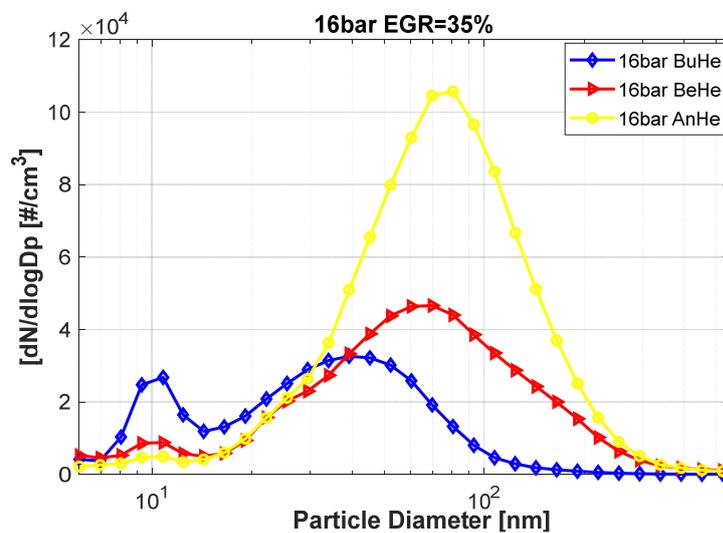


Figure 4.32: EEPS concentration and size distribution for the three fuel blends with EGR = 35 %

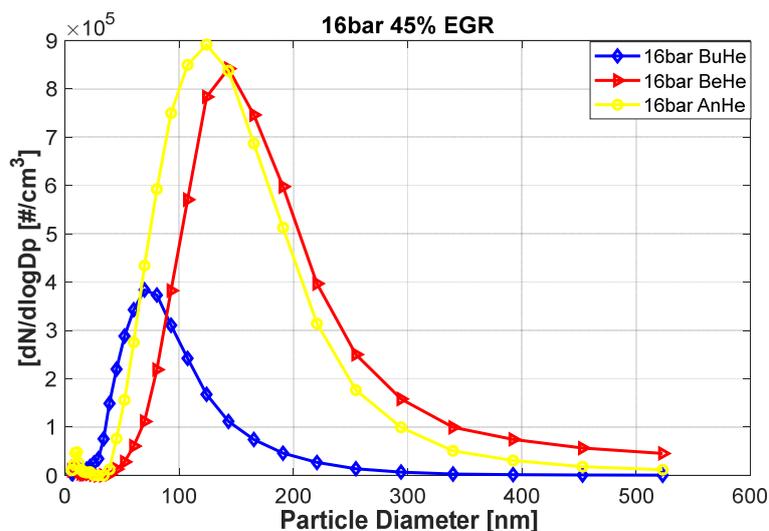


Figure 4.33: EEPS concentration and size distribution for the three fuel blends with EGR = 45 %

EEPS plots represent the particulate number concentration and size distribution: in this analysis, the dependence on EGR and load is shown. Figure (4.32) and (4.33) describes the nanoparticles characteristics at constant load with EGR rate variations. The fuel structure has a heavy influence on nanoparticles concentration: aromatic compounds, with benzene ring structure, are soot precursor and show higher size and concentration: the butanol, instead, is an alcohol and shows lower nanoparticles concentration and size. The higher the EGR rate, the higher the soot particles concentration and size (the first plot has linear X-axis while the second one has log X-axis representation). The Y-axis shows the size distribution ($dN/d\log D_p$) normalized to total particle number concentration, versus mobility diameter.

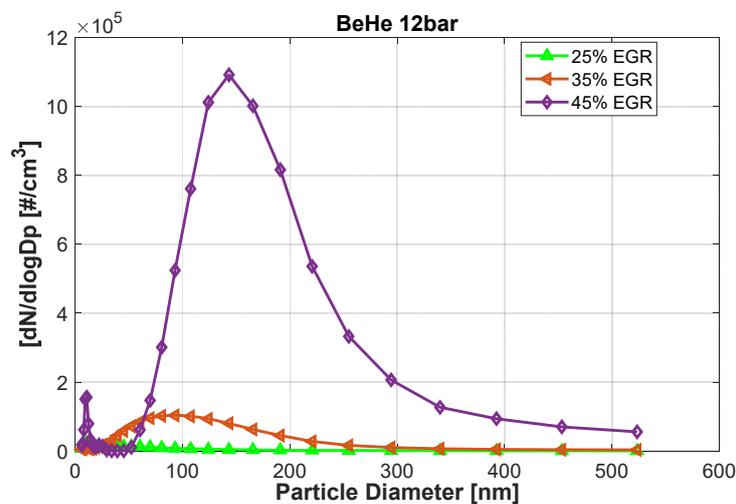


Figure 4.34: EEPS concentration and size distribution for Be-He at 12-bar load with EGR percentage variation

