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Dual-Fuel Combustion Characterization on Lean Conditions and High Loads

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Dual-fuel combustion characterization on lean conditions and high loads

Master's Thesis

Espoo, 12.10.2016

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Abstract of master's thesis

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Abstract

Unstable oil markets combined with the alarming statistics of continuously growing emission problems causes anxiety among many nations. The greatest dilemma lies in the answer about how to rationally overcome the dependency of fossil based energy sources. The truth seems to be found on utilizing renewable energy generating low emissions. Methane is suggested as one of the worthwhile solutions for substituting crude-oil based fuels. Methane as a fuel combined with modern engine technology seems to open possibilities solving the above mentioned problems.

Dual-fuel technology is suggested as a solution for effectively utilizing alternative fuels in the near future. Charge air mixed methane combined with a compression ignition engine utilizing a small diesel pilot injection seems to form a worthwhile compromise between good engine efficiency and low emission outcome. Problems concerning dual-fuel technology profitableness seems to be related to fully control the combustion in relation to lean conditions. Lean operating conditions solves the problems concerning pumping losses, but brings challenges in controlling the slow heat release of the premixed methane.

A study concerning lean operation dual-fuel combustion was executed. In the thesis, a single cylinder ‘free parameter’ diesel engine was adapted for dual-fuel (diesel-methane) usage. A parameter research related to lambda window widening possibilities was carried out. The main variables studied were diesel pilot injection timing and pressure, diesel substitution rate and the relation between combustion characteristics and charge air temperature.

Diesel pilot injection parameter optimization seems to affect less on combustion quality compared to the benefits from increasing the charge mass reactivity through preheating the charge air. Improvements in combustion quality were noted in all lean condition tests in the diesel substitution rate scale 0 – 70%. However, significant variation in combustion characteristics were noted through the whole substitution rate range. Remarkable accomplishments in combustion behavior were experienced in the misfire boundary conditions in a form of premixed mixture ignition in the end gas region (PREMIER).

Keywords dual-fuel, methane fuel, combustion character

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Öljymarkkinoiden epävarmat ennusteet sekä jatkuvasti kohoavat päästöt aiheuttavat maailmanlaajuista huolta tulevaisuuden kannalta. Suurimmat kysymykset ongelmatilanteeseen liittyvät fossiilisten raakaöljyjen korvaamismahdollisuuksiin. Ratkaisu vaikuttaisi löytyvän uusiutuvista, matalia päästöarvoja tuottavista energianlähteistä. Yhdeksi kannattavaksi raakaöljyn korvaajaksi on ehdotettu metaanikaasua. Ennusteet metaanikaasun eduista yhdistettynä modernien polttomoottorien kanssa enteilevät mahdollisuuksia ongelmien ratkaisemiseksi.

Dual-fuel-teknologia on esitetty hyväksi moottorityypiksi hyödyntämään uusiutuvia energianlähteitä. Imuilmaan esisekoitettu metaanikaasu yhdistettynä puristussytytteisen polttomoottorin kanssa, jossa polttoaine-seoksen syttymistä sekä palamista hallitaan pienellä dieselannoksella, vaikuttaisi tarjoavan ominaisuuksiltaan hyvän hyötysuhteen mataline päästötasoineen. Dual-fuel-teknologian haasteet liittyvät lähinnä laihojen seosten palamishallintaan. Laihoissa toimintapisteissä voidaan kuitenkin minimoida pumppaushäviöt. Laihat seokset aiheuttavat toisaalta samalla haasteita esisekoitetun metaanikaasun liekkirintaman etenemisnopeuden hallitsemisessa.

Kyseisessä diplomityössä tutkittiin dual-fuel-palamista laihoilla toimintapisteillä kokeellisin menetelmin. Kokeet suoritettiin yksisylinterisellä “vapaaparametrimoottorilla”, joka työn aikana muokattiin dual-fuel-käyttöön soveltuvaksi. Kokeet keskittyivät parametritutkimuksiin jotka liittyivät lambdaikkunan laajentamismahdollisuuksiin sekä lämmönvapautumisen käyttäytymisen analysointiin eri olosuhteissa. Kokeissa muutetut pääparametrit olivat: diesel pilottiruiskutuksen ajoitus sekä paine, ilmakerroin, diesel-polttoaineen korvausaste ja imuilman lämpötila.

Imuilman lämmitys näyttäisi vaikuttavan selvästi myönteisemmin laihaseospalamisen laatuun diesel pilottiruiskutuksen parametreihin nähden. Palamisen hyötysuhteen monitoroitiin nousseen läpi käytetyn diesel-korvausasteskaalan (0 – 70%). Palamiskäyttäytyminen osoittautui kuitenkin vaihtelevaksi riippuen diesel-korvausasteesta sekä ilmakertoimesta. Huomattavia saavutuksia palamiskäyttäytymisessä saatiin eteenkin laihoilla seoksilla lähellä syttymisrajaa, jossa koettiin hallittua palotilassa tapahtuvaa nakutuspalamista (PREMIER).

Avainsanat Dual-fuel, palaminen, metaanikaasu

Preface

This thesis was executed in Aalto University Internal Combustion Engine Laboratory as a research project funded by FLEXe. Also, a minor funding for finalizing the thesis was provided by Merenkulunsäätiö. I would like to thank both of the involved organizations for participating in the financial aid for the interesting project contributing to the knowledge of possible future engine technology.

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Finally, I would like to thank all my family, closest friends and Laura for at least trying to understand my enthusiasm in respect of engine technology. A particular honor belongs to my mother Ingemo for all the understanding and personal support throughout my entire life.

Helsinki, October 2016
Rasmus Pettinen

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Nomenclature

Abbreviations:

AFR	Air-fuel ratio
ATDC	After top dead center
BTDC	Before top dead center
CA	Crank angle
CAD	Crank angle degrees
CI	Compression ignited
COV	Cycle of variation
DF	Dual-fuel
E%	Diesel substitution rate
EGR	Exhaust gas recycling
EGT	Exhaust gas temperature
EHVA	Electro hydraulic valve actuator
HCCI	Homogenous charge compression ignition
HRR	Heat release rate
IMEP	Indicated mean effective pressure
LHV	Lower heating value
ppm	parts per million
PREMIER	Premixed mixture ignition in the end-gas region
PRR	Pressure rise rate
QHR	Cumulative heat release
RCCI	Reactivity controlled compression ignition
SI	Spark ignited
SOI	Start of injection

Chemical Compounds and Emissions:

CH_4	Methane
CO	Carbon monoxide
CO_2	Carbon dioxide
HC	Hydro carbon
N_2	Nitrogen
NO	Nitric oxide
NO_2	Nitrogen dioxide
NO_x	Nitrogen oxides
THC	Total hydrocarbon

Greek Letters:

λ	Lambda
λ_{CH4}	Methane lambda
λ_{DF}	Dual-fuel lambda
φ	Equivalence ratio

1 Introduction

The technological evolution during the 20th century combined with the global growth of population, human welfare and the trend of world politics have redefined the requirements of the modern energy markets. The growing problem with unstable crude oil markets together with demands of preventing the globally growing emission levels opens many opportunities for implementing alternative energy sources. Although the predictions of the earth's crude oil cache running out are arguable, the assumptions of constantly rising oil prices are not completely wrong. This phenomena has its roots in the fact that when the worthwhile crude oil spots runs out, the oil draining must be done on places that requires greater investments.

Methane gas is suggested as one of the most reasonable options substituting crude oil based fuels for many reasons. One outstanding motive is that methane as a fuel, compared e.g. with gasoline, has a reasonably good well-to-wheel efficiency leading to a reduced carbon footprint [1]. Also, its availability is commendable in form of natural gas and can be produced via several worthwhile methods (e.g. methanogenesis [2]). Due to its characteristics, several different combustion applications benefits from the usage of methane-fuel. This includes dual-fuel (DF) technology. The dual-fuel technology is basically a technique where two different fuels are combined in order to attain the strengths from both. Several studies have already proved that when used correctly, methane fuel combined with a compression ignited combustion (using diesel pilot injection) is a worthwhile technology .

Like all fuel types, methane suitability as a fuel is restricted to the boundaries of the chemical characteristics. Specifically in dual-fuel operation, this means several challenges: The low-load conditions are found tricky, since the operation on sufficient efficiency levels require lean mixture combustion that are hard to control. This easily leads to incomplete combustion originating from partial misfiring or flame quenching [3]. High load operation efficiency on the other hand is reliant of the methane characteristics during very lean combustion, as the high cylinder pressures sets threshold limits for a controlled heat-release cycle. More specifically: Uncontrolled high load combustion may lead to engine knock or other abnormal combustion phenomenon affecting negatively both the engine life-expectancy and causing low thermal efficiency [4, 5]. Other challenging features of methane have been found: The unburned methane emissions are problematic to handle using conventional exhaust gas after-treatment systems. This forces engineers to re-design the mechanical dimensions of a dual-fuel application so that the so-called "methane slip" (exhaust CH_4) is reduced to its minimum.

This thesis is a continuation of Juhani Törmänen's [6] and Alberto Murillo Hernández [7] studies concerning both variable valve actuation and exhaust gas recycling (EGR) parameter impact on a methane DF- engine. Like mentioned before, there are a few challenges with respect to methane dual-fuel combustion control in both ends of the load scale. It seems that the efficiency in DF-modes increases through the combination of raising the lambda value within controllable combustion conditions. Studies have however proved that although the ultra-lean combustion strategy might be a tool for achieving a balance between good efficiency and suitable emission levels, it is hard to implement because of the physical boundaries causing risks of unwanted combustion events. This study's main goals are to study and understand the factors that impact the DF-combustion limitations, and to find reasonable methods how to bypass these thresholds without losing the gain that DF-combustion serves.

The main goals of the study are:

1) Studying the lean condition dual-fuel combustion characteristics
2) Understand the relations between dual-fuel combustion behavior and parameter setup (diesel pilot injection, substitution rate, lambda etc.)
3) Finding possibilities to extend the lean combustion operation region

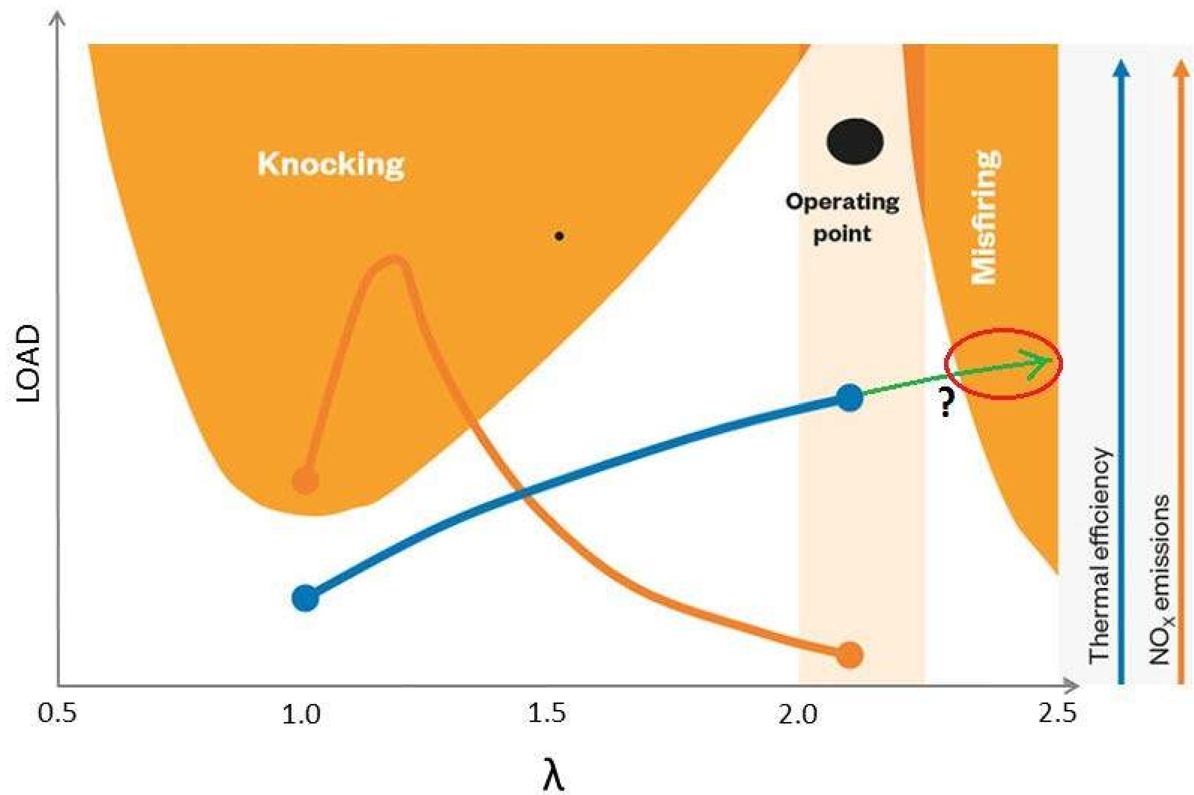

Figure 1. Wärtsilä studies in DF-efficiency and NOx emissions in relation to lambda. The green line represents the needs of further studies [8]

2 Background

2.1 Combustion

The internal combustion engine technology has evolved throughout the years in various directions. This factor makes a modern "conventional" internal combustion engine rather complex to describe. The injected fuel is generally transformed into usable energy by the piston moving in the cylinder transferring combustion originated pressure into mechanical work. The pressure rise is dependent of the thermochemical reaction behavior caused by fuel oxidization. Despite of this common working principle, the combustion phenomenon is very different between the Otto and Diesel operated engines. In order to understand the combustion characteristics of a DF-cycle a few types of combustion phenomena must be understood.

2.1.1 Premixed combustion

Ideally, a premixed combustion of fuel and air occurs when the AFR (air-fuel ratio) is precisely at a stoichiometric state ($\lambda = 1$). As the premixed combustion requires the fuel and air to be mixed beforehand, it yet seldom fulfills the criteria in an ideally homogenous form. The premixed combustion is a chain of chemically controlled processes, which are starting from an ignition source and continuing as a flame front propagation. The chemical chain reactions are taking place at the flame front releasing energy in form of heat as fuel gets oxidized. The premixed combustion flame speed for a specific fuel is severely dependent of the gas kinetics. Depending of the circumstances, premixed combustion can be divided into laminar and turbulent flame structures.

Premixed laminar flame is a result of a self-sustaining chemical process taking place in a shape of flame front, heating the unburned mixture in the laminar environment turning them into combusted products. The laminar flame front can be divided into two regions: a preheat zone where the unburned mixture temperature increases through conduction, and a reaction zone where the exothermic chemical reactions take place [9]. The fuel attributes defines the premixed laminar combustions fundamental burning properties, defining its diffusivity, exothermicity, and reactivity [10]. Also, the laminar flame characteristics are dependent on the environment temperature, pressure and equivalence ratio. A typical laminar flame front reaches a velocity between 0.5-2 m/s. Due to the high engine speed and great air mass flow levels during the gas exchange process in relation to the flow rate areas, laminar conditions seldom occur in internal combustion engines.

When a turbulent premixed air-fuel mixture is ignited at a stoichiometric state, a turbulent flame front propagation starts from the reaction point through the cylindrical volume. The turbulent flame front propagation can reach speeds of 20-25 m/s, but it severely dependens on several variables such as charge movement (swirl, tumble or squish), environment temperature, fuel/air equivalence ratio and charge density. The turbulent flame speed is also highly dependent on the turbulence intensity, which in turn is close to directly proportional to engine speed. The relation between engine speed and turbulence intensity therefore explains why an Otto-type engine operation is even possible. The premixed combustion is nowadays mostly exploited in port injected Otto (spark ignited) engines, but can also be implemented in compression ignited applications e.g. in HCCI (homogenous charge compression ignition) and in DF applications. A more comprehensive description of the turbulent premixed combustion can be read in the "*Internal Combustion Engine Handbook*" by van Basshuysen and Schäfer [11] and in "*Internal Combustion Engine Fundamentals*" by Heywood [9].

2.1.2 Non-premixed combustion

Opposite to the premixed flame phenomenon, diffusion flame is described as a reaction where the fuel-air mixing process limits the reaction rate. Specifically this means that the mixing process of the combustion occurs continuously and during the moment of injection via a turbulent flow, until the process runs out of either substance [9].

Diesel combustion is a compression ignited process where the fuel is injected to the combustion chamber with high pressure and velocity close to the piston top-dead-center (TDC). The increased charge density, temperature and pressure causes the injected diesel fuel to firstly decompose into tiny droplets (primary atomization) followed by evaporation, ignition and combustion. Commonly is the diesel combustion divided into a four phases: 1) ignition delay, 2) premixed combustion, 3) diffusion combustion and 4) late combustion phase. [9, 11, 12]

1) Ignition delay: The time between diesel injector needle opening and start of combustion. During the injection delay phase, evaporated fuel is premixing with air. The local air-fuel-ratio (AFR) varies notably depending of the injector nozzle orifice geometry and fuel spray velocity. [9]

2) Premixed combustion: At this moment, a limited portion of the injected fuel has had time to evaporate into a combustible gaseous mixture. The temperature of the evaporated mixture exceeds the fuels auto ignition temperature and starts the premixed combustion phase. When combustion occurs this premixed gas burns rapidly resulting in a high pressure rise rate (PRR) and heat release rate (HRR) that can lead into remarkably high combustion temperatures and undesired emission levels. The whole phenomena lasts commonly only a few crank angles. [9]

3) Diffusion (non-premixed) combustion: The third phase takes place after the premixed combustion phase has consumed all the available premixed fuel. The diffusion flame combustion relies on the fast flowing fuel that turbulently mixes with air. The third phase includes several processes: liquid fuel atomization, vaporization, mixing of fuel and air and preflame chemical reactions. The rate of diffusion flame burning is controlled by the mixing of fuel and air. This phase is commonly known as the dominant event during diesel combustion, also known as *mixing controlled combustion.*[9]

4) Late combustion phase: The last phase then takes place during the later expansion stroke as the combustion occurs in low oxygen and temperature environment. The late combustion phase releases the heat from unburnt fuel available present in form fuel-rich end-gas. Therefore, the late combustion phase has commonly a reasonably high influence in exhaust gas emission formation. [9]

An extensive and a deeper detailed description of diesel combustion in general can be read in "Internal Combustion Engine Fundamentals" book by Heywood [9]. Also, a widely accepted conceptual model of diesel combustion based on optical measurements is made by Dec in his SAE-paper "A Conceptual Model of DI Diesel Combustion Based on Laser-Sheet Imaging" [12] (figure 2).

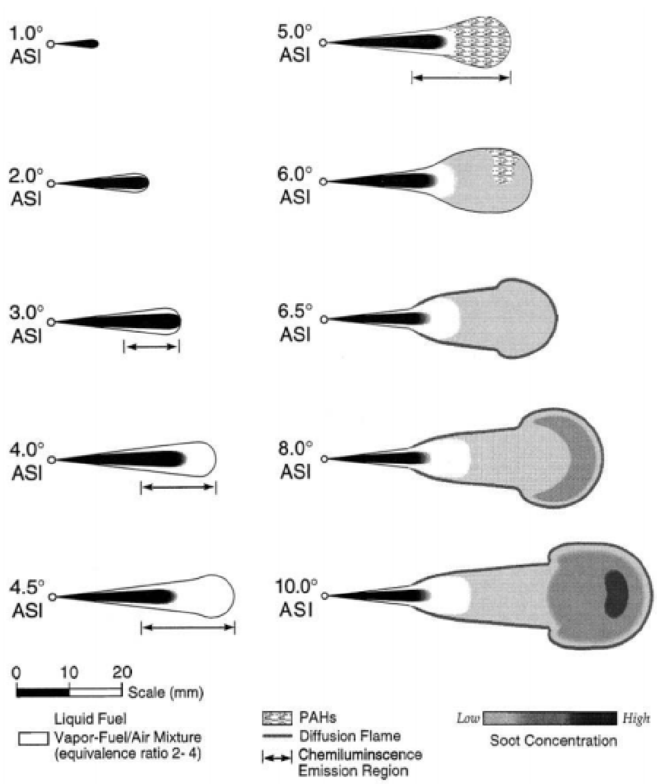

Figure 2. A conceptual model of diesel combustion. The term ASI represents "after start of injection" and PAH "polycyclic aromatic hydrocarbons"[12]

2.1.3 DF-combustion

Like the definition states, DF-technology utilizes two different types of fuels to form the combustion event. The dual-fuel concept is based on controlling the combustion of a pre-injected and premixed (e.g. port fuel injected) low reactivity fuel by igniting it with a fuel of high reactivity attributes [13]. A technology easily associated with dual-fuel is the reactivity-controlled compression ignition (RCCI) concept [14]. In general, the RCCI utilizes similar attributes of combining high and low reactivity fuels, but differs from the DF-model by the timing parameters of high reactivity injection. In DF-combustion, the high reactivity fuel is injected close to TDC, meaning that the combustion is dependent of both chemical kinetics and high reactivity spray physics. The dual-fuel combustion process is therefore highly dependent of both the diesel spray and ignition features together with the injection type and concentration of the gaseous substitute charge [15]. However, RCCI combustion relies only on the chemical kinetics of the combined fuels [13]. A modern diesel engine commonly serves as a good platform for DF-usage, as both the compression ratio and high pressure common-rail injection system makes the engine adaptable for a wide scale of operating modes [16].

