



Influences of Alternative Fuels and EGR on Thermodynamic Parameters and Emissions of a Heavy-Duty Engine

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Kurzfassung

Diese Arbeit beschreibt die Untersuchungsergebnisse zum Einfluss der AGR auf den Arbeitsprozess und die Emissionen eines Euro II-Nutzfahrzeugmotors. Für die Untersuchungen wurden unterschiedliche alternative Kraftstoffen sowie deren Blends eingesetzt. Die Ermittlung der thermodynamischen Parameter war Schwerpunkt der Versuchsauswertung.

Bei dem Motor handelte es sich um einen turboaufgeladenen Reihen-Sechszylindermotor mit Direkteinspritzung. Der Motor wurde mit einer externen Abgasrückführungsstrecke mit Wasserkühlung nachgerüstet. Die Einstellung der AGR-Rate erfolgte durch Variationen des AGR-Ventils, des Ladedrucks und des Drosselklappenwinkels im Ansaugtrakt. Es wurden Abgasrückführaten von 0%, 9% und 27% bei unterschiedlichen Temperaturniveaus des zurückgeführten Abgases eingestellt.

Für die Untersuchungen wurde als Referenzkraftstoff Dieselmotorkraftstoff (DK) und als alternative Kraftstoffe RME (Biodiesel) und GTL-Kraftstoff eingesetzt. Weiterhin wurden folgende Mischkraftstoffe untersucht: Blends aus Biodiesel und DK (B50, B10), O₂-Diesel, ein Blend aus DK und 7,8 % Bioethanol sowie ein Blend aus DK und 50 % GTL (G50). Außer GTL bzw. G50 weisen die untersuchten alternativen Kraftstoffe auf Grund ihrer schlechteren Heizwerte im Volllastbereich niedrigere Drehmomente im Vergleich zu DK auf. Weiterhin wurden bei allen alternativen Kraftstoffen eine Abnahme des effektiven Wirkungsgrades des Motors festgestellt, vor allem beim O₂-Diesel.

Die Zylinderdruckverläufe bei den Biodieselmotorkraftstoffen sowie bei O₂-Diesel zeigen höhere Zylinderdruckgradienten im Vergleich zum DK, d.h. höhere Geräuschemissionen. Mit zunehmendem AGR-Anteil und gekühltem, rückgeführtem Abgas fallen die Zylindergradienten und die maximalen Zylinderdrücke. Ähnliche Trends konnten bei allen alternativen Kraftstoffen nachgewiesen werden. Bei der Untersuchung von RME und den zugehörigen Blends konnten geringfügige Verlängerungen des Zündverzugs in der Teil- und Volllast festgestellt werden. Mit O₂-Diesel und GTL wurde eine Verkürzung der Zündverzugszeit ermittelt.

Die meisten alternativen Kraftstoffe weisen im Vergleich zu DK geringere Ruß- und HC-Emissionen und höhere NO_x-Emissionen auf. Für die GTL-Kraftstoffe waren alle Emissionen niedriger. Bei reinem RME wurde die geringste Rußzahl (FSN) gemessen. O₂-Diesel hingegen weist im Vergleich zu RME höhere Rußwerte auf. Das Ruß-Emissionsniveau von GTL-Kraftstoffen lag zwischen dem der konventionellen und der anderen alternativen Kraftstoffen (Ausnahme 27% AGR bei unterer Teillast).

Die Erhöhung des Anteils von RME im mineralischen DK auf 10% (B10) bewirkt nur eine geringfügige Verbesserung der thermodynamischen Parameter. B10 reduziert zwar NO_x und Ruß, jedoch verglichen mit DK erhöht sich die HC-Emission um bis zu 8%. Blends aus konventionellem Dieselmotorkraftstoff mit mehr als 5% RME-Anteil sind realisierbar. Zur Reduzierung von NO_x-Emissionen haben sich die Abgasrückführung bzw. eine verspätete Einspritzung als Lösungsmöglichkeiten bewährt.

Der Untersuchungen zeigten, dass der Einsatz von GTL und G50 ein gewisses Potenzial zur Verringerung der Emissionen ergibt. Darüber hinaus weist GTL auf Grund eines höheren Heizwertes einen geringeren spezifischen Kraftstoffverbrauch auf als DK. Dies ist ein wichtiger Vorteil der GTL-Kraftstoffe.

Abstract

Compression ignited (CI) engines, commonly known as diesel engines, are widely used power sources in on-road and off-road applications with complex emissions. Determining the impact of diesel exhaust emissions on human health and the environment requires evaluating both regulated and unregulated properties of exhaust gas. In particular, emissions from new fuels have to be examined thoroughly to prevent any impact from increased emissions of harmful substances. In addition, identifying and utilizing new alternative fuels to replace fossil fuels is an important task not only in Germany but also worldwide, now and in the future.

Previous research has generally focused on the composition of exhaust gas, analysis of its operating behavior and optimizations to improve operating and emissions parameters but only with neat biofuels. There has been little examination of new alternative fuels and their blends, e.g. ethanol, ethanol-diesel blends and RME-diesel blends (with ratios over 5%), and, in particular, of their use in heavy-duty diesel engines. Furthermore, investigations of alternative fuels combined with other operating conditions such as different exhaust gas recirculation ratios, retarded or earlier injection pump delivery, etc. have not been analyzed.

This study investigates the influences of properties of RME 100%, RME-diesel blends of different ratios, ethanol-diesel blends with low amounts of ethanol and gas-to-liquid and its blends on thermodynamic properties and emissions, specifically their impacts on cylinder pressure levels. To this end, different variables are combined, e.g. different alternative fuels and different blend ratios with different EGR rates, EGR temperatures and injection timing.

A cylinder pressure sensor, an injection pipe pressure sensor, evaluation software and data processing equipment were used to detect differences in cylinder pressures, heat release rates and spectrums of cylinder pressure in different operating points and under different operating conditions. Moreover, maximum heat release rates and maximum cylinder pressure peaks were calculated and evaluated in order to correctly interpret the effects of the aforementioned variables at different operating points and under different operating conditions and with different parameters, e.g. ignition delay periods, combustion duration and maximum cylinder pressure rises.

The results obtained are used to evaluate and optimize the engine's operating process with exhaust gas recirculation and different alternative fuels and their blends with conventional fuels.

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Abbreviations, Acronyms and Symbols

Abbreviations and Acronyms

ATDC	After Top Dead Center
BTDC	Before Top Dead Center
B.O.E	Beginning of Energising
BTL	Biomass to Liquid
B100	Pure Rapeseed Methyl Ester (100%)
B50	The Blend of Rapeseed Methyl Ester (50%) and Diesel Fuel (50%)
B10	The Blend of Rapeseed Methyl Ester (10%) and Diesel Fuel (90%)
B.D.C	Bottom Dead Center
CI	Combustion Ignition
CA	Crank Angle
C.D	Combustion Duration
CFPP	Cold Filter Plug Point
CNG	Compressed Natural Gas
DF	Diesel Fuel
EGR	Exhaust Gas Recirculation
ECU	Electronic Control Unit
ESC	European Stationary Cycle
ETC	European Transient Cycle
ELR	European Load Response
FSN	Filter Smoke Number
FFT	Fast Fourier Transformation
FT	Fischer-Tropsch
GTL	Gas to Liquid
G100	Pure GTL (100%)
G50	The blend of GTL (50%) and Diesel Fuel (50%)
H.R.R	Heat Release Rate
HC	Hydrocarbon
HPS	Heavy Paraffin Synthesis
HPC	Heavy Paraffin Conversion
HPL	High Pressure Loop
I.D	Ignition Delay
LPL	Low Pressure Loop
Max.	Maximum
Min.	Minimum
NO _x	Nitrogen Oxide
O ₂ -Diesel	The Blend of Diesel Fuel and Ethanol

O ₂ -1	The Blend of Diesel Fuel (without Rapseed Methyl Ester) and Ethanol
O ₂ -2	The Blend of Diesel Fuel (5% Rapseed Methyl Ester) and Ethanol
PAHs	Polycyclic Aromatic Hydrocarbons
RME	Rapseed Methyl Ester
Ref.	Reference
SMDS	Shell Middle Distillate synthesis
SGP	Shell Gasification Process
TDC	Top Dead Center
T-EGR	Temperature of Recirculated Exhaust Gas

Latin Symbol

B _o	g/h	Gravimetric Fuel Consumption
b _e	g/kW h	Effective specific Fuel Consumption
C _{pair}	J/kg K	Specific Heat of Air Charge at Constant Pressure
C _{pEGR}	J/kg K	Specific Heat of EGR at Constant Pressure
C _{pair/EGR}	J/kg K	Specific Heat of Mixture of Air Charge and EGR at Constant Pressure
[CO ₂] _{air}	ppm	Concentration of Dioxide Carbon of Air
[CO ₂] _{EGR}	ppm	Concentration of Dioxide Carbon of EGR
[CO ₂] _{aircharge}	ppm	Concentration of Dioxide Carbon of Air Charge
(CA) _{5%}	deg CA	Crank Angle at which 5% Heat Releases
(CA) _{90%}	deg CA	Crank Angle at which 90% Heat Releases
(CA) _{time}	ms	Combustion Duration
(CA) _{inject.}	deg CA	Crank Angle at which Fuel is Injected
$\frac{dp}{d\alpha}$	bar/deg CA	Cylinder Pressure Gradient
L _p	dB	Cylinder Pressure Level
\dot{m}_L	kg/h	Air Flow Rate (without EGR)
\dot{m}'_L	kg/h	Air Flow Rate (with EGR)
\dot{m}_{EGR}	kg/h	The Flow Mass Rate of EGR
\dot{m}_{air}	kg/h	The Flow Mass Rate of Air Charge
\dot{m}_z	kg/h	Mixture Flow Rate in Cylinder at the End of Intake Stroke (without EGR)
\dot{m}'_z	kg/h	Mixture Flow Rate in Cylinder at the End of Intake stroke (with EGR)
m		Polytropic Coefficient
n	rpm	Speed of Engine

n		Number of Sample
n		Interval of Crank Angle
p_{EGR}	bar	Pressure of Recirculated Exhaust Gas
$p_{i,p}$	bar	Pressure of Air Charge Measured in Inlet Port
$p_{aircharge}$	bar	Pressure of Air Charge (without EGR)
$p'_{aircharge}$	bar	Pressure of Air Charge (with EGR)
p_z	bar	Cylinder Pressure at the End of Intake Stroke (without EGR)
p'_z	bar	Cylinder Pressure at the End of Intake Stroke (with EGR)
$p_{ref.}$	bar	Reference Pressure
P	kW	Output
Q_i	$J/m^3 \text{ deg CA}$	Heat Release Rate
$R_{air/EGR}$		Constant of Mixture (without EGR)
$R'_{air/EGR}$		Constant of Mixture (with EGR)
s		Standard Deviation of Sample
$t_{i,d}$	ms	Ignition Delay Time
T_{air}	K	Temperature of Air Charge
T_{EGR}	K	Temperature of Recirculated Exhaust Gas
$T_{i,p}$	K	Temperature of Air Charge Measured in Inlet Port (without EGR)
$T'_{i,p}$	K	Temperature of Air Charge Measured in Inlet Port (with EGR)
T_z	K	Temperature in Cylinder at the End of Intake Stroke (without EGR)
T'_z	K	Temperature in Cylinder at the End of Intake Stroke (with EGR)
t		Factor for P
V_z	l	Cylinder Volume
v		Confidence Interval for Average Value

Greek Symbol

α	deg CA	Crank Angle
λ		Air Fuel Ratio
$\varphi_{i,d}$	deg CA	Crank Angle of Ignition Delay
Φ		Fuel/Air Equivalence Ratio
ρ	g/cm^3	Density

1 Introduction

The gasoline engine invented by Nicolaus A. Otto (1832-1891) and Eugen Langen (1832-1895) and then widely implemented and used [1, 2] was followed by a new type of internal combustion engine invented by the German engineer Rudolf Diesel (1858-1913). The diesel engine employed a concept of initiating combustion by injecting a liquid fuel into the air heated solely by means of compression and thus permitted doubling efficiency over other internal combustion engines [1, 2].

Today, the diesel engine's advantages over gasoline engines, e.g. best thermal efficiency among all internal combustion engines, better fuel economy, better power source, better reliability, etc., have made it the worldwide power plant of choice for trucks, trains, boats and most other heavy-duty applications. Moreover, while power sources other than the conventional combustion engine such as fuel cells, accumulated based electric drive system and the like may represent the future, they are not yet feasible. Therefore, the compression ignited (CI) engine will most likely remain the power source for the long future and just as likely be subject to emission controls to remedy the two major problems connected with it, namely high emissions of NO_x and fine particles. Diesel technology's inherently beneficial efficiency and durability make it best suited for compliance with future energy and environmental goals. Its compression ratios and excess air supply produce lower specific fuel consumption and carbon dioxide (CO_2) emission than gasoline engines [2].

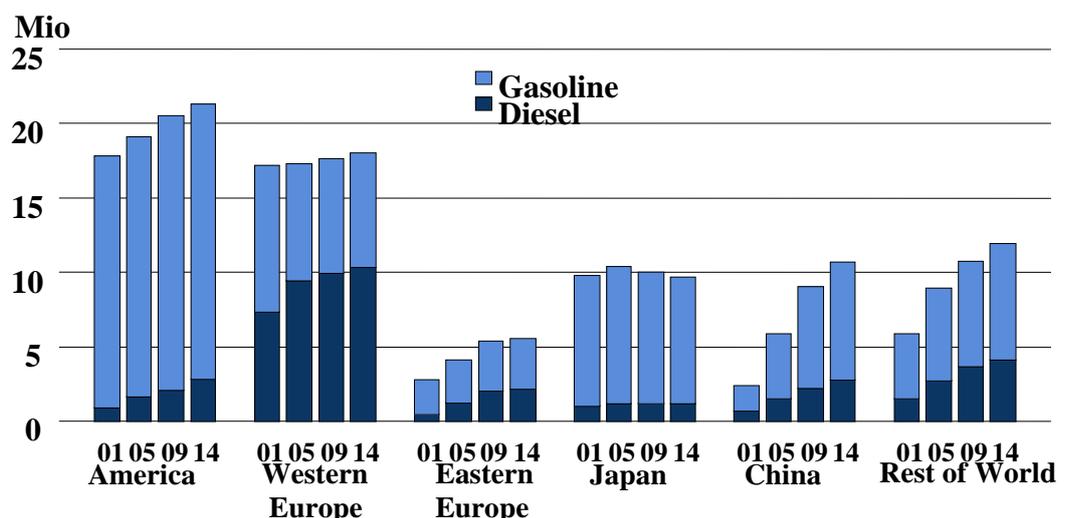


Figure 1.1: Future development of the worldwide distribution of diesel engines by 2014 [6]

Figure 1.1 indicates worldwide production of CI engines will increase in the near future, especially in the USA, Europe, China and Rest of World. This trend also underscores the importance of diesel engines in transportation worldwide, not just now but also in the near future.

Despite their many advantages, diesel engines also have some disadvantages. Diesel engines are a significant source of fine soot particles as well as hydrocarbons and nitrogen oxides. In addition, they are also a significant source of noise. Engine noise has several sources: the exhaust system, the fan used for cooling and the engine block surface [2]. Noise is generated by aerodynamic effects, caused by forces generated by the combustion process or may result from mechanical excitation of rotating or reciprocating engine components [1, 2, 8]. The modern automotive industry has focused its development work on diesel engines that meet requirements of better fuel consumption, lower noise, reduced emissions and maximum service life and lower manufacturing costs.

In response to the threats of pollution, the United Nation's 3rd Climate Conference held in Kyoto in 1997 formulated goals to reduce emissions of greenhouse gases by 2012 [1, 2, 7]. The proposals for reducing pollutant emissions have been implemented in regulations and laws all over the world. Decreased emissions have become a mandatory requirement not only for newly approved engines but also for engines already in use (see **Figure 1.2**).

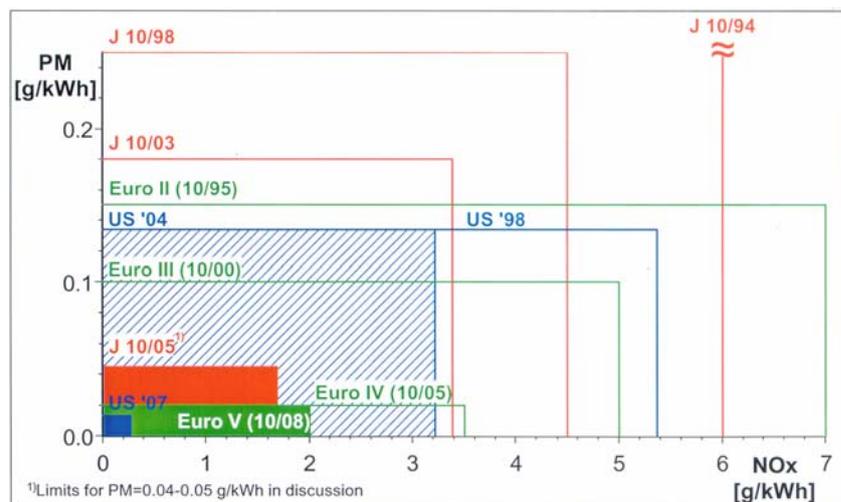


Figure 1.2: Emission limits for diesel engines [7]

Exhaust gas emitted by CI engines fueled by regular diesel fuel is a mixture of many constituents, including organic, inorganic, particulate and gaseous compounds that contribute to ambient concentrations of several critical air pollutants, e.g. fine particles, nitrogen oxides and air toxins such as aldehydes (formaldehyde, acetaldehyde, acrolein), hydrocarbons (benzene, 1,3-butadiene and polycyclic aromatic hydrocarbons or PAH). Diesel particles chiefly consist of carbonaceous material, the surface of which may also contain absorbed compounds [8]. Apart from carbonaceous materials, smaller “particles” are likely to consist of tiny droplets of volatile material, hydrocarbons, sulfuric acid or the like.

As *Figure 1.2* indicates, emission requirements can be expected to grow increasingly stringent. In 2005, nitrogen dioxides (NO_x) had to be reduced by approximately 50% more than stipulated in EURO II. In 2008, the reduction will be approximately 72%. EURO IV and V call for reducing particulate matter approximately 85% more than EURO II.

An analysis of increasing cancer mortality rates revealed that somewhere between 6 and 60 deaths per 100000 individuals in densely populated areas are caused by air pollution [9]. Several studies conducted by the World Health Organization (WHO) also concluded that mortality rises as a consequence of increased particulate emissions [10, 11]. The Swedish Environmental Protection Agency (SWEPA) studied environmental impacts in the run-up to developing an environmentally sustainable transport system. Its study identified thirteen threats [12], six of which can be directly connected to diesel exhaust gas emissions:

- Greenhouse gases, primarily CO_2
- Ground and water acidification caused by such compounds as nitrogen oxides (NO_x) and sulfur dioxide (SO_2)
- Photochemical oxidants/ground level ozone dependant on volatile organic compounds (VOC) and NO_x
- Air and noise pollution in densely populated areas and hazards from particles, NO_x and VOC
- Ground and water eutrophication caused by compounds such as NO_x
- Harmful environmental impacts caused by toxic organic substances, i.e. persistent organic compounds (POC) and persistent organic pollutants (POP), one example of which is poly aromatic compounds (PAH) in exhaust

Clearly, more stringent emission requirements are necessary.

Figure 1.3 illustrates the increase of non-road particulate emissions as on-road particulate emissions started to decrease after 1990. These differences may be caused by the longer life of non-road engines than on-road engines [12]. Moreover, non-road engines are often developed from on-road applications that have been modified for non-road applications. Generally, the most technically advanced engines are used for on-road applications. Another reason for these differences may be the stricter regulations and laws to which on-road engines are subjected at present and will be subjected in the near future [14, 15]. Nonetheless, stricter regulations for non-road engines have been passed [10]. Thus, methods to reduce the emissions of internal combustion engines, especially compression ignited engines and those producing nitrogen oxide (NO_x) and particulate matter (PM), are and will be urgently required.

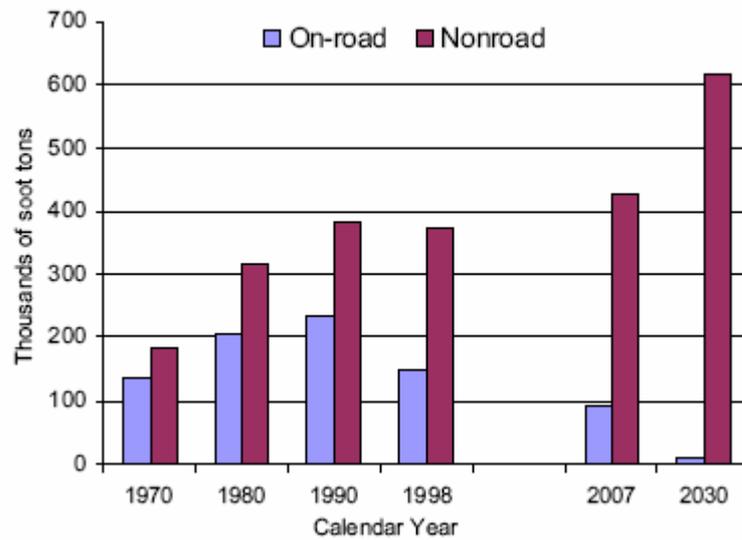


Figure 1.3: Projected development of particulate matter by 2030 [13]

Various attempts have been made with engine tuning, modifications, alternative fuels, alternative fuel composition, additives and different exhaust gas aftertreatments to reduce exhaust gas emissions. Many of these solutions have a positive effect on the majority of exhaust gas emissions. By contrast, they negatively influence the emissions of internal combustion engines, e.g. by causing modest increases of other harmful emissions.

One method to reduce emissions employs an exhaust gas recirculation system, both with and without a coolant. This system's positive impact on emissions, particularly NO_x , in both heavy-duty and light-duty engines has been verified in many investigations [16, 17, 18, 19]. EGR is not a new technology. It has been in use in gasoline fueled passenger car engines since the mid 1970s, yet EGR was only introduced in diesel passenger cars and heavy-duty engines in around the 1990s [17].

Heavy-duty applications of EGR technology have attracted the most attention because heavy-duty engines are more technically challenging. Advanced EGR systems in modern diesel cars have also adopted electronic control and EGR cooling and have become highly sophisticated. While it improves NO_x emissions, EGR also increases emissions of particulate matter (PM), hydrocarbons (HC) and CO and creates problems with fuel economy, potential engine wear and durability [17]. However, if stricter NO_x emission limits are to be met, EGR may have to be used. **Table 1.1** summarizes the history of commercial applications of EGR systems in diesel engines. Some small scale EGR applications, mostly driven by voluntary "low emission vehicle" certification programs, came earlier than the table shows.

Table 1.1: Commercial applications of EGR system in diesel engines [17]

Emissions legislation	NO_x limit	EGR application
EURO I-II	NO _x = 8-7 g/kW h	Introduced in DI and larger IDI EURO 1 engines, EGR became the main NO _x reduction strategy in nearly all EURO 2 and later diesel passenger cars and light trucks.
EURO IV	NO _x = 3.5 g/kW h	Some manufacturers of heavy-duty truck and bus engines introduced EGR; competes with urea-SCR technology.
USA 2004	NO _x + HC= 2.5 g/bhp-hr	Most manufacturers (Cummins, Volvo/Mack, DDC, International) introduced cooled EGR in heavy-duty truck and bus engines.
Japan 2005	NO _x =2.0 g/kW h	Some manufacturers (Hino, Isuzu) of heavy-duty truck and bus engines introduced EGR; competes with urea-SCR technology.

The introduction of EGR technology in passenger cars almost went unnoticed. Indeed, the introduction of EGR technology in automobile engines was not considered a major breakthrough for good reason. Since the NO_x reduction required was quite modest, the system allowed very little EGR back into the cylinder and, since the NO_x reduction was small, there was also no need for EGR cooling. In addition, typical passenger car engines usually operate at part load conditions where temperatures are relatively low and may lead to slight engine wear. Higher demands on passenger car EGR systems were only came about when EURO 3 limited NO_x to 0.5 g/km and EURO 4 legislation limited it to 0.25 g/km. Cooled EGR was introduced in larger engines in EURO 3, and became the standard in EURO 4 passenger cars [1, 7, 17]. The application of EGR technology to heavy-duty engines was more conspicuous and drew a great deal of attention. HPL (high pressure loop) cooled EGR has been, is and will be the most expedient technology for NO_x reduction. To meet the NO_x reduction aimed for in the experimental investigation, this study used a heavy-duty DEUTZ 1013 EC compliant with the EURO2 standard (the original engine had no EGR system), retrofitted with a cooled external EGR system controlled by compression air and capable of changing cooling temperatures.

In addition to reducing emissions to meet stringent requirements, finding, applying and substituting alternative fuels for fossil fuels are important and urgent tasks for researchers and manufacturers because fossil oil is running out. **Figure 1.4** presents the development of alternative fuel use. Even though these fuels have some drawbacks (e.g. higher specific fuel

consumption), alternative fuels not only represent a solution to counter the mounting scarcity of raw materials but also to reduce some central elements of emissions.