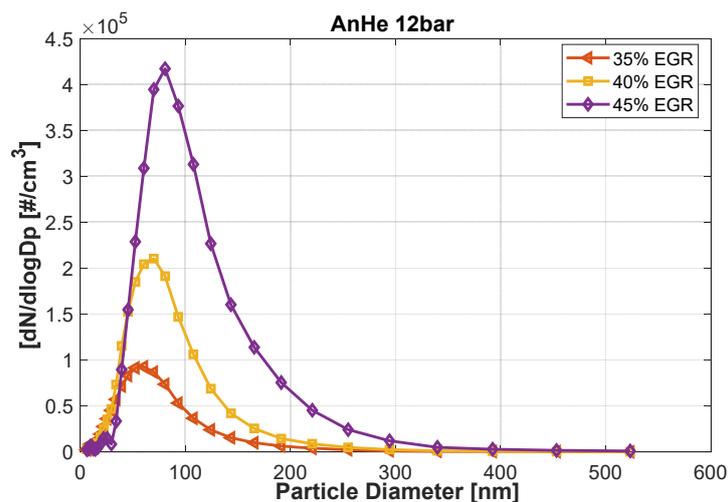


Figure 4.35: EEPS concentration and size distribution for An-He at 12-bar load with EGR percentage variation

Figure (4.34) and (4.35) shows the trend for 12bar load related to Be60 – He40 and An60 – He40: there is no representation for Bu80 – He20 due to its lower emissions at 12bar.

The following plots, figure (4.36), shows the two aromatic molecular structured fuels trend and the alcohol structured fuel trend at higher load of 16bar: the nanoparticles distribution is higher and related to bigger sizes, compared to the lower load.