The dual-fuel combustion shares characteristics with both spark ignited (2.1.1) and compression ignited (2.1.2) engines. According to the studies of G. A. Karim [15] the heat release phasing can traditionally be divided into three overlapping components (figures 3, 4): The first heat release component occurs as the diesel pilot spray is injected, evaporated and auto-ignited, resulting in a diffusion flame (I). The start of the second phase is considered as the moment after the surrounding temperature of the pilot spray caused combustion overcomes the auto-ignition point of the premixed lean gas mixture and initially starts the flame propagation. This results in a reaction where the gaseous methane fuel combusts in the vicinity of the premixed diesel pilot-region (II). The last phase then results in a turbulent flame propagation through the remaining lean gas mixture in the cylinder (III). An interesting additional detail G. A. Karim stated in his journal is concerning very lean gaseous operating modes: the bulk energy release mainly originates from the ignition and sequential prompt combustion of the primary diesel pilot spray zone (I). Moreover, some of the lean gaseous fuel-air mixture combusts entering a relatively fuel rich, high temperature area where the diffusion flame zone affects during the first phase (II). It seems that in cases like this very little heat release is derived from the bulk mixture located further away from the pilot spray. Thus there will be very little coherent lean mixture flame propagation taking place in regions (III). Specifically: An increased pilot injection mass flow will cause a higher energy release rate that contributes higher combustion chamber temperature throughout the cylinder. Higher combustion chamber temperatures increases the ignitibility of the surrounding secondary fuel component resulting in improved combustion quality.

Note that the figures presented below (figures 3, 4) are only schematic, as the size of the pilot injection quantity affects more than proportionally in relation to the total energy released.

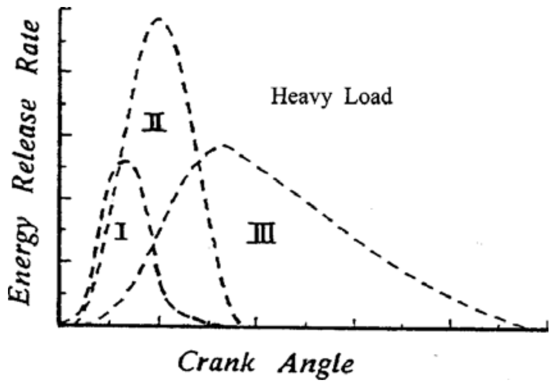

Figure 3. A dual-fuel HRR trend on heavy load [15]

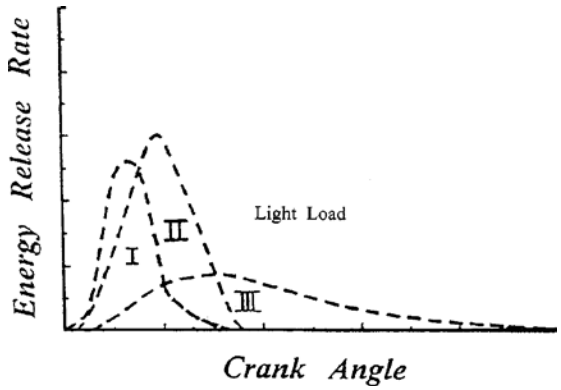

Figure 4. Dual-fuel HRR on light load [15]

PREMIER combustion

A possible additional attribute in DF-combustion was later discovered by Azimov et al. [17, 18]. According to this study, an additional possible component of heat release could be achieved: a bulk ignition of the end gas region. The premixed mixture ignition in the end-gas region (PREMIER) event is a lean condition dual-fuel heat release phenomenon where the highest heat release rates are caused by a controlled autoignition of the end gas. Therefore, PREMIER combustion can be defined as a controlled knock-combustion event. The concept is based on lean burning kinetics where the slow moving, methane flame front ignited by the diesel fuel leaves unburned hydrocarbon-mixtures in the surrounding while proceeding through the combustion chamber. The ongoing heat release caused by both diesel and premixed methane combustion eventually rises the environment temperature above the leftover methane-air mixtures autoignition point. This causes a simultaneous autoignition of the end gas in several limited locations (known as exothermic centers), resulting in a rapid heat release rise. A requirement of a PREMIER event is that the combustion of the end gas cannot occur during or following the moment of the rapid release of diesel pilot energy. Crossing these boundaries could possibly result in engine knock, decreasing the thermal efficiency or even engine failure. Optimizing the combustion conditions seems to be the key for controlling a PREMIER typed combustion. The main affecting parameters can be considered as: methane lambda, diesel pilot injection timing, pressure, quantity and EGR rate. The challenge of controlling the PEMIER combustion is to control the relatively sensitive methane lambda as both knock and misfire can occur. Properly accomplished PREMIER combustion generally increases the engine performance levels by improving the indicated mean effective pressures (IMEP) and thermal efficiency. The combustion process generates therefore higher HRR values increasing the in cylinder temperatures, resulting in decreased CO and HC values, simultaneously increasing NO_x emissions by promoting oxidation reaction levels of nitrogen and oxygen.

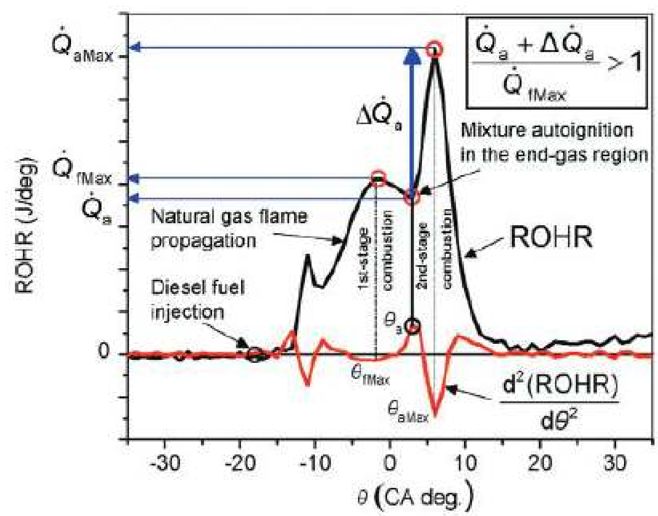

Figure 5. A schematic heat release trend of a PREMIER combustion. [17]

Diesel substitution rate E%

The relation which determines the amount of injected energy replaced by the secondary fuel, methane (figure 6) is called diesel substitution rate, E% CH_4. This ratio therefore defines the specific combustion characteristics and heat release behavior compared to CI- and SI- engine combustion. The substitution rate is calculated from the required energy share replacing the original total injected diesel energy content, and does not therefore take the required mass flow into account. The required substituting mass flow is calculated from the fuels energy density using the lower heating value LHV. [19]

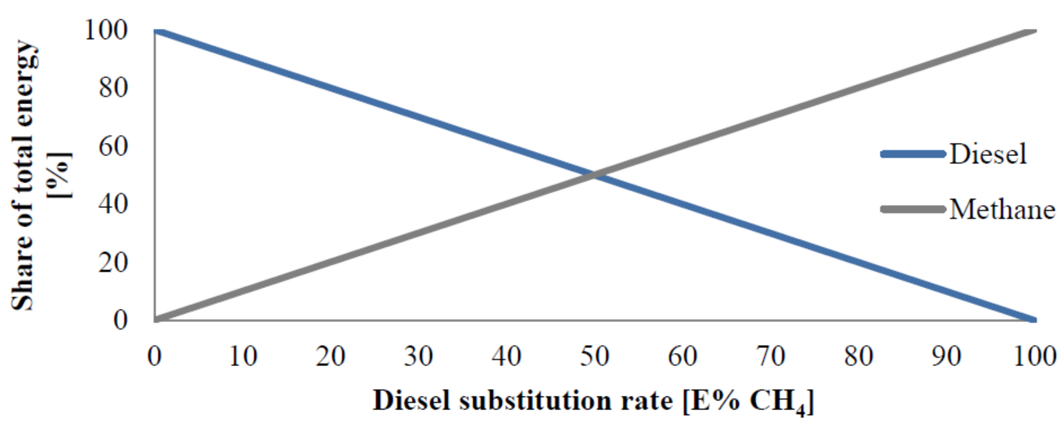

Figure 6. Share of energy in relation to the diesel substitution rate. [19]

Optical demonstration of a DF-combustion

An image of a DF- (diesel-syngas) combustion taken with a high speed camera through a Bowditch type piston is seen the figure (figure 7) below. The diesel pilot is injected before top dead center (BTDC) using a four-hole injector and was used as an ignition source for the syngas bulk mixture. The DF-combustion characteristics described earlier in this paragraph can clearly be seen: the combustion starts after an ignition delay from the areas where the diesel rich areas ignite. The pilot injection originating heat release ignites the surrounding air-syngas mixture, forming a flame front propagation throughout the entire surface. The glowing sections represent soot particles formed in the combustion event.

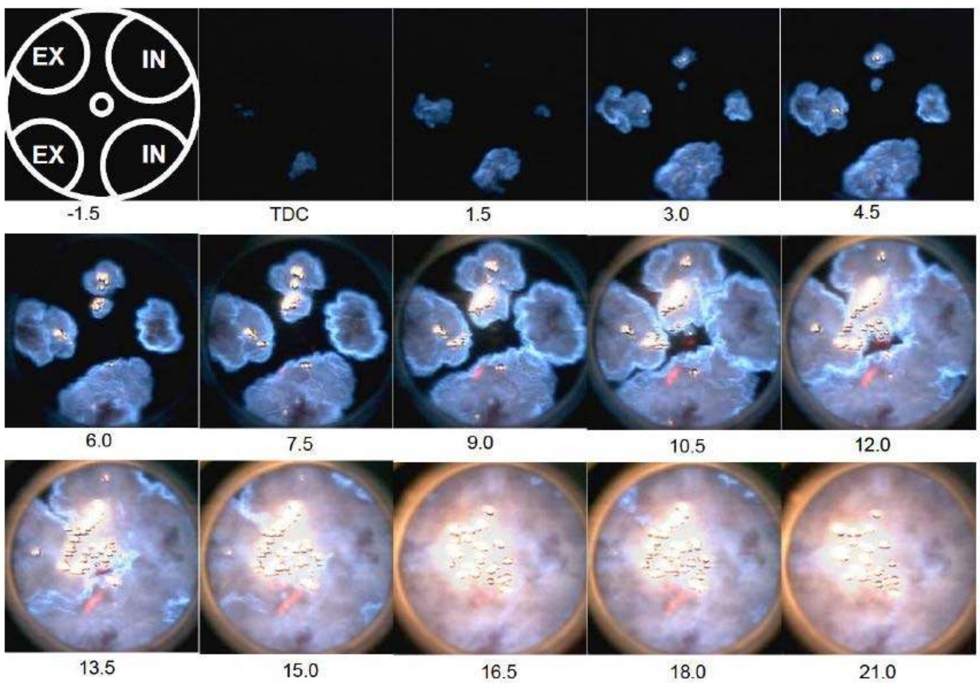

Figure 7. An optical demonstration of a DF-combustion [5]

2.1.4 Emissions in dual-fuel applications

Main emission components of a DF-engine can be categorized as: CO, NO_x and THC. Emission formation on a DF-application is greatly dependent on the combustion conditions. The emission characteristics shares similarities with CI- and SI-engine and is generally determined by the used substitution rate E% and equivalence ratio ϕ. The emission formation can be divided into subcategories following:

CO

Carbon monoxide (CO) and carbon dioxide (CO_2) emissions are formed in oxidization process between oxygen and fuel. An ideal combustion occurs at a stoichiometric state where only hydrocarbons and oxygen are present [9]. A combustion process is however seldom perfectly ideal and is controlled by both equivalence ratio and combustion temperature. CO is formed as a combustion by-product in fuel rich areas where the local oxygen supply is insufficient. These conditions often occurs in SI-applications as the required AFR is close to or below the stoichiometric level. CO emissions can also be formed during lean combustion. Since the combustion process is highly dependent of environment temperature, CO may be formed as a partial oxidization e.g. during late expansion stroke or close to low temperature cylinder walls. [5]

CO emissions always indicates incomplete combustion. Compared with a conventional Diesel engine, DF-applications in general produces significantly larger CO levels as a result of charge portion of premixed combustion and lower combustion temperatures [16].

NO_x emissions

The general definition of nitrogen oxides (NO_x) consists of components both from nitric oxide (NO) and nitrogen oxide (NO_2). NO_x emissions are formed in internal combustion engine-applications in high temperature conditions [9]. NO_x emissions are considered as hazardous emission components, destroying ozone layer, forming nitric acid rains and causing respiratory health issues. NO_x formation can originate due to different phenomena: Thermal NO_x, prompt NO_x, fuel origination NO_x and NO from N_2O. Thermal NO_x is generally seen as the main source of total NO_x emission, forming approximately 90-95% of all engine nitrogen oxide emissions [5]. Due to its high activation energy of the rate limiting reaction, mechanism of thermal NO_x formation is highly temperature dependent. Prompt NO_x on the other hand is formed in locally fuel rich areas where N_2 reacts with HC radicals [5].

Thermal NO is composed both in the flame front area and in postflame gases. Formation mechanism of NO in near stoichiometric fuel-air mixtures are [9]:

$$O + N2 = NO + N \tag{2.1}$$

$$N + O2 = NO + O \tag{2.2}$$

$$N + OH = NO + H \tag{2.3}$$

And NO2 formation mechanism is mainly originating from the flame front originating NO [9]:

$$NO + HO_2 \rightarrow NO_2 + OH \tag{2.4}$$

Due to the apparent dependency of combustion and environment temperature, NO_x emission formation can be manipulated by controlling the combustion and HRR trends. DF-combustion has been found to produce less total NO_x emissions than a conventional Diesel engine. The overall combustion temperature in DF-modes is significantly lower, decreasing the main NO_x origin, thermal NO_x. NO_x formation using dual-fuel technology is however highly dependent of the diesel pilot injection zone where the local temperature and longer reaction times promotes the formation of nitric oxides. Increasing the pilot injection quantity therefore enlarges the pilot injection area, contributing larger NO_x formation. Locally formed NO_x caused by hot combustion chamber zones are however trickier to solve as it in general requires combustion chamber geometry modifications. [3, 15]

THC (total hydrocarbon) emissions

Besides formed CO, incomplete combustion results in unburned hydrocarbons (HC). HC emission are formed by partly- or unreacted fuel or by lubrication oil escaped from engine lubrication system. HC emission formation can be caused by several reasons and is often divided into subcategories dependent of the type causing the emission formation. Great shares of HC emission causing problems originates from: cold cylinder walls and combustion chamber surfaces (flame quench), combustion chamber crevices, abnormal combustion (e.g. misfire) or fuel slip caused by the valve overlap period during the gas exchange process [5].

Largest contribution of HC-emission in applications where premixed fuel mixtures is used seems to be originating from combustion chamber crevices: Some of the premixed air-fuel gas escapes during the compression stroke into cold, narrow volumes where the fuel is unable to ignite. Some examples of where most of crevice emissions form are piston ring packs and cylinder gasket crevices. Also, engine types where premixed combustion takes place, flame quench in cold combustion chamber areas can occur. The low temperature in these regions causes the flame front to quench as the temperature decreases below the fuels self-ignition point. [5]

Diesel engines are known for producing low HC emissions. Due to the high reactivity attribute of diesel fuel most of the HC is oxidized into other emission components. Diesel pilot injection caused HC emission are therefore not generally considered as a prominent source of HC emissions in DF-applications.

Uncontrollability of the port injection timing of the secondary fuel in relation to valve overlap timing forms HC emissions of fresh fuel slip. Premixed fuel is able to escape during the overlap period if the pressure difference over the gas exchange process is positive. Manipulating the pressure difference, valve overlap period or start of injection (SOI) of the port injected fuel affects greatly the amount of fuel escaping from the intake manifold directly to the exhaust port. [20]

Both substitution rate E% and equivalence ratio ϕ affects the unburned hydrocarbon concentration in DF-applications. The share of unburned CH_4 is referred as methane slip. Methane slip can be caused by all the above mentioned reasons depending of the used engine application geometry in relation to equivalence ratio [19]. The equivalence ratio in general also affects the methane slip by controlling the methane flame front propagation quality, speed and determining the secondary fuels misfire-limit [19]. Karim G. states however that combustion originating methane slip is highly dependent of the heat distribution originating from the diesel pilot injection [15]. The diesel pilot injection characteristics depends on the used substitution rate E% and diesel pilot injection parameters combined with the used injector. A greater pilot injection quantity contributes an even and a sufficient temperature for methane to completely oxidize.

2.2 Parameter influence

Engine parameters are obviously a central key for combustion and emission control. Since DF-combustion is a multidimensional phenomenon dependent of several variables, it is important to understand the basic consequence of each main parameter. This section is therefore a summary of known combustion behavior affected by individual engine variables.

2.2.1 Lambda, λ

Although the usable lambda range in DF- mode is moderately narrow, combustion characteristics, ignition delay and emission formation vary significantly depending of the air factor λ. A general rule concerning DF-combustion character in relation to lambda states: Lambda values close to stoichiometric improves the flame front propagation quality outside the diesel pilot burning zone. In the opposite, lean air-fuel mixtures deteriorates the flame front propagation until a lambda limit is exceeded and no flame front is seen. During above mentioned lean conditions combustion will occur only in the immediate pilot flame diffusion flame region causing a *partial burning* state [11]. The unburned methane residue in these areas leads to methane emission slip that is easily detected as an ascending trend of exhaust CH_4. The combustion quality in these conditions can be improved by lowering the substitution rate. [4, 15]

Also, Karim G.A. states in his research that ignition delay behavior between DF- and diesel combustion differs significantly [15]. As pictured below (figure 8), diesel ignition delay decreases when ϕ is increased. Meanwhile in DF-mode, starting from leanest lambda conditions show a moderately low ignition delay. Richer conditions on the other hand makes the delay longer until a maximum is reached. If still enrichened after the maximum peak, the delay will drop down before reaching the fuels stoichiometric value.