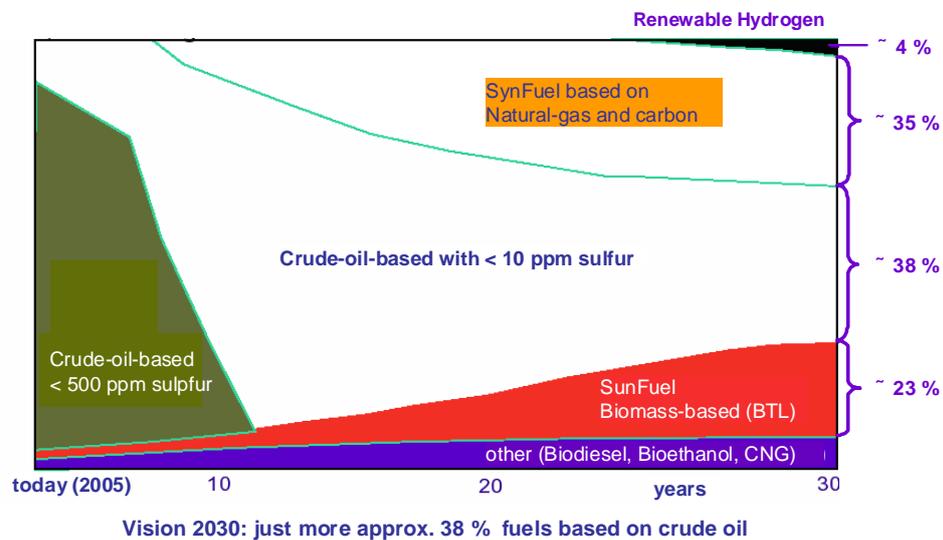


Figure 1.4: Fuel scenario for Europe [7]

The trend toward using biofuel will increase significantly in the near future. Biofuels used in internal combustion engines and especially in compression ignited engines have many forms. They may be plant oils or upgrading products, e.g. methyl esters. They may be neat or blended.

As the use of biofuels increases, numerous investigations of the influences of biofuels on engine process and emissions have been and are being conducted, for example [12], [27], [28], [29], [34], [35], [37], [38], [39], [40] and many more. While this research work has delivered numerous findings on ecological and economic effects, it has essentially focused on light-duty engines as in *Cole, R. L.* 1997 [63], on injection spray and fuel-air mixture generation without EGR or only on rapeseed oil and neat RME as in *Hopp, M.*, 2005 [27], on light-duty engine without alternative fuels as in *Raffael, A. G.*, 2001 [31], only on heavy-duty engines without EGR and using ethanol fuel as in *Nord, K.* September 2005 [12], only on light-duty diesel engines with EGR as in *Röpke, B.* 1996 [33] or only on effects of EGR as in *Timothy, J.* [22]. *Spessert, B. M.* [20] conducted experiments with RME and presented findings on the influence of RME on exhaust emission and noise emission but only in small diesel engines. *Shih, L. K.-L.* (1998) [64] tested seven additives and noted that a blend containing 20% ethanol was one of two fuels that produced the greatest reduction in smoke opacity. In addition, several reports have focused on various oxygenated fuels in diesel engines. Some conclusions (*Yukio, A. and Takanobu, S.* 1990 [75]; *Frank, J. L. and Daniel, M. M.* 1993 [76]) have elucidated the remarkable effects of the addition of oxygenated organic compounds in fuel to make it burn cleaner. *Kramer, K.* 1980 [29] investigated the

influence of methanol and methyl fuel on gasoline engines. *Likos, B.* 1982 [74] studied the addition of lesser alcohols such as methanol and ethanol to diesel fuel, which effectively reduces particulate emission without sacrificing other emission components. *Murayama, T.* 1982 [73] investigated effects of non-alcohol oxygenated organic compounds to improve diesel emission reduction. Some studies have addressed the effect of the addition of liquid oxygenated agents to fuel on diesel combustion and emissions (*Yukio et al.*, 1990; *Oh, Y. T.* and *Choi, S. H.* 2005) [28]. Some papers have presented experimental findings on emissions and performance with blends of ethanol (up to 15%) and diesel fuel (puranol additive up to 2%) (*Ahmed. Irshad.*, 2001) [35]. Some research such as *Verbeek, R.* [36] has focused on impacts of EGR associated with an aftertreatment device. Some studies of gas-to-liquid (GTL) have been carried out focusing primarily on effects on emissions. *Allerman, T. L.* and *Eudy, L.* (2004) at the National Renewable Energy Laboratory (USA) [85] tested fuel properties and the effects of GTL on emissions on a fleet of class 6 vehicles. *Munack, A.* and his assistants (2005) in Germany [69] focused on the influences of GTL and its blends on emissions, particularly particulate matter, in a Mercedes Benz engine OM 906 LA. *Herrmann, H.-O.* of DaimlerChrysler AG, Stuttgart (2004) studied the effects of GTL and its blends (20% and 50% GTL) on emissions and performance of the Mercedes-Benz passenger car engine E220 CDI [86]. *Clark, N.* [89] and *Norton, P.* (1999) [87] at West Virginia University researched the effects of Fischer Tropsch diesel blends on the emissions of an on-road truck engine. Most of these studies of GTL have only focused on influences on emissions and impacts on the health of humans and animals. Effects on thermodynamics and combinations of the effects of GTL and other factors such as EGR ratios or EGR temperatures have not been considered.

This study concentrates on the effects of neat and blended (varying ratios) rapeseed methyl ester, the blend of ethanol and conventional diesel fuel with a low percentage of ethanol (7.7%) and different diesel fuels and gas-to-liquid (GTL) and its blend on the engine process. To this end, a EURO 2 norm heavy-duty engine DEUTZ 1013EC equipped with an external cooled exhaust gas recirculation system controlled by external compression air and external cooled water (the prototype engine was not equipped with an EGR system) was used for experimental tests. Different EGR rates and EGR temperatures are variables in the investigations. Modifications of the injection strategy, e.g. retarding or maintaining early injection timing, are subjects of investigation. Sensors, measuring devices and analysis devices are used to calculate and analyze thermodynamics and emissions and cylinder pressure levels (noise emission) in particular. The findings are applied to determine the influences of some new alternative fuels on the process of a heavy-duty engine and to optimize working process of engine with the different alternative fuels.

2 Research Objectives

This study focuses on research intended to produce findings to explain what influences exhaust gas recirculation with different EGR ratios, different temperatures of cooled recirculated exhaust gas and retarded or earlier injection timing combined with different alternative fuels such as Rapeseed Methyl Ester (RME), a blend of ethanol and diesel fuel (O₂-1: diesel fuel without RME), a blend of ethanol and diesel fuel (O₂-2: diesel fuel with 5% RME), different blend rates of RME and conventional diesel fuel and GTL and its blend have on the engine process of a heavy-duty direct injection diesel engine (based on thermodynamic and emission parameters).

The properties of RME, including blends with diesel fuel and O₂ Diesel with a low ethanol ratio and blends of RME with diesel fuel in different percentages of RME and GTL and its blend, which differs from conventional diesel fuel, not only cause an engine's working process but also its parameters to change. Findings obtained from an experimental test bench station and emissions analyzing devices were used to evaluate and adjust the working process to meet more stringent emission requirements and to provide a basis for making decisions on the applying new alternative fuels in practice.

Investigations based on an ESC test cycle were conducted in five operating points, e.g. nominal power, maximum torque, two part load points (higher part load and lower part load) and low idle point. A heavy-duty engine (EURO II norm), originally without any exhaust gas recirculation system, was retrofitted with an externally cooled exhaust gas recirculation system. The exhaust gas recirculation system is operated by external compression air, thus allowing modifications of exhaust gas recirculation ratios. Moreover, the EGR system's external water coolant also permits adjusting EGR temperatures to experimental requirements.

Combining different operating conditions of exhaust gas recirculation, different alternative fuels and different blend ratios with changes in the injection timing of the unit-pump fuel system which will be fulfilled by an application controlling unit at steady-state operating points will be variable of this investigation. Other parameters, e.g. ignition delay, maximum cylinder pressure, maximum heat release rate, combustion duration, start of combustion, maximum cylinder pressure gradient, were calculated with an experimental data processing software. These parameters are fundamental to evaluating and clarifying various characteristics of an engine's working process at different operating points and under different conditions.

This study contributes to better understanding of the different characteristics of a direct injection heavy-duty engine's working process, thermodynamics parameters, emissions and,

in particular, spectrum of cylinder pressure (dB) using an EGR system and the different alternative fuels such as RME and blends with different percentages of RME (B10-10% RME, B50-50% RME), O₂-Diesel, Gas-to-Liquid and its blend (G50-50% GTL).

The experimental results such as specific fuel consumption, energy consumption and emissions (NO_x and FSN) are also used to optimize working process of engine with the different alternative fuels at different operating conditions. Findings from this study also contribute to making decisions on the applying alternative fuels and their blends and increase of RME's percentage in blend in practice.

3 State of the Art

3.1 Exhaust Gas Recirculation System

3.1.1 State of Technology

Engine designers, researchers and regulators have been interested in exhaust gas recirculation (EGR) for quite some time. EGR facilitates significantly reducing NO_x emission in light and heavy-duty diesel engines. EGR was initially used in gasoline fueled passenger car engines [17]. Later, EGR was also introduced in diesel passenger cars and, in around the 1990s, in heavy-duty diesel engines [1, 2]. The heavy-duty applications of EGR technology attracted the most attention because heavy-duty engines are more technically challenging. In modern diesel cars, EGR systems also adopted electronic control and EGR cooling and became highly sophisticated. Recently, EGR has emerged as a means to meet more stringent requirements for heavy-duty engines, in particular NO_x requirements. Nevertheless, effectively introducing EGR into the combustion chamber of a multi-cylinder engine remains a considerable challenge. There are two types of EGR: internal and external. Internal EGR systems use variable valve timing or other devices to retain a certain fraction of exhaust gas from a preceding cycle. External EGR systems use piping to route exhaust gas to the intake system where it is inducted into succeeding cycles. This is the preferred approach at present. While internal EGR provides very short response time, its practical application will be impossible until variable camshaft technology becomes widely available. What is more, internal EGR cannot be cooled, whereas external EGR can. Cooling the residuals improves fuel economy and engine performance and enables further reducing NO_x .

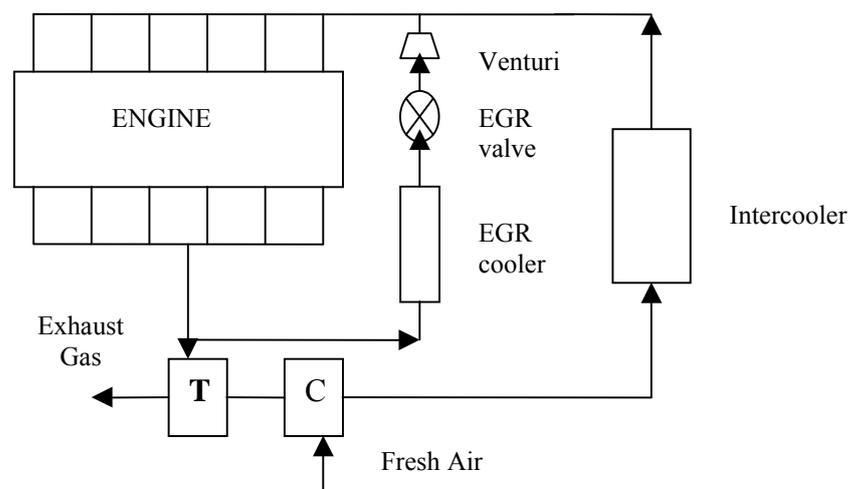


Figure 3.1: Layout of the short-part EGR (HPL) system with EGR cooler [17]

However, the application of external EGR gives rise to several system issues that need to be dealt with. In particular, the high efficiency of state-of-the-art turbochargers often establishes conditions in which intake manifold pressure is higher than exhaust gas manifold pressure. If residuals flow from the exhaust manifold directly to the intake manifold (the so called short-path EGR [22] or high pressure loop [17]), the exhaust back pressure must be higher than the intake back pressure. Branching at a point upstream from the turbocharger (see *Figure 3.1*) where exhaust pressure is higher than intake manifold pressure solves this dilemma. Another approach, more common in heavy-duty engines, is the use of a venturi to increase the kinetic energy of EGR allowing more to flow with lower pumping losses. Indeed, some studies have confirmed that the use of venturi improves fuel economy. (See *Figure 3.1* and *Figure 3.2* for the EGR system layout).

Another method for conducting external EGR, pipes the EGR after the turbine into the fresh air intake upstream from the compressor (long-path EGR [22] or the low pressure loop [17]). Although this method has many advantages, e.g. lower fuel consumption and better preserved engine durability with use of particulate filter, it is not favored over the HPL (short-path) system for a number reasons:

- Even though the LPL system uses a particulate filter (LPL EGR is sourced downstream from the particulate filter), it is not entirely free from carbonaceous material because the filter trapping efficiency is less than 100%. With carbonaceous material still remaining in the recirculated exhaust stream, it can potentially erode the compressor wheel as it turns at a high rate of speed.
- Unburned oil vapors as well as any unburned fuel adsorbed on the surface of carbonaceous particles accumulating in the inlet system can emit carbon monoxide (CO) gas when exposed to high temperatures. If this happens, CO displaces fresh and cooled air to the engine and this is detrimental to combustion efficiency.
- Plumbing for the long-path (LPL) EGR system arrangement is often awkward and cumbersome.
- As EGR temperature drops, acidic condensation develops. This problem appears to be more pronounced in LPL (long-path) than HPL EGR (short-path).

In light of these disadvantages, the DEUTZ engine in this study was equipped with external EGR, cooled EGR and a short-path (HPL) EGR system (see *Figure 3.1*).

3.1.2 Methods for Calculating EGR Ratios

The EGR ratio is the ratio of the amount of EGR to the charge aspirated into an engine cylinder. In this study, the EGR ratio was calculated with the following equation (3.1).

$$\text{EGR} = \frac{\dot{m}_{\text{EGR}}}{\dot{m}_{\text{air}} + \dot{m}_{\text{EGR}}} \quad (3.1)$$

\dot{m}_{EGR} is the flow mass rate of EGR

\dot{m}_{air} is the flow mass rate of fresh air

To calculate correct EGR ratios, the flow EGR rate has to be defined exactly. However, this is very difficult because of the high temperature and contamination by ash, soot and unburned hydrocarbon [21]. There are many methods to calculate the EGR ratio.

- **Calculating the EGR Ratio from the Mixture Temperature**

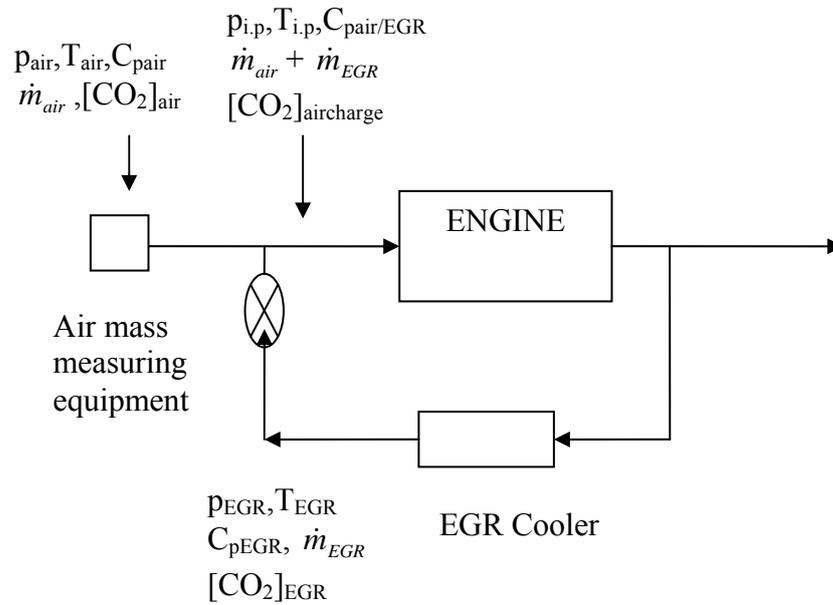


Figure 3.2: Schematic for calculating the EGR ratio

The energy balance can be used to calculate the EGR ratio with the following equation [21]:

$$\dot{m}_{\text{air}} \cdot C_{\text{pair}} \cdot T_{\text{air}} + \dot{m}_{\text{EGR}} \cdot C_{\text{pEGR}} \cdot T_{\text{EGR}} = (\dot{m}_{\text{air}} + \dot{m}_{\text{EGR}}) \cdot C_{\text{pair/EGR}} \cdot T_{i,p} \quad (3.2)$$

\dot{m}_{air} Flow mass rate of air

\dot{m}_{EGR} Flow mass rate of EGR

C_{pair} Specific heat of air at constant pressure

C_{pEGR} Specific heat of EGR at constant pressure

$C_{\text{pair/EGR}}$ Specific heat of air and EGR mixture at constant pressure

T_{air}	Air temperature
T_{EGR}	EGR temperature
$T_{\text{i,p}}$	Inlet port temperature
p_{air}	Air pressure
p_{EGR}	EGR pressure
$p_{\text{i,p}}$	Inlet port pressure

The following ensues from equation (3.1):

$$\dot{m}_{\text{EGR}} = \dot{m}_{\text{air}} \cdot \frac{\text{EGR}}{(1 - \text{EGR})} \quad (3.3)$$

The following ensues from equations (3.2) and (3.3):

$$\text{EGR} = \frac{1}{1 + \frac{C_{\text{pEGR}} \cdot (T_{\text{EGR}} - T_{\text{i,p}})}{C_{\text{pair}} \cdot (T_{\text{i,p}} - T_{\text{air}})}} \quad (3.4)$$

Since the differences between the specific heat of EGR (C_{pEGR}) and the specific heat of fresh air (C_{pair}) are negligible, the following equation can be applied:

$$\text{EGR} = \frac{T_{\text{i,p}} - T_{\text{air}}}{T_{\text{EGR}} - T_{\text{air}}} \quad (3.5)$$

Equation (3.5) can be used to easily calculate the EGR ratio. However, its precision decreases as the temperature difference ($T_{\text{EGR}} - T_{\text{air}}$) decreases. This equation cannot be used to calculate the EGR ratio for cooled EGR because of the modest temperature difference [21]. Temperature sensors have to be calibrated to a common zero point.

• Calculating EGR Ratio from the CO₂ Balance

The CO₂ balance in the inlet port can be used to calculate the EGR ratio with the following equation [21]:

$$[\text{CO}_2]_{\text{air}} \cdot \dot{m}_{\text{air}} + [\text{CO}_2]_{\text{EGR}} \cdot \dot{m}_{\text{EGR}} = [\text{CO}_2]_{\text{aircharge}} \cdot (\dot{m}_{\text{air}} + \dot{m}_{\text{EGR}}) \quad (3.6)$$

$[\text{CO}_2]_{\text{air}}$	Concentration of carbon dioxide in air
$[\text{CO}_2]_{\text{EGR}}$	Concentration of carbon dioxide in EGR
$[\text{CO}_2]_{\text{aircharge}}$	Concentration of carbon dioxide in the air charge

The following ensues from equations (3.3) and (3.6).

$$\text{EGR} = \frac{[\text{CO}_2]_{\text{aircharge}} - [\text{CO}_2]_{\text{air}}}{[\text{CO}_2]_{\text{EGR}} - [\text{CO}_2]_{\text{air}}} \quad (3.7)$$

If the carbon dioxide concentration $[\text{CO}_2]_{\text{air}}$ (approx. 500ppm) is disregarded, the EGR ratio can be calculated with the following equation:

$$\text{EGR} = \frac{[\text{CO}_2]_{\text{aircharge}}}{[\text{CO}_2]_{\text{EGR}}} \quad (3.8)$$

- **Calculating the EGR Ratio from the Air Mass**

Using air mass to calculate EGR ratios requires measuring equipment. The pressure and temperature of fresh air and the mixture of EGR and charge must be measured.

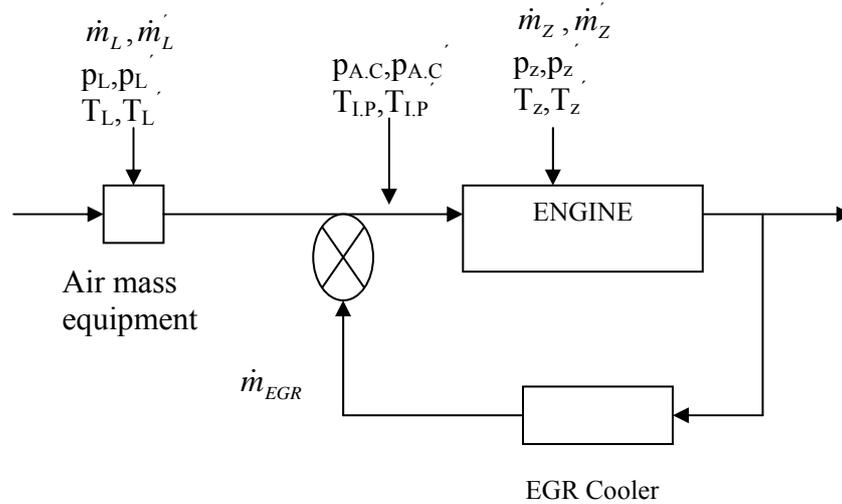


Figure 3.3: EGR system schematic

When there is no EGR, the mixture flow mass rate in the inlet port (or in the cylinder at the end of the intake cycle) is the same as the flow mass rate of fresh air measured by the equipment measuring air mass. Thus, the following equation can be formulated:

$$\dot{m}_z = \frac{\dot{V}_z \cdot p_z}{R_{\text{air/EGR}} \cdot T_z} = \dot{m}_L \quad (3.9)$$

When EGR is used, the following equation can be formulated:

$$\dot{m}'_z = \frac{\dot{V}_z \cdot p'_z}{R'_{\text{air/EGR}} \cdot T'_z} = \dot{m}'_L + \dot{m}'_{\text{EGR}} \quad (3.10)$$

The following ensues from equations (3.9) and (3.10):

$$\frac{\dot{m}'_z}{\dot{m}'_z} = \frac{\dot{V}_z \cdot p'_z}{R'_{\text{air/EGR}} \cdot T'_z} \cdot \frac{R_{\text{air/EGR}} \cdot T_z}{\dot{V}_z \cdot p_z} \quad (3.11)$$

\dot{m}_z	Mixture flow rate in cylinder at the end of intake stroke (without EGR)
\dot{m}'_z	Mixture flow rate in cylinder at the end of intake stroke (with EGR)
V_z	Cylinder volume
p_z	Cylinder pressure at the end of intake stroke (without EGR)
p'_z	Cylinder pressure at the end of intake stroke (with EGR)
T_z	Temperature in cylinder at the end of intake stroke (without EGR)
T'_z	Temperature in cylinder at the end of intake stroke (with EGR)
$R_{\text{air/EGR}}$	Constant of mixture (without EGR)
$R'_{\text{air/EGR}}$	Constant of mixture (with EGR)

If the difference between the constants of mixture with EGR and without EGR is disregarded ($R'_{\text{air/EGR}} \approx R_{\text{air/EGR}}$), the following equation can be formulated:

$$\frac{\dot{m}'_z}{\dot{m}_z} = \frac{p'_z}{p_z} \cdot \frac{T_z}{T'_z} \quad (3.12)$$

$$\dot{m}'_z = \dot{m}_z \cdot \frac{p'_z}{p_z} \cdot \frac{T_z}{T'_z} = \dot{m}'_L \cdot \frac{p'_z}{p_z} \cdot \frac{T_z}{T'_z} \quad (3.13)$$

\dot{m}'_L Air flow rate (without EGR)

\dot{m}_L Air flow rate (with EGR)

As is known:

$$\text{EGR} = \frac{\dot{m}'_{\text{EGR}}}{\dot{m}'_z} = \frac{\dot{m}'_z - \dot{m}'_L}{\dot{m}'_z} = 1 - \frac{\dot{m}'_L}{\dot{m}'_z} \quad (3.14)$$

\dot{m}'_{EGR} Flow rate of EGR

The following ensues from equations (3.13) and (3.14):

$$\text{EGR} = 1 - \frac{\dot{m}'_L}{\dot{m}'_L} \cdot \frac{p'_z}{p'_z} \cdot \frac{T'_z}{T'_z} \cdot 100 [\%] \quad (3.15)$$

$$\text{If } \frac{T'_z}{T'_z} \approx \frac{T'_{i,p}}{T'_{i,p}} \text{ and } \frac{p'_z}{p'_z} \approx \frac{p'_{\text{aircharge}}}{p'_{\text{aircharge}}} :$$

$$\text{EGR} = 1 - \frac{\dot{m}'_L}{\dot{m}'_L} \cdot \frac{p'_{\text{aircharge}}}{p'_{\text{aircharge}}} \cdot \frac{T'_{i,p}}{T'_{i,p}} \quad (3.16)$$

$p'_{\text{aircharge}}$ Air charge pressure (without EGR)

$\hat{p}'_{\text{aircharge}}$ Air charge pressure (with EGR)

$T'_{i,p}$ Inlet port temperature (without EGR)

$T'_{i,p}$ Inlet port temperature (with EGR)

If $\frac{p'_{\text{aircharge}}}{\hat{p}'_{\text{aircharge}}} \cdot \frac{T'_{i,p}}{T'_{i,p}} \approx 1$, then the EGR ratio can be calculated with the following equation:

$$\text{EGR} = 1 - \frac{\dot{m}'_L}{\dot{m}'_L} \quad (3.17)$$

This study uses equation (3.16) to calculate the EGR ratio.

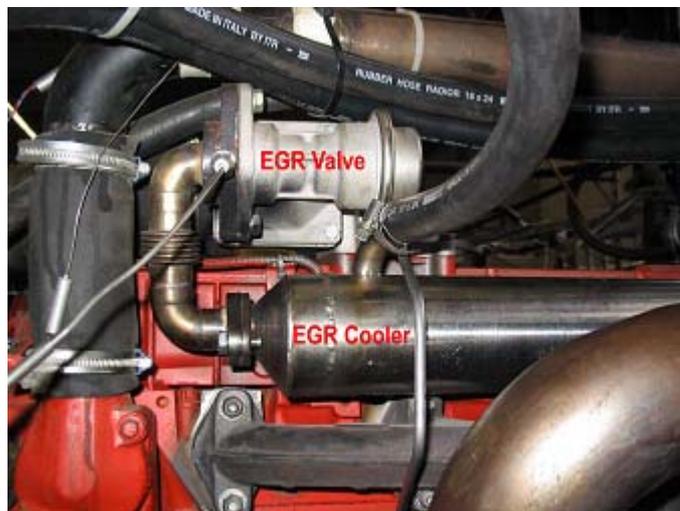


Figure 3.4: Exhaust gas recirculation system

Figure 3.4 shows a real EGR system. When equations (3.16) and (3.17) are used to calculate the EGR ratio, the maximum difference between the two equations is 10%.