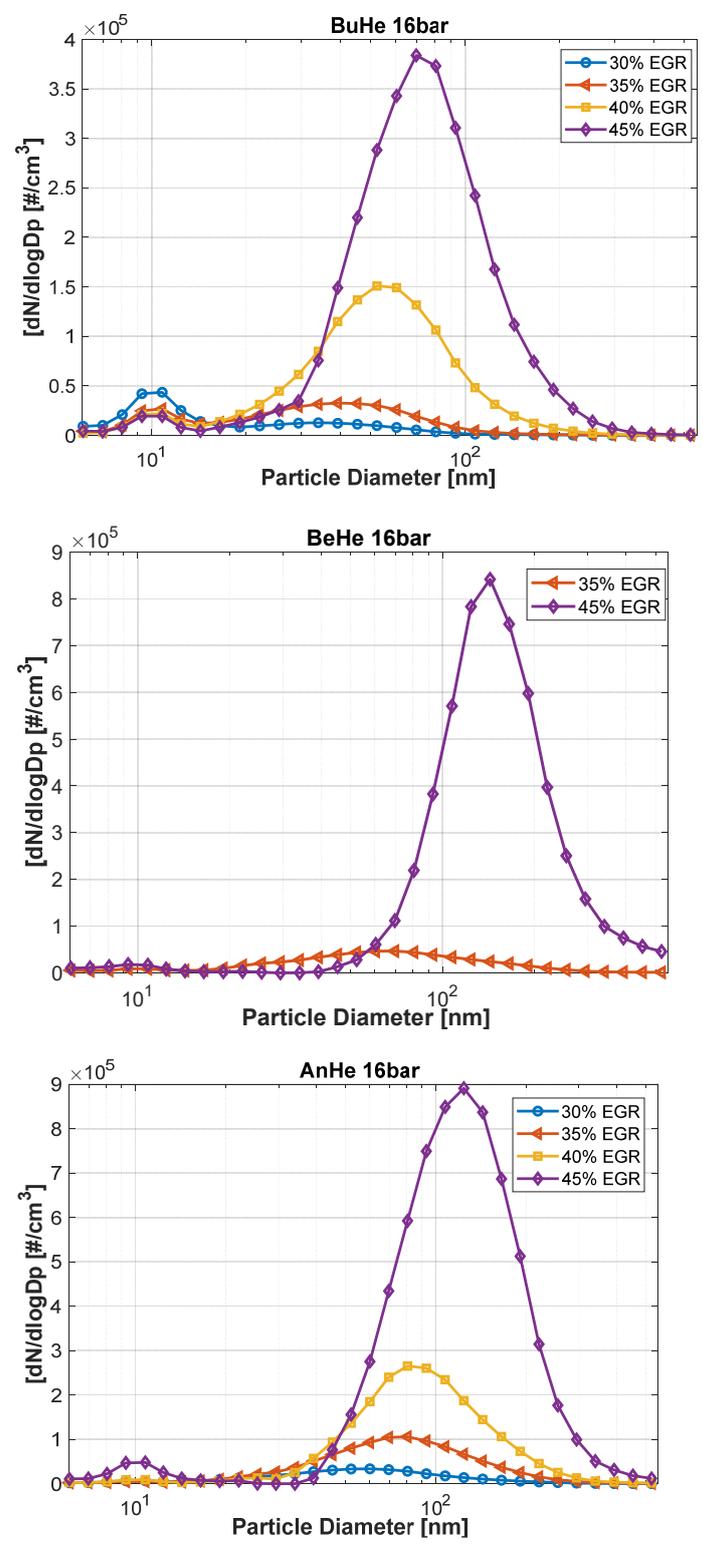


Figure 4.36: EEPs concentration and size distribution for the three fuel at 16-bar load with EGR percentage variation

CHAPTER 5 - CONCLUSIONS AND FUTURE IMPROVEMENTS

Low temperature combustion (LTC) has a great potential to achieve high engine efficiency and low NO_x and soot emissions. This research investigates combustion and emission characteristics of three different fuel blends, all having a RON value around 70: the tests have been conducted under different loads, EGR rates and injection pressures, with single injection strategy.

At the end of this analysis, several conclusions can be assumed:

- Ignition delay has a strong influence on the emissions: longer air-fuel mixing leads to lower combustion temperature with soot increase and NO_x emissions reduction, while lower ignition delay means higher NO_x and less soot. The three high-octane alternative fuels show different ignition delay: the longest is given by the butanol blend, due to its less reactivity and alcoholic molecular structure. Benzaldehyde and anisole show shorter ignition delay, because of their higher reactivity. It's interesting to see that these fuels provide longer ignition delay than diesel, resulting in both less NO_x and soot emissions, due to their structure which provides cleaner combustion.
- Combustion phasing is a key factor for these fuels: the more retarded the CA50 position, the lower the NO_x and the soot emissions. The best CA50 values for an optimal combustion are located between 0 and 10 CAD/ATDC, due to unacceptable heat losses before TDC and low combustion efficiency after 10 CAD/ATDC. For these fuels, the optimum CA50 is retarded around 9 CAD/ATDC, showing a substantial decrease in NO_x, low soot emissions and longer ignition delay, providing also lower PRR inside the cylinder.
- The EGR rate influence on these fuels is clearly visible on combustion efficiency and emissions. Higher EGR rate leads to a sharply decrease in combustion efficiency, starting from 35% of EGR. The EGR effect is lowering the in-cylinder temperature: NO_x decrease until almost zero increasing the EGR rate up to 45%, while soot emissions increase sharply for an EGR rate higher than 35%. Butanol shows the best behaviour in terms of emissions, probably due to its molecular structure: butanol is an alcohol while benzaldehyde and anisole are aromatic compounds with a benzene ring in their structure as a soot precursor.

EGR rate between 35% and 40% is a great trade-off for NO_x and soot emissions, showing the lowest value for both at the same time.

- The injection pressure influences mainly the soot emissions, determining the atomization of the fuel particles. The higher the injection pressure, the lower the particulate matter. These fuels show incredibly low soot emissions, even at high load and high EGR rate: anyway the soot emissions decrease with an increasing injection pressure, for all the three fuel blends. In addition, higher injection pressure enhanced the air-fuel mixing, providing shorter ignition delay, shorter burn duration and high combustion efficiency.
- Engine exhaust particle sizer (EEPS) analysis shows the effects of EGR rate, load and fuel structure on the soot formation: benzaldehyde and anisole, with their structures, provide much more soot emissions than butanol, in any condition. The concentration and size distribution of nano-particles in the exhaust gas increase with the load and the EGR rate.

In conclusion, the high-octane fuel structure highly influences the combustion: it is proved that alcohols like butanol are better than aromatics hydrocarbons like benzaldehyde and anisole, based on the emissions and performance on a heavy duty diesel engine.

These fuels also present some limitations, such as their unstable and difficult to control combustion, mostly at lower loads. It is impossible to keep the combustion stable: high EGR rate and too retarded SOA always lead to misfire, due to the low reactivity of these alternative fuels.

Future improvements

Alcohols as fuels can seriously become a great discover in the next years, mostly because of their lower cost related to other alternative fuels and their clean combustion. In this research, a high percentage of butanol blended is used, resulting in a certain complexity in keeping the combustion stable at low loads. As a solution, the investigation of lower alcohol percentage in the fuel blend could be interesting.

In addition, emission trade-off is not so good at high loads for high-octane fuels with PPC: additional studies with multiple injections, increasing fuel-air mixing and reducing soot emissions, could be a solution.

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