Note that the equivalence ratio is defined: $\Phi = 1 / \lambda$

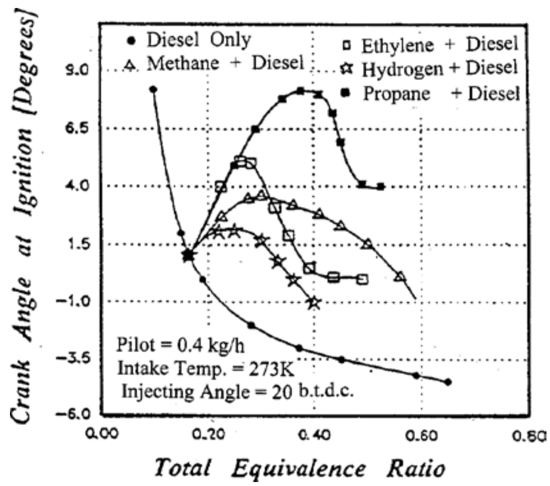

Figure 8. Dual-fuel ignition delay time in relation to Φ [15]

2.2.2 Diesel Substitution rate, E%

The diesel substitution rate E% is a central parameter concerning controllability of DF-combustion. Substitution rate determines the share of energy replaced by the secondary fuel and therefore significantly affects the heat release trend. In lower substitution rate conditions, diesel combustion and its affecting flame area is dominant and controls the combustion of the premixed methane fuel. In such conditions, methane ignition and combustion is strictly governed of the diesel pilot injection attributes. The heat release components of each fuel in lower substitution rates tends to overlap one another. The overlapping heat release components results in high HRR values and high combustion temperature as methane and diesel fuel combusts simultaneously. Higher diesel substitution rate conditions are however rather dependent on the premixed fuel's lambda. The diesel pilot injection quality only determines the size of ignition area from where the methane flame propagation start. In these conditions, heat release components are generally separated, resulting in lower HRR trend and combustion temperatures. [15]

The E% rate is not only determining the DF-combustion behavior between SI- and CI-combustion, but also specifies the required air mass demand. As methane and diesel fuels have different stoichiometric AFR- values ($AFR_{CH4} = 17$, $AFR_{Diesel} = 14.5$ [21]), the required air mass used be defined from the stoichiometry based on the combined fuels masses, $AFR_{DF,}$. The stoichiometric dual-fuel AFR value is therefore a variable that is dependent of the fraction of diesel mass compensated by the gaseous fuel, methane. In order to completely understand the control parameters during dual-fuel operation, the relation between the desired lambda, fuel- and air mass flows be known. A calculation of each lambda component in relation to total fuel mass flow is plotted below (figure 9).

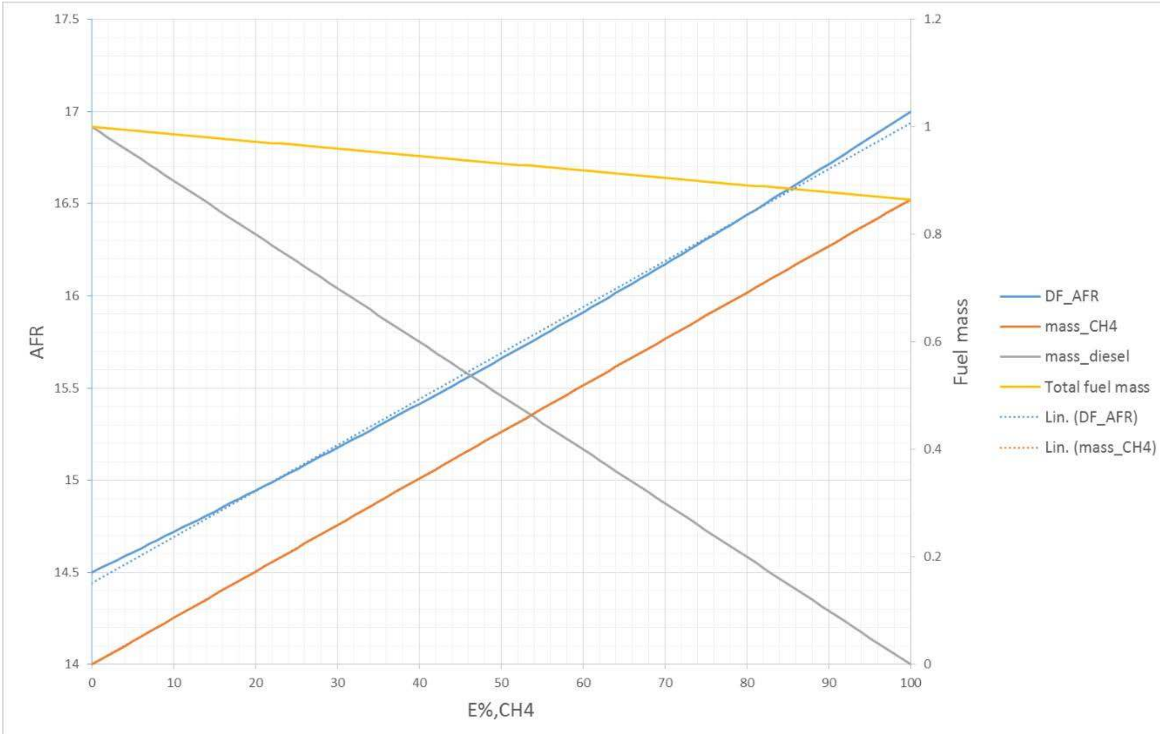

Figure 9. DF-massflows in relation to AFR

2.2.3 Diesel pilot injection parameters

In DF-mode, diesel pilot injection parameters controls the combustion of the secondary fuel, methane. The requirements of pilot injection parameters are dependent of both substitution rate E% and equivalence ratio ϕ [19]. As complete combustion, controlled exhaust emission levels and good thermal efficiency is desired, a controllable diesel pilot injection combustion is a requirement.

The diesel injection parameters determine both the heat released by diesel fuel and affects severely on local overall combustion chamber temperature defining the area where methane ignition is possible. Using a constant nozzle geometry, three variables can be manipulated:

(1) Diesel pilot injection timing (SOI)

(2) Diesel pilot injection pressure

(3) Diesel pilot injection duration

Diesel pilot injection timing, also known as start of injection (SOI) is the time when diesel injection is started in relation to TDC. SOI affects both the time of ignition and therefore impacts directly to the time of HRR in relation to piston movement. The total heat released in relation to TDC defines the cylinder temperature and cylinder pressure trend. Manipulation of SOI therefore also affects the emission formation as the combustion temperature is known as one of the main factors affecting emissions. Furthermore, diesel pilot injection timing influences directly the engine efficiency as optimization of HRR in relation to piston movement improves the heat transferred to piston work. In DF-applications: Higher cylinder temperatures promotes methane and promotes faster flame front propagation. [3, 9]

Diesel pilot injection pressure and duration are interdependent parameters specifying the required time to inject the diesel energy into the combustion chamber. Greater injection pressures decreases injection duration and vice versa. Increased injection pressures affects the HRR raising the combustion temperature and improving methane combustibility. Higher injection pressures also promotes fuel evaporation and spray penetration affecting positively both on ignition delay and diffusion flame formation. Longer diesel pilot injection durations on the other hand evens out the heat release in a longer period in relation to crank angle. Cylinder pressure trends are less steep and prevents temperature exceeding methane autoignition limit. [9, 19]

In general greater pilot injection quantities contributes improvements in dual-fuel application by increasing the combustion temperature. This results in higher thermal efficiencies lowering both CO and HC emissions. Due to higher reaction temperatures, larger diesel injection quantities enchases NO_x formation. High diesel pilot injection quantities also leads to rapid heat release, therefore increasing risks of early engine knock. [4]

2.3 Limitations and boundary conditions in DF-combustion

Since the dual-fuel combustion is a combination of different heat release phenomenon where the combustion behavior depends on several factors described in part 2.2, different limitations and boundaries may appear typical for both SI- and CI- engines. Running an engine longer periods over its limitations will typically result in either lower thermal efficiency, high emission levels or cause severe damage of internal parts of an engine. A dual-fuel engine can reach these limitations mostly by incorrect parameters that leads to uncontrolled combustion. Also, engine design affects the combustion behavior since the gas exchange phase, combustion chamber geometry and compression ratio plays a severe role in the heat release trend. In order to understand the limitations that guides the controlling parameters, several phenomena must be taken into account. In order to be able of avoiding these conditions, the cause of the different phenomenon are essential to understand. In this section, most common the DF -combustion and operation limiting factors and phenomena are explained.

2.3.1 Knock

Knock, also known as autoignition of the end gas, is a phenomenon often related to SI- engines. Knock conventionally occurs during the compression stroke as the local compression temperature exceeds the premixed fuels autoignition limit. This causes the air-fuel mixture to self-ignite before a controlled flame front has reached the end gas region, resulting in rapid high frequency local pressure and heat release rates fluctuating in the cylinder. The pressure waves caused by knock travels with the speed of sound through the combustion chamber which may cause the chamber to resonate at its natural frequency. Specifically, the pressure distribution in the combustion chamber is no longer uniform, resulting in high local pressure and temperature peaks generated by the cross travelling pressure waves until it gets dampen out. The knock phenomenon can therefore cause severe damage to engine internal components such as piston ring breakage, cylinder head, piston crown and top land erosion and piston melting, which in turn leads in low thermal efficiency and high emission levels. [9, 11]

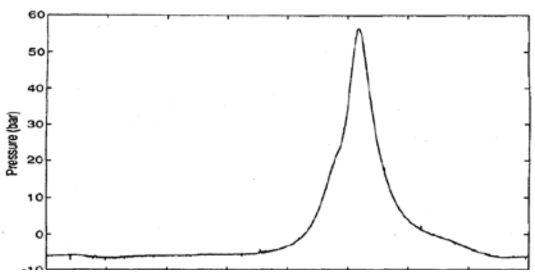

Figure 10. A common cylinder pressure curve

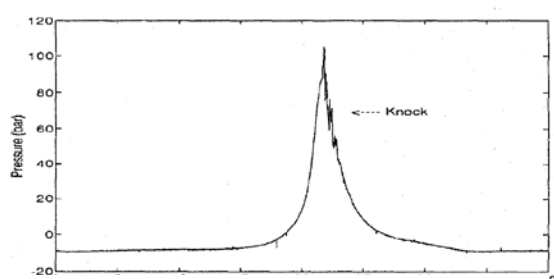

Figure 11. Cylinder pressure curve illustrating knock

Knock can be caused from various reasons, and is therefore defined by the origin:

Surface-ignition:

Surface ignition is caused by *hot spots*, as the local temperature of fuel-air mixture is heated by local high temperature areas in the combustion chamber area such as overheated valves, spark plugs or glowing combustion chamber deposit e.g. hot soot, lubrication oil, glowing carbon deposits or residual exhaust gases and even by diesel injector leakage. This form of knocking can occur both before (*preignition*) or after (*postignition*) the actual moment of controlled ignition. Surface ignition produces a turbulent flame propagation through the combustion chamber in an analogous manner to normal spark ignition. An early stage surface ignition caused by ignition of deposit particles forming one or more erratic sharp cracks is referred as *wild ping. Wild ping* is not continuous and disappears when the particles are exhausted and will reappear only when new particles enters the combustion chamber. The *preignition* phenomenon can be prevented by cooling the hot parts of combustion chamber i.e. changing the engine design (lowering compression ratio, valve cooling etc.) or fixing possible source of hydrocarbon leakage that accesses to the combustion are (injector and piston ring repairing). [9]

Spark-knock:
When autoignition occurs repeatedly during otherwise ordinary conditions in the combustion cycle is the effect called *spark knock*. *Spark knock* occurs after the fuel is ignited outside the flame front area, when the in cylinder temperature reaches the fuels auto ignition temperature caused by rapid pressure rise during compression. This results in an uncontrolled combustion state, resulting in local high energy release sections. *Spark knock* can start simultaneously from several locations during each stroke, and often ends up in a chain reaction caused by the pressure oscillating shockwaves travels in the combustion chamber. Spark knock can be evaded by either increasing the flame speed via precipitating the in-cylinder turbulence or by controlling the peak cylinder pressure and temperature. [9]

Knock caused by *spark-knock* can also easily be avoided in SI-engines by either retarding the spark advance or by reducing engine load. Since the start of the combustion is dependent of the pilot injection timing, similar knock avoiding strategy can be applied with DF-applications. [19].

Diesel Knock: Knock can also occur in diesel combustion. The diesel knock originates from the premixed combustion phase as the premixed fuel autoignites in the spray region. During diesel knock, the time between ignition delay and combustion is too long in relation to the already premixed diesel mass. The flame propagation of the premixed fuel increases the local temperature in the surrounding, simultaneously uncontrolled igniting the premixed fuel in several areas. [22, 23]

Knock in dual fuel engines:
Three types of *knock* can be identified in DF-combustion: *diesel-knock, spark-knock* and *erratic-knock*. *Erratic-knock* is a definition of a secondary ignition delay of the primary fuel, comparable with an uncontrolled PREMIER combustion. *Erratic knock* sensitivity is highly dependent of the diesel substitution rate and equivalence ratio used. [23]

To summarize: Depending on knock-type, DF-knock in general can be avoided by the equal manners described in each knock-description above: By retarding diesel pilot injection timing, manipulating engine combustion chamber geometry, decreasing combustion temperature or by lowering diesel substitution rate E%. [9, 19, 23]

2.3.2 Misfire

When the unburned premixed air-fuel gas is leaned ($\lambda > 1$), the combustion quality decreases. The poor combustion quality is a result of an increased flame development period, slowing down the flame propagation speed. If the mixture is leaned enough, the premixed flame propagation will not reach all unburned gas before exhaust valve opens. This phenomena is often referred as *partial burning*. When the mixture is further leaned, a limit of no notable combustion i.e. *misfire* will be noted. A stable operation condition of a conventional internal combustion engine is defined by the cycle-by-cycle variation. If the combustion process between several cycles are unequal, the engine *stable operation limit* has been exceeded. The variation of cycle-by-cycle indicates a *partial burning* or even *misfire*. [9]

2.3.3 Cylinder pressure and PRR

In order to be able to control and monitor the DF-combustion as desired, it is important to understand the in-cylinder pressure behavior during different engine operation conditions. Limitations of the pressure rise ratio and peak cylinder pressure must be set so that the boundary conditions can be detected without running into engine malfunction or high emission levels. The maximum cylinder pressure levels drastically affects the engines life expectancy. The acceptable maximum peak cylinder pressure is therefore naturally dependent of the engine design. The maximum cylinder pressure levels should be chosen low enough so that the risk of engine failure is reasonably small. However, high power output levels together with a good efficiency requires relatively high cylinder peak pressures so that most of the heat release from combustion can be transferred to work. A classic pressure rise rate (PRR) boundary used for determining stable combustion is 10bars/CAD [14]. Cylinder pressures exceeding above the applications recommended values may cause engine damage, such as deformation of connection rods, piston failure etc. The maximum cylinder pressure level should be chosen on basis of the engine design and purpose, but also by the used cylinder pressure sensor. In order to be able of monitor possible abnormal combustion e.g. knock, the cylinder pressure boundary should be set 10% below the maximum measurable value.

3 Methods

3.1 Test engine setup

The test engine is based on an AGCO 84AWI 6-cylinder common rail diesel engine converted into single-cylinder research application, also referred as LEO1. Similar 6-cylinder applications are generally used in off road vehicles such as tractors, harvesters and other agricultural implementations. The geometrical measures of the single-cylinder engine was held original and can be specified (Table 1):

Table 1. LEO1 test engine specifications

LEO 1 research engine

Cylinder bore (mm)	111
Stroke (mm)	145
Displacement total (l)	8.42
Displacement 1-cyl	1.4
Compression ratio	16.5

The LEO1 research engine can be defined as a free parameter system, practically meaning that all the physical variables can be manipulated freely without any restrictions of normal engines operation points. These all parameters are controlled by a field-programmable gate array supplied by National Instruments via a LabView-based software specially customized for the specific research engine. A deeper explanation of the engine setup can therefore be listed:

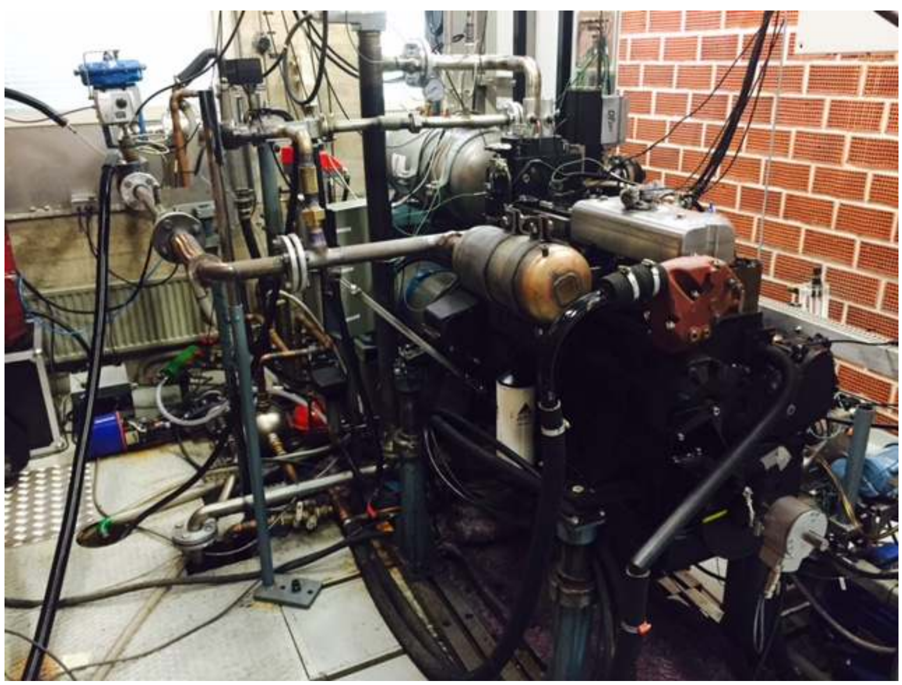

Figure 12. Test engine setup

3.1.1 Air mass feed, exhaust system and gas exchange

The LEO1 charge air feed is externalized by an industrial air compressor with a theoretical maximum pressure of 8 bars. Charge air temperature can be increased and controlled with an external heating core located in the charge air feed pipe located approximately 1m from the intake manifold. The charge air mass flow is computer controlled by an electronic butterfly valve positioned before the air mass heater. The butterfly valve position can be commanded by three different manners: by a manual position request or by a PID controller in relation to either boost or air mass flow targets.

Missing the turbocharger, the exhaust manifold is a relatively trivially built element. The system consists of the main tubing, a pulse absorber and a back pressure valve. An option for EGR-distribution also exists, but was left plugged due to thesis limitations. The pulse absorber is a parallel mount reservoir positioned approximately 20cm from the cylinder head port. The exhaust back pressure valve is by practical standards very similar to the charge air valve, commanded using a PID-controller integrated in the LabView system.

LEO1 has a camshaft-free gas exchange valve system EHVA (Electrohydraulic valve actuator), that controls the valve lift slope by high pressure hydraulic fluid controlled by the user demands. This feature allows the engine not only to run normal camshaft imitating configurations, but can also be configured to run unusual camshaft profiles physically not possible for normal camshaft designs. To particularize, EHVA capable of running multiple valve configurations such as standard timings, internal EGR preferences and Miller-cycles without the need of changing any hardware or physical components in the engine.

Figure 13. EHVA control solenoids

3.1.2 Diesel injection system

The diesel injection system is a simplified version of a multi-cylinder configuration, practically consisting of: fuel tank, two low pressure pumps, fuel filter, high pressure pump, high pressure rail and injector.

Both rail pressure and diesel injection parameters are controlled with the LabView-based ECU. The rail pressure is generated with an AC-motorized pump using a frequency transformer. The pressure level can be accurately manipulated with a PID-controller from 800 to 2000bars. Diesel injection parameters are controlled through a high-current, peak and hold type DI-driver (diesel injection-driver) capable of up to 5 injections/cycle.

The diesel injector used in the experiments is a Tier4final standard fulfilling, high pressure direct-injection injector commonly used in commercial off-road applications. The injector is equipped with a 9-hole nozzle. Recommended operation pressure for the injector is 250-2000bars. Due to the operating range of the pump, the operation pressure is however limited to a minimum of 800 bars. The DI-driver is capable of a minimum duration of 350μs (including both peak and hold currents).

According to the injector characteristics data, nozzle tip temperatures exceeding over 300 ^{o}C are not recommended. Exceeding the recommended nozzle temperature, a possibility of injector tip boiling resulting in potential clearance changes or even complete nozzle failure can occur. When substituting diesel with methane, diesel injection mass flow is decreased. As the injector cooling is completely reliant on the diesel mass flow, a significant risk of exceeding the recommended nozzle tip temperature exists. Since the injector is remained unmodified lacking nozzle tip temperature monitoring and additional cooling elements, the possibility of nozzle overheating must be taken into consideration when contemplating the research parameters [19].