The experiments in this study used an EGR system with an EGR cooler with water as the coolant and an EGR valve controlled by compressed air (see **Figure 3.4**). Experimental findings also indicated the maximum EGR ratio is about 8 to 9% at most operating points and under most operating conditions, except for lower part load (1100 1/min-213Nm). In the operating point for lower part load, the maximum EGR ratio reached 27% when throttling was applied in the intake manifold. (See the test bench in **Figure A.1** in *Appendix*).

3.2 Alternative Fuels

3.2.1 Biodiesel, Bioethanol and Shell Middle Distillate - Gas-to-Liquid (GTL) as Fuels for Modern Diesel Engines

The basic principles behind the use of plant oils as a fuel for diesel engines were already known and applied by Rudolf Diesel himself and mentioned in his patent application in 1912 [60]. New impulses came from the oil crisis in the 1970s and again from the Gulf War in 1991 [60]. Highly dependent on huge imports oil, a finite resource, not only the European Union but also the rest of the world is facing increasing risks to the security of the energy supply for transportation. The International Energy Agency (IEA) has named two primary risks:

- The gap between the production of and demand for fossil oil is growing worldwide.
- The energy demand of developing countries such a China and India is growing dramatically.

According to the IEA, the need for all and any alternative fuels will grow in the transportation sector. Biodiesel is one of these alternative fuels [61]. In 1982, the Institute for Agricultural Engineering in Braunschweig, Germany tested Rapeseed Oil Methyl Ester (RME) in diesel engines for the first time. A small pilot plant at the agricultural University in Austria had already tested the production of RME with a new technology in 1985 and the first farming cooperative in Austria started commercially producing biodiesel derived from rapeseed and sunflowers in 1990. To date, plant oil methyl ester, produced from rapeseed oil in Germany, has been the only alternative fuel produced from plant oil that has succeeded in acquiring a certain standing on the diesel fuel market. Use of this fuel is associated with a number of environmental advantages. Of the various options for utilizing plant oils as fuel, only the transesterification of rapeseed oil into rapeseed oil methyl ester (RME) has hitherto been implemented in Germany on a commercial scale [62]. A blend of diesel fuel and an unspecified addition of up to 5% RME (B5) have been used for most diesel engines. Some 500,000 tons of biodiesel was used for blending in 2005 [47]. Commercial use of biodiesel

and a percentage of biodiesel in blends can be expected as fossil fuels prices increase, emission requirements are tightened and biodiesel blend will be introduced by legislation.

Along with the many alternative fuels such as CNG, LPG, biodiesel and BTL, researchers and manufacturers have also focused on another alternative fuel: ethanol. Ethanol has been used in vehicles since the early 1990s to bring about more complete combustion of fuel and thereby reduce emissions of carbon monoxide (CO), a regulated pollutant hazardous to human health. The reduction of CO can be significant (20% to 30%), particularly in cold weather in older or poorly maintained vehicles [48, 52]. Moreover, many emissions tend to decrease when a low level blend of ethanol and conventional fuel is used. The blend of ethanol and gasoline was widespread earlier, especially in USA and Brazil. Ninety-eight ethanol plants throughout the USA have the capacity to produce nearly 3.7 billion gallons annually. Moreover, while blends containing up to 85% ethanol may be approved in theory, gasoline blends with up to 10% ethanol (E10) are approved for use in all gasoline vehicles and have been in use across the nation to improve air quality for many years [52]. In the 1980s and early 1990s, there was considerable interest in using ethanol and methanol as the primary fuel for diesel engines (*Moses et al.* 1980 [77]; *Toepel* 1983 [78]; *Bechtold* 1991 [79]). Although these engines produced significantly lower particulate matter emissions than engines fueled with neat diesel fuel, the disadvantages outweighed the advantages. The engines required modifications such as a higher compression ratio and fuel injectors with larger volumetric delivery. They were hard to start and did not always run well at light load. Starting aids and cetane enhancers were required and the fuel economy was poor because alcohol has a lower heat value than diesel fuel.

An ethanol and diesel fuel blend has recently been receiving more attention not only because of more stringent emissions requirements but also because modern technology permits manufacturing engines that run with many different fuels without any modifications (flexible-fuel vehicles FFV) and producing many additives that effectively enhance the properties of fuels. The literature confirms that diesel blends with up to 15% ethanol, known as E-diesel or O₂-diesel, can reduce certain regulated diesel emissions, especially diesel particulate matter. Its disadvantages include a low flash point, which may present a safety issue. *Shih* (1998) [64] and *Ahmed and Marek* (1999) [35] tested and reported that an ethanol and diesel blend reduced NO_x emissions at light load and low speeds. They also observed an increase in unburned HC. Ethanol is made from agricultural crops, predominantly corn but also sugar beets and sugar cane. As it grows, a corn plant captures carbon dioxide from the atmosphere, converting it to sugars and starches through photosynthesis. These sugars and starches are then fermented to produce ethanol, thus transforming the building blocks of a potent greenhouse gas into a renewable fuel. Numerous techniques for using diesel and ethanol concurrently in CI engines have been evaluated. Some of these techniques are alcohol evaporation, dual injection, alcohol-diesel fuel emulsions and alcohol-diesel fuel blends. Of

these approaches, only alcohol-diesel emulsions and blends are compatible with most commercial diesel engines. Since emulsions are difficult to attain and tend to be unstable, blends, either as micro-emulsions or with co-solvents, hold the most promise since they are stable and can be used in engines with relatively few modifications.

Synthetic fuels such as GTL (Gas-to-Liquid or Shell Middle Distillate) diesel fuel have enjoyed a significant rise in interest in recent years. Traditionally, diesel fuels and other petroleum products are manufactured by refining crude oil. However, they can also be produced synthetically from various carbon bearing feedstocks. The feedstock attracting the most attention today is natural gas, for example: CNG, but synthetic fuels can be also produced from such sources as coal or biomass.

The first and best known synthetic fuel technology is the Fischer-Tropsch (FT) process developed in Germany in the 1920s. Apart from two historical episodes, namely Germany during World War II and South Africa during the economic embargo, commercial use of FT fuels has been extremely limited. Nevertheless, several companies have continued research on FT, leading to the development of a mature technological process to improve fuel economy. Today, the key FT technology players are large oil companies such as Shell, ExxonMobil and Sasol [88].

By virtue of their emphasis on the natural gas, synthetic fuel processes are frequently referred to as Gas-to-Liquid or GTL technologies. This connotation does not cover every synthetic fuel technology since liquid fuels may be and have been produced from almost any gaseous, liquid or solid carbonaceous feedstock. Coal is a good example of a solid feedstock that has been used in the past to manufacture FT diesel fuel. The term GTL is sometimes also used to refer to non-FT fuels, e.g. dimethyl ether, which can be produced from natural gas feedstock too [88]. This study is limited to fuels produced by Fischer-Tropsch synthesis and focuses on natural gas as the most important feedstock, i.e. the Shell Middle Distillate.

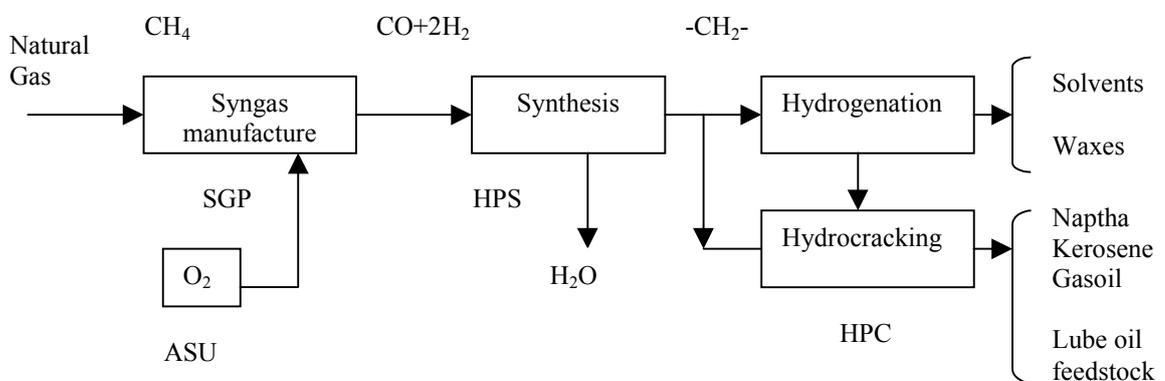


Figure 3.5: Schematic of Shell's GTL production process (SMDS) [84]

The Shell Middle Distillate Synthesis or SMDS process is well documented [84, 85]. Therefore, this study only describes it briefly. *Figure 3.5* illustrates this process, which was developed at the Shell Research & Technology Centre Amsterdam [84] and is essentially comprised of three stages:

- Manufacture of synthesis gas (hydrogen + carbon monoxide- with an H₂ to CO ratio of approximately two to one, H₂: 35%-75%, CO: 15%-45%) from natural gas by non-catalytic auto-thermal partial oxidation using, for example, the Shell Gasification Process (SGP).
- Wax synthesis from reactions between CO and H₂ by Heavy Paraffin Synthesis (HPS), followed by flash distillation to separate light ends (e.g. liquefied petroleum gas). This stage is the heart of SMDS process.
- Wax cracking to distillates by Heavy Paraffin Conversion (HPC), where the boiling range and quality of the products can be adjusted to produce either kerosene or atmospheric gas oil (diesel).

A recent modification to this process, designated as SMDS-2, provides an improved HPS catalyst, which will enable manufacturers to increase production capacity considerably. In addition, adjusting the severity of the hydrocracking/isomerization (HPC) stage enables controlling the n- paraffin to iso-paraffin ratio in the final product. Two types of catalysts have been used: iron-based catalysts and cobalt-based catalysts. The choice of catalyst is very important for determining what feedstocks can be converted into synthetic fuel. Iron has mostly been used with coal feedstock (in Germany and South Africa). Cobalt is considered more suitable for natural gas processes and is currently being pursued by both Shell and ExxonMobil [84].

Synthesis fuel-GTL is an extremely clean fuel without sulfur, aromatics or odors and has a cetane number above 70. Hence, once it has been certified as a road fuel, it may facilitate further reductions of engine emissions required to meet future emission regulations. Key oil companies such as Shell, Sasolchevron and ConocoPhillips have announced several global-scale GTL plants will be in operation around 2010 [86]. They could produce quantities exceeding 1 million barrels per day. It is estimated that the GTL diesel supply could correspond to 5% - 10% of the demand for diesel in the EU by 2010 [86].

Therefore, of the various alternative fuels currently being considered for road transport, GTL diesel fuel is considered the most likely to become available in significant quantities in the near future. Another advantage of GTL diesel fuel is large-scale production would not be constrained by feedstock limitations in the near future (as would be the case with bioderivative fuels such as biodiesel or bioethanol) and widespread market launch would not be constrained by any need for a new distribution infrastructure (as would be the case with gaseous fuel such as CNG and hydrogen). Production of GTL diesel fuel is already

economically viable today and significantly less expensive than many other alternative fuel technologies.

3.2.2 Flexible Fuel Vehicles

A flexible fuel vehicle or dual fuel vehicle can typically alternate between two or more sources of fuels. One common example is a vehicle that can accept blends of conventional fuel and alternative fuels or 100% alternative fuels without modifications. In practice, only flexible fuel vehicles can use blends with more than 10% ethanol. These vehicles have special controls and materials to accommodate higher ethanol levels. More than four million flexible fuel vehicles are on the road today and these vehicles can use virtually any proportion of ethanol (up to 85%) with gasoline [4, 66, 67].

3.2.3 Properties and Specifications of Alternative Fuels and their Blends

- **RME and Blends**

Biodiesel is produced in a pure form (100% biodiesel fuel referred to as B100 or neat biodiesel) and may be blended with petroleum-based diesel fuel. Several standard-setting organizations worldwide have recently adopted biodiesel specifications. German authorities have issued a provisional specification for FAME in DIN 51606 [60]. The European Committee for Standardization (CEN) fixed a technical standard for biofuels (EN14214) [72].

European specifications include more stringent limits for sulfur and water as well as a test for oxidation stability, which the current ASTM specification lacks. Recently, ASTM international approved a specification for biodiesel referenced as D6751 [72]. Depending on the biomass feedstock and the process used to produce the fuel, B100 fuels should meet the requirements of either DIN 51606 or approved American and European specifications, such as ASTM D 6751 or EN 14214 [72]. The European Commission's action plan for the promotion of biofuels calls for a medium-term increase in consumption. The directive on the Promotion and Use of Biofuels and Other Renewable Fuels in the Transport Sector requires each member state to ensure that biofuel accounts for 2% of total fuel consumption (measured by energy content) by the end of 2005 and at least 5.75% by the end of 2010 [47].

Public and private bodies have recently adopted positions on the use of biodiesel blends. With up to 5% RME (B5) added, these "diluted" fuel are found in most filling stations in Germany. The standard EN 590 [72] lays out specifications for the use of biodiesel blends. EN 590 (specifically EN 590:2000) describes the physical properties all diesel fuel must have if it is to be sold in the EU. It allows blending up to 5% biodiesel with a "normal" 95/5 mix. The directive 2003/30/EG states that "as a result of technological advances, most vehicles

currently in circulation in the European Union are capable of using a low biodiesel blend without any problem. The most recent technological developments make it possible to use higher percentages of biofuel in the blend⁹. The most popular biodiesel blends in the USA are B20 (20% biodiesel), compliant with the Energy Policy Act of 1992 (EPAAct) and B5 or B2 [50]. Any diesel engine can operate on these blends with few or no modifications. When used in low-level blends of 5% biodiesel (B5) or lower, biodiesel is invisible to the user. When biodiesel is used as B20, the user may experience a 1-2% decrease in power, torque, and fuel economy. However, these changes are usually not noticeable.

Table 3.1: Properties of neat RME [68]

	Test procedure	Unit	DF	RME
Heat value	DIN 51606	MJ/kg	43.866	37.6
Cetane number	DIN EN ISO 5165		55.5	45...59
Density in 15°C	ISO 3675	g/cm ³	0.8276	0.875...0.900
Sulfur content	ISO 4260	mg/kg	8	9...12
Flash point	ISO 2719	°C	56.9	110
Coke residue (of 10 % distillations residue)	DIN EN ISO 10370	%	0.004	0.05
Water content	DIN 51606	mg/kg	130	300
Oxide ash	DIN EN ISO 6245	%mass	0.001	0.01
Total contamination	DIN EN 51419	mg/kg	27.3	20
Copper corrosion (3h at 50°C)	EN ISO 2160		1	1
Viscosity in 40°C	ISO 3104	mm ² /s	2.71	3.5...5
CFPP	DIN EN 116	°C	-8	-9...-7

The first lesson learned from tests of plant oils as a fuel was that pure oils, even fully refined, are not suited for modern fast running diesel engines with high efficiency and low emission profiles. Methyl esters became the plant oil derivative of choice. They are rather simple to produce and very nearly approximate the fuel properties of diesel. Differences in density and viscosity are slight and acceptable. The higher flash point is a beneficial safety feature and plant oil's complete lack of sulfur is the reason biodiesel has excellent low SO_x emissions.

Table 3.1 lists the chief properties of neat biodiesel (RME). Biodiesel (RME) is already a naturally oxygenated fuel with a molecular oxygen content of about 10% that varies depending on the length of the fatty acid chain. Oxygen is a key factor in emission levels as well as lower caloric value. One significant quality criterion for plant oil methyl esters is their water content. Unlike diesel fuel, which absorbs virtually no water, biodiesel can absorb up to approximately 1700 ppm of water at room temperature [62]. With such high water content, plant oil methyl esters may experience biological growth. If this phenomenon occurs in a vehicle, it results in engine failure. DIN 51606 limits the water content to 300 mg/kg. This narrow tolerance range for water is intended to make it unnecessary to add biocides to fuel to resist biological growth. Relatively large quantities of water may cause corrosion phenomena, e.g. the chromate film on chromated components may dissolve and brass components may dezincify. This phenomenon is associated with gumming, which can cause blockages in the fuel injection system.

At low temperatures, elevated water content may cause water separation both in pure biodiesel and in blends with diesel fuel. Such water separation may cause fuel line and fuel filter blockages.

Plant oil methyl esters have a flash point above 100 °C. This is an important quality criterion. Lower values always indicate contamination with methanol from the transesterification process or mixing with diesel fuel or gasoline. Unlike diesel fuel, which has a flash point above 55°C, plant oil methyl esters may be stored without being subject to particular safety measures, a feature which may be of particular interest to small users with their own fuel storage facilities.

Started as the CFPP (Cold Filter Plugging Point), approximate temperature limits for low temperature flow behavior are required for plant oil methyl esters just as for diesel fuel. However, experience has shown that components of solid plant oil methyl esters may be precipitated even if the specified CFPP values are observed. This may be the case particularly if plant oil methyl esters are stored at low temperatures for an extended period, even if the temperatures are still above the lower limit specified in the CFPP test method [60]. Trouble-free engine operation throughout the year requires special flow improvers, which, like plant oil methyl esters, should be biodegradable.

- **Ethanol Diesel Fuel (O₂-Diesel)**

O₂-Diesel, also called ethanol diesel in the USA, is a blend of ethanol and conventional diesel fuel. The USA used ethanol as fuel with percentages of ethanol as high as 85% (blended with gasoline) for a long time in gasoline engines [53]. DIN EN 590 outlines O₂-Diesel specifications.

The analysis of diesel fuel and O₂-Diesel used for the experiments in this study was performed at the University of Rostock [4]. ASG analytics service Ltd determined the specific heat value and the cetane number as well as the effects of corrosion on copper. **Table 3.2** presents the results of the analyses.

Table 3.2: Properties of O₂-Diesel (ethanol-diesel blend) [4]

Parameter	Test procedure	Unit	DIN EN 590	DF	O ₂ -Diesel
Heat value	DIN 51 900-2	MJ/kg	approx. 42	43.866	42.360
Cetane number	DIN EN ISO 5165		min. 51	55.5	54.8
Cetane index	DIN EN ISO 5165		min. 46	54.6	52.8
Density in 15°C	EN ISO 12185	g/cm ³	0.820-0.845	0.8276	0.8252
PAK	DIN EN 12916	%(m/m)	max. 11	2.0	1.9
Sulfur content	E DIN 51400-10	mg/kg	max. 10	8	3
Flash point	DIN EN 22719	°C	> 55	56.9	12.9
Coke residue (of 10% distillation residue)	DIN EN ISO 10370	%(m/m)	max. 0.30	0.004	0.08
Water content	EN ISO 12937	mg/kg	max. 200	130	250
Oxide ash	DIN EN ISO 6245	%(m/m)	max. 0.01	0.001	0.004
Total contamination	DIN EN 12662	mg/kg	max. 24	27.3	27.8
Copper corrosion (3h at 50°C)	EN ISO 2160		1	1	1
HFFR Lubricity with 60°C	DIN EN ISO 12165-1	µm	max. 460	317	284
Viscosity in 40°C	DIN 53015	mm ² /s	2.00-4.50	2.71	2.03
Distillation % [V/V] - 250°C	DIN EN ISO 3405	Vol.-%	< 65	41	45
% [V/V] - 250°C		Vol.-%	min. 85	94	95
95% -Point		°C	max. 360	352.9	353.3
Fatty acid methyl ester content	EN 14078	%(V/V)	max. 5	0.17	0.16
CFPP	DIN EN 116	°C	max.0	-8	-7

O₂-Diesel fuel is a mixture that consists of the following components [4]:

Water-free ethanol: 7.7% Vol.

Diesel fuel: 91.6% Vol.

Additive 2EHN:	0.2% Vol.
Additive O2DO5:	0.5% Vol.

The percentage of water in O₂-Diesel should be less than 0.05% Vol (250mg/kg) [4]. In order to offset ethanol's lower cetane number, additive packages for ethanol diesel blend contain cetane improver additives. If the additive package contains sufficient cetane improver, the cetane number of the final ethanol diesel blend can be restored to that of the baseline diesel fuel or, in some cases, be even higher (see **Table 3.2**).

In experiments with the ethanol-diesel blend, the additive 2EHN with 0.2% Vol. was used to enhance the cetane number for a number of reasons including the fact that little water can be introduced into the fuel system because ethanol has an affinity for water (the experimental ethanol diesel blend had a water content of less than 0.5%) and because blend levels of up to 15% are often desired. One way to achieve this is to use additives. Two common types of additives are surfactants and co-solvents. Blends formed with these additives are very stable. Co-solvents have immediate polarity between ethanol and diesel fuel and act as a bridging agent to produce a homogeneous blend.

In addition, surfactants used to ensure blend stability of micro-emulsions can also enhance fuel lubricity. The additives package may comprise anywhere from 0.2% to 0.5% of the blend. Consequently, the additive O2DO5 was used to enhance the solubility of ethanol with diesel fuel. In the USA, the additive Chimec 9432 is used to improve the lubricity of the ethanol and diesel fuel mixture.

- **Gas-to-Liquid (GTL)**

Produced by the Shell Oil Company (SMDS), gas-to-liquid or GTL is one of the new alternative fuels used for diesel engines without any modifications. DIN EN 116 applies to this alternative fuel.

A wide range of properties of GTL diesel fuel were tested, e.g. composition, energy content, cetane number, cold flow properties, density and viscosity. The PAE lab of Shell Global Solutions (Deutschland) GmbH in Hamburg performed the testing of the fuel properties, specifically the properties of summer GTL. **Table 3.3** presents the test results.

SMDS fuel-GTL diesel fuel promises a reduction of emissions because its sulfur and aromatics content is very low. GTL diesel fuel has a higher cetane number than conventional diesel fuel. An increased cetane number has been linked to decreased or unchanging NO_x emissions. A reduced T90 temperature has been demonstrated to have little impact on

emissions [85]. For GTL fuel, any impact based on T90 temperature is likely to be obscured by other fuel properties such as paraffin content and cetane number.

One very important feature of synthetic fuels is their compatibility with existing diesel engines. The only adjustment that may be required is an increase in fuel lubricity in order to prevent excessive wear on the fuel injection system. This can be achieved with commercial lubricity additives. FT fuel can also be doped with biodiesel, which has excellent lubricity properties [85].

Table 3.3: Properties of summer GTL diesel fuel [85]

Properties	Test method	unit	results
Density	EN ISO 12185	g/cm ³	0.7853
Viscosity at 40°C	DIN EN ISO 3104	mm ² /s	3.5871
Cloud point	DIN EN 23015	°C	0
CFPP	DIN EN 116	°C	-3
Sulfur content	DIN 51400 T11	mg/kg	< 5
Heat value	ASTM D240	MJ/kg	44.398
Cetane number			74
Distillation	DIN EN ISO 3405		
Boiling point		°C	202.3
T10		°C	244.7
T50		°C	295.2
T90		°C	340.9
Boiling end point		°C	351.3
Residue		% v/v	2.0
Flash point	ASTM D93	°C	89
Carbon to Hydrogen ratio			2.13
Aromatics	ASTM D5186	mass %	1.4
Gum content	ASTM D381	mg/100 ml	5.9
Ash	ASTM D482	mass%	< 0.001
Carbon residue	ASTM D524	mass %	0.03

3.2.4 Advantages and Disadvantages of Alternative Fuels

• **RME and Blends**

RME and blends have the following advantages and disadvantages:

- The energy content of neat biodiesel fuel is about 11% lower than that of petroleum-based diesel fuel (on a per gallon basis) [30]. This results in power loss in engine operation. The actual percentage power loss varies depending on the percentage of biodiesel blended in the fuel.
- The viscosity range of biodiesel fuel is however higher than that of petroleum-based diesel fuel (1.9-6.0 centistokes as opposed to 1.3-5.8 centistokes) [30]. This tends to reduce barrel/plunger leakage and thus slightly improves injector efficiency.
- Biodiesel's superior lubricating properties can reduce wear in diesel engines. In particular, biodiesel fuel is blended with ultra-low sulfur diesel fuel because it can have poor lubricating properties. Low levels of biodiesel used as a lubricating additive can help solve this problem.
- Biodiesel also has a higher cetane number than most diesel fuel. Cetane measures the tendency of diesel to autoignite in an engine and is comparable to the octane number of gasoline. Fuels with higher cetane numbers have shorter ignition delay periods than lower cetane fuels. Fuels with a cetane number lower than an engine's minimum requirements can cause rough engine operation and more difficult starting, especially in cold weather or at high altitudes. Low cetane fuels may increase engine deposits resulting in more smoke, increased exhaust emissions and greater engine wear.
- Pure biodiesel and higher percentage biodiesel blends can cause a variety of engine performance problems, including filter plugging, injector coking, piston ring sticking and breaking, elastomer seal swelling and hardening/cracking and severe engine lubricant degradation. At low ambient temperatures, biodiesel is thicker than conventional diesel fuel. This would limit its use in certain geographic areas. Moreover, the compatibility of elastomer with biodiesel remains unclear. Therefore, the condition of seals, hoses, gaskets and wire coatings should be monitored regularly when biodiesel fuels are used.
- Available information on the effect of neat biodiesel and biodiesel blends on engine durability during various environmental conditions is limited.
- Pure biodiesel and biodiesel blends reduce visible smoke, particulates and HC and CO emissions and increase NO_x emissions compared with petroleum-based diesel fuel used in an unmodified engine. Neither B100 nor biodiesel blends should be used as a means to improve air quality in ozone non-attainment areas.
- Biodiesel fuels have exhibited poor oxidation stability. This can result in long-term storage problems. When biodiesel fuels are used at low ambient temperatures, filters may plug and the fuel in the tank may thicken to the point that it will not flow sufficiently for proper engine operation. Therefore, it may be prudent to store biodiesel fuel in a heated

building or storage tanks. Additives may also be needed to improve storage conditions and enable using biodiesel fuel in a wider range of ambient temperatures. A technical statement issued by the Engine Manufacturers Association (EMA) stipulated that biodiesel fuels should have a minimum of 80% reflectance after aging for 180 minutes at a temperature of 150 °C.

- Biodiesel is an excellent medium for microbial growth [47]. Inasmuch as water accelerates microbial growth and is naturally more prevalent in biodiesel fuels than in petroleum-based diesel fuels, care must be taken to remove water from fuel tanks. The effectiveness of using conventional anti-microbial additives in biodiesel is unknown. The presence of microbes may cause operational problems, fuel system corrosion, premature filter plugging and sediment build-up in fuel systems.
- Pure biodiesel fuels have been tested and found to be nontoxic in animal studies. Emissions from engines using biodiesel fuels have no biologically significant short term effects on test animals other than minor effects on lung tissue at high levels of exposure. Biodiesel fuels are biodegradable. This may advance their use in applications where biodegradability is desirable (e.g. marine or farm applications).
- The cost of biodiesel fuels varies depending on the base stock, geographic region, seasonal variability in crop production and other factors. A report released by the “Union zur Förderung von Öl und Proteinpflanzen e.V.” (UFOP) 2005, the year end price of diesel at the fuel supplier was 83.08 cents/liter including petroleum tax as compared with 65.78 cents/liter for biodiesel without tax [47].

- **Ethanol Diesel Fuel (O₂-Diesel)**

Ethanol diesel fuel consists of diesel fuel, ethanol and additives. It too has advantages and disadvantages.

- A special advantage of bioethanol is its considerable anti-knocking properties (high octane number). It permits a higher compression ratio in gasoline engines.
- Lower emissions of PM and CO for example without engine modifications are another advantage of using O₂-Diesel.
- The cost of bioethanol is commercially competitive [52, 54]. It is a clear stable liquid with the same clouding point as diesel fuel.
- Ethanol comes from renewable raw materials, thus providing farmers an additional source of income.
- The lower viscosity of ethanol diesel can increase pump and injector leakage resulting in reduced maximum fuel delivery and lower peak power. Hot restart problems may be encountered as well if insufficient fuel is injected at cranking speed.
- Based on flammability limits and vapor pressure data for ethanol, flammable mixtures of ethanol and air in a closed container such as a fuel tank can easily form at temperatures of approximately 10 °C – 40 °C (the flammable temperature range at equilibrium in a closed

container for diesel is approximately 64 °C – 150 °C) [48, 52]. With O₂-Diesel, ignition could easily propagate down a fill neck and into the fuel tank at typical ambient temperatures, causing the fuel tank to fail catastrophically. Low flash point and vehicle tank vapor flammability have been identified as the most important technical barriers to the commercialization of O₂-Diesel. That is why flame arrestors are critical safety devices for the fill necks of fuel tanks for O₂-Diesel.

- Neat ethanol has a low cetane number. Estimates place it between 5 and 15. It is 8 of the part of ethanol in the O₂-Diesel in this study [53]. Blending ethanol into diesel fuel reduces the cetane number because ethanol is a short chain molecule that is more resistant to free-radical scission than longer molecules, alcohol functional groups tend to generate lower cetane numbers and ethanol has a high enthalpy of vaporization with associated evaporative cooling that tends to cool the entire fuel charge during injection. The low cetane number results in long ignition delays and unacceptable deterioration of combustion quality.
- The literature also confirms that ethanol diesel blends formed without additives have very low tolerance of water [52, 54]. Even small amounts of water can cause phase separation. In addition, lubricity and wear are concerns with ethanol diesel and additives are critical to ensure proper lubricity.
- Since ethanol has a very low freezing temperature it is expected to improve cold-flow properties when blended with diesel fuel. While very significant depressions in pour point have been observed with e-diesel blends, the cloud point can actually increase significantly. It is believed that upon cooling, the micelles in e-diesel micro-emulsions grow in size to the point that they become visible and make the fuel cloudy. These ethanol micelles are liquid and apparently flow through a fuel filter. This contrasts with the cloud point of a conventional diesel, which indicates the onset of the formation of solid wax crystals that can plug a fuel filter.

- **Gas-to-Liquid (GTL)**

GTL is product synthesized from natural gas. Since a number of FT catalyst and reactor technologies exist, the properties of GTL diesel fuel vary. The properties of GTL fuel can be also designed to suit a particular application by selecting the parameters of the production process. Generally, GTL diesel fuel has the following qualitative advantages over petrodiesel:

- Ignition characteristics are very good and cetane numbers may reach 75 or higher (with experimental SMDS, the cetane number was as high as 74) [85]. GTL fuel is almost entirely composed of paraffins and n-paraffins, which are known to have very high cetane number.
- It has no or a very low sulfur content. Thus, reductions of PM can be expected. In engines with newer technology, GTL fuel's sulfur content of nearly zero may prove more beneficial by making sulfur-sensitive emission control devices possible.

- It has very low aromatics content and a higher H/C ratio (2.1, i.e. around 16% greater than conventional diesel fuel) [85]. These advantages promise a reduction of emissions.
- GTL fuels have reduced densities compared with conventional diesel fuel. This has been shown to reduce the PM emission in engines with older technology [85]. In addition, lower density fuels such as GTL may alter fuel mass flow rates.
- It is colorless and virtually odorless.
- Its heat value is similar to that of petrodiesel.
- Synthetic fuel-GTL diesel fuel is compatible with existing engines, making engine modifications unnecessary. GTL is compatible with conventional diesel fuel because, apart from its comparable heat value, it can be mixed with petroleum diesel fuel and transported as a liquid in the existing infrastructure.
- Its cold-flow characteristics meet all relevant specifications, even the stringent freezing point requirements for aviation turbine fuel. SMDS products are excellent blending components for upgrading refinery fractions that would otherwise only be used in fuel oil.
- Other properties such as gum, ash, water and sediment are in line with other diesel fuels and are not expected to impact operations [86].
- Typically, their lack of polar molecules causes unadditized GTL diesel fuels to have poor lubricities but they respond well to additives.

4 Experimental Methodology

4.1 Experimental Test Bench

4.1.1 Experimental Engine

Research was carried out on a DEUTZ BF6M 1013 EC heavy-duty engine applying the main technological parameters presented in **Table 4.1**. The DEUTZ BF6M 1013 EC is a four-stroke direct injection engine with six in-line cylinders and turbocharger and are retrofitted with exhaust gas regulations (EURO II).

Table 4.1: Features of the experimental engine

Engine manufacturer	DEUTZ AG
Identification	DEUTZ BF6M 1013 EC
Type	6 cylinders in-line
Valve per cylinder	2
Stroke number	4-stroke
Features	DI; turbocharged with waste-gate-valve and charge cooler
Injection system	Unit-pump; one pump per cylinder; magnetic valve control
Nominal power	148 kW at 2300 1/min
Maximum Torque	852 Nm at 1400 1/min
Exhaust gas emission standard	EURO II
Bore/stroke	108/130 mm
Displacement	7.14 l
Compression ratio	17.6
Engine controlling device	Bosch-serial control device EDC 15 04199551
Application control device	ETAS ETK 3.1; 5007

The software INCA and an ETAS ETK 3.1 control unit were used to change controlling parameters such as injection timing.

4.1.2 Test Bench Schematic

A schematic of the engine test bench is presented in *Figure A.1* in the *Appendix*. A Schenk W400 eddy current brake produces the engine load. The brake and fuel cooling systems operate with external water. Given its low boiling temperature, the use of O₂-Diesel fuel requires a cooling system for a portion of the unused O₂-Diesel fuel (returning from the engine). Unused O₂-Diesel fuel recirculated to the tank becomes too hot and can begin to boil while passing through an engine's fuel system. Overheated O₂-Diesel substantially influences an engine's power losses and fuel consumption. This problem is solved by retrofitting a small cooler to reduce the temperature of the fuel before it reaches the injectors. The fuel temperature of O₂-Diesel fuel should constantly be maintained at approximately 30 °C under all operating conditions. Such a fuel cooling system is not needed for other fuels because they have high boiling temperatures.

Ethanol's flash point is around 13°C, while diesel fuel's is much higher. Diesel fuel blends with up to 20% ethanol have nearly the same flash points as neat ethanol [4, 35]. When ethanol is added to diesel (E-Diesel), storage requirements must be more stringent and include such measures as locating storage tanks farther away from property lines, buildings, other tanks and vent terminals and installing flame arrestors on all vents. In short, e-diesel must be stored and handled like gasoline. According to flammability limits and vapor pressure data for ethanol, flammable mixtures of ethanol and air can easily form in a closed container such as a fuel tank at temperatures of approximately 10 °C-40 °C. With e-diesel, ignition could easily propagate down a fill neck and into the fuel tank at typical ambient temperatures, causing the fuel tank to catastrophically fail. Thus, the fuel system needs a flame safety valve (see *Figure A.1* in *Appendix*). An air mass meter is installed to measure the air mass aspirated into the engine in order to calculate the EGR ratio. The temperature and pressure of the fresh air are measured after filtering. External compression air controls the EGR valve. A throttle valve is also used to increase the EGR ratio in lower part load (1100 1/min-213 Nm).

4.2 Measuring Equipment and Method of Evaluation

4.2.1 Emissions and Exhaust Gas Emission Devices

4.2.1.1 Emissions

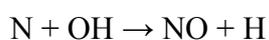
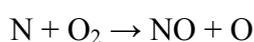
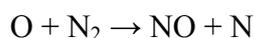
The combustion of fuels with air containing 21% O₂ by volume, less than 1% inert gases by volume and N₂ releases energy as heat in an exothermic reaction. The heat released in internal combustion engine by hydrocarbon-based fuels such as gasoline and diesel fuel is determined by numerous incomplete reactions dependant on the compositions of the hydrocarbons in the fuel. Important fuel components are paraffins, olefins and aromates. Besides the main

components of such as CO₂, water vapor and N₂ we can find also carbon monoxide, unburned and partly burned hydrocarbons (HC-aldehydes, ketones, etc.) and nitrogen oxides (NO_x) as pollutants in exhaust gas of an IC engine. These pollutants are limited by the law. Pollutants are primarily produced by an interruption of the reaction chain when its dwell time in the combustion chamber is short. Hence, the equilibrium no longer exists. Inhomogeneities in the mixture caused by different air-fuel ratios (λ), combustion chamber wall effects and impurities and additives in the fuel also contribute to undesired byproducts. Depending on the type of the fuel and combustion process, solids may also be present as particle emissions. Unrestricted exhaust components produced by the thermal cracking of hydrocarbons and its byproducts are increasingly attracting attention since they are either potentially hazardous or cause a discernible odor.

On the other hand, the concentrations of pollutants from internal combustion engines differ from values calculated assuming chemical equilibrium. The detailed chemical mechanisms by which pollutants form and the kinetics of these processes are important when determining emission levels. The formation and destruction reactions of some pollutants, e.g. carbon monoxide, organic compounds and particulates, are intimately coupled with the primary fuel combustion process. Understanding the formation of these pollutants requires knowledge of the combustion chemistry. The formation and the destruction processes of nitrogen oxides are not part of the fuel combustion process. However, the reactions which produce these pollutants occur in an environment created by the combustion reaction. Thus, the two processes are intimately linked. The following summarizes the mechanisms by which these pollutants form as well as their effects on humans and the environment.

- **Nitrogen Oxides (NO_x)**

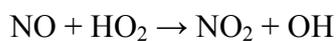
Unlike CO and HC, NO_x, which consists of nitrogen monoxide (NO) and nitrogen dioxide (NO₂), is a product of complete combustion. Nitrogen monoxide is the predominant nitrogen oxide produced inside an engine cylinder. The principal source of NO is the oxidation of atmospheric (molecular) nitrogen. However, if the fuel contains a significant amount of nitrogen, the oxidation of the fuel compounds containing nitrogen is an additional source of NO. It is generally accepted that the following are the principal reactions in the combustion of near stoichiometric fuel-air mixtures. Often called the extended Zeldovich mechanism, these reactions govern the formation (and destruction) of NO from molecular nitrogen:



NO forms in both the flame front and postflame gases. However, since combustion in engines occurs at high pressure, the flame reaction zone is extremely narrow ($\approx 0.1\text{mm}$) and the

residence time within the zone is brief. Also, due to the cylinder pressures that increase during the combustion process the gas are compressed to a temperature higher than immediately after combustion. Thus, NO formation in the postflame gases almost always dominates any formation of NO in the flame-front. Furthermore, the NO formation rate is a function of the gas temperature and fuel-air equivalence ratio in postflame gases. In an engine, a noteworthy NO level can first be detected above around 1900 °C [90]. Overall, more than 90% of NO_x emissions are nitrogen monoxide. Fuel nitrogen is also a source of NO by means of a different and not yet fully explained mechanism. The nitrogen in distillate fuels can exist as amines and ring compounds (e.g. pyridines, quinolines and carbazoles). During combustion, these compounds are likely to undergo some thermal decomposition prior to entering the combustion zone. Therefore, the precursors of NO formation are nitrogen-containing compounds with low molecular weights such as NH₃, HCN and CN. Detailed information on the kinetics of NO formation from these compounds is limited. A characteristic of NO is its reactivity with oxygen and particularly quick reactivity with ozone to form NO₂. Pure NO does not irritate the lungs but if NO₂ remains unconverted, methemoglobin forms after resorbed by the respiratory tract. NO is an endogenous modulator of blood vessel tone and thus a well studied substance in terms of physiology and metabolism [91].

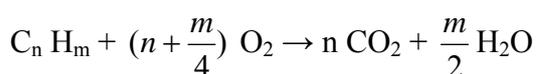
NO₂ has a pungent odor and a reddish brown color. A plausible mechanism for the persistence of NO₂ is:



Chemical equilibrium considerations indicate that NO₂/NO ratios should be negligibly small and NO₂ can be 10% of total exhaust of nitrogen oxides emissions [1, 2]. As a free radical, NO₂ is basically in a position to abstract hydrogen from fatty acids and thus to cause lipid peroxidation, which ultimately leads to a loss of function in biological membranes. In the presence of water, NO₂ develops as a cellular poison in the respiratory tract, through the nitric acid (HNO₃).

- **Hydrocarbon (HC)**

Ideally, an internal combustion engine's combustion process corresponds to the following reaction [69]:



In practice, the combustion process does not produce carbon dioxide and water alone. Thus, combustion in an internal combustion engine is usually incomplete process. As a result, incomplete combustion of the hydrocarbon-based fuel results in hydrocarbons or more organic emissions. The level of unburned hydrocarbon (HC) in the exhaust gases is generally

specified in term of the total hydrocarbon concentration expressed in parts per million. Engine exhaust gases contain a wide variety of hydrocarbon compounds. Diesel fuel contains hydrocarbon compounds with higher boiling points and hence higher molecular weights than gasoline [1, 2, 55]. Also, substantial pyrolysis of fuel compounds occurs within the fuel sprays during the diesel combustion process. Thus, the composition of the unburned and partly oxidized hydrocarbons in the diesel exhaust is much more complex than in the spark-ignition engine and extends over a larger molecular size range. HC constituents range from methane to the heaviest hydrocarbons that remain in the vapor phase in the heated sampling line (which is usually maintained at about 190 °C) [2]. Levels of hydrocarbon emission from diesel engines vary widely depending on operating conditions.

So far it has not been easy to assess the direct impact of hydrocarbon emissions on humans. Hydrocarbon emissions are sometimes slightly relevant toxicologically (alkanes or alkenes etc.) and can also be carcinogenic (benzene) [69]. Hydrocarbons from engine emissions contribute to the development of summer smog.

- **Soot**

Soot particles form primarily from the carbon in the diesel fuel. The formation process begins with a fuel molecule containing 12 to 22 carbon atoms and having an H/C ratio of about 2 and ends with particles that are typically a few hundred nanometers in diameter, composed of spherules with 20 to 30 nm in diameters, each of which contains some 10^5 carbon atoms and has an H/C ratio of about 0.1. Soot forms in the diesel combustion environment at temperatures between around 1000 and 2800K, at pressures of 50 to 100 bar and with sufficient air to fully burn all the fuel [2]. Solid soot particles form from a fraction of the fuel in milliseconds. Soot formation has two stages. In the first stage, particles form. The initial condensed phase material is produced by the fuel molecules by means of oxidation and pyrolysis products. These products typically include various unsaturated hydrocarbons, particularly acetylene's higher analogues ($C_{2n}H_2$) and polycyclic aromatic hydrocarbons (PAH). These two types of molecules are considered the most likely precursors of soot in flames. In the second stage, particles grow. This involves surface growth, coagulation and aggregation. Surface growth, which generates the bulk of the solid-phase material, involves the attachment of gas-phase species to the surface of the particles and their incorporation into the particulate phase [2]. Surface growth reactions lead to an increase in the amount of soot but the number of particles remains unchanged. These stages of particle generation and growth constitute the soot formation process. In each stage, oxidation can occur, during which soot or soot precursors are burned with oxidizing species to form gaseous products such as CO and CO₂. The real emission of soot from the engine will depend on the balance between these processes of formation and burnout [1, 2].

- **Particulate Matter (PM)**

Diesel particulates primarily consist of carbonaceous material (soot) generated in combustion, which have absorbed some organic compounds [1, 2]. Most particulate material results from incomplete combustion of fuel hydrocarbons, but also from the lubricating oil. Other sources are hydrocarbon compounds (some of which are bound to soot) and a few sulfates in the form of aerosols. In a combustion engine, the development of particles is related to the development of soot, which in turn basically depends on the local temperature of the zone the fuel enters and a low oxygen supply. According to the Acetylene Theory (*Klingenberg et al.*, 1992) [92], the combustion of different hydrocarbons runs through several intermediate substages such as cracking and dehydration. This increases the share of carbon molecules until the first particle with a diameter larger than 0.01 μm develops, around which so-called primary particles coagulate to form larger units with diameters from 0.01 to 0.08 μm (forming secondary particles). Their large specific surfaces enable uncombusted and partly combusted hydrocarbons, especially aldehydes, to bond to the secondary particles. When combustion occurs, the secondary formation phase is soot reoxidation governed by the dwell time and oxygen concentration. A specific problem of diesel engines is the conflict between particles and NO_x . The conditions for low particle formation and low HC emissions (achieved with higher temperatures) conflict with the preconditions for low nitrogen oxide emissions. Therefore, attention is concentrated on the secondary formation phase of soot reoxidation. Soot reoxidation necessitates a large amount of mixture formation energy in the last phase of combustion. This can be attained by a specific swirl and tumble in the combustion chamber, higher injection pressure and a faster injection rate at the end of the injection process. Unfortunately, these conditions are prerequisites to high NO_x emissions.

Numerous environmental studies have concluded that increases of particle concentration in the air correlate with increases in the number of patients with respiratory and circulatory illnesses (*Dockery and Pope*, 1994; *Samet et al.*, 1995; *Katsouyanni et al.*, 1997) [93, 94, 95]. A study published in 1999 found a link between the risk of lung cancer and particles as well as ozone and sulfur dioxide (*Abbey et al.*, 1999) [96]. In general, particle emissions have a negative impact on human health, especially particularly sensitive individuals, e.g. children, elderly and invalids.

- **Carbon Monoxide (CO)**

Carbon monoxide is likewise a product of an incomplete combustion process. CO emissions from internal combustion engines are primarily controlled by the air fuel ratio. CO concentrations in the exhaust of fuel-rich mixtures steadily increase as the air fuel ratio decreases, i.e. as the amount of excess fuel increases. Since the fuel-air mixture must be

enriched when an engine is cold, CO emissions during engine warm-up are much higher than emissions in the fully warmed-up state.

Carbon monoxide is a colorless and odorless gas and attaches itself to hemoglobin 250 times more strongly than oxygen. As CO concentrations increase in inhaled air, the suppression of oxygen induces symptoms from suffocation to death. Acute poisoning occurs above 2000 ppm. All in all, CO emissions from internal combustion diesel engines are insignificant.

• **Unregulated Exhaust Gas Components**

Important unregulated exhaust gas components in untreated exhaust from diesel engines are cyanide, ammonia (NH₃), sulfur dioxide (SO₂) and sulfates. Of the specific hydrocarbons, methane, ethane, ethene, ethine, benzene and toluene are of particular interest. Of the PAH, phenanthrene, pyrene, fluorene, fluoranthene and anthracene predominate and in this order. The concentration of these components is at least six times higher than that of other individual PAH substances and forms about 90% of the PAH [1]. The components cited are formed from trace substances in the fuel, in the lubricant and, to a degree, from secondary reactions in the exhaust system.

Combustion exhaust gases contain a multiplicity of different toxic elements. The bulk of the over 150 PAH substances identified are mutagenic and carcinogenic. Many studies have been carried out on the influences of diesel engine emissions on the health of humans and the environment. In 1978, an Ames Test (Salmonella Microsome Test) performed by *Huisinigh et al.* [97] demonstrated for the first time the capability of diesel engine emissions to cause genetic damage. Since then, many further studies (*Clark and Vigil*, 1980; *Claxton and Barnes*, 1981; etc.) have confirmed these findings [98, 99].

4.2.1.2 Exhaust Gas Emission Measuring Device

A CEB II emission analysis system (AVL) was used to determine emissions components. The CEB II (Combustion Emission Bench) measures most exhaust gas components, e.g. NO, NO_x, CO₂, CO, O₂ and total HC. A gas sampling probe extracts the exhaust gas from the exhaust gas line. The exhaust gas measuring system is normally connected to a test cell host computer, which controls the measuring system according to the measuring jobs, saves the measured values from analyzers and performs any calculations.

Correct measurement of the exhaust gas components necessitates calibrating the analyzers. This is done before experiments are performed. **Table 4.2** show the different analysers of CEB II.

Table 4.2: Emission measuring device [32]

Gas component	Measuring equipment	Measuring range
THC	Multi-FID 14	10-10000ppm
NO, NO _x	CLD EcoPhysics 700 RH ht GY 0544	50-10000ppm
O ₂	Oxygen analyzer module Magnos 16	0-25%Vol
CO ₂	NDIR MLT1 IR1	0-20%Vol
CO	NDIR MLT1 IR50	50-2500ppm

4.2.2 Smoke Meter

The soot content in the exhaust gas of a combustion engine emitting particles is determined by a variably sampling smoke meter AVL 415 and based on the filter paper method. A portion of hot exhaust gas sucked from the exhaust gas line flows into the device where the filter paper is blackened by the soot component in exhaust gas. An optical measuring head defines the blackness of the filter paper. In addition, the quantity of exhaust gas sucked through the filter paper is also defined and the effective length is determined. The aforementioned parameters indicate the filter smoke number (FSN).

4.2.3 AVL 670 Indimaster Thermodynamics Analyzer

The INDIMASTER AVL 670 is employed for thermodynamic tests of gasoline and diesel engines. Both the 670 Indimaster and the 660 Indistation share the same base system, the 6704. The 670 system permits connecting one or more data acquisition units to the workstation with a special bus link. Signals from a cylinder pressure piezo-sensor installed in the third cylinder head and a crank angle sensor installed in the flywheel detect changes of cylinder pressure and crank angle, respectively. The measurement parameters, e.g. engine parameters, TDC values, channel allocation through signal name input, etc. have to be set before any measurement is taken. When all the necessary settings are stored in a parameter file, this file can be downloaded before a measurement series is started. Data can also easily be saved on hard or floppy disks and transferred to the host computer. Stored data can be reloaded into the system at any time. Thermodynamic offset must be corrected in order to obtain correct readings from cylinder pressure signals. The method of offset is based on a polytropic compression between crank angles X_1 and X_2 and a constant polytropic coefficient

(m). Input is required in the two fields X_1 and X_2 beneath the calibration table and in the field with the polytropic coefficient. The equation:

$$p_1 \cdot V_1^m = p_2 \cdot V_2^m \quad (4.1)$$

can be expressed as follows:

$$p_{\text{abs}} = \frac{\Delta_p}{\left(\frac{V_1}{V_2}\right)^m - 1} \quad (4.2)$$

Thus the absolute value can be derived from the pressure differential between points (1) and (2). The user can redefine the Angles X_1 and X_2 and the polytropic coefficient (m) [58].

In addition, the signal from the injection pressure sensor installed in the injection pipe between the injection pump and the injector detects the behavior of injection pressure in relation to the crank angle measured by the Indimaster 670.

4.2.4 Cylinder Pressure Sensor and Charge Amplifier

In this study, a cylinder pressure sensor installed in the third cylinder head and an amplifier, manufactured by AVL, were used to measure the behavior of cylinder pressure in relation to the crank angle.

The GU 21 D cylinder pressure sensor is an uncooled piezoelectric sensor. This sensor functions in high temperatures. The sensor's measuring elements are primarily made of gallium orthophosphate (GaPO_4). This material gives the sensor the following advantages [82]:

- Temperature resistant up to over 900 °C
- Double sensitivity, compared with quartz-based sensors
- Stable sensitivity to over 500 °C
- High insulation resistance
- No pyroelectric effect

Thus, the sensor can be used to measure cylinder pressure without cooling. The membrane's elasticity enables detecting signs of changes in cylinder pressure. Voltage is proportional to pressure that acts on the membrane. As a result, changes of cylinder pressure are indicated. Important technical properties of the sensor are presented in **Table 4.3**.