3.1.3 Methane feed system

Due to the need of the methane injection parameters being controlled with the ECU, a commercial methane system was modified and adapted to fulfill the requirements. Because of the high pressure difference between the methane tank and the injectors, no actual pumps are needed. The methane feed system therefore consists of: fuel tank, high pressure reducer, main pressure line, Coriolis meter, relative low pressure regulator/reducer, gas feed pulse absorber, methane gas rail, gas injectors and intake manifold.

As a fuel tank, a 50 liter, 200 bar methane bottle was used. During the tests, only industrial methane consisting of 99.8% CH_4 was allowed. A manual pressure reducer was attached to the bottle for lowering the main line pressure to approximately 20 bars. Before entering the low pressure regulator, a Coriolis current meter was installed to monitor the gas mass flow. After the Coriolis meter, a retro-fit intended Keihin pressure regulator/reducer was placed to reduce the rail pressure to 2.4 bar relative to the intake pressure.

Due to the small volume between the injectors and the regulator compared to the injection mass flow, a methane pulse absorber was applied to suppress pressure fluctuations. The injection rail is located on the in-house built intake manifold, distributing the gas for two methane injectors positioned in the intake manifold. The selected injectors are Europegas/Hana engineering inc. manufactured units specifically intended for of nature gas. The given specifications for the injectors can be listed as in Table 2:

Table 2. Hana engineering injector specifications

Hana Engineering, Inc (H2000)	
Voltage (V DC)	12
Impedance (Ω)	1.3
Operating Pressure (bar, relative)	2.4
Maximum Gas Pressure (bar, relative)	5.8
Flow Rate (l/min, 1.2 bar)	130
Opening Time (ms)	2.2
Closing Time (ms)	1.2
Draw Current (A)	4
Hold Current (A)	2

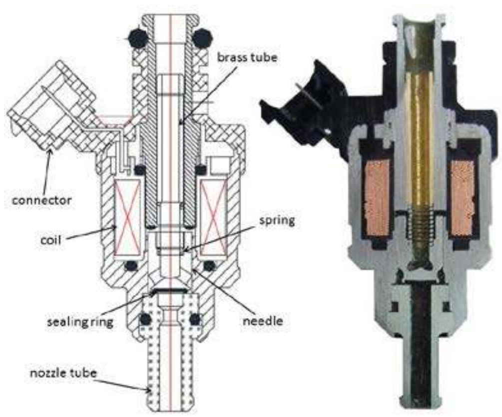

Figure 14. A cross-section of the methane injector

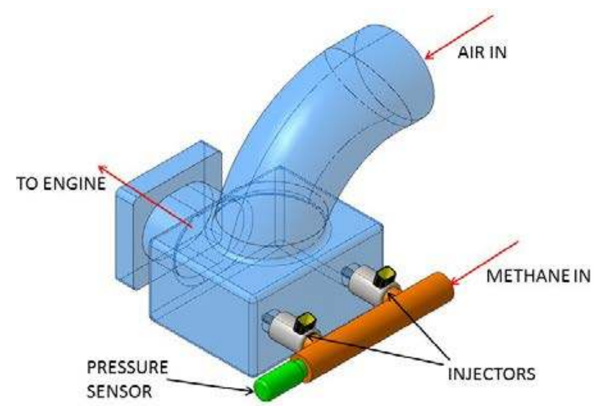

Table 15. The methane injector setup and intake manifold

2.1.1 Engine braking/Dynamometer

The single cylinder research application is somewhat vulnerable to engine speed deviation during different abnormal combustion events. The engine is furthermore not always able of producing the required work to keep up the requested speed. Therefore, an AC-motor based unit together with a speed controlling frequency transformer was used. The user defined speed request was controlled by the LabView- engine control unit, sending the signal to the frequency transformer. The greatest advantages of an AC-dyno is that it can either distribute or receive energy to or from the engine crankshaft. Specifically, the dyno can be operated as a motoring unit rotating the engine even though no combustion occurs - or when the engine produces work greater than the losses - be used as a braking component. This feature allows the user to run the engine on desired speeds before injecting any fuel, making it easier to start the combustion on right injection parameters. With the above mentioned setup, parameter combinations can be studied that cannot normally maintain engine speed requirements - such as misfire conditions or engine braking.

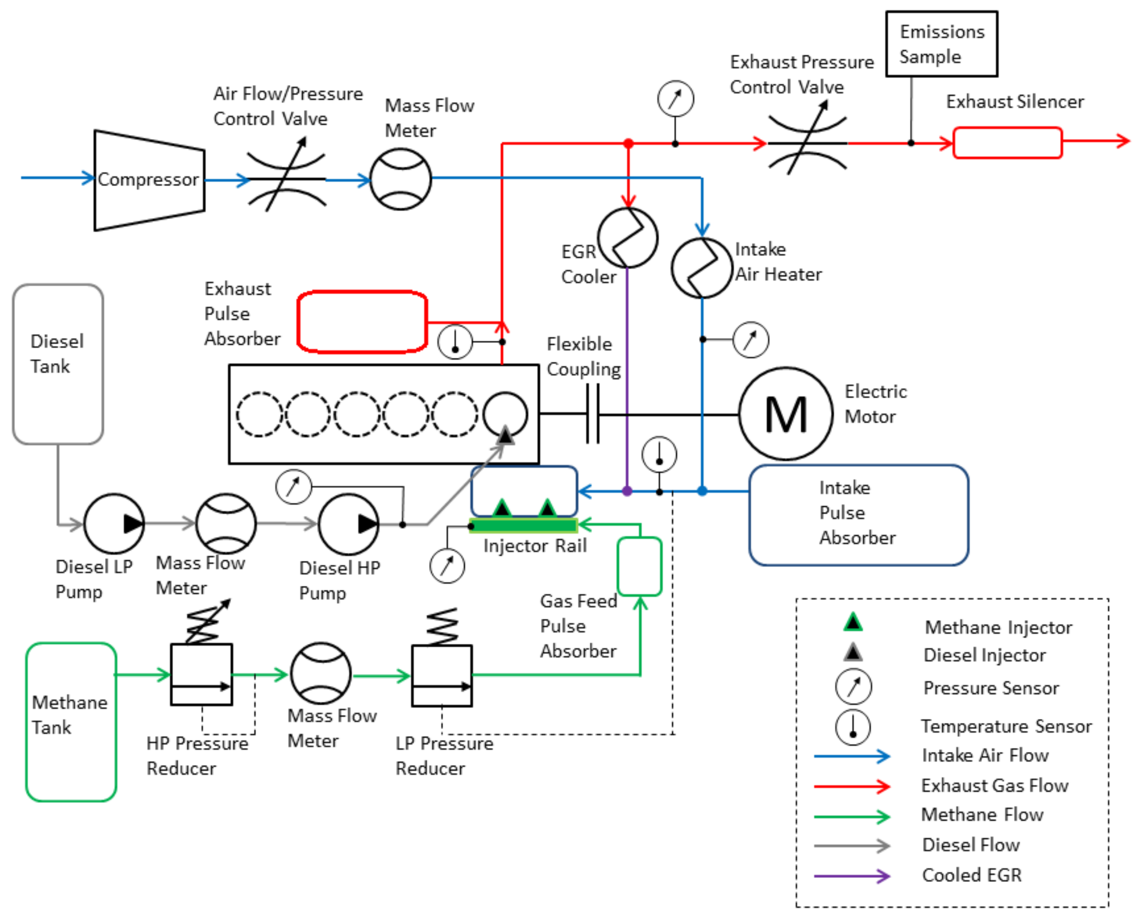

Figure 16. LEO1 - test configuration [6]

3.2 Experimental methods

3.2.1 Parameter setup

Due to a large range of feasible parameter combinations, a simple, relatively safe, well repeatable but wide scaled base matrix was needed to be defined. In order to roughly determine the tests load and fuel consumption levels, a reasonable reference had to be discovered. Some physical quantities were therefore referred from an earlier in-house test engine, an Agco 84AWI 6-cylinder factory specified platform. Since one of the thesis main goals were to analyze the lean mixture combustion behavior, a coherent DF-lambda range had to be used. Changes in each fuels specific lambda in relation to the variable substitution rate E% was also needed to be taken into account. As the physical limitations of the engine boundaries are somewhat known such as e.g. maximum cylinder pressure and charge air mass flow, the directional test matrix setup could be concluded via old data trends, calculations and premade iterations. The determination of the load level specified for the tests were therefore defined on basis of pretested and rational maximum air mass flow rates combined with old cylinder pressure data monitored on high load diesel points. In order to prevent the matrix of growing too large and time consuming, as many parameters as possible were intentionally kept constant.

To simplify the parameter scaling for the entire base matrix, a parameter calculator platform in excel was formed. The load level was set in relation on the maximum usable air mass flow. Known from earlier studies, the DF- combustions misfire appears in the $\lambda_{DF} > 2$ region. The dual-fuel lambda parameter was originally set to 2.5, but due to approaching the maximum air mass flow rates concerning cylinder pressure levels and air compressor load the apex value was decreased to $\lambda_{DF} = 2.2$. In consequence of same issues, load and fuel mass flow rates were rescaled to 84.5% from maximum, equaling a total diesel mass flow to 6kg/h. A cross section of the combustion behavior utilizing different diesel substitution rates E% was also needed. The main limiting factor concerning diesel substitution rate is the minimum possible injector opening time. Furthermore, as the fuel mass flow through a diesel injector normally cools the injector tip, risks of overheating the nozzle appears as the diesel substitution rate is increased too high [19]. Based on these factors, a maximum substitution rate E% on the given load was determined as 70%. A total of four different substitution rates (E%) were selected: 0 (reference), 30, 50 and 70.

The reference values for the test matrix were chosen from a factory setup Agco 84AWI (table 3):

Table 3. Test matrix reference values

Engine speed	1500	rpm
Lower heating value, Diesel	43.4	MJ/kg
Diesel mass flow, reference	6	kg/h
Calc. inj. energy	260.4	MJ/h
Load	84.507	%
Valve timing	AGCO 84AWI factory	

Based on the reference values, engine parameters were chosen following (table 4):

Table 4. Research parameters

Substitution rate (E%)	0	30	50	70
Lambda, dual-fuel (λ_DF)	1.5, 2, 2.2			
Diesel pilot inj. timing (CA BTDC)	8, 9			
Diesel pilot inj. pressure (bar)	1000, 1500			
Charge air temperature	25C, 70C			

Forming a 4x3 test cluster from where the calculated physical quantities can be presented in detail (table 5):

Table 5. Detailed physical quantities

Diesel mass flow (kg/h)

E%/λ	0	0.3	0.5	0.7
1.5	6	4.2	3	1.8
2	6	4.2	3	1.8
2.2	6	4.2	3	1.8

Methane mass flow (kg/h)

E%/λ	0	0.3	0.5	0.7
1.5	0	1.56	2.6	3.65
2	0	1.56	2.6	3.65
2.2	0	1.56	2.6	3.65

Total fuel mass flow (kg/h)

E%/λ	0	0.3	0.5	0.7
1.5	6	5.76	5.6	5.45
2	6	5.76	5.6	5.45
2.2	6	5.76	5.6	5.45

Diesel energy content (MJ/h)

E%/λ	0	0.3	0.5	0.7
1.5	260.4	182.3	130.2	78.1
2	260.4	182.3	130.2	78.1
2.2	260.4	182.3	130.2	78.1

Methane energy content (MJ/h)

E%/λ	0	0.3	0.5	0.7
1.5	0	78.12	130.2	182.3
2	0	78.12	130.2	182.3
2.2	0	78.12	130.2	182.3

DF_AFR_stoich

E%/λ	0	0.3	0.5	0.7
1.5	14.50	15.23	15.7	16.3
2	14.50	15.23	15.7	16.3
2.2	14.50	15.23	15.7	16.3

Diesel Lambda

E%/λ	0	0.3	0.5	0.7
1.5	1.5	2.16	3.04	5.1
2	2	2.88	4.06	6.8
2.2	2.2	3.17	4.46	7.48

Methane Lambda

E%/λ	0	0.3	0.5	0.7
1.5	inf	4.96	2.99	2.15
2	inf	6.61	3.99	2.87
2.2	inf	7.27	4.39	3.15

DF-lambda

E%/λ	0	0.3	0.5	0.7
1.5	1.5	1.5	1.5	1.5
2	2.0	2.0	2.0	2.0
2.2	2.2	2.2	2.2	2.2

Charge air mass flow (kg/h)

E%/λ	0	0.3	0.5	0.7
1.5	131	131.6	132	133.2
2	174	175.5	177	177.5
2.2	191	193.1	194	195.3

Table 6. Test cluster studying the effect of heated charge air

DF-lambda

E%/λ	0	0.3	0.5	0.7
1.5				1.5
2				2.0
2.2	2.2	2.2	2.2	2.2

Executing the last cluster (heated charge air temperature), it seemed irrational to run through the whole entity of the given matrix. As the aim was studying the possibilities of widening the operation window of lean total mixtures, it was reasonable to choose a set containing interesting points concerning misfire and flame propagation behavior. These specially selected points can be listed above (table 6).

The studied combustion affecting parameters were iterated and concluded after every repeated and successfully executed test of the base matrix. This allowed some space in the safety margin as the PRR and HRR behavior could be followed and decided in what direction the subsequent parameter should be chosen. The requirements for the baseline parameter combination were simple: they needed to be safe but compatible within a normal common rail diesel setup. Throughout the test procession, a total of four different parameter combinations were used. The first three clusters were chosen to characterize the combustion phenomenon, meanwhile the fourth was executed for studying lambda window widening possibilities.

3.2.2 Test methods and analyses

In order to monitor the recorded data from the whole matrix as trustworthy as possible, a reliably and repeatable path through the research was a requirement. Recording and processing a great quantity of data based on several different test points, a systematical routine is highly valued. Every test point was therefore executed equally throughout the whole process, including engine and matrix preparations, averaging, sampling and data processing. Generally these different test phases can be divided by following:

(1) Matrix overview and parameter preparation

(2) Experiment preparation

(3) Engine parameter tuning

(4) Stabilization, averaging and data recording

(5) Data processing

(6) Combustion analyses

(1) Matrix overview and parameter preparation

Before running any essential tests, the upcoming mass flows were calculated and inspected for logical abnormalities. If a similar point was run in beforehand, PRR and cylinder pressures were taken into account in order to predict future trends and possible boundary points. In addition, estimations based on old data regarding future parameter alterations were made, particularly concerning the charge air density dependable methane feed parameters.

(2) Experiment preparation

A daily convention before performing any tests, the engine had to be prepared and checked from aberrations. The daily process included more specifically: engine cooling water system de-airing, coolant water preheating, lubrication oil level inspection, ensuring of sufficient fuel stocking, emission sample calibration, EHVA calibration and numerous other more straight forward tasks. When the routines were completed, the engine was ready to be rotated to the desired engine speed.

(3) Engine parameter tuning

As the desirable engine speed level was reached, the required charge air mass flow was set. In order to quickly attain a stable and a constant level of the air mass flow, the charge air controlling butterfly valve was manually adjusted close to the craved extent before changed to a PID-controlled mode. The PID-controller parameters were adjusted to follow the target with a reasonably large averaging (long integral time) to avoid deviations caused by the single cylinder caused pulse fluctuations. When the targeted air mass flow level fulfilled the tolerances of deviation, the injection parameters could be set. Yet known in beforehand, but likewise discovered in practice, the combustion environment (combustion chamber and cylinder walls) temperature has a great influence concerning the combustion quality when moving into the combustion mode (starting injection). Although the coolant water was preheated to 70^{o}C, the charge air mass flow was set in beforehand decreasing the cylinder wall temperature prominently enough to de-stabilize the beginning of the combustion mode. The unstable combustion could clearly be heard by bare ear, but was also detected in form of high cylinder pressure fluctuations. To prevent the above-mentioned phenomenon were the tests always started on diesel mode, from where the DF-parameters could rapidly be switched and roughly tuned to the desired scale. When the combustion environment temperature reached normal operating conditions, the injection parameters could be fine-tuned matching the desirable mass flow values. Tuning patience was needed especially concerning diesel injection duration control, as the injector behavior was largely dependent of the nozzle temperature, often overshooting the desired mass flows before reaching a stable operation temperature.

(4) Stabilization, averaging and data recording

When all parameters were set and the desired mass flows were reached, the point required to be stabilized. The recording of the measured so-called ‘slow’ values could be executed after both the temperatures had moderated and the emissions had overcome the delay time. A stabilization interval of 5-7 minutes was discovered to be optimal with a good balance between effectivity, fuel consumption controllability and sample averaging time. To prevent errors caused by sensor noise, the recorded ‘slow measurement’ values were sampled with a one minute average. Immediately after, cylinder pressure values (defined as ‘fast measurements’) were recorded separately with a 5 second period using a sample frequency of 20Hz.

(5) Data processing

After every lambda cluster had been accomplished, all data recorded by the LabView software were transferred to a personal data processing workstation. All received values from the monitoring software were automatically saved in text file form, from where the necessary data were migrated to excel-based tables for future data processing. In order to get a reliable general view of the cylinder pressure data, an averaging of all the gathered pressures were made. To clarify the pressure trends by smoothening the remaining noise caused by the sampling and averaging, the data was additionally filtered with a Matlab coded filtering program.

(6) Combustion analyses

As all the needed data from one cluster was gathered to one single excel file, the data could then easily be redefined and processed into an illustrative form. Based on the cylinder pressure data and engine geometry, PRR, HRR, QHR (*cumulative heat release*), IDT (*ignition delay time*) and IMEP (*indicated mean effective pressure*) could then be calculated. This excel layout was copied and data transfers were repeated as many times as there were test days (i.e. lambda clusters on different starting parameters). Lastly, all the calculated data were gathered in one main file used as a tool to compare the differences in combustion quality trends depending on the different parameter combinations.

3.3 Calculations

3.3.1 Dual-fuel calculations

When the fuels stoichiometric AFR and lower heating values are known, both energy flow and substitution rate can be calculated through the fuel mass flows.

Energy flow $\dot{E}$ can be expressed:

$$\dot{E} = LHV\ x\ \dot{m} \tag{3.1}$$

where LHV stands for lower heating value and $\dot{m}$ [J/kg] for fuel mass flow rate [kg/h].

When the energy flows are known, diesel substitution rate E% can be calculated:

$$E\% = \frac{\dot{E}_{CH4}}{\dot{E}_{CH4}+\dot{E}_{Diesel}}\ x\ 100\% \tag{3.2}$$

The DF -lambda (total lambda), λ_{DF} is an expression for the combination of both fuels lambda values, λ_{CH4} and λ_{Diesel}. As known, lambda (λ) in general is expressed as a relation between the air-fuel-ratios (the stoichiometric AFR_{st} and AFR):

$$\lambda = \frac{AFR}{AFR_{st}} \tag{3.3}$$

The same rule applies for calculating each fuels lambda component e.g. λ_{CH4}:

$$\lambda_{CH4} = \frac{AFR_{CH4}}{AFR_{CH4,st}} \tag{3.4}$$

Knowing each fuels stoichiometric AFR values, the stoichiometric dual-fuel $AFR_{DF,st}$ can be solved:

$$AFR_{DF,st} = AFR_{CH4,st}\ x\ m\%_{CH4} + AFR_{Diesel}\ x\ (1 - m\%_{CH4}) \tag{3.5}$$

And λ_{DF} can therefore be expressed from the measured fuel mass flows:

$$\lambda_{DF} = \frac{AFR_{DF}}{AFR_{DF,st}} \tag{3.6}$$

3.3.2 Heat release calculations

The recorded cylinder pressure was used for studying PRR, HRR and QHR trends. Cylinder pressure data can be used for solving the heat released in the combustion process. Modeling heat release, the cylinder charge is assumed as a single zone and ideal gas law is applied, first law of thermodynamics for an open system can be used. When the cylinder pressure and the change in volume in relation to crank angle are known, the heat release during combustion, dQ_{gr} can be expressed [24]:

$$\frac{dQ_{gr}}{d\theta} = \frac{\gamma}{\gamma-1}\left[p\frac{dV}{d\theta} + V\frac{dp}{d\theta} + (u - c_v T)\frac{dm_c}{d\theta})\right] - \sum h_i \frac{dm_i}{d\theta} + \frac{dQ_{ht}}{d\theta} \tag{3.7}$$

Where m_c stands for mass of cylinder charge, c_v for rate specific heat at constant volume, u for specific internal energy, T for mean charge temperature, p for cylinder pressure, V for cylinder volume, γ for ratio of the specific heats (c_p/c_v), $dQ_{ht}$for charge-to-wall heat transfer and $\sum h_i m_i$ for enthalpy flux across the system boundary.