Table 4.3: Technical properties of the GU 21 D sensor [82]

Measuring Range	0...250 bar
Lifetime	> 10 ⁸ load changes
Overload	300 bar
Sensitivity (normal)	38 pC/bar
Linearity	< ± 0.3 (0.1)* %FSO
Natural Frequency	85 kHz
Acceleration Sensitivity	< 0.0002 bar/g
Shock Resistance	> 2000 g
Operating Temperature Range	Up to 400°C
Thermal Sensitivity Shift	20...400°C < ± 2% 200...300°C < ± 0.5%
Insulation Resistance at 20°C	> 10 ¹³ Ω
Capacitance	8 pF
Mass (without cable)	6 g
Mounting Torque	3 Nm
Sensitivity	20 bar/V

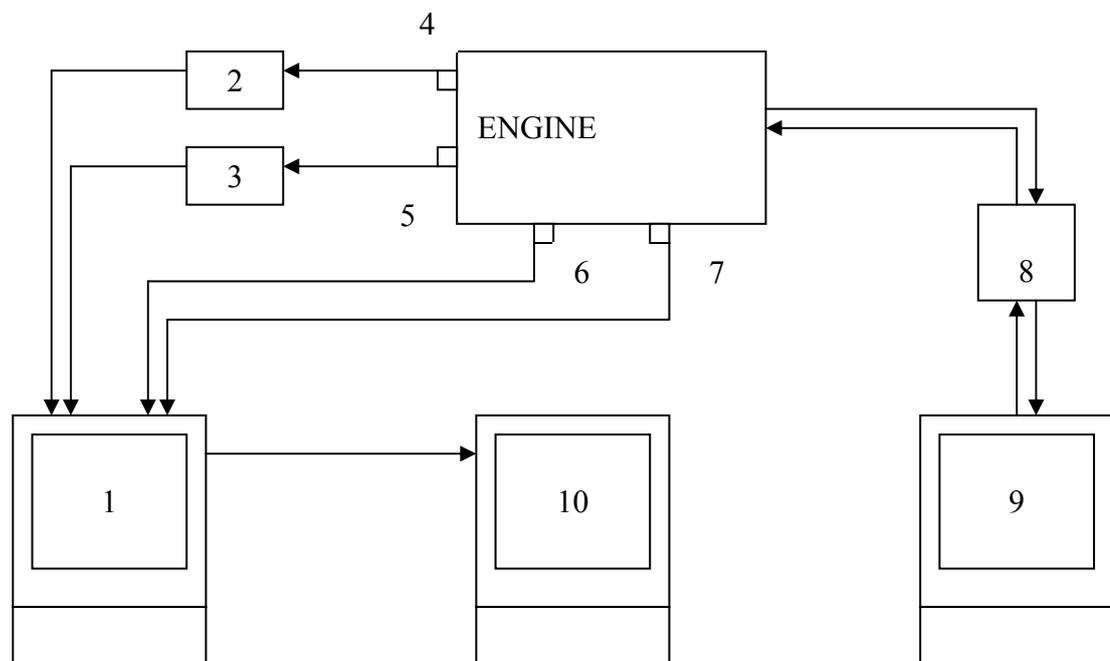
The GU 21 D cylinder pressure sensor was integrated with an AVL 3057 charge amplifier. Before the cylinder pressure sensor is installed into the head of an engine, it is calibrated by balancing the pressure.

Top dead center and crank angle signals are required for an analysis of cylinder pressure signals. The TDC sensor and crank angle sensor were used (see **Figure 4.1**).

The TDC mark and crank angle sensor operate according to the principle of optical measurement. By using light emitting and receiving diodes and a mark disk which has 360 slits and one hole for trigger, signals of crank angle and Top Dead Center (TDC) of the first cylinder will be sent to Indimaster (see **Figure 4.1**). Crankshaft position is an important factor for the analysis of cylinder pressure. The angular resolution is one to one degree of crank angle. Thus, 360 symmetrical square-wave impulses are received per crankshaft revolution.

One important task is to define TDC before experiments are performed. Since the measurement table inputs and the Indimaster data outputs are both related to TDC, a TDC value must be input for each specified cylinder. This value is the differential angle between the trigger mark of the crank angle marker and the top dead center of the relevant cylinder. There are many methods for determining TDC value, e.g. using a stroboscope, calculating the symmetric maximum of the compression pressure curve or using the TDC mark sensor.

In this study, controlling signal of unit pump of fuel injection system in third cylinder is sampled to calculate top dead center position from the symmetry of the compression pressure curve around the peak pressure. The TDC value calculated with the aforementioned method can be compared with the TDC value determined by the TDC mark. The thermodynamic loss angle is added to the TDC value determined.



- 1: Indimaster AVL 670
- 2: Amplifier for cylinder pressure sensor
- 3: Amplifier for injection pressure pipe sensor
- 4: Cylinder pressure sensor
- 5: Injection pipe pressure sensor

- 6: TDC mark sensor
- 7: Crank angle sensor
- 8: ECU or application ECU
- 9: Computer
- 10: Computer

Figure 4.1: Layout of the thermodynamics measuring system

4.2.5 Injection Pipe Pressure Sensor and Charge Amplifier

Injection pipe pressure was determined with a SL31D-2000 high pressure sensor (AVL). It was installed in the injection pipe between the high pressure pump and the injector of the third cylinder. The pressure sensor's design is based on the strain gauge bridge principle. The technical parameters of the SL31D-2000 sensor are presented in *Table 4.4*.

Table 4.4: Technical parameters of the injection pipe pressure sensor [81]

Measuring range	0...2000 bar
Lifetime	$> 10^8$ load changes
Overload	4000 bar
Burst pressure	6000 bar
Sensitivity (at 24°C)	$4.5 \cdot 10^{-4}$ mV/V. bar
Linearity	$\pm 0.5\%$
Natural frequency	> 100 kHz
Operating temperature range	20...120°C
Thermal sensitivity shift	$< \pm 1\%$ at 20...120°C
Thermal zero point shift	$< \pm 1\%$ at 20...120°C
Excitation voltage	4 V
Measuring element	Full bridge 1000 Ω
Mass (without cable)	19 g
Mounting torque	15 Nm

This strain gauge sensor is designed to take precise dynamic measurements in injection lines and hydraulic systems. It is suitable for pressures up to 2000 bar and can easily be mounted in injection lines with a solder-on adapter with a very small dead volume. The SL31D-2000 sensor works with any strain gauge amplifier with sense line support. In this study, the DMS amplifier AVL 3009A02 was used. The DMS amplifier 3009A02 amplifies very weak signals. The amplifier has two channels that can be operated independently. Its maximum range of 100 kHz is suitable for very rapid pressure developments in injection lines. The injection pipe pressure sensor is also calibrated by balancing the pressure.

4.2.6 Main ECU and Application ECU

The DEUTZ 1013 EC engine has a BOSCH electric control unit. When speeds, intake flow rate, temperature of coolant, etc., change, main ECU will adjust operating parameters of engine to be suitable for changes of operating conditions.

An engine electric control unit, ETAS ETK 3.1, was connected to a PC and the application tool, INCA, was used to change injection timing (B.O.E).

4.2.7 Confidence Interval for Average Value

The confidence interval for the average values in the table of results is the range of the true average value of the measured values. It can be calculated with the following equation:

$$V = \pm \frac{t \cdot s}{\sqrt{n}} \quad (4.6)$$

t: factor for P= 95%

s: Standard deviation of sample

n: Number of samples

The confidence interval is an important value for the evaluation of measurement readings.

4.2.8 Data Processing Software (Concerto Software)

CONCERTO version 3.6 (AVL) was the software used to process the experimental data. This software calculates the changes of heat release rate based on the developments of cylinder pressure. A simplified process is used to calculate the heat release by computing the energy the cylinder pressure effectively delivers to the gas. Surface losses (approximately 20%) are disregarded. Therefore, the heat release calculated is lower than the actual energy released. The calculation is based on the first law of thermodynamics. The heat release rate is calculated using the change of cylinder pressure in the following equation [59]:

$$Q_i = \frac{K}{m-1} [m \cdot p_i \cdot (V_{i+n} - V_{i-n}) + V_i \cdot (p_{i+n} - p_{i-n})] \quad (4.7)$$

n: Interval (1 deg.CA)

m: Polytropic exponent

p: Cylinder pressure

V: Volume

K: Constant

It is also used to calculate other thermodynamics parameters, e.g. combustion start, ignition delay period and maximum cylinder pressure rise. One of important tasks in this study was to calculate the spectrum of cylinder pressure. This is evaluated based on the cylinder pressure levels, the noise of engine combustion in different operating points and the operating conditions. Cylinder pressure levels are calculated with the Fast Fourier Transformation (FFT) and the following equation:

$$L_p = 20 \cdot \lg \frac{p}{p_{\text{ref}}} \text{ (dB)} \quad (4.9)$$

with $p_{\text{ref}} = 2 \cdot 10^{-10}$ (bar)

4.3 Test Program

4.3.1 ESC Test Cycle and Stationary Operating Points for Experiments

Starting in 2000, the European Stationary Cycle (ESC) test cycle (also known as the OICA/ACEA cycle) and the European Transient Cycle (ETC) and European Load Response (ELR) tests were introduced in Europe to certify heavy-duty diesel engines for their emissions [80]. The ESC is a 13-mode, steady-state procedure that replaces the R-49 test. An engine is tested on an engine dynamometer over a sequence of 13 steady-state modes. The specified speed is to be held to within ± 50 rpm and the specified torque to within $\pm 2\%$ of the maximum torque at test speed.

Based on ESC test modes (see *Table 4.5*) and this study's research objectives, which did not include emission certification, experiments were performed with five operating points: maximum torque, nominal power, two part load points and low idle. Along with the two part load points and low idle (Mode – 3, Mode – 9 and Mode - 1) chosen for experiments because of very high weighting factors (see *Table 4.5*), nominal power and maximum torque were also subjects of experiments. This required determining the B speed (experimental speed).

Speeds had to be calculated in experimental operating points based on the ESC test cycle directive. Experimental engine speeds were determined as follows [80]:

- The high speed n_{hi} was determined by calculating 70% of the declared maximum net power. The highest engine speed at which this power value appears on the power curve is defined as n_{hi} .
- The low speed n_{lo} was determined by calculating 50% of the declared maximum net power. The lowest engine speed at which this power value appears on the power curve is defined as n_{lo} .

The ESC test involves high average load factors and very high exhaust gas temperatures. The experimental speed selected was calculated with the following equation [80]:

$$n_{\text{experiment}} = B = n_{lo} + 0.5 \cdot (n_{hi} - n_{lo}) \quad (4.11)$$

The experimental results of a full load curve were:

$$n_{hi} = 1197 \text{ (1/min) at } P_{hi} = 0.7 \cdot 148 = 103.6 \text{ (kW)}$$

$$n_{lo} = 967 \text{ (1/min) at } P_{lo} = 0.5 \cdot 148 = 74 \text{ (kW)}$$

$$n_{\text{experiment}} = B = 967 + 0.5 \cdot (1197 - 967) = 1082 \text{ (1/min)}$$

Table 4.5: ESC test modes [74]

Mode	Engine Speed	% Load	Weight Factor %	Duration (minutes)
1	Low Idle	0	15	4
2	A	100	8	2
3	B	50	10	2
4	B	75	10	2
5	A	50	5	2
6	A	75	5	2
7	A	25	5	2
8	B	100	9	2
9	B	25	10	2
10	C	100	8	2
11	C	25	5	2
12	C	75	5	2
13	C	50	5	2

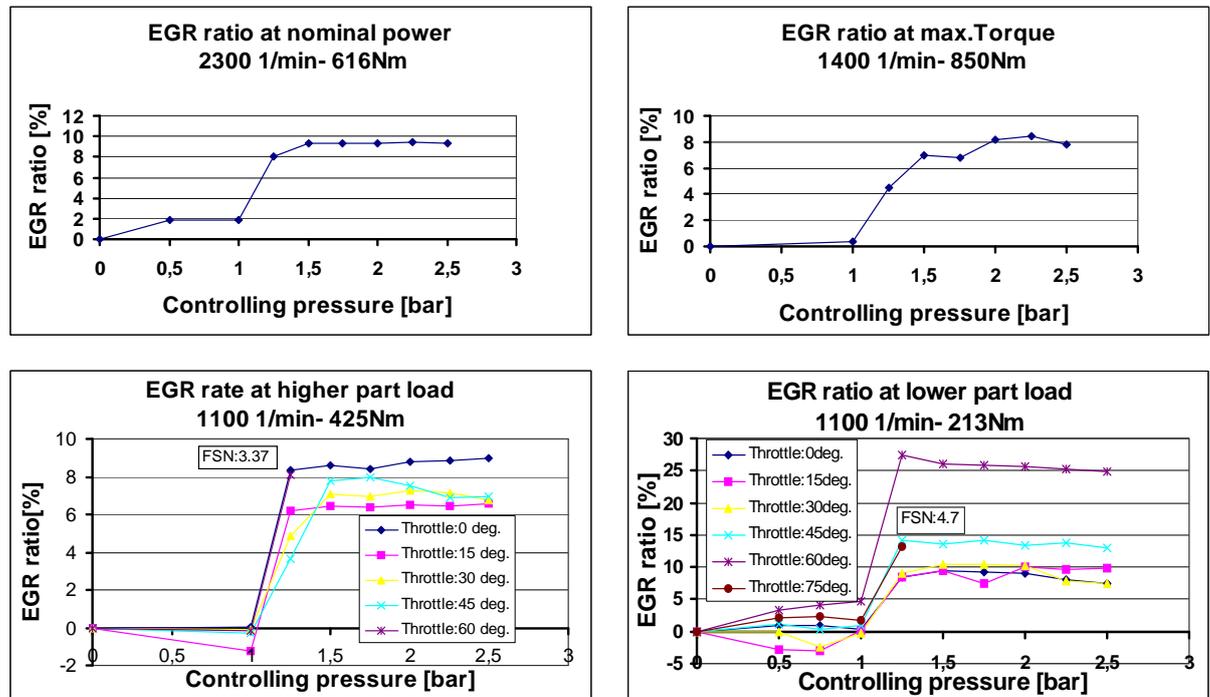
According to the ESC test cycle directive, speed $n = 1100$ 1/min is the experimental speed for two part load points. The five operating points for experiment are listed in **Table 4.6**. They are nominal power (**A**), maximum torque (**B**), higher part load (**C**), lower part load (**D**) and Idle. Of these five operating points, only four operating points, i.e. maximum torque, two part load operating points and a low idle point, were selected for experiments under all the operating conditions, i.e. EGR rates, temperatures and alternative fuels. The nominal power operating point was only tested with conventional diesel fuel.

Table 4.6: Operating points for experiments

WORKING POINTS	POWER (kW)	TORQUE (Nm)	SPEED (rpm)
Nominal Power (A)	148	616	2300
Max. Torque (B)	125	850	1400
Higher Part Load (Mode 4) (C)	49	425	1100
Lower Part Load (Mode 3) (D)	24.5	213	1100
Idle (Mode 1)	–	–	650

4.3.2 Adjustments of EGR Ratios for Experiments

First, experiments were conducted to determine the maximum exhaust gas recirculation ratios. The EGR rates were varied by changing the controlling pressure of external compression air, which controls the EGR valve's stroke. Experiments to define maximum EGR ratios were conducted at four operating points, i.e. nominal power, maximum torque and at two part load points (high and low part load). The results are presented in *Figure 4.2*:

**Figure 4.2:** Variation of EGR ratios at different points

The definition of maximum EGR ratios (see **Figure 4.2**) is based on equation (3.17). **Figure 4.2** indicates that throttling in the intake manifold (see **Figure A.1** in **Appendix**) is only applied at a lower part load of 1100 1/min-213 Nm point to increase the EGR ratio. These experiments stop when the FSN exceeds the FSN limits. Experimental results for determining EGR ratios demonstrate that when the throttle closes about 60 deg at higher part load(1100 1/min-425 Nm) and around 75 deg at lower part load (1100 1/min-213 Nm), the FSN reaches up to 3.37 and 4.7, respectively (exceeding the FSN limit of 2.5). Thus, 60 deg and 75 deg. are the limits on the throttle for the two part loads.

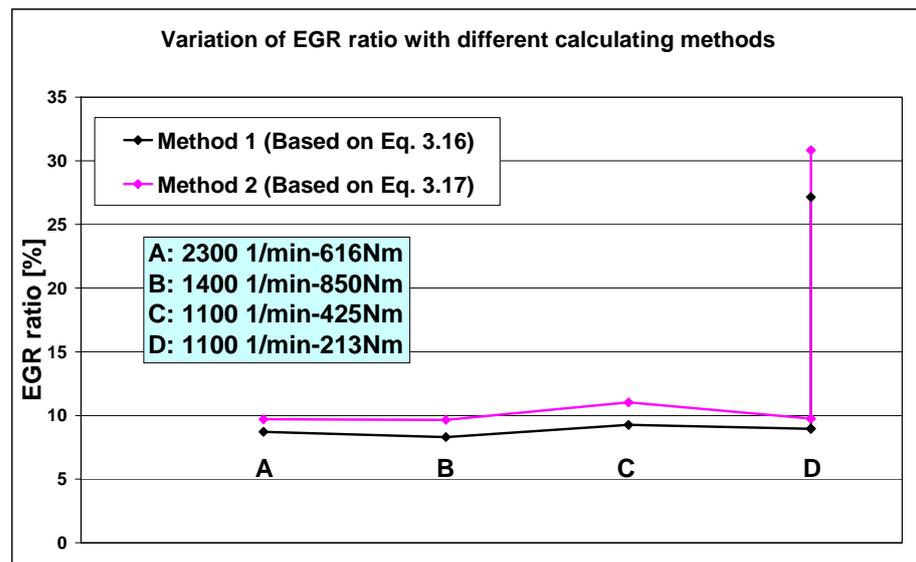


Figure 4.3: Variation of EGR ratio calculated by different methods

As explained in **section 3.1.2**, EGR ratios can be calculated with equation 3.16 or equation 3.17. Experimental results calculated with these equations exhibited acceptable differences in EGR ratios. The maximum difference reached up to 10% (see **Figure 4.3**). Consequently, equation 3.16, which depends on charge flow rates, temperature and pressure, was chosen to calculate EGR ratios in this study.

4.3.3 Operating Conditions for Experiments

Applying the aforementioned equipment and fuels, operating points and EGR ratios at different operating points, experiments were conducted under the operating conditions presented in **Table 4.7**. Experiments were conducted for three cases, i.e. earlier, original and later injection timing (B.O.E), at all experimental operating points. In the part load point (1100 1/min-213 Nm), tests were executed with 9% and 27% EGR (using the throttle in the intake manifold).

Two exhaust gas recirculation temperatures were selected for tests in every operating point with EGR. These are normal EGR temperature (the coolant being water in the engine) and 50°C. Experiments in the low idle operating point were performed without EGR.

Normal EGR temperature:

- At nominal power point-**A** (2300 1/min – 616 Nm): 200°C
- At maximum torque point-**B** (1400 1/min – 850 Nm): 150°C
- At higher part load-**C** (1100 1/min – 425 Nm): 130°C
- At lower part load-**D** (1100 1/min – 213 Nm): 100°C (9% EGR) and 140°C (27% EGR)

The experimental fuels tested were DF (diesel fuel), B100 (100% RME), B50 (50% RME with 50% DF), B10 (10% RME with 90% DF), O₂-1 (ethanol-diesel fuel blend without RME), O₂-2 (ethanol-diesel fuel blend with 5% RME), G100 (100% GTL) and G50 (50% GTL with 50% DF).

Taking the experimental results, the engine's processes were optimized for every alternative fuel at different operating points, based on NO_x concentration, FSN, specific fuel consumption and energy consumption.

Table 4.7: Various operating conditions for experiments

	B.O.E	EGR Ratio	EGR Temp.	DF	O₂-1	O₂-2	B10	B50	B100	G50	G100
A	-16	0		X							
		9	200	X							
			100	X							
	-11.7	0		X							
		9	200	X							
			100	X							
	-4	0		X							
		9	200	X							
			100	X							
B	-12	0		X			X		X	X	X
		8	150	X			X		X	X	X
			50	X			X		X	X	X
	-7.9	0		X			X		X	X	X
		8	150	X			X		X	X	X
			50	X			X		X	X	X
	-2	0		X			X		X	X	X
		8	150	X			X		X	X	X
			50	X			X		X	X	X
C	-10	0		X	X	X	X	X	X	X	X
		9	130	X	X	X	X	X	X	X	X
			50	X	X	X	X	X	X	X	X
	-6.5	0		X	X	X	X	X	X	X	X
		9	130	X	X	X	X	X	X	X	X
			50	X	X	X	X	X	X	X	X
	0	0		X	X	X	X	X	X	X	X
		9	130	X	X	X	X	X	X	X	X
			50	X	X	X	X	X	X	X	X

A: 2300 1/min- 616 Nm**B:** 1400 1/min- 850 Nm**C:** 1100 1/min- 425 Nm**D:** 1100 1/min- 213 Nm**E:** IDLE- 650 1/min

Table 4.7: Various operating conditions for experiments (cont.)

	B.O.E	EGR Ratio	EGR Temp.	DF	O₂-1	O₂-2	B10	B50	B100	G50	G100	
D	-12	0		X	X	X	X	X	X	X	X	
		9	100	X	X	X	X	X	X	X	X	
			50	X	X	X	X	X	X	X	X	
		27	140	X	X	X	X	X	X	X	X	
			50	X	X	X	X	X	X	X	X	
	-7.7	0		X	X	X	X	X	X	X	X	
		9	100	X	X	X	X	X	X	X	X	
			50	X	X	X	X	X	X	X	X	
		27	140	X	X	X	X	X	X	X	X	
			50	X	X	X	X	X	X	X	X	
	-2	0		X	X	X	X	X	X	X	X	
		9	100	X	X	X	X	X	X	X	X	
			50	X	X	X	X	X	X	X	X	
		27	140	X	X	X	X	X	X	X	X	
			50	X	X	X	X	X	X	X	X	
	E	-12			X	X	X	X		X	X	X
		-8			X	X	X	X		X	X	X
		-2			X	X	X	X		X	X	X

5 Results and Discussions

5.1 Influences of Alternative Fuels and EGR

For some 100 years, diesel engines have been being developed for optimal utilization of fossil diesel fuels. Apart from a few refinements (less sulfur, better ignition capability), the “gas oil” has essentially remained unchanged throughout this time. Increasingly stricter environmental legislation has led to correspondingly stricter emission regulations that engine modifications alone can only meet with great effort. This underscores the need to further develop the key partner in the energy conversion process, namely the fuel. Even if considerable effort is invested, the potentials of fossil fuels remain rather limited. Thus, alternative fuels represent a significant solution. Breeding, genetic technology, plant selection, modern production methods and process control, blending, aftertreatment and the use of additives substantially expand the potentials of alternative fuels, especially plant oil. A wide range of plant oils and other products derived from agricultural products such as: sugar beets, wheat, rye, triticale (a mixture of rye and wheat), etc., have already been modified, for example: transesterification, to optimize plant oil for use as an engine fuel.

Initially, the obvious approach was to perform experiments intended to optimize every conceivable fuel mixture. However, we already know that interactive effects of doing this are so extremely complex that a final assessment of the global results may involve very lengthy investigations. Thus, the initial excitement about simple fuel mixtures has dissipated. In most cases, the objective that was to produce raw alternative fuels suitable for use in modern mass produced engines without resorting to transesterification allegedly requires considerable effort. So far, these fuels have not proven successful. Apart from problems with engine operation, maintaining adequately high ignition characteristics and complying with exhaust gas emission limits is virtually impossible. Developments in the chemical engineering of additives that improve fuel properties have made pure and blended alternative fuels more suitable for use as engine fuel. In this study, experiments were conducted with pure (100%) alternative fuels and their blends with conventional diesel fuel (see *Table 4.7*).

5.1.1 Thermodynamic Parameters

5.1.1.1 Full Load Curve and the Effects of Fuel Temperature

Changes of engine torque (M_d) at full load speeds and with different alternative fuels are presented in *Figure 5.1*. Many published studies have concluded that alternative fuels such as RME, O2 Diesel, blends and GTL reduce exhaust gas and noise emissions. The experimental

results of this study not only confirm these findings but demonstrate the influence of fuels on thermodynamics parameters.

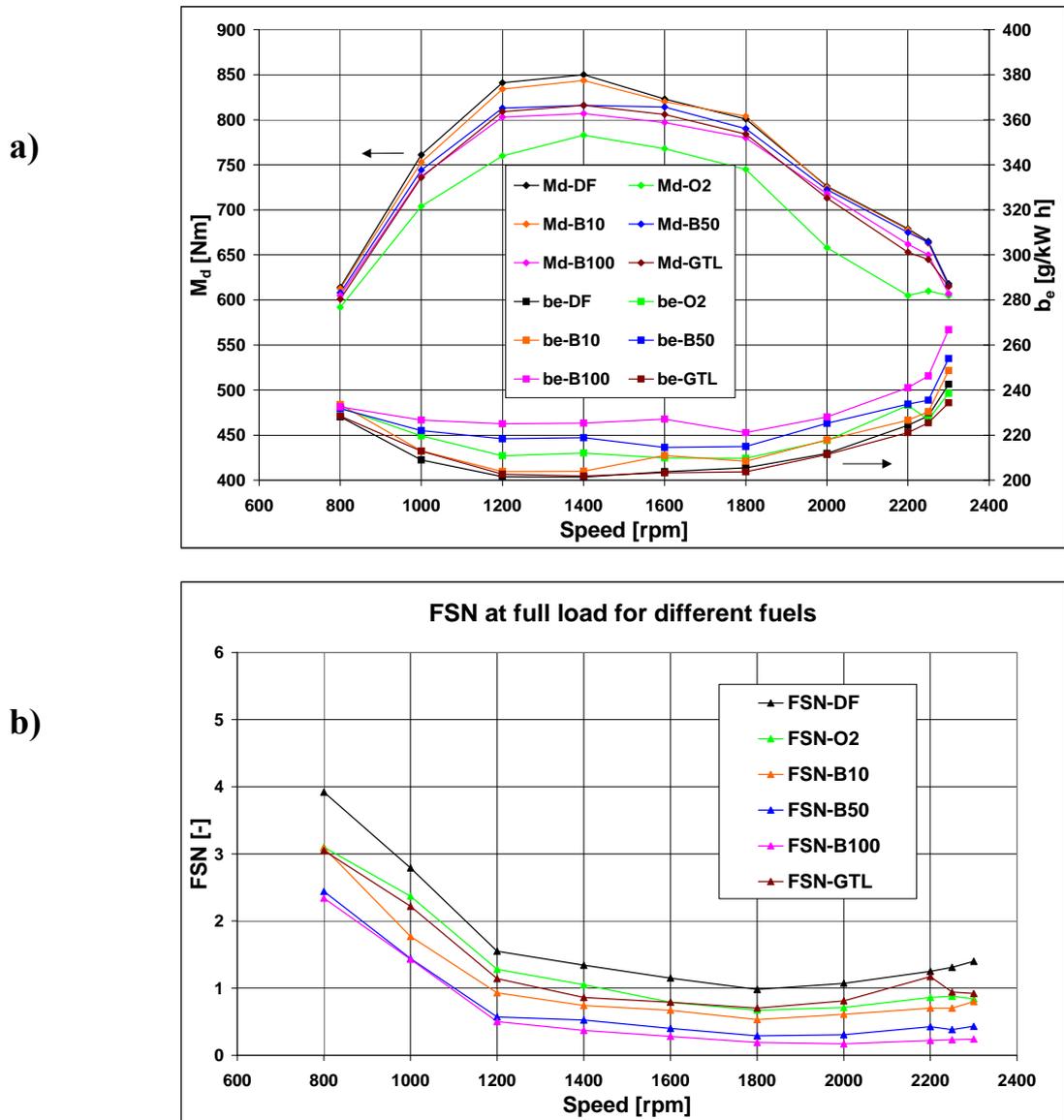


Figure 5.1: M_d , b_e and FSN at full load for different fuels: **a)** M_d and b_e ; **b)** FSN

Figure 5.1 a shows the development of decreasing engine torque (M_d) and increasing specific fuel consumption (b_e) with every experimental alternative fuel. At every point (the same speed) and full load, due to the unchanged injection duration with different fuels, different chemical properties of fuels lead to changes of injection mass and b_e , compared with diesel fuel. Logical trends, compared with neat RME and GTL, in the blends between RME and conventional diesel fuel and the blend of GTL and diesel fuel are also indicated from experimental results. Engine torque (M_d) is highest with conventional diesel fuel followed by B10 (the blend of 10% RME), B50 (the blend of 50% RME), B100 (pure RME), GTL 100 % and O₂-Diesel because RME has the lowest specific heat value of the fuels, a lower

stoichiometric air requirement (L_{st}) than other fuels and a very high density compared with other fuels.

Specific fuel consumption is highest with B100 followed by B10, O₂ Diesel, B50, GTL 100% and conventional diesel fuel (see **Figure 5.1 a**). The admitted fuel energy per time unit is smaller with O₂-Diesel and RME than with diesel fuel. Thus, the alternative fuels and their blends have lower power (work per time unit) than diesel fuel. All this explains the behavior of the different blends and the development of M_d . The difference in specific fuel consumption between GTL 100% and diesel fuel is very slight because the difference between the heat values of the two fuels is modest (43.866 MJ/kg for diesel fuel and 44.398 MJ/kg for GTL 100%).

M_d decrease to 7.8% in the worst case (with O₂-Diesel) and 2.6% in the best case (with B10). It is 4% for B50 and 5% for B100. M_d of G100 decreases around 4% compared with diesel fuel. By contrast, specific fuel consumption increases with alternative fuel. This drawback is acceptable though. The maximum increase in specific fuel consumption was approximately 11.8% with B100, while only 1.3% with B10. The increase was 8.5% with B50 and 5% with O₂-Diesel. The difference in specific fuel consumption between GTL 100% and diesel fuel is negligible (about 0.2%).

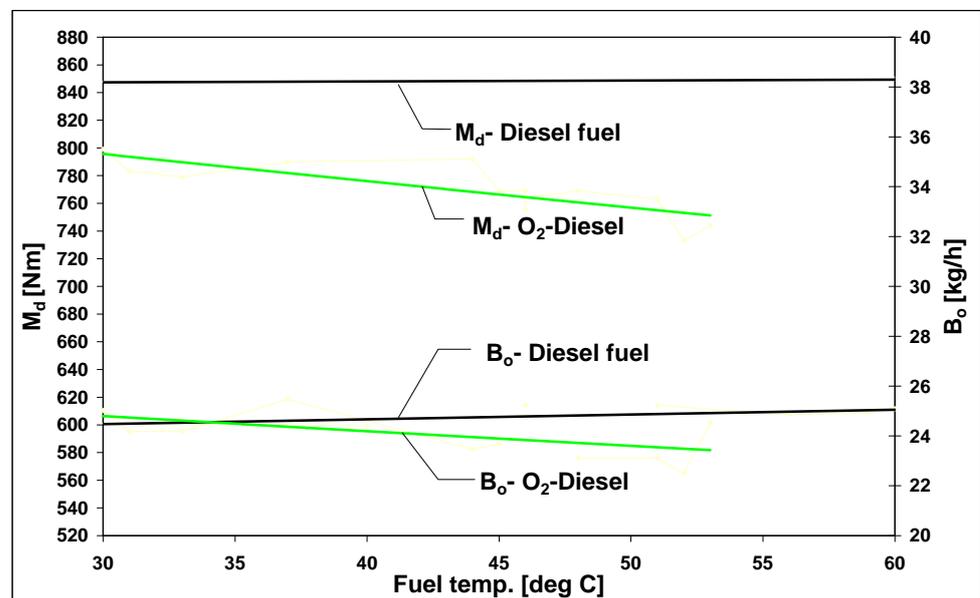


Figure 5.2: Impact of O₂-Diesel temperature on M_d and fuel consumption (B_o kg/h)

Alternative fuels and their blends point to better emissions trends, especially the FSN at full load (see **Figure 5.1 b**). The FSN decreases roughly 40% with B100 and at least 21% with O₂-Diesel and B10. The FSN falls around 38% with B50. GTL 100% has a lower FSN than O₂-Diesel, except at speeds above 1800 rpm. Compared with diesel fuel, GTL 100%

decreases about 6.4% in the minimum point (2200 rpm) and around 22% in the maximum point (800 rpm). All the alternative fuels and their blends have generally been found to decrease the FSN.

Experimental results also confirmed that fuel temperatures had less of an impact on engine torque and fuel consumption (B_o g/h) when RME, G100 and diesel fuel were used than when O_2 Diesel was used. The temperatures of RME, GTL and conventional diesel fuel were similar under most operating points and conditions. By contrast, the temperature of O_2 -Diesel significantly influences M_d and B_o (see **Figure 5.2**). Experimental results indicate the engine runs unstably when the temperature of O_2 -Diesel is around 55 °C (the same working temperature as diesel fuel, GTL and RME). When the temperature increases from around 30°C to 55 °C, engine torque changes approximately 10% and B_o approximately 8.3%. The evaporation of ethanol in O_2 -Diesel and its lower boiling temperature range (see **Table 3.2**) are responsible for these effects. Experimental results also confirmed the engine runs stably when the temperature of O_2 -Diesel is approximately 30 °C. Therefore, this was the temperature maintained in experiments with O_2 -Diesel.

5.1.1.2 Cylinder Pressure

- **Effects of Alternative Fuels**

Alternative fuels and EGR affect cylinder pressure. Changes in cylinder pressure with different fuels are presented in **Figure 5.3**.

Higher viscosity, lower density and thus higher injection pressure peaks and injection mass rates make the combustion process with RME “faster” and “harsher”. Consequently, less heat is lost through fuel evaporation than when diesel fuel is combusted and the highest peaks in cylinder pressure are produced. **Figure 5.3** shows that maximum cylinder pressure increases with most alternative fuels except GTL and its blend. Maximum cylinder pressure is highest with B100 (for the aforementioned reasons) followed by B50, O_2 -Diesel (both O_2 -1 and O_2 -2), B10, G50 and G100 (see **Figure 5.3 c**). The maximum cylinder pressure increase of approximately 3 bar occurs with B100 and the minimum of approximately 0.5 bar with B10. These changes are similar in all operating points.

Maximum cylinder pressure with GTL and its blend decreases moderately at higher load (0.6 bar in 1100 1/min - 425 Nm and 0.1 bar in 1400 1/min - 850 Nm) and significantly at lower part load (3.74 bar in 1100 11/min - 213 Nm). GTL’s high cetane number of around 74 (diesel is around 55) produces a shorter ignition delay period. As a result, the combustion process starts earlier (see **Figure 5.3 a** and **b**). Hence, combustion rates drop, causing decreases in maximum cylinder pressure and the heat release rate.

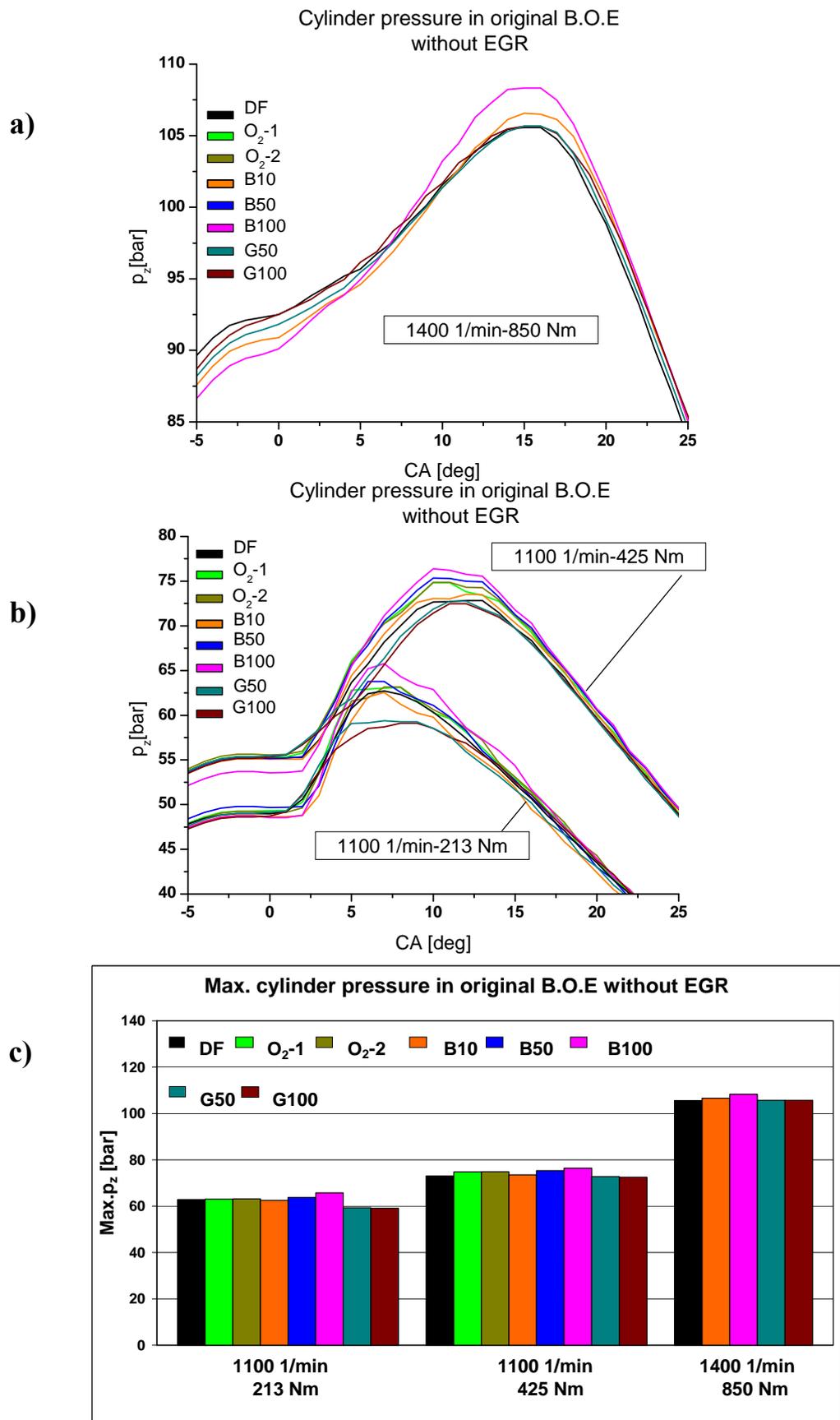


Figure 5.3: Cylinder pressure with different fuels (without EGR): **a)** at maximum torque; **b)** at two part loads; **c)** max. p_z in different operating points (without EGR)

The experimental results also display increases in maximum cylinder pressure when loads are higher. While the same speed (1100 rpm) was maintained, the maximum cylinder pressure increased around 10 bar as the load increased from 213 Nm to 425 Nm (see *Figure 5.3 c*).

• Effects of EGR

Experimental results confirm that maximum cylinder pressure decreases when the EGR ratio increases. By contrast, maximum cylinder pressure increases when EGR is cooled more strongly (see *Figure 5.4 a* and *Figure A.2* in *Appendix*).

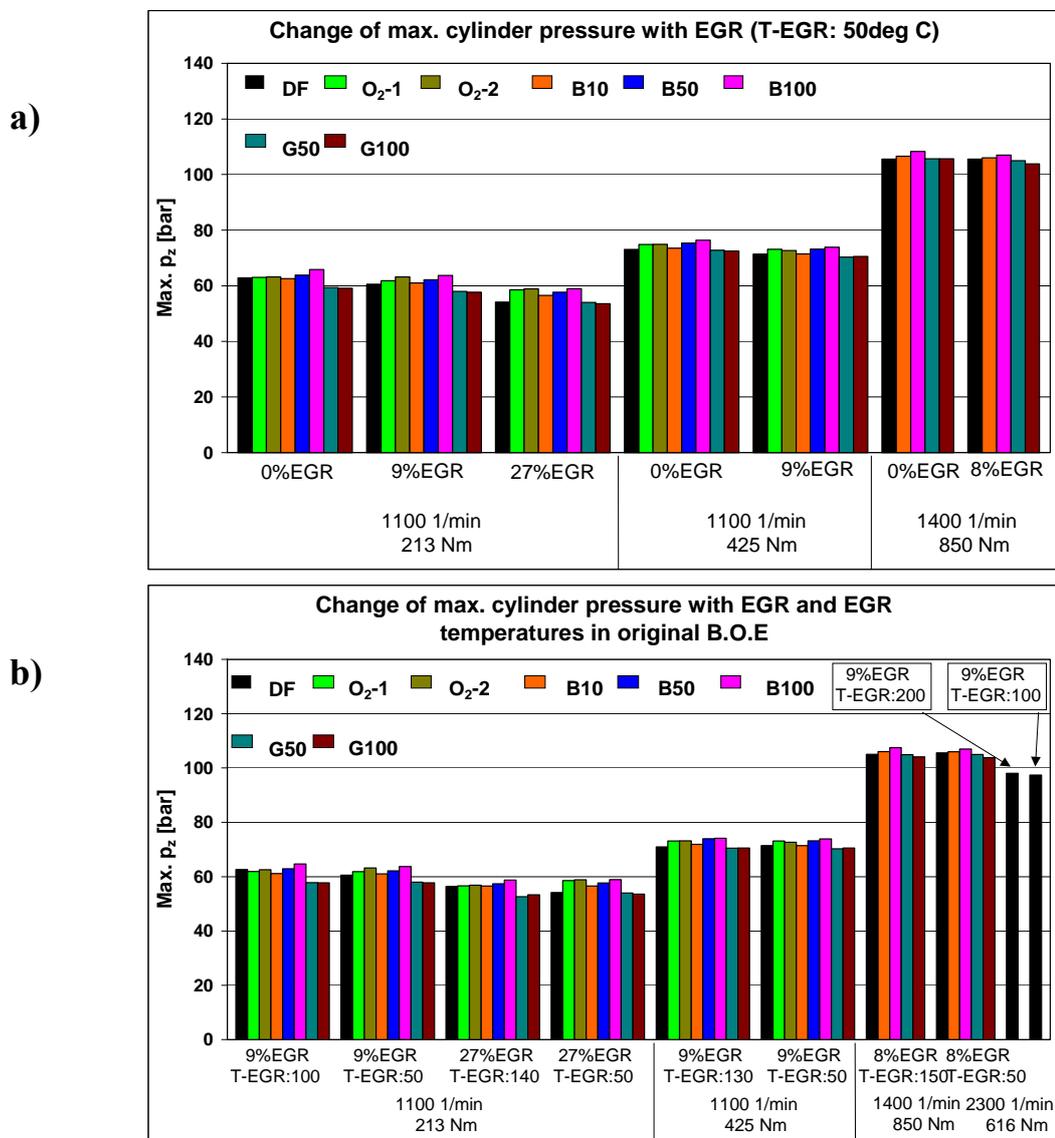


Figure 5.4: Max. cylinder pressure: **a)** with different EGR ratios; **b)** with different EGR temperatures

EGR routes exhaust gas from preceding engine combustion cycles into the combustion chamber for succeeding combustion cycles. Therefore, the initial composition of the succeeding cycle's mixture contains concentrations of burned products [1, 22]. These products primarily contain CO₂ and H₂O, and much smaller concentrations of carbon monoxide, nitric oxides, hydrocarbons, particulates, sulfur dioxides, sulfates, etc. As the concentrations of exhaust gas components in the intake charge increase, the concentration of oxygen decreases, resulting in longer periods of ignition delay. The combustion process starts later and the combustion rate decreases. As a result, maximum cylinder pressure drops as EGR ratios rise.

Decreases in cylinder pressure peaks are dependent on the EGR ratios. The greatest changes (with diesel fuel) occur at around 8 bar when EGR is 27% and around 2 bar when EGR is 9%. The maximum cylinder pressure with EGR changes very little in the nominal power point because the EGR ratio is low and the effect of the higher speed is shorter (see **Figure A.2 a** in **Appendix**).

When the EGR ratio is 27%, changes in maximum cylinder pressure are around 6.5 bar with O₂-Diesel and around 5.5 bar with GTL and thus smaller than with diesel fuel. (See **Figures 5.4 a** and **Figures A.2 b** and **c** in **Appendix**) When the EGR ratio is 9%, the maximum cylinder pressure of both O₂-Diesel and GTL changes approximately 1.4 bar. The maximum cylinder pressure in all alternative fuels has been found to change little at higher loads. At maximum torque, these changes are 0.57 bar with diesel fuel and around 1.8 bar with GTL.

Experimental results also indicate that changes in maximum cylinder pressure are minor at different EGR temperatures (see **Figure 5.4 b**). The differences increase at lower loads (the maximum difference being 2 bar when the EGR ratio is 27%). The difference is 0.7 bar at 425 Nm, 0.5 bar at 850 Nm. With the exception of a few operating conditions, maximum cylinder pressure decreases when the EGR temperature drops.

Combined effects of fuels and EGR are presented in **Figure 5.4 a**. The percentage changes in maximum cylinder pressure are presented in **Figure 5.5**. **Figure 5.5 a** presents increased trends of maximum cylinder pressure with O₂-Diesel, RME and its blends and decreased trends with GTL and its blends in all operating points and conditions. Compared with diesel fuel, percentage change is the highest with B100 followed by B50, O₂-Diesel and B10. The maximum is over 8% when the EGR is 27%. The minimum is 0.11% with B10 (in higher part load, 1100 1/min - 425 Nm). The percentage changes decrease when engine loads increase. By contrast, the percentage changes of G100 decrease more than those of G50 (see **Figure 5.5 a**). The maximum cylinder pressure of G100 decreases about 6% in the maximum case (1100 1/min - 213 Nm and without EGR) and 0.3% in the minimum case (1100 1/min - 213 Nm and 27% EGR).

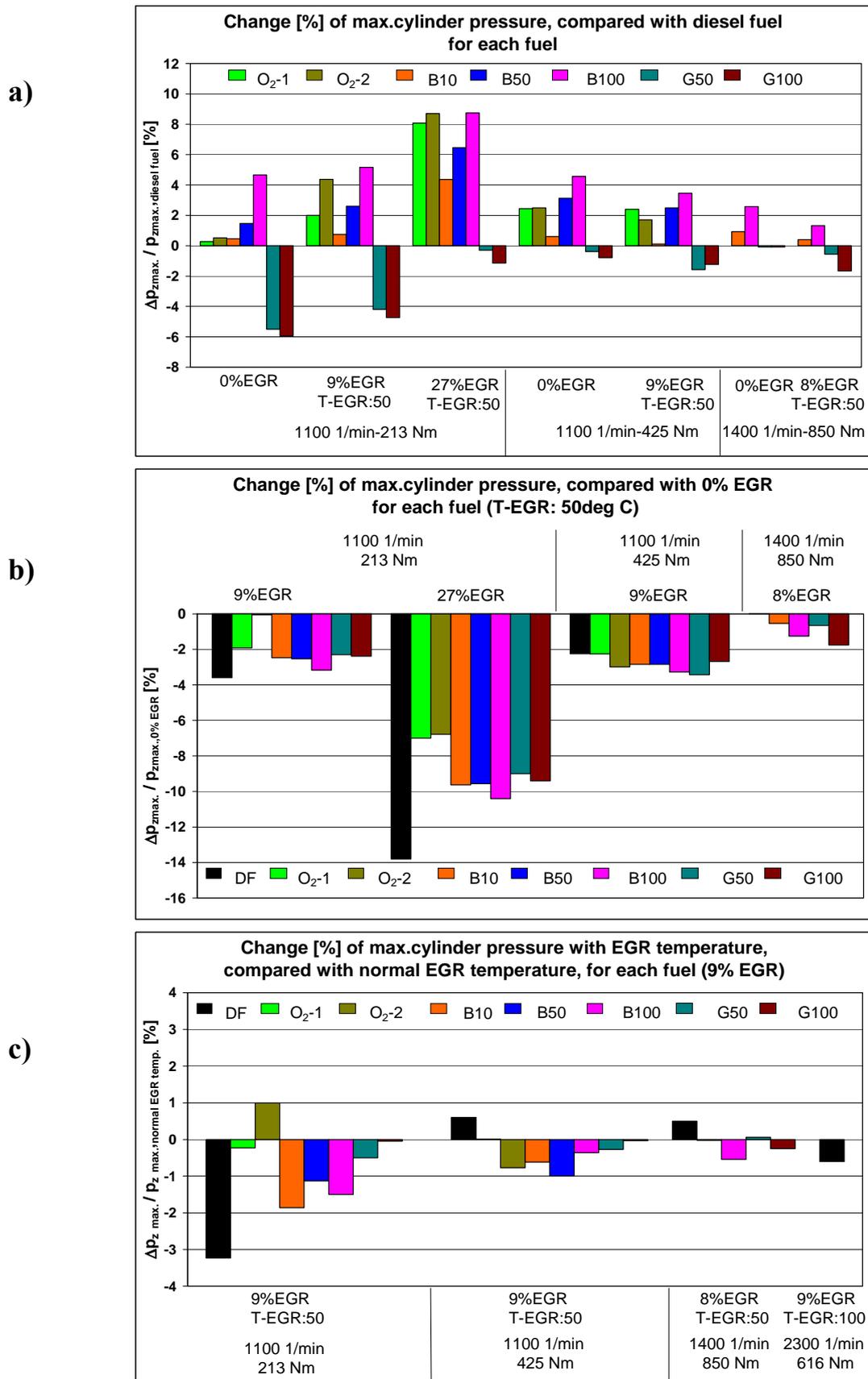


Figure 5.5: Percentage change of max. cylinder pressure: **a)** compared with diesel fuel in each operating condition; **b)** compared with 0% EGR for each fuel; **c)** with decreases of EGR temperature (compared with normal EGR temperature)

Figure 5.5 b plots the decreased trends with EGR for all fuels and under all operating conditions. When EGR ratios increase, the reduction of maximum cylinder pressure also increases. The maximum reduction of 14% occurs with diesel fuel with 27% EGR. Minimum reductions of approximately 0.06% occur with diesel fuel at maximum torque.

Figure 5.5 c plots the decreased trends when EGR temperatures are reduced from nominal EGR temperatures (see Page 48) to 100 °C at nominal power and 50 °C at other points, the exception being the operating conditions of diesel fuel at higher load and O₂-2 in lower part load (1100 1/min - 213 Nm). Percentage changes are smaller with higher loads. The maximum reduction occurs with diesel fuel a lower part load. The minimum reduction occurs with G100.

5.1.1.3 Heat Release Rate (H.R.R)

- **Effects of Alternative Fuels**

As for the changes in cylinder pressure, experimental results provide data on changes in heat release rates with various EGR rates, EGR temperatures, alternative fuels and injection timings. Under the same operating conditions, the trends of heat release rates resemble with the trends of cylinder pressure when alternative fuels are used instead of diesel fuel.

Figure 5.6 presents the increases in heat release rates when alternative fuels, except for GTL and its blend, are used instead of conventional diesel fuel. These trends occur in the experiments in all operating points and under all conditions. The physical and chemical properties (see *section 3.2.3*) prolong the period of ignition delay. This causes a higher heat release rate because heat release rate peaks essentially depend on a premixed combustion phase. Thus, the maximum heat release rate is highest with B100, followed by B50, B10 and both types of O₂-Diesel. Just as with cylinder pressure, the differences in the maximum heat release rate between O₂-1 and O₂-2 are very small under all operating conditions (see **Figure 5.6 a** and **b**). The maximum heat release rate increases 12 kJ/m³ deg CA when B100 is used (maximum case: lower part load, 1100 1/min - 213 Nm) and 0.07 kJ/m³ deg CA when O₂-1 is used (minimum case). The maximum H.R.R increases about 1 kJ/m³ deg CA with B10 and about 3 kJ/m³ deg CA with B50. The increases are 9.9 kJ/m³ deg CA and around 2 kJ/m³ deg CA with B100 in higher part load (1100 1/min - 425 Nm) and maximum torque (1400 1/min - 850 Nm) respectively with B100. In the case of smaller increase (O₂ Diesel), it is around 1.8 kJ/m³ deg CA with higher part load (1100 1/min - 425 Nm). Increase is around 1.5 kJ/m³ deg CA in both higher part load (1100 1/min - 425 Nm) and maximum torque (1400 1/min - 850 Nm) with the case of B10. With B50, change is about 3 kJ/m³ deg CA in higher part load (1100 1/min - 425 Nm) (see **Figure 5.6 c**).