If the heat transfer, blow-by, crevice volume and fuel injection effects are neglected, the net heat release, dQ_{app} can be expressed:

$$\frac{dQ_{app}}{d\theta} = \frac{dQ_{gr}}{d\theta} - \frac{dQ_{ht}}{d\theta} = \frac{1}{\gamma-1}\left[\gamma p\frac{dV}{d\theta} + V\frac{dp}{d\theta}\right] \tag{3.8}$$

Cumulative heat release (QHR) is an expression of the total heat released in the process:

$$QHR = \sum_{i=m}^{n}\left[\frac{dQ_{app}}{d\theta}\right]_i \tag{3.9}$$

And pressure rise rate (PRR) expresses the pressure rise in relation to one crank angle:

$$PRR = \frac{dp}{d\theta} \tag{3.10}$$

4 Results

4.1 Effect of substitution rate E%

The initial test were specifically executed with diesel substitution rate E% as a variable. To ensure comparable results with complete combustions throughout all test points, lowest lambda conditions ($\lambda_{DF} = 1.5$) were chosen to be analyzed. Averaged and filtered cylinder pressure data can be seen (figure 17) below. Relatively large cylinder pressure peak differences can clearly be noted through the whole diesel substitution range. The DF operating conditions clearly follows a different cylinder pressure behavior compared with the pure diesel mode. The differences in combustion behavior can better be examined from the HRR curve in figure 18.

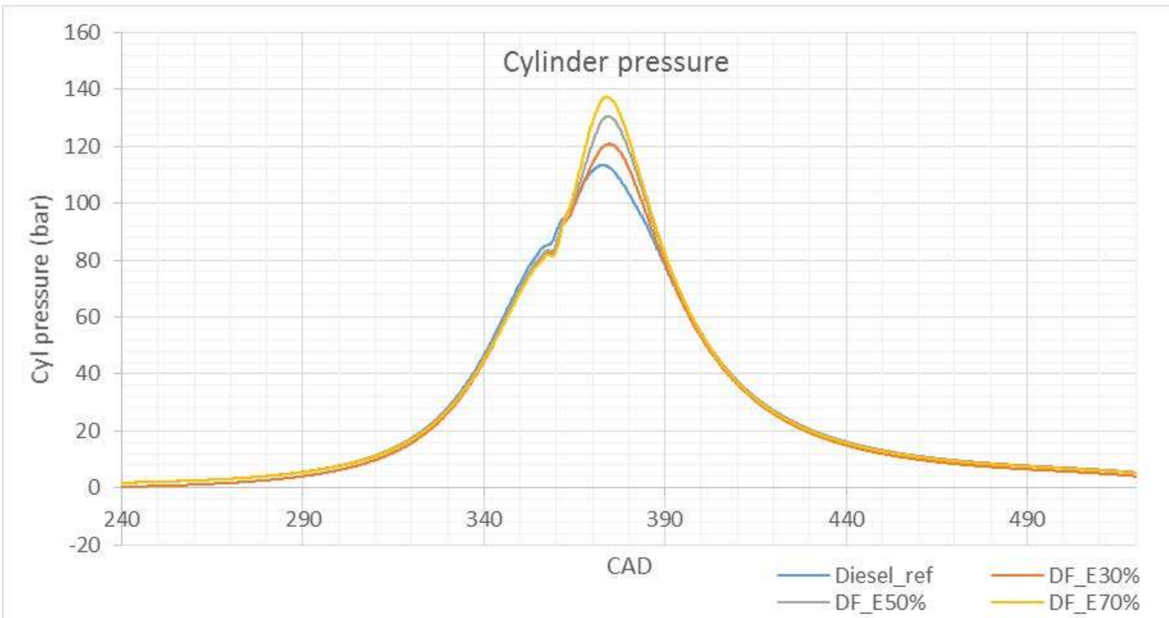

Figure 17 Measured cylinder pressures in Df-lambda 1.5 conditions

Comparing the combustion through the heat release curves, notable differences can be seen throughout the whole event (figure 18). After SOI, the premixed combustion phase clearly amplifies itself when increasing the substitution rate. This phenomenon can be explained by the dynamics of diesel spray behavior entering a lean but already premixed environment consisting of methane and air. The locally rich diesel spray close to the nozzle merges with the lean premixed mixture. Since the premixed combustion region of a diesel spray in general is independent of the diesel injection duration, the trend with increased substitution rates will be amplified close to proportionally to the amount of total energy replaced by the port injected, already premixed methane fuel. This proposition is supported by comparing the share of energy released between the diesel only and the three different substitution rates (figure 18).

Considering the fact that the HRR values are calculated from averaged and filtered results, the energy release difference matches closely to the substitution rates used. The theory of a wider combustion zone in the premixed region is supported by examining exact time of the premixed combustion peak. The wider the combustion area, the longer delay occurs for all the premixed fuel to completely burn.

Studying the combustion after the premixed phase, the trends of heat release are also notably different and dependent of the share of energy substituted by methane fuel. On pure diesel mode, the diffusion flame heat release trend is a result of injector characteristics and injection parameters. The heat release caused by the diffusion flame dynamics and propagation follows the given duration within a few degrees, lasting 23 CAD compared to the injection duration of 21 CAD. The maximum energy released per crank angle is dependent of the fuel mass flow through the injector. Running on DF -mode, a faster heat release can clearly be seen. The premixed methane combined with beneficial lambda conditions promotes a rapid, turbulent flame propagation ignited by the diesel pilot injection. Interestingly however, the combustion phenomenon is clearly independent of the pilot injection length as the burning methane-air mixture continues reacting much longer than the diffusion flame takes place (e.g. E70% pilot duration 5.7 CAD). The difference in HRR amplitude using different E% can be explained by studying the local in cylinder lambda before the time of ignition. Richer premixed mixtures promotes faster flame propagation throughout the combustion chamber but also increases the possibilities for an improved combustion quality. The latest mentioned phenomenon can be seen in the HRR figure in form of increased heat release closer to SOI and as a faster and a steeper drop caused by a rapid, complete combustion.

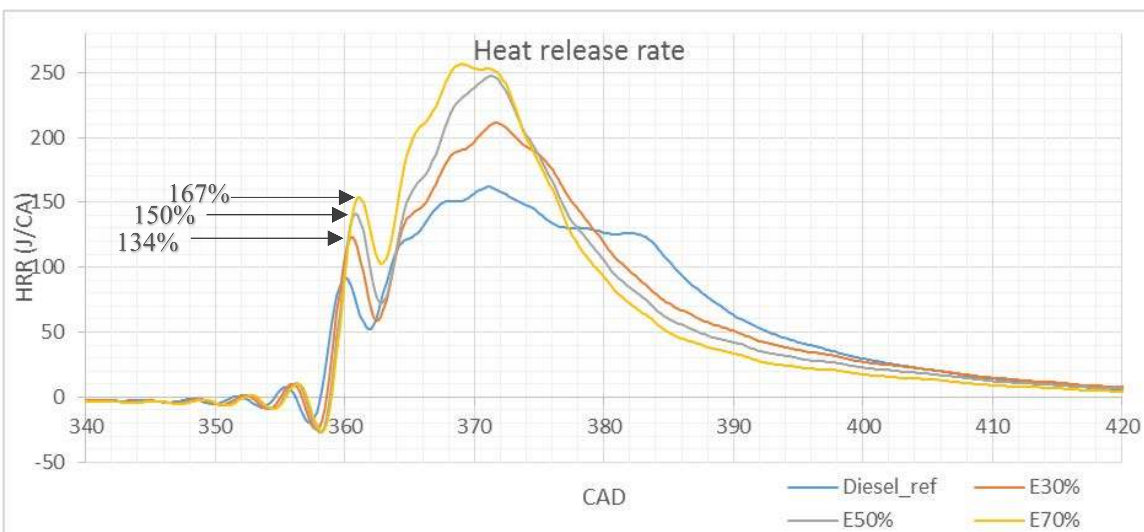

Figure 18. Calculated HRR on lambda 1.5

4.2 Effect of lambda

As the relation between the combustion trends concerning substitution rate were understood, the leaner combustion conditions could be analyzed. Since leaner operation conditions require more air charged into the cylinder increasing the motorized pressure, direct cylinder pressure curves could not be qualified for determining the combustion quality. HRR calculations were therefore seen as a more reliable tool to understand combustion processes during variable lambda conditions. The calculated HRR curves for both E50% and E70% can be seen in figure 19 below. Because of the similarities between HRR trends in E30% and E50% conditions, figures of E30% results were not shown for brevity.

Examining the combustion of E50%, HRR -trends for conditions λ_{DF} = 1.5 – 2.2 seems somewhat similar (figure 19). The diesel pilot injection premixed combustion phases follows an equal trend with only a small difference in shift during the moment of ignition. In the later combustion phase however, λ_{DF} =1.5 combustion reaches a higher peak in HRR in the 365-375deg region followed by a steeper drop, indicating a faster combustion. The λ_{DF} = 2 and λ_{DF} = 2.2 combustions in general is very similar without any notable difference in combustion speed. An explanation for the equal combustion can be detained from the relation between methane lambda (Table 7.) and diesel pilot injection attributes. As the premixed flame propagation speed is highly dependent of lambda, an indication of a faster heat release trend in relation to the methane lambda should be seen. This is however not the case. However, using the specific diesel substitution rate (diesel injection quantity) with the particular diesel injector configuration causes a wide, combustion chamber filling diesel pilot spray. Most of the methane-air fuel therefore mixes with the diesel spray during the diffusion flame event. More specifically: the very lean premixed gaseous fuel is mixed with the diesel spray and is combusted together with the diesel flame inside or in the immediate vicinity of the diffusion flame.

Table 7. Test matrix in relation to methane lambda

Methane Lambda

E%/λ_{DF}	0.5	0.7
1.5	2.99	2.148
2	3.99	2.865
2.2	4.39	3.151

In this case, the lean methane combustion is independent of the lambda value, implying there will be no significant premixed flame front propagation taking place. This why in very lean methane conditions DF-combustion is more reliant on the diesel injection spray behavior. This explains the phenomenon of premixed flame propagation actually existing on DF-lambda 1.5 conditions: the combustion is faster due to some premixed flame front propagation originating from the outer layers of the diffusion flame.

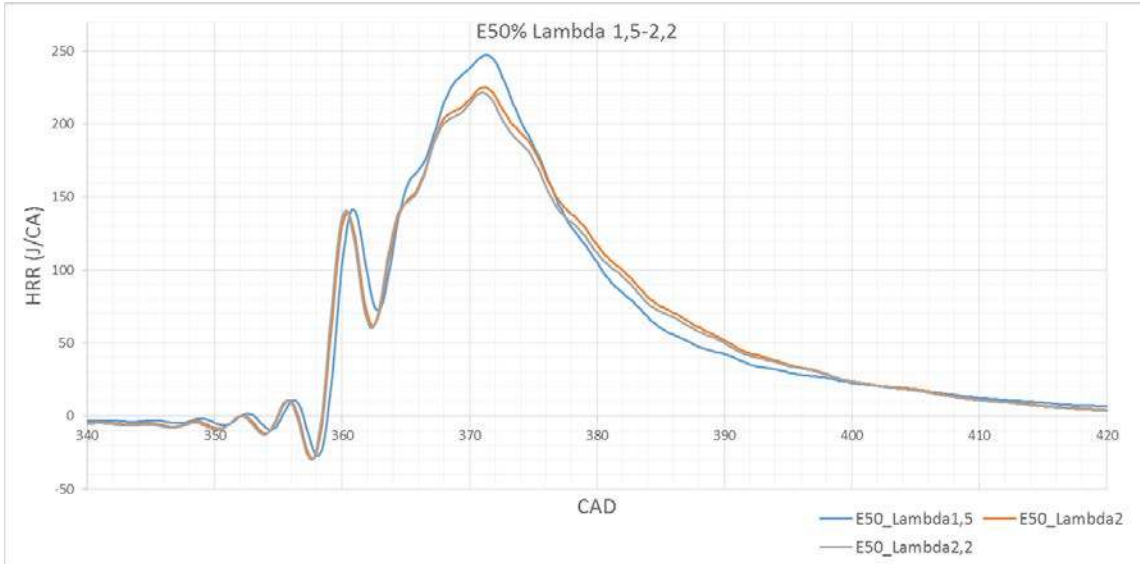

Figure 19. HRR plot of conditionsE50%, lambda 1.5-2.2

The above mentioned HRR behavior in relation to the size of the diesel pilot injection combustion zone can be confirmed comparing the results with the E70%, λ_{DF} = 1,5 - 2,2 conditions (figure 20). Since increasing the substitution rate decreases the injected diesel mass flow, the pilot injection size, shape and penetration will be affected. These variables determines the heat release area, which during higher diesel substitution rates signifies a smaller volume of diffusion flame. Comparing the different lambda conditions, the combustion behavior follows the exact same trend: in the lowest lambda value (λ_{DF} = 1.5) a clear premixed flame propagation can be noted. Both λ_{DF} = 2 and λ_{DF} = 2.2 however seems to miss the steep HRR trends completely. The combustion in these last mentioned states therefore also lasts longer, with poor or non-existing flame propagation.

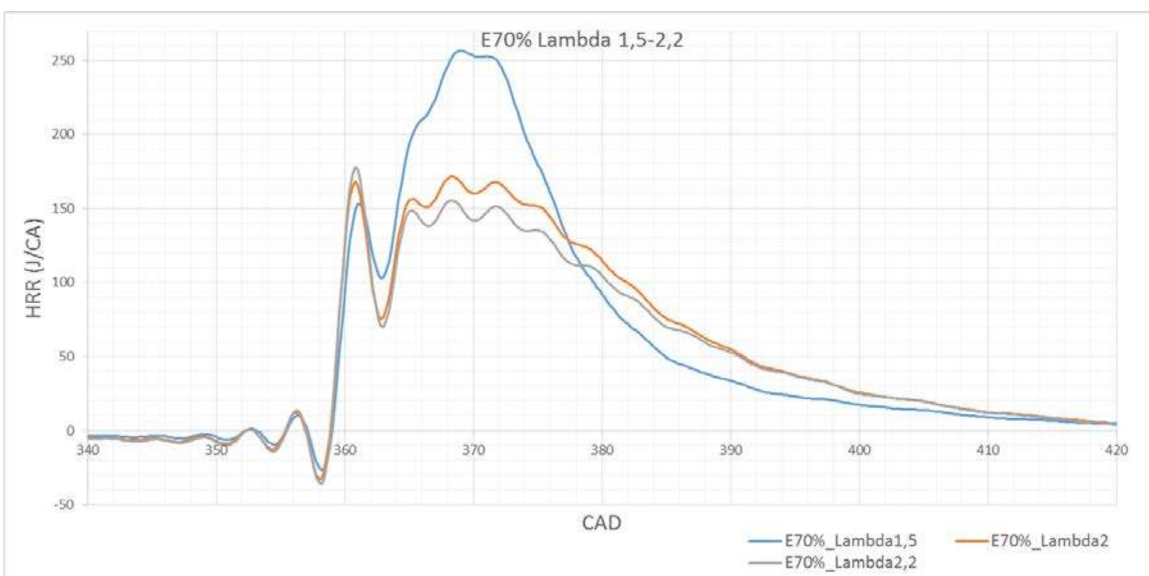

Figure 20. HRR of E70%, lambda 1.5-2.2

4.3 Effect of diesel injection parameters

4.3.1 Diesel pilot injection timing

Understanding the relation between DF-combustion and diesel pilot injection parameters supports both combustion characterization and is furthermore a requirements for controlling lean conditions combustion. A study concerning DF-combustion behavior in relation to diesel pilot injection timing was therefore executed. The diesel pilot injection timing was varied between 8 and 9 CA BTDC. An example of averaged and filtered cylinder pressure results from both 8CA BTDC and 9CA BTDC timings using E30% parameters can be seen in the graph below (figure 21). Nearly a 10bar gain in cylinder pressure peak between the tested parameters can be seen. A deeper study of the combustion phenomena was however made through HRR calculations.

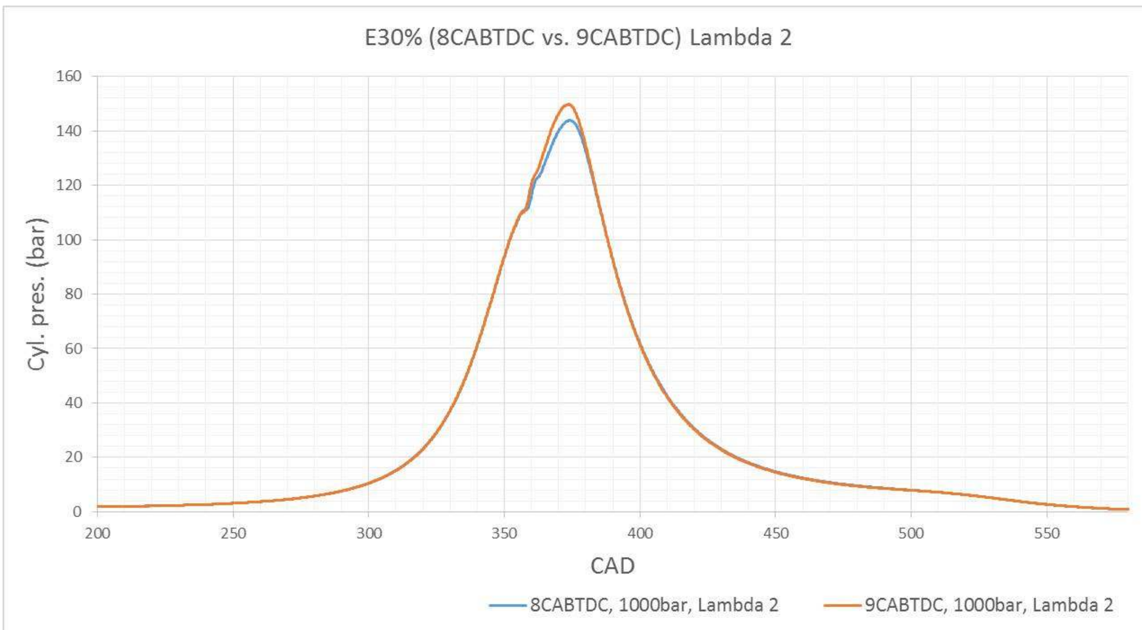

Figure 21 Cylinder pressure figures with variable diesel pilot injection timing

Although the results of cylinder pressure behavior differences between the studied pilot injection timings followed predictions based on knowledge from the past, an interesting detail of the DF-combustion was found. Studying the HRR graphs, no notable difference in HRR trends between the pilot-injection timings could be found. The 10 bars higher cylinder pressure did not affect the charge temperature enough to cause changes to the lean burn methane combustion. To confirm this assumption, each HRR plot for all three substitution rate were shifted to match the exact time of ignition.