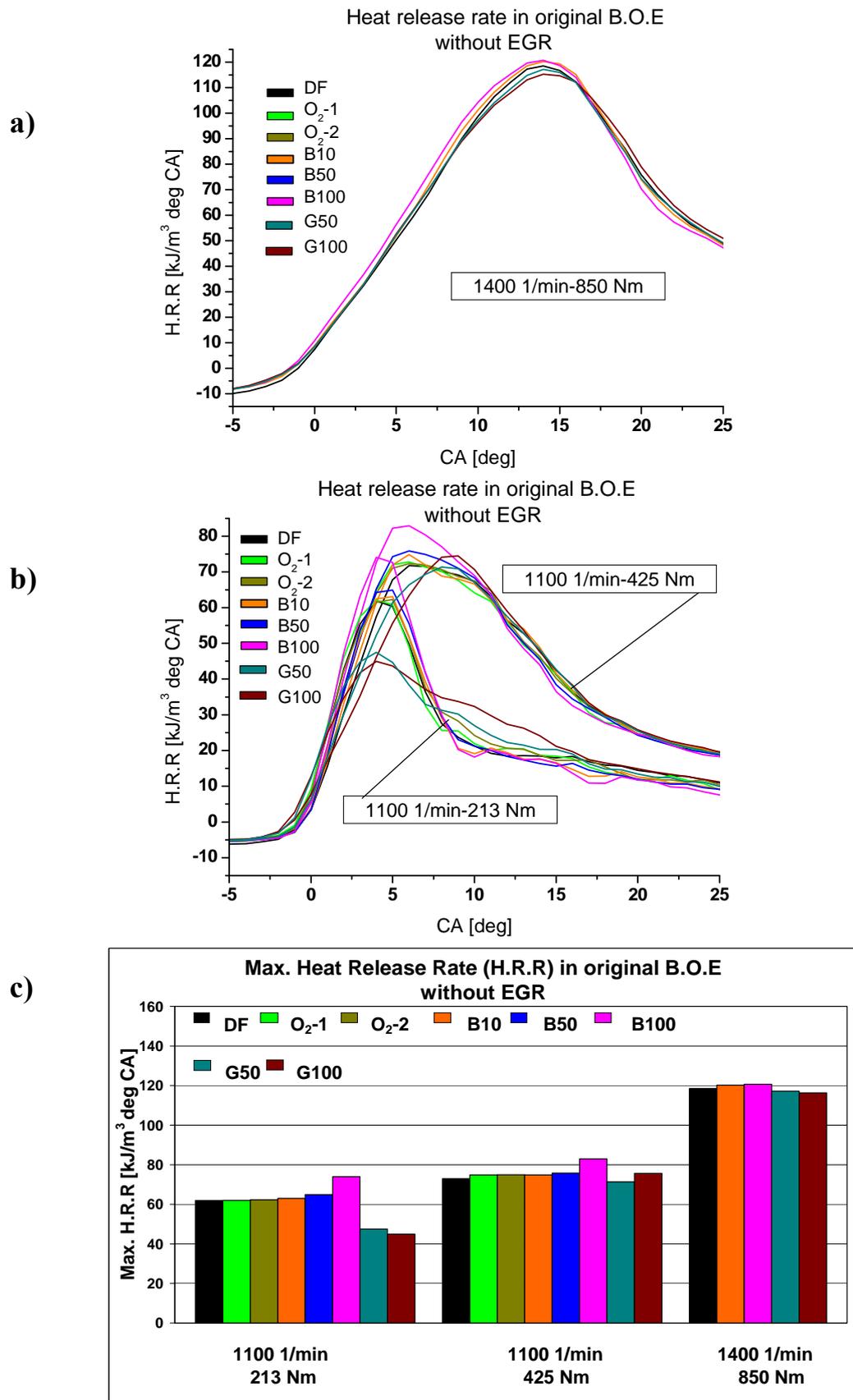


Figure 5.6: Heat release rate with different fuels (without EGR): a) at maximum torque; b) at two part loads; c) max. H.R.R. in different operating points (without EGR)

With the exception of G100 at higher part load (1100 1/min - 425 Nm), maximum heat release rates decrease when G100 and its blend are used instead of diesel fuel, (see *Figure 5.6 a* and *b*). A likely explanation for these trends is the earlier start of combustion (see *Figure 5.6 a* and *b* and *Figure 5.12 b*). The very high cetane number causes combustion rates to decrease.

Shorter periods of ignition delay allow less fuel to mix with air. Consequently, the maximum heat release rate drops with G100 and G50. Logically, the maximum heat release rate is lower with G100 than with G50, except at higher part load (1100 1/min - 425 Nm) (see *Figure 5.6 b*). Compared with diesel fuel, it decreases as much as 17 kJ/m³ deg CA with G100 and 14 kJ/m³ deg CA with G50 (lower part load 1100 1/min - 213 Nm). Decreases are smaller at higher load (2.2 kJ/m³ deg CA with G100 and 1.3 kJ/m³ deg CA with G50 at maximum torque). At higher part load (1100 1/min - 425 Nm), the maximum heat release rates increase around 2.5 kJ/m³ deg CA with G100 and around 1.5 kJ/m³ deg CA with G50 (see *Figure 5.6 c*). The maximum heat release rate is higher with GTL than with conventional diesel fuel. This could be because, at the same speed (1100 1/min) and higher load (425 Nm rather than 213 Nm), the initial temperature and pressure in cylinder are higher and the injection mass is greater. Furthermore, the ignition delay lasts longer. Hence, far more fuel mixes with air. This leads to increases in the maximum heat release rate when GTL is used instead of diesel fuel, although maximum cylinder pressure decreases modestly.

• Effects of EGR

Just as with cylinder pressure, the changes in the maximum heat release rate are minor with EGR. With the exception of a few operating conditions (e.g. more cooled EGR), maximum heat release rates decrease when EGR ratios increase.

EGR involves displacing some of the oxygen inducted into the engine as part of its fresh charge air with inert gases. The higher EGR rates are the smaller oxygen concentrations are. Reducing the oxygen concentration increases the mixing time of direct-injected fuel and fresh oxygen. This is intended to increase the ignition delay period and to decrease the burn rate once diffusion combustion starts. This leads to a reduction of the maximum heat release rate when EGR ratios increase. These reductions are modest in these experiments. The maximum heat release rates of most fuels are larger when the EGR is 27% than that when it is 9% because a throttle is used in the intake manifold. However, these differences are very slight (see *Figure 5.7 a* and *Figure A.3* in *Appendix*). At lower part load (1100 1/min - 213 Nm) and with an EGR ratio of 9%, the maximum heat release rate decreases around 1.9 kJ/m³ deg CA with diesel fuel, about 1.2 kJ/m³ deg CA with O₂-Diesel and around 1 kJ/m³ deg CA with G100. With the same EGR ratio of 9%, reductions of most alternative fuels reach around 1.5 kJ/m³ deg CA and around 2 kJ/m³ deg CA at higher part load (1100 1/min - 425 Nm) and

maximum torque (1400 1/min - 850 Nm). This decrease is about $3.5 \text{ kJ/m}^3 \text{ deg CA}$ when the engine is run under the condition of nominal power (2300 1/min - 616 Nm).

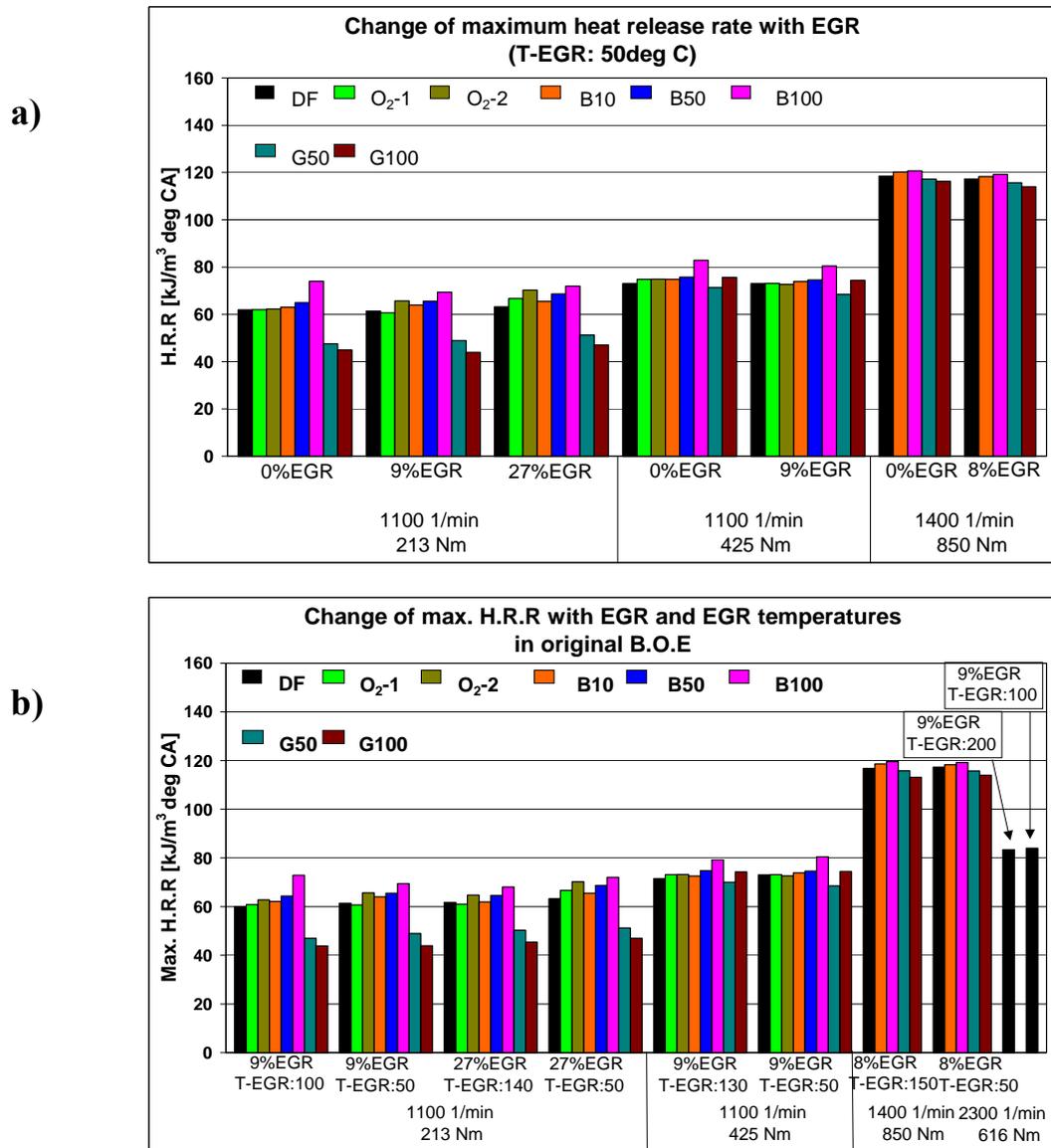


Figure 5.7: Maximum H.R.R. with different fuels and under different operating conditions: **a)** with different EGR ratios; **b)** with different EGR temperatures

Maximum heat release rates increase when EGR is cooled more strongly, even though they are higher than the maximum heat release rates when there is no EGR. This is brought about by the very late start of combustion and the longer periods of ignition delay when the throttle is applied to increase the EGR rate (27%) and the EGR cools more because the oxygen concentration drops significantly. When EGR is 27% and EGR temperature is 50°C , the maximum heat release rates increase more than when there is no EGR, e.g. $1.27 \text{ kJ/m}^3 \text{ deg CA}$ with diesel fuel, $4.5 \text{ kJ/m}^3 \text{ deg CA}$ with O₂-Diesel and around $2 \text{ kJ/m}^3 \text{ deg CA}$ with G100. When EGR temperatures reach 50°C in most operating conditions and the EGR is

approximately 9%, the decreased maximum heat release rates range from $0.5 \text{ kJ/m}^3 \text{ deg CA}$ to $1.2 \text{ kJ/m}^3 \text{ deg CA}$. They are higher with 27% EGR. When the EGR temperature drops from 140°C to 50°C and the EGR is 27%, the maximum heat release rates increase around $1.5 \text{ kJ/m}^3 \text{ deg CA}$ with diesel fuel and G100, around $5.5 \text{ kJ/m}^3 \text{ deg CA}$ with O₂-Diesel and around $4 \text{ kJ/m}^3 \text{ deg CA}$ with B100 (see **Figure 5.7 a**).

The combined influence of all alternative fuels and EGR on percentage changes in the maximum heat release rate is presented in **Figure 5.8**.

Figure 5.8 a) presents increasing trends for H.R.R with alternative fuels, with the exception of G100 and its blend. The increase is greatest with B100, followed by B50, O₂-Diesel and B10. The highest percentage increase is approximately 20% with B100 at lower part load (1100 1/min -213 Nm) and without EGR. The lowest percentage increase is around 1% with B10 at maximum torque (1400 1/min - 850 Nm). The maximum heat release rate decreases when G100 and its blend are used instead of diesel fuel. Maximum H.R.R essentially depends on the premixed combustion phase or the period of ignition delay. The high cetane number of GTL reduces ignition delay. The shorter ignition delays are the less fuel mixes with air in the period of ignition delay. Thus, maximum H.R.R decreases when G100 and G50 are used instead of diesel fuel. The greatest decrease is around 28% when the engine runs at lower part load (1100 1/min - 213 Nm) and the EGR ratio is about 9%. At higher part load (1100 1/min - 425 Nm), the maximum heat release rate with G100 increases in comparison with diesel fuel. The highest percentage increase is around 3.8%. When engine load increases, the percentage changes of all alternative fuels decrease. Maximum H.R.R and cylinder pressure significantly influence engine processes and parameters, especially NO_x emission because the temperature peak rises as the maximum H.R.R. rises.

With the exception of 27% EGR, maximum heat release rates drop when EGR phenomena occur (see **Figure 5.8 b**). The largest percentage reduction is over 5% with B10 with 9% EGR at lower part load (1100 1/min - 213 Nm). By contrast, the largest percentage increase is over 12 % with O₂-2 with 27% EGR at lower part load (1100 1/min - 213 Nm). The percentage decrease of the maximum heat release rate diminishes at high engine load when EGR ratios are around 9%.

With the exception of a few points, maximum heat release rates increase when EGR temperatures are reduced from normal EGR temperature (see *Page 48*) to 100°C at nominal power and 50°C at other operating points (see **Figure 5.8 c**) because the initial temperature of the combustion process decreases. The largest percentage increase is over 4% when the engine runs at lower part load (1100 1/min - 213 Nm) with 9% EGR. These percentages decrease when engine load increases. If, at the same speed (1100 1/min), engine

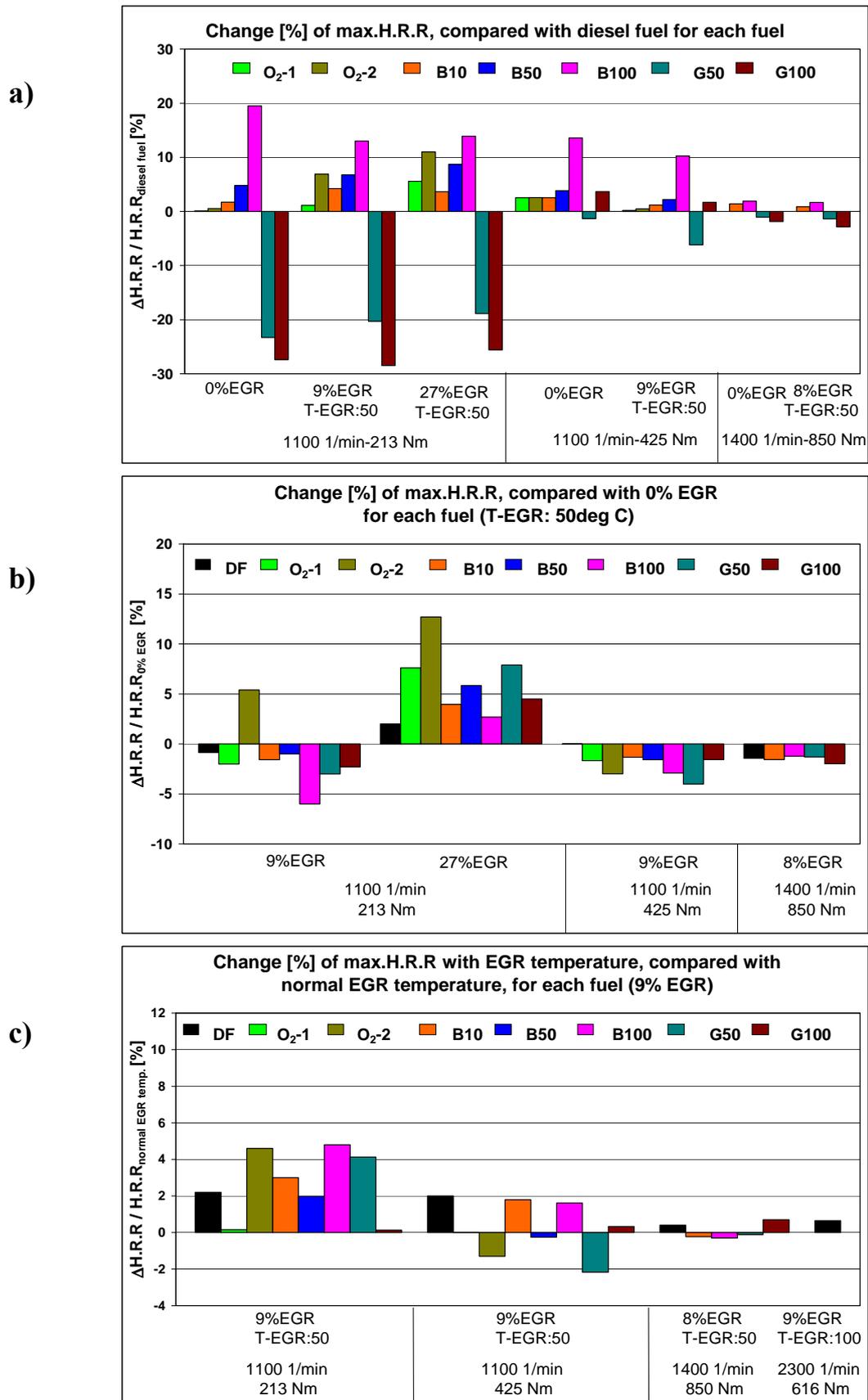


Figure 5.8: Percentage change of maximum H.R.R: **a)** with diesel fuel under each operating condition; **b)** with 0% EGR for each fuel; **c)** with decreases of EGR temperature (compared with normal EGR temperature)

load changes from 213 Nm to 425 Nm, the percentage increase of the maximum heat release rate changes from around 5% to 1.6% with B100.

5.1.1.4 Inlet Charge Temperature

Inlet charge temperature is an important parameter that significantly influences an engine's processes as well as its emissions. In this study, inlet charge temperature, also sometimes called inlet port temperature, is measured in the inlet port (see *Figure A.1* in *Appendix*). Inlet charge temperatures increase drastically when exhaust gas recirculate in the inlet manifold and hot EGR causes the EGR temperature to rise. These changes are also dependent on the EGR ratios and engine load.

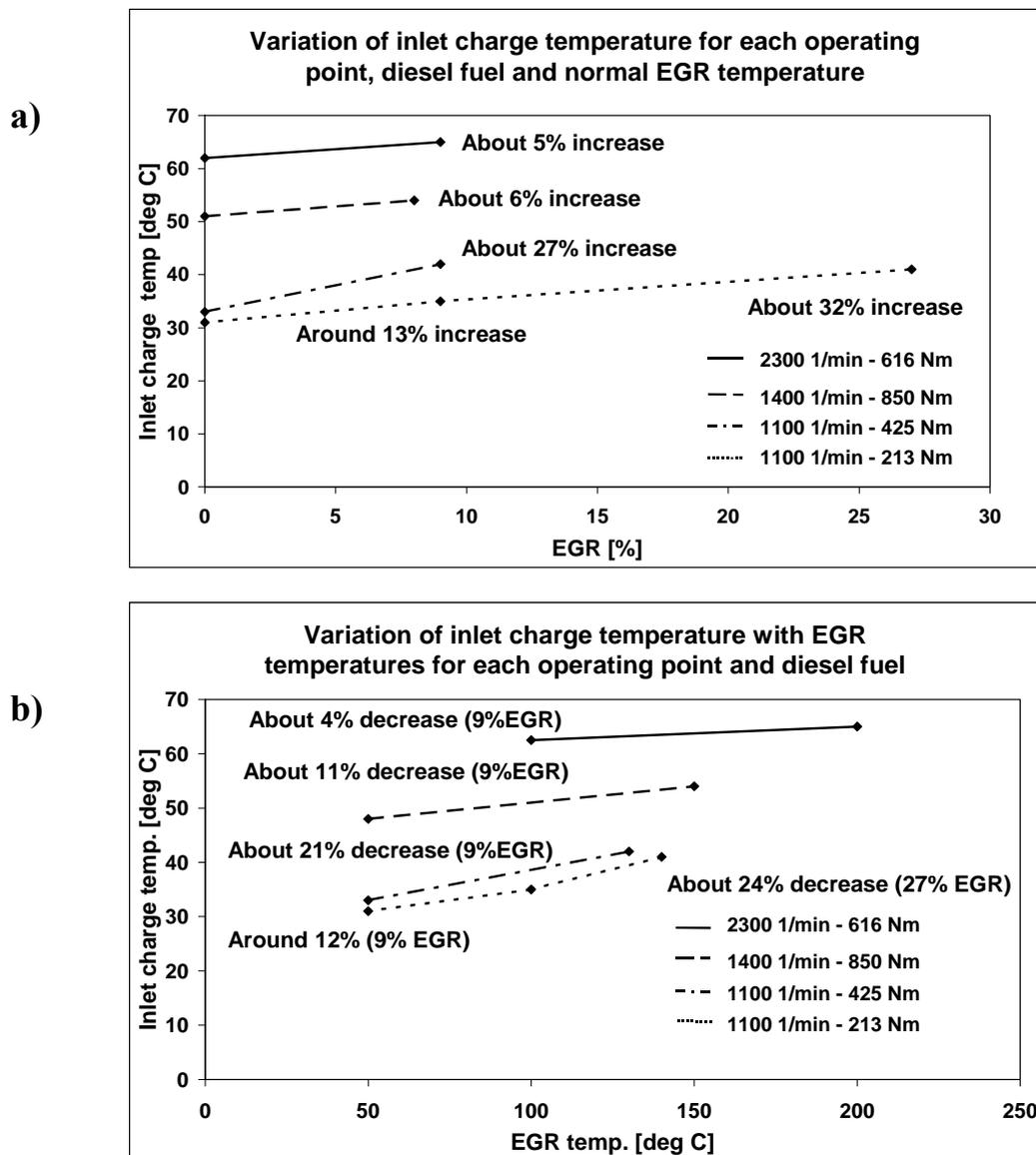


Figure 5.9: Variations of inlet charge temperature of diesel fuel: **a)** with EGR ratios; **b)** with EGR temperatures

Because changed ranges of inlet charge temperature are as same with all experimented fuel, **Figure 5.9** shows only results of diesel fuel. Experimental results also confirm that inlet charge temperature increases as EGR ratios increase (see **Figure 5.9 a**). When the EGR ratio remains the same (around 9%), the increase of inlet port temperature at lower load is greater than at higher load (5% with nominal power, 6% with maximum torque compared with 27% at higher part load). This is one explanation for the stronger effects when the engine runs at part load. At the same speed (1100 1/min), inlet port temperature increases (13% compared with 32%) when the EGR ratio increases from around 9% to about 27%.

In addition, inlet charge temperature has been found to increase as EGR temperature increases (see **Figure 5.9 b**). Like changing trends of inlet charge temperature with EGR ratios and load, with the same EGR ratios (around 9%) and the approximate change of EGR temperature (around 80 °C – 100 °C), changing percentages of inlet charge temperature with more strongly cooled EGR increase when engine load and speed are lower (4%, 11% and 21% with nominal power, maximum torque and higher part load respectively - see **Figure 5.9 b**). These changes not only significantly affect the thermodynamic parameters of the engine process but also engine emissions, especially NO_x emissions, because combustion temperatures are also dependent on initial conditions (inlet charge temperature and pressure).

5.1.1.5 Air Charge Pressure

Along with inlet port temperature, air charge pressure also plays an important role in the engine process and the formation of emissions.

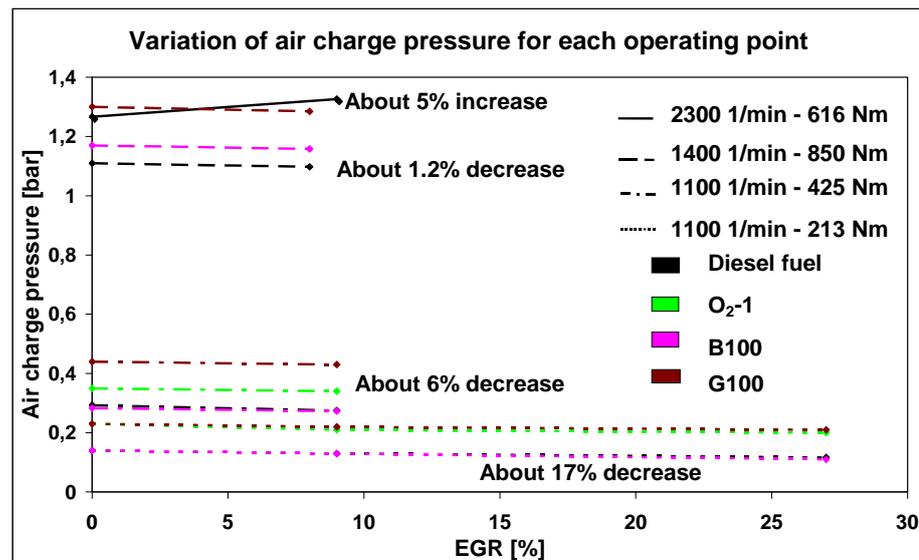


Figure 5.10: Air charge pressure over EGR ratio

Figure 5.10 presents the variation of air charge pressure with different EGR ratios, different fuels and different operating points. The same changes occur with all the fuels in the experiments. At nominal power point, experiments are carried out with only diesel fuel. Hence, **Figure 5.10** shows only experimental result of diesel fuel at this point. With the exception of the nominal power point due to performance of waste-gate valve of turbocharged, air charge pressure decreases when the EGR ratio increases. The changes of air charge pressure increases from 1.2% to 6% when the engine load drops from 850 Nm to 425 Nm while the EGR ratio remains the same (around 9%). Moreover, if the EGR ratio increases, the air charge pressure decreases (see **Figure 5.10**).