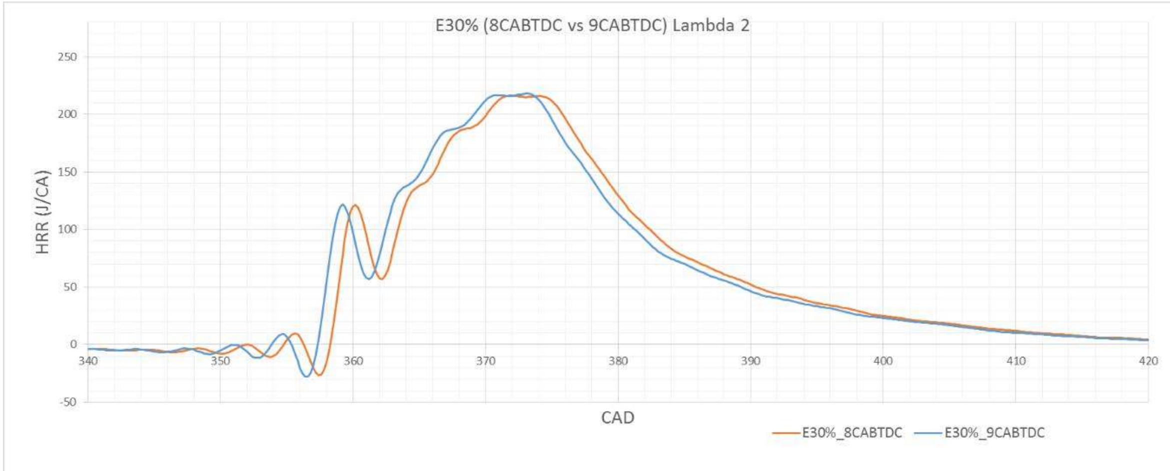

Figure 22 HRR of E30%, unshifted

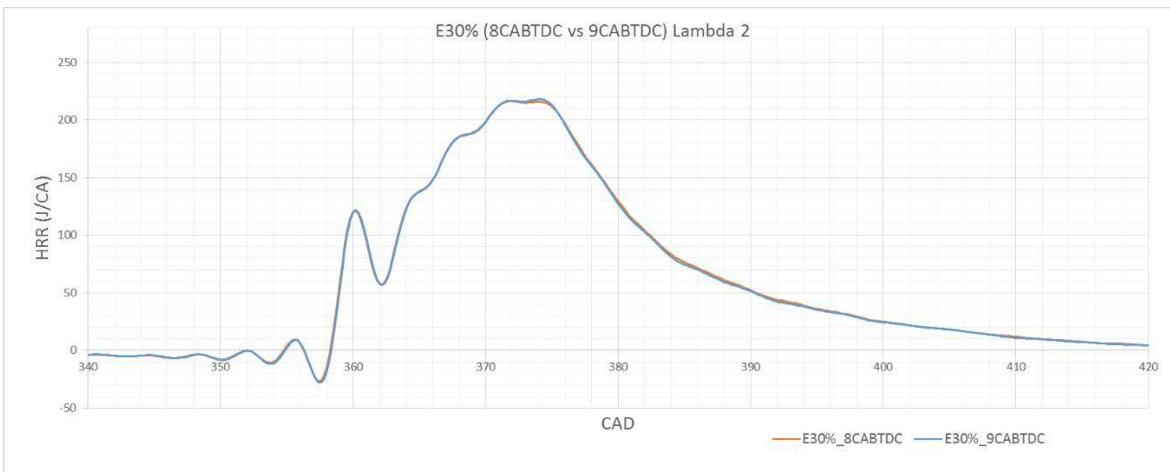

Figure 23 HRR of E30%, shifted

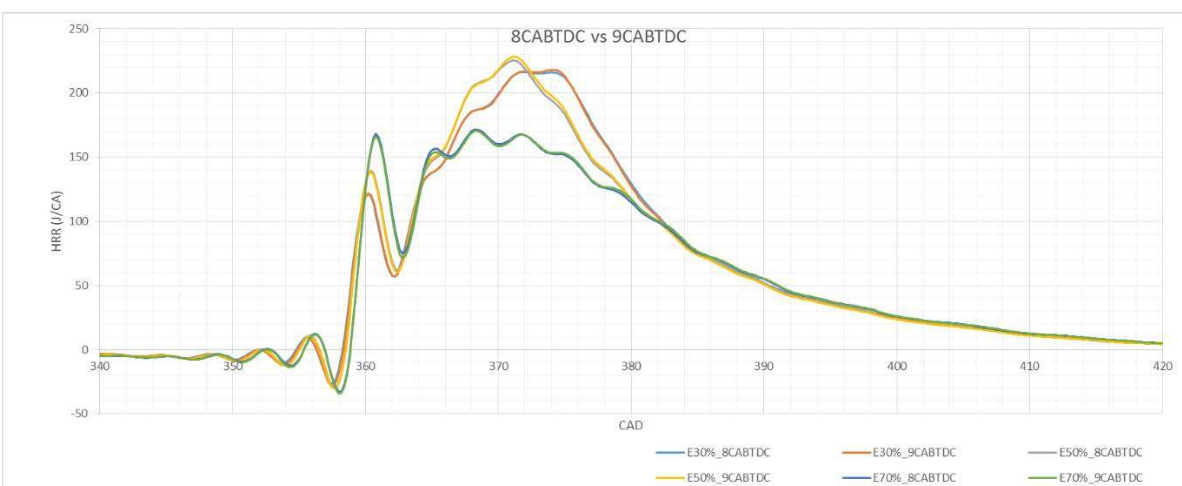

Figure 24 HRR, all test points

4.3.2 Diesel injection pressure

Further studies concerning diesel pilot injection parameters in different DF-conditions were made by examining the impact of diesel injection pressure. In order to get sufficient differences in pilot spray behavior was the injection pressure elevated from 1000bars to 1500bars. Due to the increased mass flow through the injector, the injection durations were dropped on all substitution rates approximately with 20-25%. Improvements were made in all lambda and diesel substitution rate conditions concerning cylinder pressure and power output behavior. Two monitored examples (E50%, E70%) of lean combustion cylinder pressure curves can be seen below (figure 25). The increased cylinder pressures could possibly indicate that the DF-combustion would be severely improved by the better injection penetration and fuel vaporization caused by the higher injection pressure. It would nevertheless be irrational to draw any conclusions without being reminded that higher injection pressures in general also speeds up the diesel combustion in similar manners, making the HRR analysis along with QHR study a central key of defining combustion efficiency. Assuming that the methane slip during the valve overlap period is remain constant, the emission results could therefore be used as a tool of DF-combustion quality indicator.

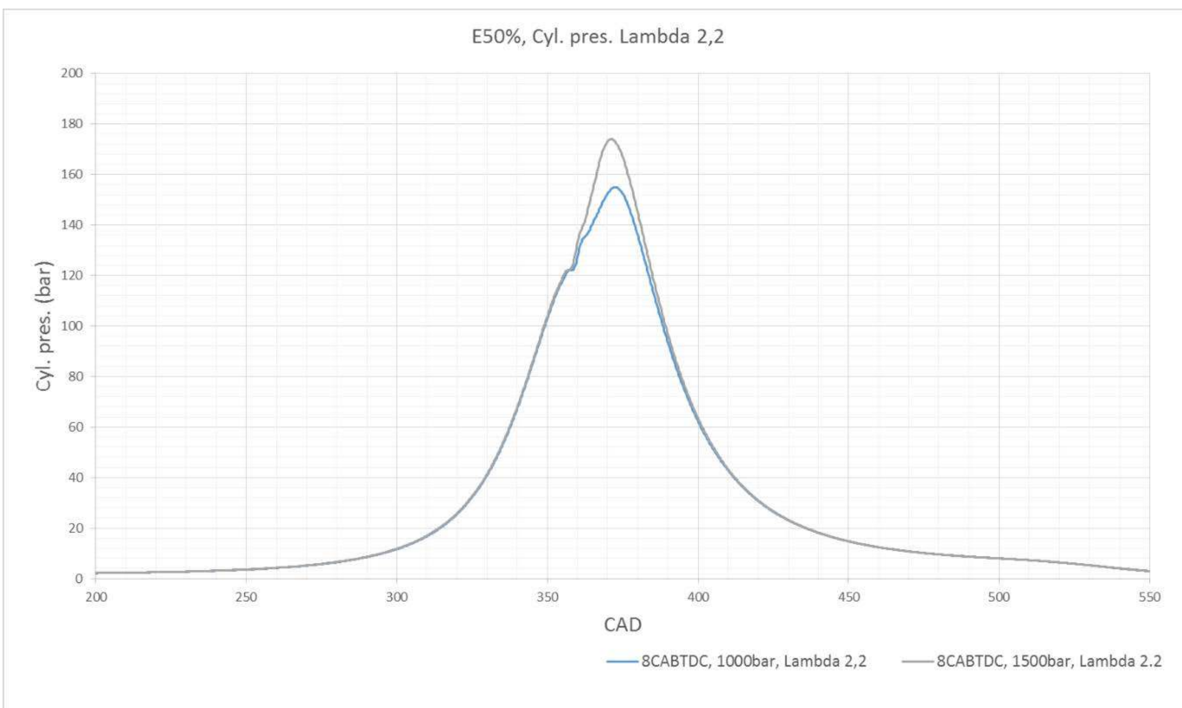

Figure 25 Recorded cylinder pressures with diesel pilot injection pressure as a variable. (E50%)

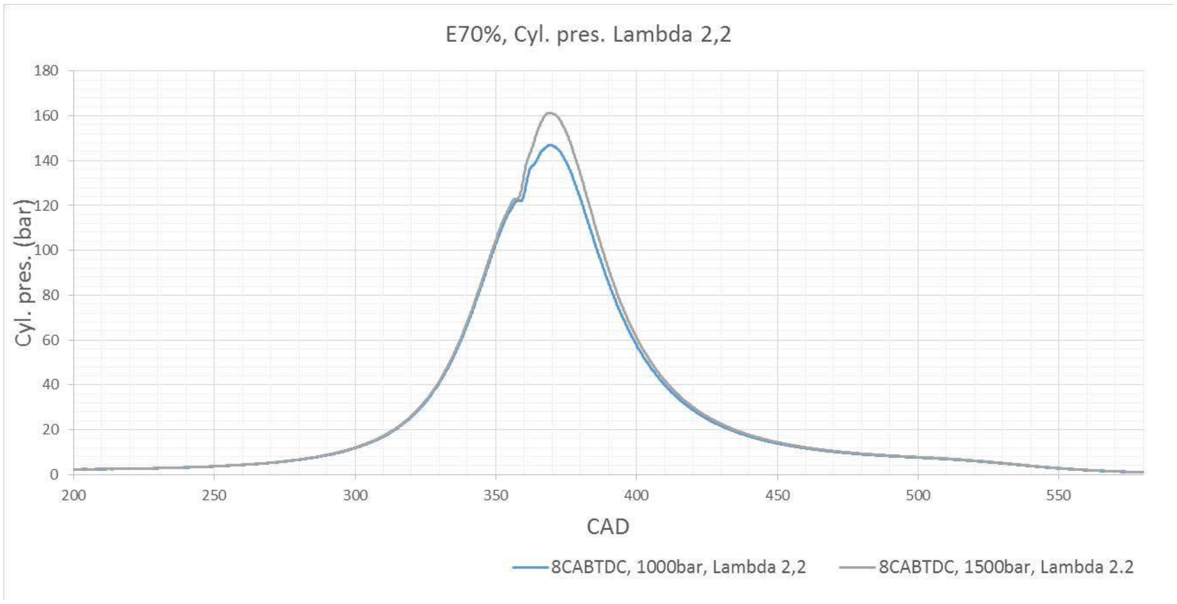

Figure 26 Recorded cylinder pressures with diesel pilot injection pressure as a variable. (E70%)

Examples of HRR plots of each used E% were chosen based on a lambda conditions were both complete and abnormal combustion occurs. The λ_{DF} = 2 results through the whole diesel substitution rate scale shows clearly these two above mentioned conditions and are therefore chosen to be analyzed.

The improved evaporation of the diesel spray results in higher HRR in the premixed combustion phase. The greater injection pressure also expedites the ignition of the premixed gas in general. The diffusion flame is completing faster, requiring less time for heating and igniting the surrounding methane-air mixture. Concluded earlier in section 3.3.2. *Lambda studies*, lean premixed methane combustion behavior (within slow premixed flame propagation speeds, λ_{CH4} > 2) is mainly dependent of the size of diffusion flame heat release area. Greater pilot injection pressures improves the spray penetration through the cylinder resulting in larger diffusion flame area. This claim can be proven by comparing the HRR results between the E70% combustions using 1000bar and 1500bar injection pressures. Even though the combustion does not reach any similar HRR levels compared with lower E% points on neither parameter-combinations, a significant improvement of the HRR trend can be seen.

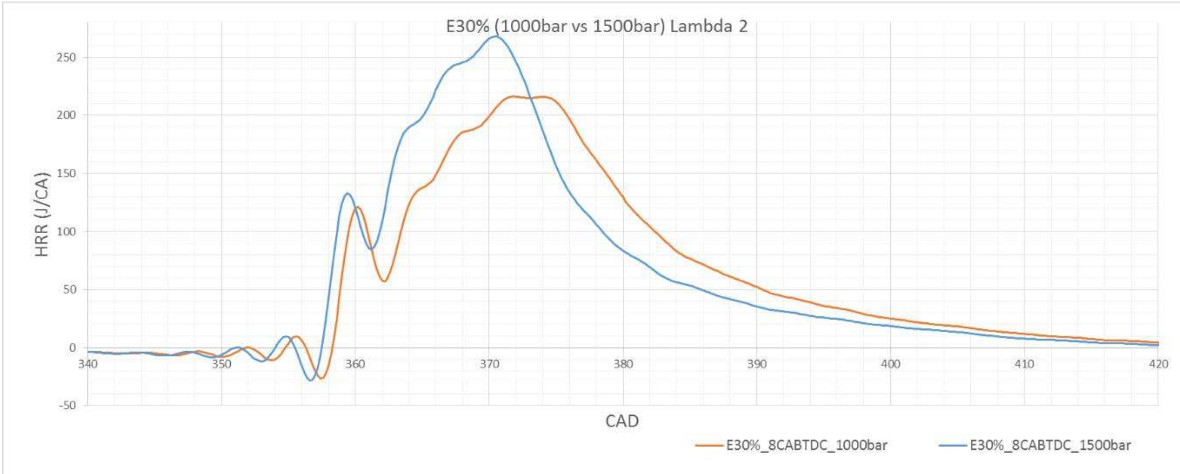

Figure 27. HRR figures of E30% conditions with diesel pilot injection pressure as a variable

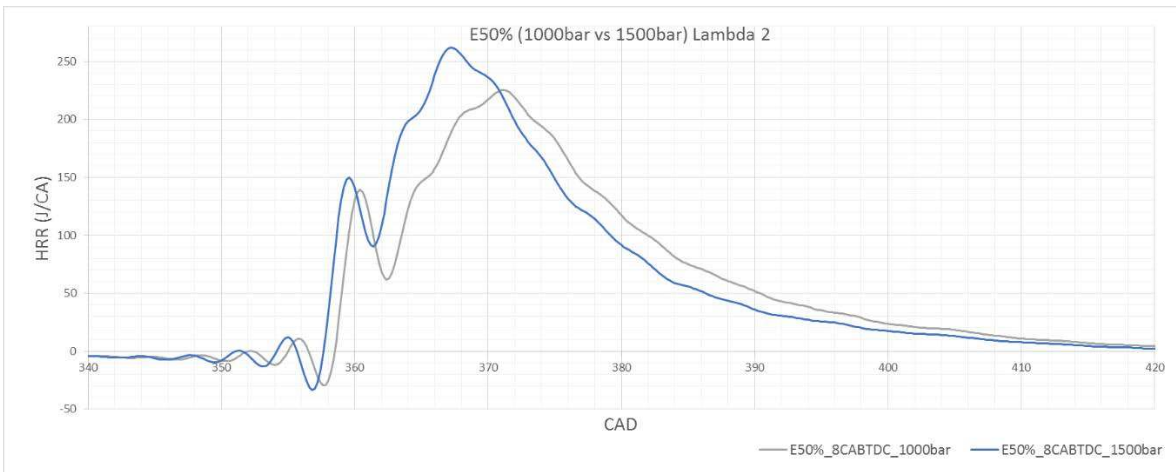

Figure 28. HRR figures of E50% conditions with diesel pilot injection pressure as a variable

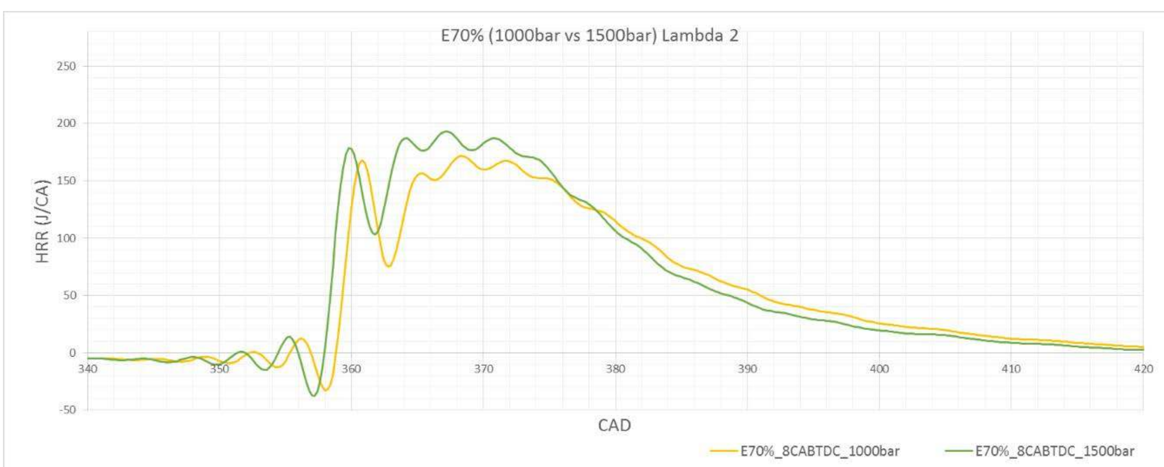

Figure 29. HRR figures of E70% conditions with diesel pilot injection pressure as a variable

The QHR analysis was in this case be used as a tool to study the combustion efficiency in the previously mentioned test conditions. Each QHR-trend is calculated from the earlier plotted HRR-data, expressing the total heat released by the overall combustion. The calculations also confirms which parameters only affects the character of heat release, and which actually improves the quality of combustion.

The below presented QHR-graph represents the cumulative heat release for both injection pressures of 1000bar and 1500bar. The calculations confirms the hypothesis of the correlation between DF-combustion quality and diesel pilot injection spray behavior in the E30%-E70% scale. In E30% and E50% conditions, the increased diesel injection pressure mainly speeds up the heat release effect because of the higher spray and combustion velocity. In both 1000bar and 1500bar circumstances, the diffusion flame can be defined as a fully combustion chamber filling heat release phenomenon. This can be proven for example through comparing the calculated HRR and QHR results. Both HRR and QHR trends with the substitution rates of 30-50% increases faster using the higher pilot injection pressure indicating a faster combustion speed, but will eventually culminate in the equal region of total released heat. In the case of E70% however, the increased injection pressure widens the spray area so that a greater share of the very lean methane mixture is allowed to ignite and combust.

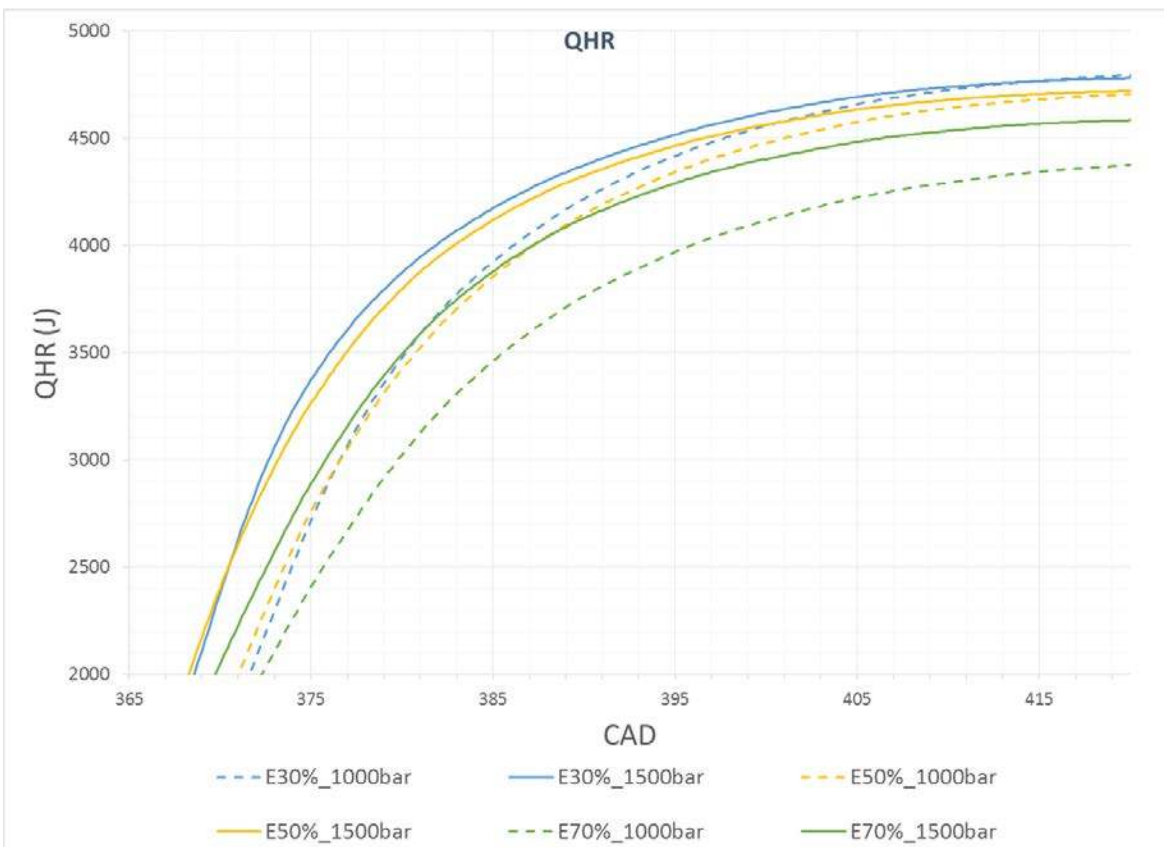

Figure 30. Calculated QHR-values for given combustion conditions

The same conclusions can be made through studying EGT (exhaust gas temperature), torque and emission data. Increased torque values were monitored through all tested parameters using higher pilot injection pressure. Faster heat release rate increases the in cylinder temperature resulting in higher cylinder pressures, increasing the work done by the piston. This can be monitored as lower EGT values in the exhaust channel simultaneously as the measured torque is increased. Misleading however might be the detail that during E70% conditions, EGT rises using higher injection pressures. The reason for this is the phenomenon of an increased quality of the partial burning situation that has to be taken into account. Furthermore, an indication of faster heat release resulting in risen cylinder temperatures can also be seen from the promoted NO_x formation. Paradoxical however is the fact that even though the QHR calculations indicates an equal level of heat released i.e. equal combustion efficiency, the measured CH_4 emissions are decreased during the 1500bar diesel injection conditions. An indication of a higher combustion quality can also be concluded from decreased CO-emission levels.

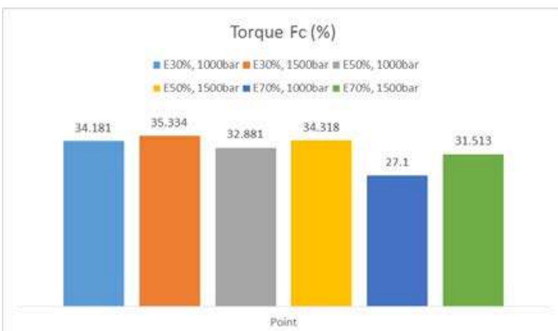

Figure 31. Monitored torque values (%) of all test conditions

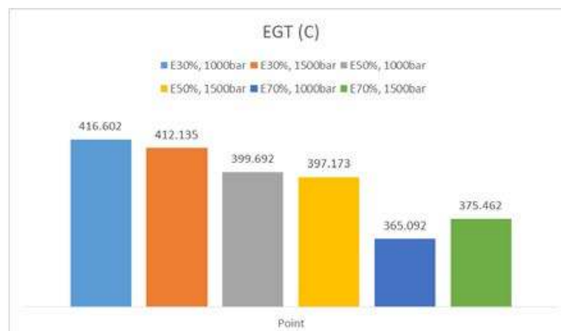

Figure 32. Monitored EGT

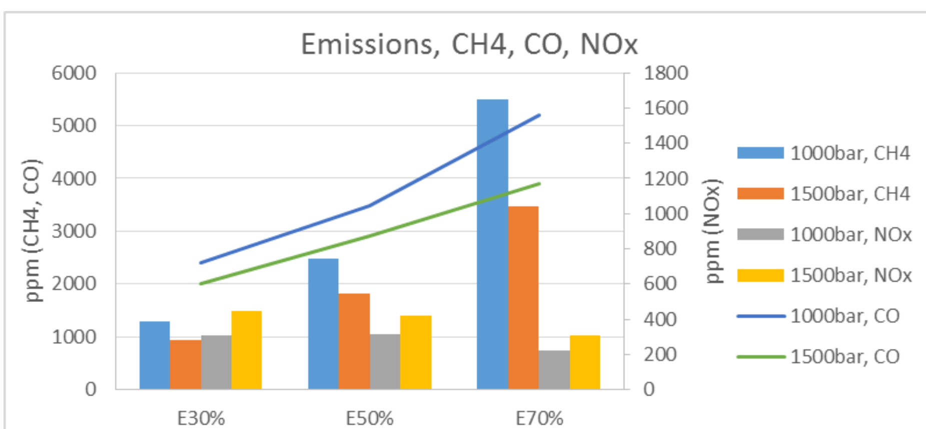

Figure 33. Exhaust CH4 slip, NOx and CO

4.4 Charge air temperature

Several occasions of misfire were monitored in the lean, high substitution rate conditions. The misfiring mostly occurred during the stabilizing period, suddenly dropping cylinder pressures and performance values. It was noted that this only occured during highest DF-lambda conditions where great charge air pressures and substitution rates were used. Already in early studies, the relation between low cylinder temperature and misfiring, were understood. The transition from pure diesel mode to dual-fuel required therefore to be performed fast, preventing the combustion chamber to cool down. Equal effects were experienced in steady state conditions close to misfire boundaries. If the EGT dropped below a certain level, a sudden misfire appeared. As the problematic areas concerning misfiring were identified in the highest E% conditions combined with lean DF-parameters, the λ_{DF} = 2.2 and E70% point was chosen to be further studied. Raising the in cylinder temperature before ignition was predicted to be a key factor for improved combustion behavior. In order to understand the effect of different combustion environments in relation to lean condition combustion quality, charge air temperature was chosen to be increased by ~45°C, from 25°C to 70°C.

The increased charge air temperature indicated significant improvements in combustion behavior in the most lean lambda conditions. Both a faster rate of heat release and an improved combustion quality were experienced. The monitored cylinder pressures also indicated enhancements in combustion stableness (COV). Using richer DF-mixtures however resulted in drastic PRR peaks and evident knock. The pressure rise in relation to the DF-lambda could clearly be heard with the bare ear. When PRR was exceeding the limit of 10bar/CAD, the engine started to make sharp metal-to-metal type noise.

Calculated HRR-figures of all executed lean condition tests using preheated charge air can be seen below (figure 34). As a reference, a misfire point is included (λ_{DF} = 2.2 and E70%). A clear difference in combustion quality and speed compared with the unheated test point can be concluded. The combustion characteristics is in general highly dependent of the substitution rate used. Increasing the initial pre-combustion temperature seems to fade away the premixed diesel combustion phase to unrecognizable levels in all conditions. The highest heat release rate is experienced with E50%, combining the good combustibility of diesel with the rapid premixed methane-air flame propagation. The central factor for this phenomenon seems to be a result of the combination of favorable diesel spray behavior, evenly distributing the heat caused by the diffusion flame in the cylinder combined with convenient methane lambda conditions (table 7).

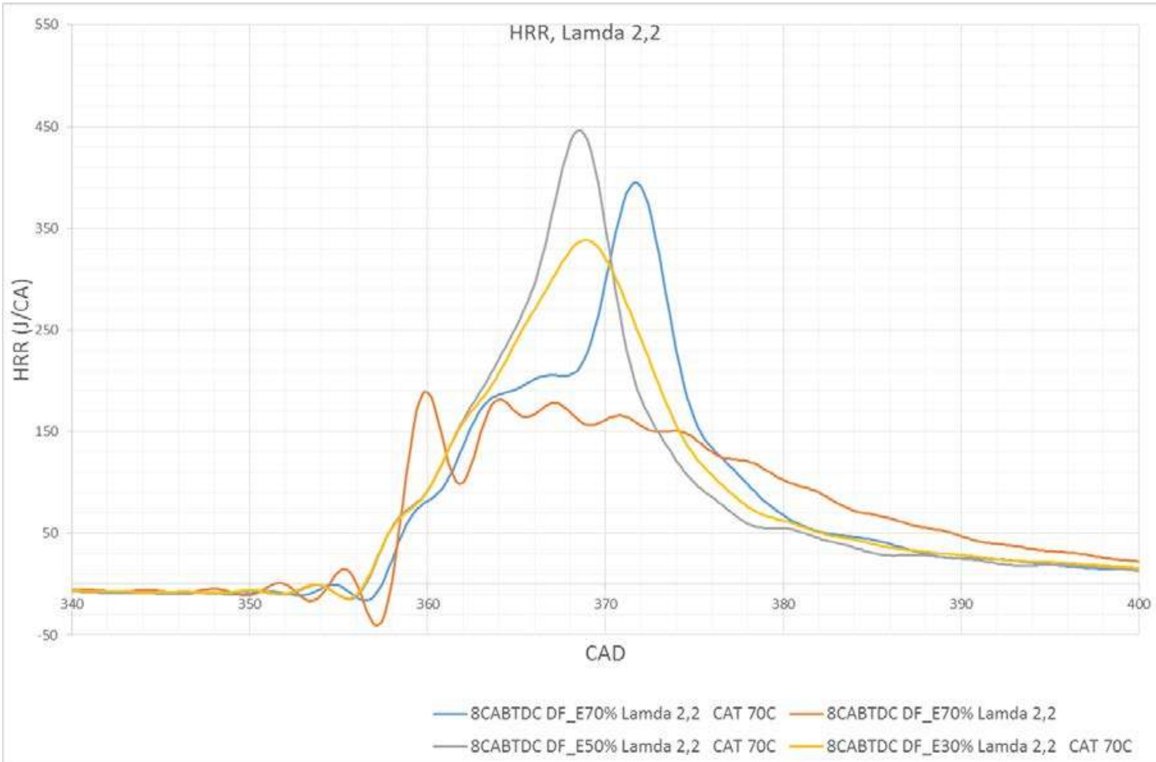

Figure 34 HRR-figures of lean condition heated CAT combustion

The general improvements of lean combustion behavior can also be proven through emission and QHR data. The higher substitution rate used, the greater relative enhancements in combustion quality can be seen. This indicates that the charge air temperature as a parameter influences the premixed methane-air combustion more using higher diesel substitution rates. Below is presented a table of calculated QHR values from both heated and unheated charge air tests on lean DF-parameters. Note that the maximum energy released is still however lower in the greater substitution rate condition, meaning that lean combustion still causes degradations concerning thermal efficiency.

Table 8. Calculated QHR-values of heated charge air test points.

	DF_E30%	DF_E50%	DF_E70%	DF_E70%
	Lamda 2,2	Lamda 2,2	Lambda 2	Lamda 2,2
	QHR_MAX	QHR_MAX	QHR_MAX	QHR_MAX
Charge air temp 70C	4829.6	4798.1	4698.3	4757.6
Charge air temp 25C	4838.6	4723.1	4591.8	4413.6

The same conclusions can be drawn from the emission data below (figure 35). Heating the charge air decreases drastically the methane slip caused by the poor ignitibility and slow flame propagation on lean conditions. Improved combustion quality in form of higher cylinder temperatures can also be seen from a rising NOx trend simultaneously as CO levels diminish.

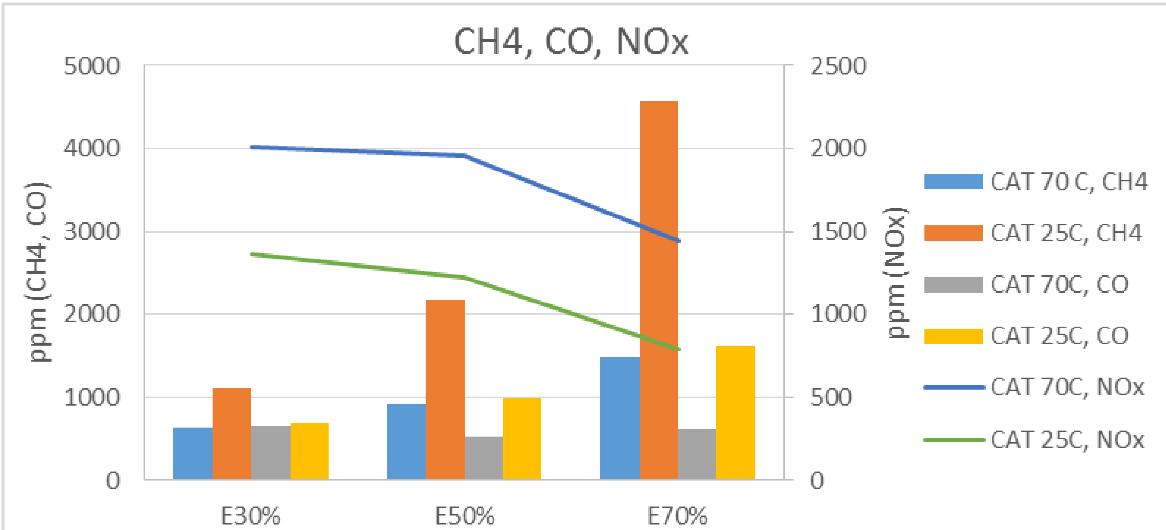

Figure 35. Emission data of heated CAT

Comparing the results of E50% parameters between heated and unheated charge air temperatures indicates very different HRR-trends (figure 36). Higher in cylinder temperatures contributes improved reactivity of the lean methane-air mixtures, requiring less energy from the diesel diffusion flame to be released for methane combustion to occur. Furthermore seems the heated CAT to promote faster flame propagation, making the methane-mixture to combust rapidly but controlled, increasing the combustion effectivity. A lack of a visible premixed diesel combustion phase is an interesting phenomenon requiring further studies to be understood. The exact similar trend appears in all heated CAT conditions. The premixed diesel combustion phase should logically be increased as a result of faster evaporation and promoted ignition conditions. The combustion in turn misses the premixed combustion completely, increasing the energy released instead evenly throughout the combustion. This phenomenon of HRR -behavior could be explained by approaching towards a limit of a RCCI typed combustion. A semi-rapid premixed flame propagation of the bulk mixture starts therefore immediately from the ignition source, diesel combustion. More specific: the ignition environment temperature approaches to a boundary close to the methane self-ignition point where only little energy is required for the bulk mixture to ignite. The combustion reacts therefore more or less spontaneously without being dependent of the diesel diffusion flame energy.

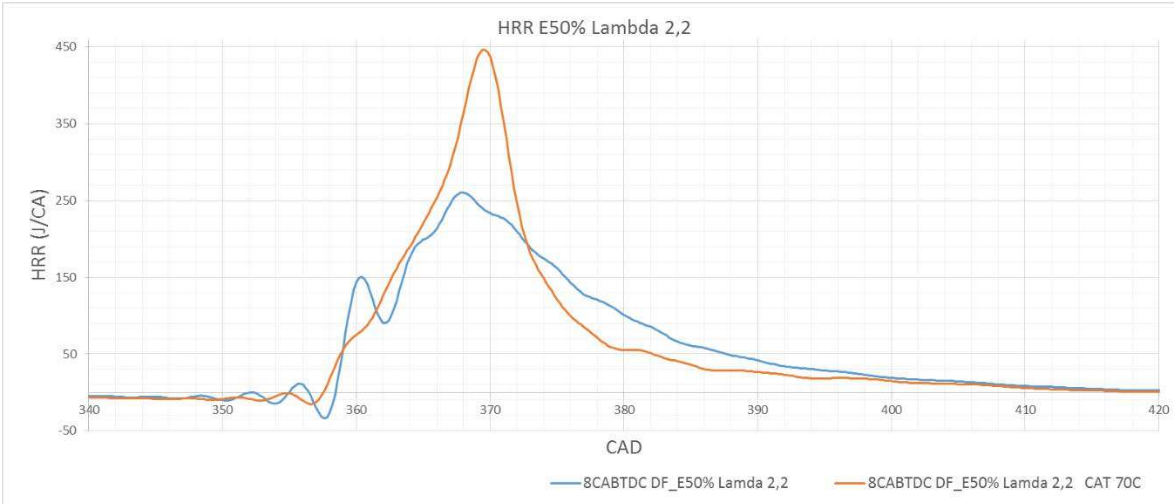

Figure 36. HRR of E50%, DF-lambda 2.2

Studying the problematic areas where misfiring clearly appears (E70%, $\lambda_{DF} = 2.2$), significant improvements in ignition behavior and combustion quality can be seen. The early part of the heat release behavior is somewhat equal to the above described E50% point. Using the higher substitution rate of 70% will however manipulate the later part of heat release identified as a PREMIER combustion (figure 37). In PREMIER combustion, the increased combustion chamber temperature caused by the rapid heat release from both diesel and methane combustion causes a controlled combustion of the end gas. The rapid methane combustion in the end gas region causes a high HRR-peak in the later part of the DF-combustion. Based on the HRR, a rapid heat release phase originating from bulk ignition in the end gas region can be seen [17]. The bulk ignition starts in the 368 degree region forming a rapid HRR caused by the simultaneous combustion of the end gas. In addition, improved combustion quality avoiding misfire in the E70% and lean lambda conditions can be proven by the captured emission data, reducing drastically methane slip levels with approximately 70%.

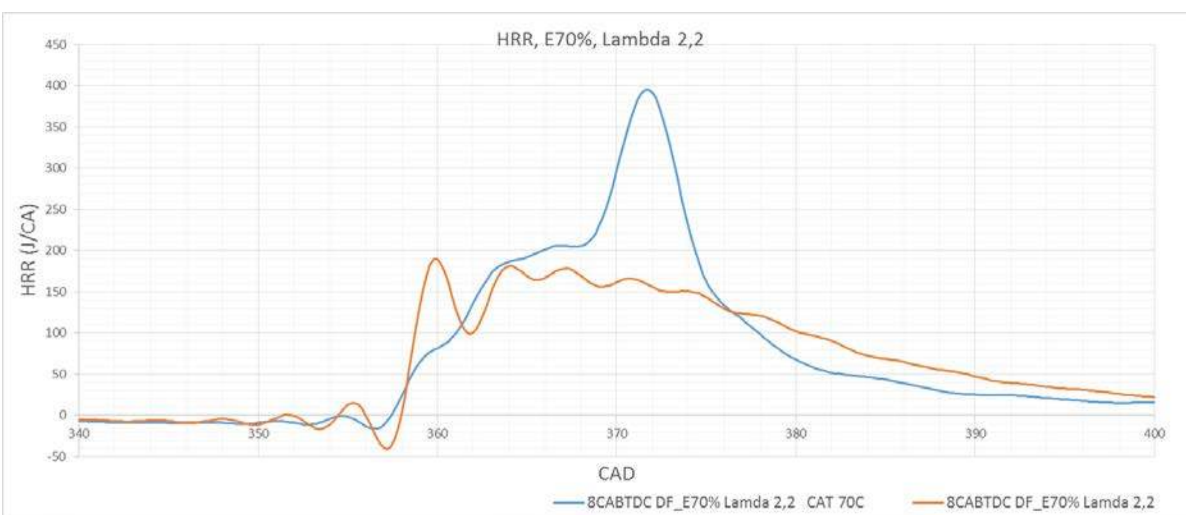

Figure 37. HRR of E70%, DF-lambda 2.2

Heating the charge air on richer DF-lambda conditions caused immediately unwanted combustion behavior. During the tests, loud noise and uneven cylinder pressure were experienced. Due to these mentioned occasions, the test point of $\lambda_{DF} = 1.5$ was aborted before any averaging or data monitoring was done. Fortunately, cylinder pressure data was managed captured from the unsuccessful test point. The HRR-plot in figure 38 describes the results between CAT of 25C and respectively 70C. A very rapid and uneven HRR trend can be noted after TDC. The combustion with these parameters can obviously be defined as uncontrolled as the heat release fluctuations and rapid peaks indicates severe irrationality of flame propagation.

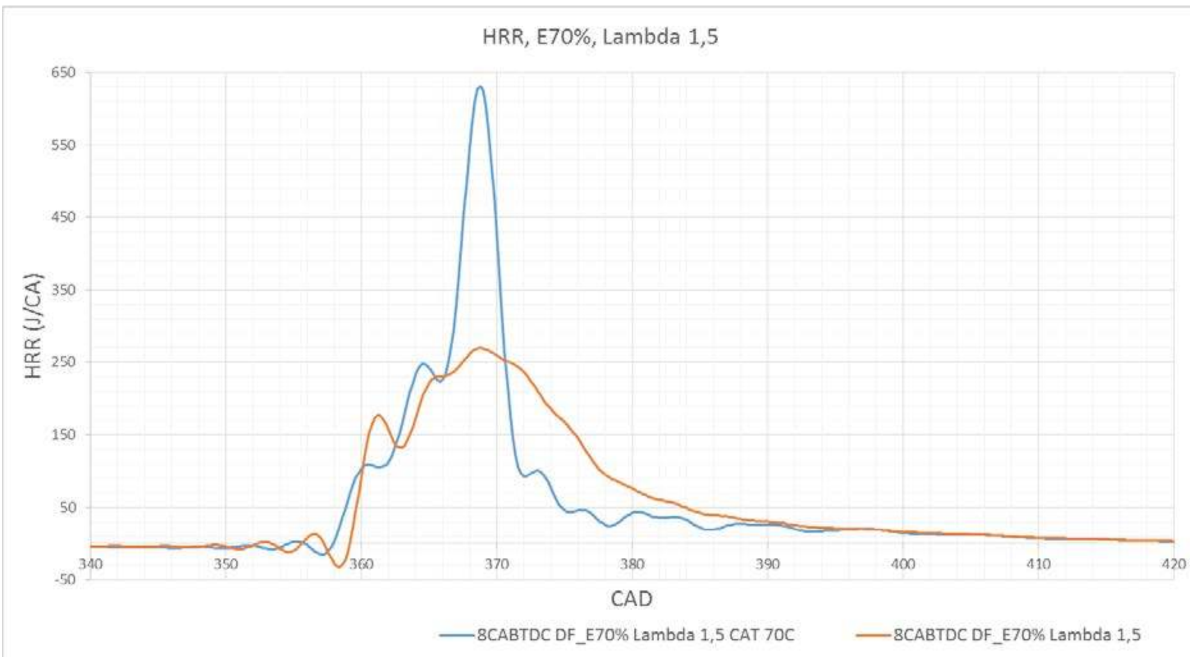

Figure 38. HRR-trends from both heated and unheated CAT in DF-lambda 1.5

A graph of an unfiltered cylinder pressure curve where the abnormal combustion occurs is presented in figure 39. Prominent pressure rise levels can be noted before turning into sharp, descending sawtooth typed pressure fluctuations. Based on the knock trends described in chapter *2.3.1 Knock*, the phenomenon can be defined as *spark knock:* the knock phenomenon starts well after SOI as the pressure level i.e. cylinder temperature exceeds the limit of the fuels self-ignition limit, spontaneously combusting uncontrolled in several areas of the combustion chamber. At approximately 8CAD ATDC, a pressure level exceeding 165bars can be experienced before actual knock occurs in the 370 degree region. During the test point where knock occurs, PRR levels exceeding 17 bar/CAD were monitored. The above mentioned knock event could possibly be avoided by retarding the pilot injection timing. Retarding the ignition timing will delay the combustion event, decreasing the temperature peaks by increasing the combustion volume available.

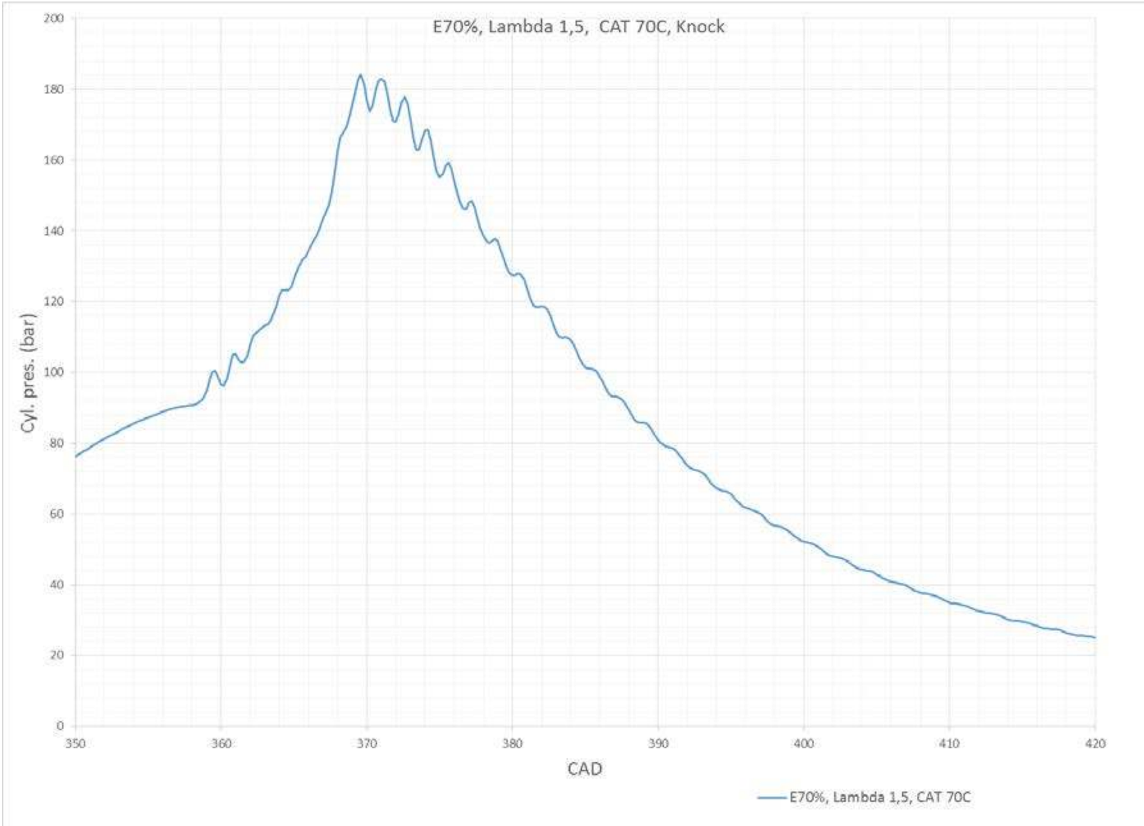

Figure 39. A recorded cylinder pressure figure indicating knock

4.5 Combustion characterization

The dual-fuel combustion character is evidently dependent of the substitution rate used, defining both the combustion speed and share of combustion attributes reminding either Otto or Diesel engine typed heat release behavior. Greater diesel substitution rates also amplifies the premixed combustion phase originated from diesel injection, enrichening locally the lambda and distributing more combustible products in the flame area. Methane-lambda values closer to its stoichiometry enhances the overall methane-air flame propagation speed. Therefore, high diesel substitution rates promotes engine performance less dependent of pilot injection parameters through fast heat release on DF-lambda conditions where complete, good premixed combustion can occur.

The methane-lambda, dependent of DF-lambda and diesel substitution rate, on the other hand directly affects the premixed methane fuels ignitibility, combustion quality and flame propagation velocity. The premixed methane-air combustion during ultra-lean lambda conditions are therefore highly depending of the diesel pilot injection, distributing sufficient energy for the premixed fuel to ignite and combust. This relation can clearly be seen in e.g. the leanest DF-lambda test. The methane mixture is too lean for any proper, rapid and independent flame propagation to originate. In these cases a wide, completely combustion chamber filling diffusion flame would be needed to completely ignite the methane fuel. Increasing diesel substitution rate however decreases both the diesel energy, but also reduces injection penetration and diffusion flame volume directly affecting the effective ignition area in relation to the combustion chamber. It can be deduced that the diffusion flame area can be manipulated both via physical diesel injector geometry e.g. nozzle hole pattern and size combined with optimizing the diesel pilot injection pressure. Higher diesel pilot injection pressures also releases the required diesel energy faster, compressing the separate dual-fuel heat release phases closer on to the other, promoting engine performance through fast heat release rates resulting in higher cylinder pressures close to TDC.

By summarizing the test results made in this thesis, controlling the methane lambda in relation to the substitution rate seems to be a key requirement for good a quality dual-fuel combustion. It seems irrational that low substitution rate conditions (E30%-E50%) with very lean methane lambda values combusts far better compared with the high substitution rate (E70%) parameters. Even though the E70% forms richer methane lambda values, a logical explanation can be drawn: in conditions where very lean methane mixtures occur, premixed methane-air flame propagation is unable to proceed. A wide diffusion flame area of the diesel pilot is therefore required. In these conditions, the combustion of premixed methane-air mixture is highly dependent of the diesel pilot injection parameters. On high diesel substitution rates, the diesel pilot injection quantity will not be sufficient enough for igniting all surrounding methane, resulting in partial or complete misfire. A methane lambda boundary value can be concluded from the test results: a value somewhere in the λ_{CH4} = 2.2 region is nevertheless sufficient enough for a proper premixed flame propagation to occur.

Based on this hypothesis, a prediction of functional lean combustion points using higher diesel substitution rates can be summarized. A calculated methane lambda table can be seen below. Cells marked with green can be stated as "complete combustion" qualified points, cells marked with red indicates a tested and predicted misfire region. Note that only E% levels of 0-70% are confirmed by the experiments made in this thesis. The bolded methane lambda points in the E90%-E99% scale are conclusions drawn from the combustion character hypothesis.

Table 9. DF-operation conditions in relation to methane-lambda

Methane Lambda

E%/λ	0	0.3	0.5	0.7	0.9	0.99
1.5	0	4.95	2.99	2.15	**1.68**	**1.53**
2	0	6.60	3.99	2.86	**2.24**	**2.04**
2.2	0	7.27	4.39	3.15	**2.46**	**2.24**

4.6 Lambda window widening possibilities

During the tests executed in the study, several lean condition misfire events were experienced. In order to find combustion improving factors, many different parameter combinations were studied. Concluded in the previous chapter, the dual-fuel combustion is highly dependent of the methane lambda. The combustion is characterized by two different types of phases: combustion where premixed flame propagation can and cannot occur.

Running the engine in the region outside the boundary of possible premixed methane flame propagation, lambda window can be widened by:

1. Increasing the diesel pilot injection volume through:
    a. raising diesel injection pressure
    b. increasing diesel injection quantity or by lowering the diesel substitution rate

2. Increasing charge reactivity by increasing the charge air temperature through:
    a. preheating the charge air temperature
    b. raising the compression ratio (hypothetical)

Additionally, increasing the charge reactivity (e.g. through higher temperature) is a key factor for improving combustion the quality in cases where premixed flame propagation can occur. Controlling the charge air temperature in relation to methane lambda, PREMIER combustion can be achieved. In order to avoid knock, the charge reactivity needs to be controlled depending of the diesel pilot injection parameters. It seems that if the charge air temperature could be varied constantly depending of the load and air mass flow used, an optimized combustion with a maximum efficiency could constantly be attained through the whole lambda scale.

5 Conclusions

The heat release behavior of a dual-fuel application can be defined as a rather complex chain of events severely dependent of the engines fuel injection and charge air parameters. Based on the experiments made in the thesis, many variables affect the combustion character and should therefore be chosen depending of the engine application and performance requirements. A deep understanding in DF-combustion is required by choosing suitable injection and lambda combinations for good combustion quality and engine efficiency.

The lean dual-fuel combustion can be divided into two different types of heat release trends: Combustion where spontaneous premixed methane flame propagation cannot occur (1), and conditions where the premixed methane flame propagation is able to take place (2). These two combustion forms must be controlled in unequal manners dependent of separate parameters. The general dependency determining the type of combustion behavior is the methane lambda λ_{CH4}. A boundary level between these two types of combustions using unheated charge air temperatures (~25°C) is approximately in the $\lambda_{CH4} = 2.2$ region. A quick brief of combustion character and their dependencies can be summarized:

(1) When no proper premixed flame propagation can take place, the ignition and combustion of methane is highly dependent of heat released by the diesel pilot injection. Methane combustion takes place only inside and in the immediate vicinity of the pilot injection diffusion flame. The methane combustion is therefore only reliant of the diesel pilot injection parameters. A wide sprayed diesel injection covering as much area in the combustion chamber as possible is therefore desired, determining the DF-combustion quality in general.

(2) Known from the Otto-engine, premixed flame propagation speed and quality is highly dependent of the fuels stoichiometry. Compared to the pure Otto-engine, a dual-fuel application can operate efficiently and controlled on lean conditions due to the larger energy released by the ignition source, the diesel diffusion flame. In order to attain a sufficiently rapid premixed flame propagation, the methane lambda should be aimed below $\lambda_{CH4} < 2.2$. Desiring high efficiencies with lean DF-lambda values above $\lambda_{DF} > 2$, a high substitution rate (E% > 90%) is a requirement. Due to the unbalance of lambda components in the mid-region of diesel substitution rate (E70%-E90%), challenges in the lean conditions can be concluded (table 9). In addition, pre-heating the charge air using high substitution rates may bring remarkable benefits in form of PREMIER combustion.

The dual-fuel lean operation window can be improved by manipulating the in-cylinder pre-ignition reactivity e.g. by increasing charge air temperature or by optimizing diesel pilot injection parameters. A balance between a suitable methane lambda and the increased reactivity should be held at controllable levels as a risk of knock always exists.

6 Future work

The diesel injectors minimum mass flow rate used in the present experiments limited the diesel substitution operation range to a scale of 0-70%. Future experiments concerning lean methane lambda boundaries are however possible to be studied even with the exact same engine configuration used in this thesis. The natural way of future studies would nevertheless be researching the top end of the diesel substitution rate scale (> E90%). Running the engine above approximately E80% of diesel substitution range requires heavy diesel injector modifications both by increasing the nozzle tip cooling and by decreasing the diesel mass flow rates matching suitable duty times. The injector modification concerning minimum diesel mass flow manipulation could be made through reducing number of nozzle holes or by downsizing the nozzle-hole area. In addition, converting for a pure duel-fuel application, piston and cylinder head geometry modifications may be considered.

Bibliography

1. Curran, Scott J., et al. Well-to-Wheel Analysis of Direct and Indirect use of Natural Gas in Passenger Vehicles. *Energy*, 10/1, 2014, vol. 75. pp. 194-203. ISSN 0360-5442.

2. Vertes, Alain, et al. *Biomass to Biofuels Strategies for Global Industries.* Chichester, West Sussex: Wiley, 2010. ISBN 9780470750025; 0470750022.

3. Chaichan, T. C. Combustion of Dual Fuel Type Natural Gas/Liquid Diesel Fuel in Compression Ignition Engine. *Journal of Mechanical and Civil Engineering*, 2014. ISSN 2278-1684.

4. Abd Alla, G. H., et al. Effect of Pilot Fuel Quantity on the Performance of a Dual Fuel Engine. *Energy Conversion and Management*, 4, 2000, vol. 41, no. 6. pp. 559-572. ISSN 0196-8904.

5. Königsson, F. *Advancing the Limits of Dual Fuel Combustion.* Licentiate Thesis ed. Sweden: Royal Institute of Technology, 2012. ISBN ISSN 1400-1179.

6. Törmänen, J. *Variable Valve Actuation and Dual-Fuel Combustion; Muuttuva Venttiiliajoitus Ja Dual-Fuel-Palaminen.* , 2015-08-24. Available from <http://urn.fi/URN:NBN:fi:aalto-201509184411 [urn]> Aaltodoc85 + 17.

7. Murillo Hernández, A. *Exhaust Gas Recirculation Study on Dual-Fuel Methane Combustion.* , 2015-03-23. Available from <http://urn.fi/URN:NBN:fi:aalto-201505132619 [urn]> Aaltodoc78+7.

8. Wärtsilä Finland Corporation. *"Wärtsilä Medium-Speed Engines"-Brochure [Online].* [Cited 5.1.2015]Available from:<http://pdf.nauticexpo.com/pdf/wartsila-corporation/wartsila-medium-speed-engines/24872-69921.html>.

9. Heywood, John B. *Internal Combustion Engine Fundamentals.* 1st ed. New York, NY, USA: McGraw-Hill, 1988. ISBN 0-07-100499-8.

10. Hu, E., et al. Laminar Flame Speeds and Ignition Delay Times of Methane–air Mixtures at Elevated Temperatures and Pressures. *Fuel*, 10/15, 2015, vol. 158. pp. 1-10. ISSN 0016-2361.

11. Van Basshuysen, R.; and SCHÄFER, F. *Internal Combustion Engine Handbook.* 3rd ed. Wiesbaden, Germany: SAE International, 2002. ISBN ISBN 0-7680-1139-6.

12. Dec, J. A Conceptual Model of DI Diesel Combustion Based on Laser-Sheet Imaging. Society of Automotive Engineers. 1997. SAE Paper 970873. ISSN 0148-7191.

13. Sarjovaara, Teemu. *Studies on Heavy Duty Engine Fuel Alternatives.* . Aalto-yliopiston insinööritieteiden korkeakoulu. Energiatekniikan laitos ed., Helsinki: Aalto University, 2015. ISBN 978-952-60-6561-8.

14. Curran, Scott J.; HANSON, Reed M.and WAGNER, Robert M. Reactivity Controlled Compression Ignition Combustion on a Multi-Cylinder Light-Duty Diesel Engine. *International Journal of Engine Research*, June 01, 2012, vol. 13, no. 3. pp. 216-225. ISSN DOI: 10.1177/1468087412442324.

15. Karim, G. A. Combustion in Gas-Fueled Compression Ignition Engines of the Dual Fuel Type, 2003, vol. Journal of Engineering for Gas Turbines and Power DOI: 10.1115/1.1581894. ISSN 9783527628148.

16. Banapurmath, N. R., et al. Performance, Emission Characteristics of Dual Fuel (DF) & Homogeneous Charge Compression Ignition (HCCI) Engines Operated on Compressed Natural Gas (CNG) – Uppage Oil Methylester (UOME). *Universal Journal of Renewable Energy 2*, 2014, vol. 32-44.

17. Azimov, U., et al. Premixed Mixture Ignition in the End-Gas Region (PREMIER) Combustion in a Natural Gas Dual-Fuel Engine: Operating Range and Exhaust Emissions. *International Journal of Engine Research*, October 01, 2011, vol. 12, no. 5. pp. 484-497.

18. Azimov, U., et al. Diesel Engine – Combustion, Emissions and Condition MonitoringDr. Saiful Bari ed. , 2013. *Combustion and Exhaust Emission Characteristics of Diesel Micro-Pilot Ignited Dual-Fuel Engine*. ISBN 978-953-51-1120-7.

19. Königsson, F. *On Combustion in the CNG-Diesel Dual Fuel Engine.* KTH, Internal Combustion Engines; KTH, Internal Combustion Engines, 2014, 98. ISBN ISSN 1400-1179 ; 2014:08.

20. Järvi, A. Methane Slip Reduction in Wärtsilä Lean Burn Gas Engines. *CIMAC Congress*, 2013, no. Paper No. 106. Bergen.

21. Owen, K.; and TREVOR, C. *Automotive Fuels Reference Book.* 2nd ed. Warrendale U.S.A: Society of Automotive Engineering Inc., 1995. ISBN 1-56091-589-7.

22. Nwafor, O. Effect of Advanced Injection Timing on the Performance of Natural Gas in Diesel Engines. *Sadhana,* 2000, vol. 25. pp. 11. ISSN 0256-2499.

23. Nwafor O. Knock Characteristics of Dual-Fuel Combustion in Diesel Engines using Natural Gas as Primary Fuel. *Sadhana*, 2002, vol. 27. pp. 375. ISSN 0256-2499.

24. Usman, A.; and MING, Z. Fast Heat Release Characterization of a Diesel Engine. *International Journal of Thermal Sciences*, December 2008, 2008, vol. 47, no. 12. pp. 1688-1700. Available from <http://www.sciencedirect.com/science/article/pii/S1290072908000318>. ISSN 1290-0729.