5.1.1.6 Air Fuel Ratio (Lambda λ)

The combustion process and engine emissions are dependent on several important factors such as fuel properties, initial temperature and pressure, combustion chamber shape, injection pressure, etc. One important factor, denoted as λ , indicates the ratio of air to fuel. It is the only factor that determines whether an engine's combustion process is complete or incomplete and contributes to decreased engine emissions.

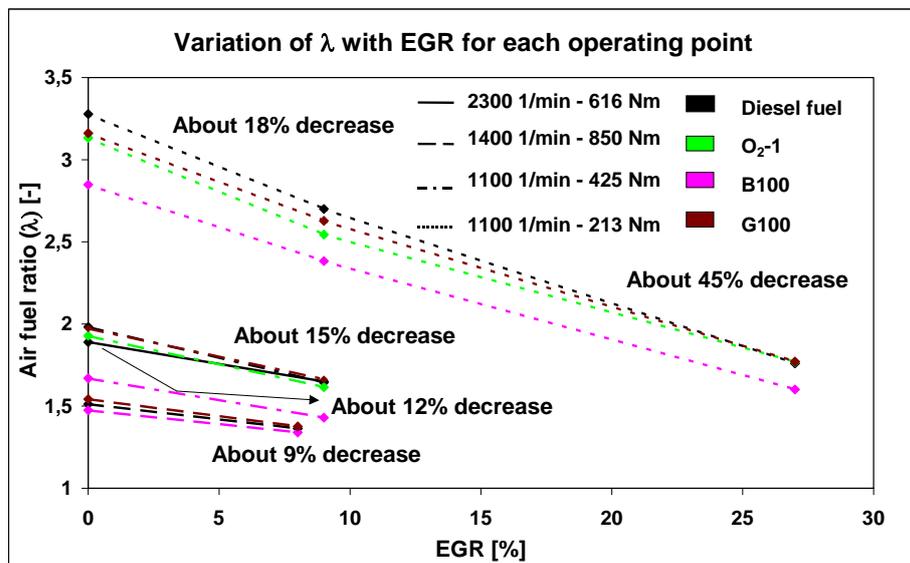


Figure 5.11: Variations of air-fuel ratio (λ) with EGR ratio

Figure 5.11 plots variations of the air fuel ratio (λ) with EGR ratios from experimental results. EGR replaces some oxygen in the fresh air charge with inert gas. Decreases in oxygen concentrations lead to decreases in the air-fuel ratio (λ) when exhaust gas is recirculated into the intake manifold (see **Figure 5.11**).

Decreasing trends with similar percentages have been found with all fuels at all operating points. Like the trends of inlet charge temperature, changing percentages of air-fuel ratios (λ)

increase strongly when engine load is reduced. When the EGR ratio remains the same (around 9%) and engine load shifts from 850 Nm to 425 Nm, changing percentage of air-fuel ratios change from 9% to 15%. They are 12% at nominal power (2300 1/min - 616 Nm) and 45 % at lower part load (1100 1/min - 213 Nm) (see **Figure 5.11**). At lower part load (1100 1/min - 213 Nm), changing percentages of air-fuel ratios (λ) change from about 17% to around 45% (compared with instances without EGR) when the EGR ratio shifts from 9% to 27%. The air-fuel ratio (λ) is highest with diesel fuel, followed by GTL, O₂-Diesel and RME.

5.1.1.7 Ignition Delay

One of the most important parameters of an engine's combustion process, ignition delay is the time (or crank angle) between the start of injection and the start of a detectable heat release rate. Based on the crank angle, ignition delay ($\varphi_{i,d}$), is determined with the following equation:

$$\varphi_{i,d}(CA) = (CA)_{5\%} - (CA)_{inject}. \quad (5.1)$$

$\varphi_{i,d}(CA)$: Ignition delay (based on crank angle)
 $(CA)_{5\%}$: Crank angle at which 5% heat is released
 $(CA)_{inject}$: Crank angle at which fuel is injected into the combustion chamber

The definition of $(CA)_{5\%}$ and $(CA)_{inject}$ is based on a mathematical model of injection start and developments of cylinder and injection pressure generated with the software Concerto (AVL). Based on time, ignition delay ($t_{i,d}$) is calculated with the following equation:

$$t_{i,d}(ms) = \frac{\varphi_{i,d}(CA)}{0.006 \cdot n} \quad (5.2)$$

n: Engine speed (rpm)

In this study, ignition delay is calculated with **Equation (5.2)**. Ignition delay is mainly a function of mixture temperature, pressure, equivalence ratio and fuel properties.

With the exception of GTL and its blend and, in some cases, O₂-Diesel at lower part load (1100 1/min - 213 Nm) and GTL and its blend at higher part load (1100 1/min - 425 Nm), ignition delay increases when alternative fuels are used instead of conventional diesel fuel. By virtue of the physical and chemical properties of RME, ignition delays are longer with RME and its blends than with other fuels. The ignition quality of a fuel, defined by its cetane number, affects the delay. The cetane number is a function of a fuel's molecular structure. Straight-chain paraffinic compounds (normal alkanes) have the highest ignition quality, which improves as the chain length increases. Aromatic compounds have poor ignition quality as do

alcohols. (Certain difficulties are connected with the use of the potential alternative fuels methanol and ethanol in compression-ignition engines.) Thus, O₂-Diesel, with additives to improve the cetane number (2 EHN-0.2%) and its evaporation of ethanol, has a shorter ignition delay than diesel and other fuels (see *Figure 5.12 a*). Thus, with a lower cetane number than diesel fuel (49 as opposed to 55), RME and its blends have a longer ignition delay period. By contrast, except at higher load, the ignition delay of GTL and its blend is shorter than that of most other fuels because it has the highest cetane number (74) (see *Figure 5.12 a*).

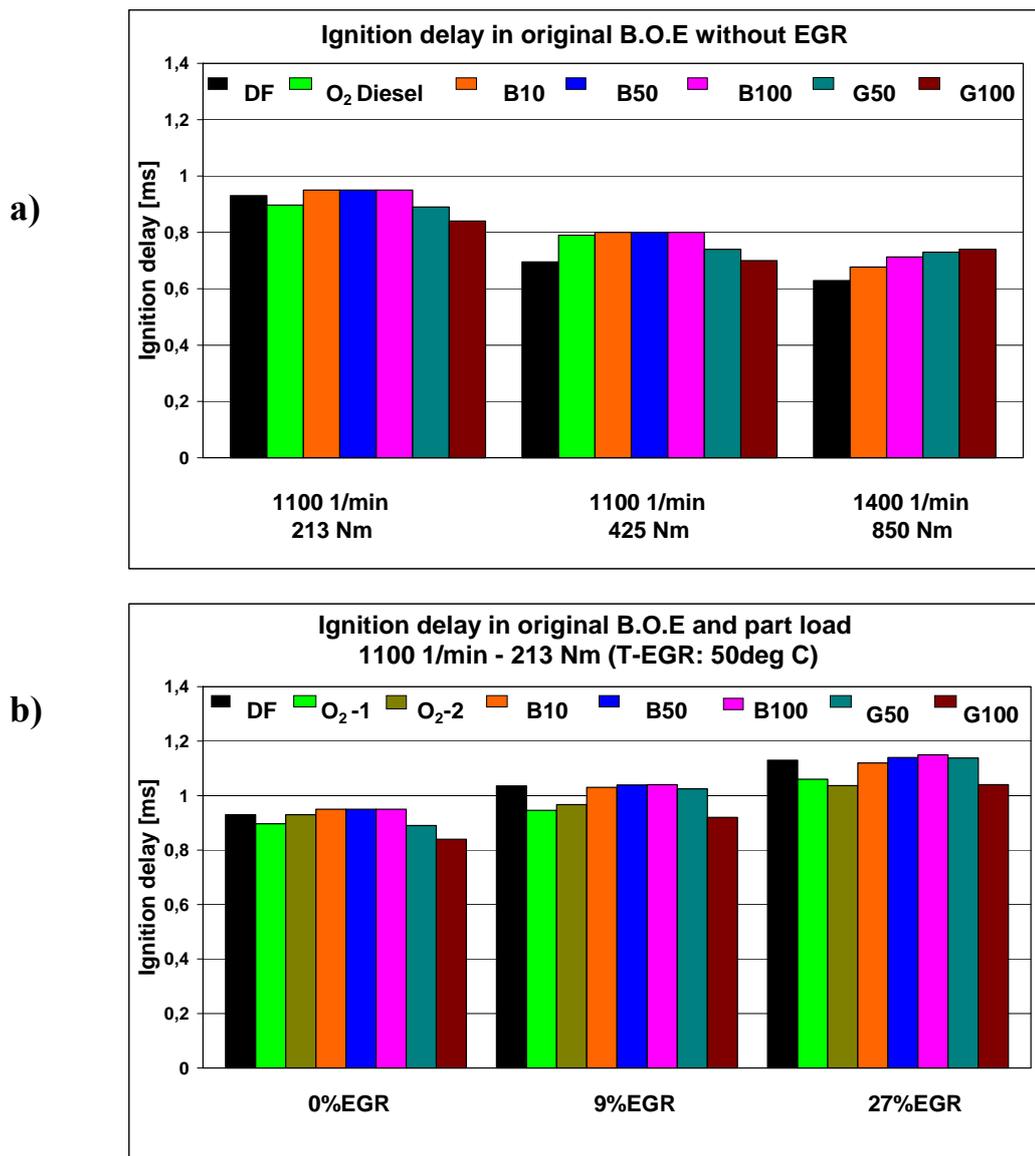


Figure 5.12: Variation of ignition delay **a)** with different fuels without EGR; **b)** with EGR ratios at lower part load (1100 1/min - 213 Nm)

The experimental results presented in **Figure 5.12 b** also demonstrate that the inert gas in the intake manifold and the insufficient oxygen concentration cause the period of ignition delay to last longer when exhaust gas is recirculated. The higher the EGR ratios are, the longer the period of ignition delay lasts. When EGR temperatures remain the same (50°C) and EGR ratios change from 9% to 27%, ignition delay increases around 10% with diesel fuel and B100 and around 12% with O₂-Diesel and G100. With EGR, similar changes have been found at all operating points and under all operating conditions.

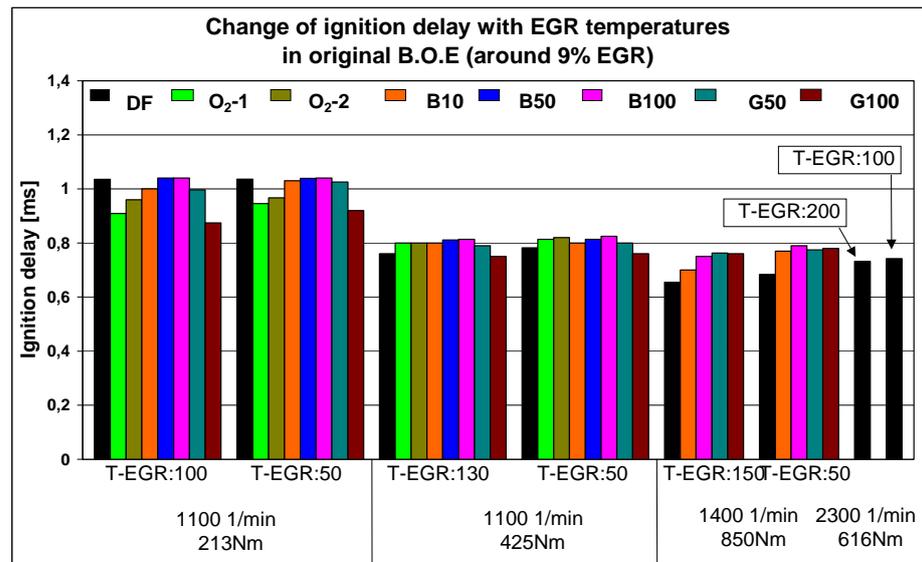


Figure 5.13: Variation of ignition delay with EGR temperatures

Periods of ignition delay increase when EGR is cooled more strongly (see **Figure 5.13**). When EGR is cooled more strongly, decreases of initial temperature lead to modest increases of ignition delay. As **Figure 5.9 b** indicates, inlet charge temperature drops when EGR is cooled more strongly. Thus, the intake air charge temperature effect on charge conditions affects ignition delay. The higher initial temperature and pressure are, the shorter periods of ignition delay are. **Figure 5.12** and **5.13** also confirm that ignition delay shortens when engine load increases. As the load increases, the residual gas temperature and the wall temperature rise. This results in a higher charge temperature (and, to a lesser extent, charge pressure) at injection, thus shortening ignition delay.

Figure 5.14 presents the percentage changes in ignition delay as a function of different fuels and EGR. Compared with diesel fuel at most operating conditions and with the exception of O₂-Diesel and GTL at lower part load (1100 1/min - 213 Nm), most alternative fuels prolong the period of ignition delay. The sharpest increase is with B100 (around 15%) and the sharpest decrease is with G100 (over 10%). At lower part load (1100 1/min - 213 Nm) and a

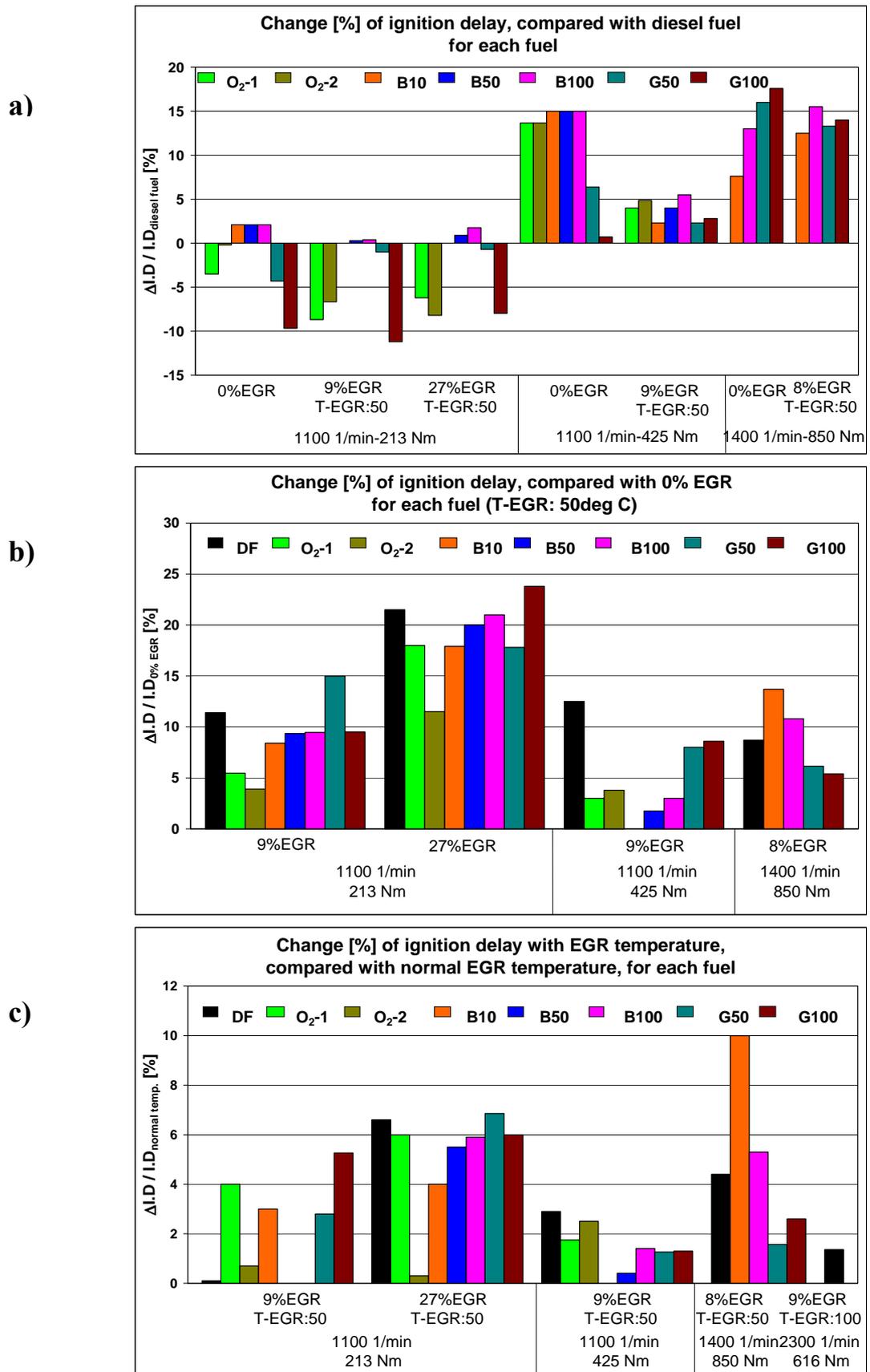


Figure 5.14: Percentage change of ignition delay: **a)** compared with diesel fuel at each operating condition; **b)** compared with 0% EGR for each fuel; **c)** with decreases of EGR temperature (compared with normal EGR temperature)

constant EGR temperature (50 °C), when the EGR ratio increase (from 9 % to 27 %), percentage changes are from 0.4% to 1.77% with B100 and from 11% to around 8% with G100 (see *Figure 5.14 a*).

The oxygen concentration in the charge into which the fuel is injected would be expected to influence delay. When exhaust gas is recirculated into the intake, the decreases in oxygen concentration lead to significant increases of ignition delay (see *Figure 5.14 b*). If the EGR ratio increases, the period of ignition delay increases. Compared with the case without EGR, when the EGR ratio changes from 9% to 27%, increased percentages of ignition delay change from about 12 % to around 22% with diesel fuel and from about 9.5% to over 24% with G100 (these changing ratios are the highest compared with other fuels). Compared with the case of 0% EGR, the increased percentages of all fuels' ignition delay decrease at higher load. When the speed (1100 1/min) and the EGR ratio (around 9%) remain the same and load increases from 213 Nm to 425 Nm, increased percentages drop from about 9.5% to 3% with B100 and from 15% to 8% with G50. Experimental results also indicate that ignition delay does not change with B10 at higher part load (indicated by the gap in *Figure 5.14 b*), when exhaust gas is recirculated.

Experimental results confirm that ignition delay increases when EGR temperatures are reduced from normal EGR temperatures (see 4.3.3) to 100 °C at nominal power and 50 °C at the other operating points (see *Figure 5.14 c*). The largest increase is up 10% when B10 is used at maximum torque. Compared with normal EGR temperature, when the EGR ratio increases (from 9% to 27%) and the EGR temperature changes to about 70 °C, increased percentages change from around 0% to 6.6% with diesel fuel and from about 3% to around 7% with G50. Thus, if the EGR ratio increases at the same point and under the same conditions, increased percentages of ignition delay caused by decreases in EGR temperature also increase (see *Figure 5.14 c*). Significant changes of ignition delay have not been detected with B50 and B100 at lower part load or with B10 at higher part load (see *Figure 5.14 c*).

5.1.1.8 Maximum Cylinder Pressure Gradient ($dp/d\alpha$)

Just as for the developments of cylinder pressure and heat release rate, experimental results provide data on the changes in the maximum cylinder pressure gradient ($dp/d\alpha$) (see *Figure 5.15a, b* and *c*). Compared with diesel fuel, the maximum cylinder pressure gradients of most alternative fuels increase at most operating points and under most operating conditions. Like cylinder pressure, they decrease when GTL and its blend are used instead of diesel fuel (see *Figure 5.15 a*). Compared with diesel fuel, the increased percentage of B100 is highest, followed by B50, B10 and O₂-Diesel. The physical and chemical properties of B100, e.g. higher viscosity, lower density and thus higher injection pressure peaks (see *Figure 5.16*) and

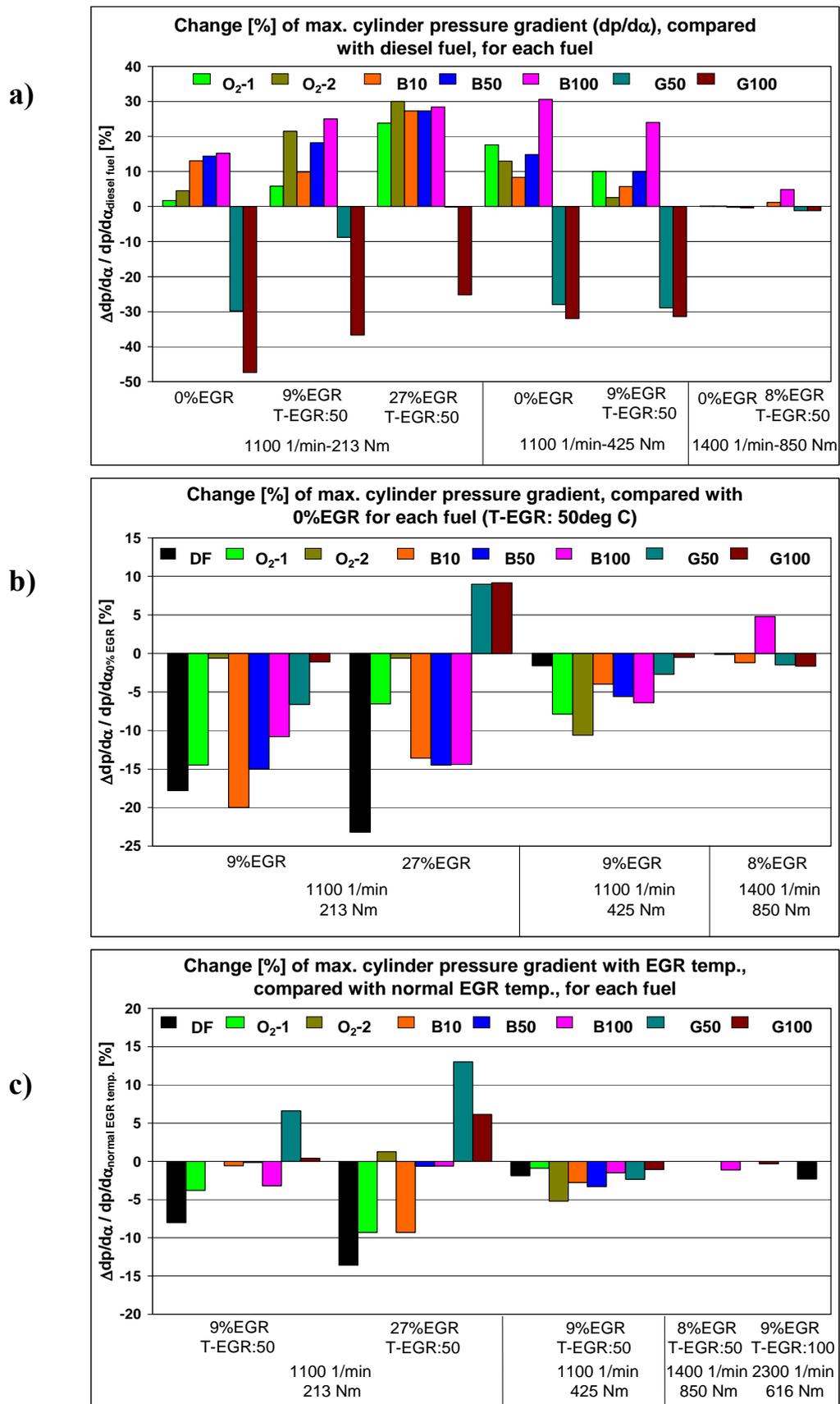


Figure 5.15: Percentage change of max. cylinder pressure gradient: **a)** compared with diesel fuel; **b)** compared with 0% EGR; **c)** with decreases of EGR temperature (compared with normal EGR temperature)

greater injection mass rates, make the combustion process “faster” and “harsher” and its duration shorter (see **Figure 5. 20**).

Compared with diesel fuel, G100 usually has a higher decreased percentage of $dp/d\alpha$ than G50 at most operating points and under most operating conditions because the combustion process occurs earlier than with diesel fuel and, therefore, combustion lasts longer (see **Figure 5. 20**). Ignition delay also influences the cylinder pressure gradient.

Longer ignition delay leads to better mixing of the air charge and the injected fuel. Hence, the combustion rate increases. This causes increases in the maximum cylinder pressure and cylinder pressure gradient, which in turn explains the higher cylinder pressure gradient of RME and its blends and the lower cylinder pressure gradient of G100 and G50, compared with diesel fuel, since RME and its blends increase ignition delay and G50 and G100 reduce it (see **Figure 5.12 a** and **5.14 a**). Although ignition delay is shorter with O₂-Diesel than with diesel fuel (see **Figure 5.12 a** and **5.14 a**), the maximum cylinder gradient increases because the evaporation of ethanol improves the combustion rate. Compared with diesel fuel, the largest increased percentage of maximum $dp/d\alpha$, is over 30% with B100 at higher part load and without EGR and the largest decreased percentage is around 48% with G100 at lower part load and without EGR. Percentage changes decrease when engine load increases. Consequently, the lowest percentage changes are found when the engine runs at maximum torque (1400 1/min - 850 Nm) and without EGR.

Figure 5.15 b presents the changes in the maximum cylinder pressure gradient with EGR. With the exception of a few points and fuels, maximum cylinder pressure gradients decrease when exhaust gas recirculate into the engine’s intake manifold. Decreases in the oxygen concentration caused by the presence of more incombustible components are a reason for this phenomenon. Compared with case without EGR, the largest decreased percentage is about 23% with diesel fuel and 27% EGR. Changing percentages of the maximum cylinder pressure gradient also increase at higher engine load (see **Figure 5.15 b**).

Figure 5.15 c illustrates that, when EGR is cooled more strongly (EGR temperatures are reduced from normal EGR temperatures (see *Page 48*) to 100 °C at nominal power and 50 °C at the other operating points), maximum cylinder pressure gradients decrease with most alternative fuels and under most operating conditions. The decrease is greatest (approximately 14%) with diesel fuel when EGR is 27%. Changes in percentage increase when the EGR ratio increases. Compared with normal EGR temperature, when diesel fuel is used, decreased percentages of $dp/d\alpha$ change from around 8% with 9% EGR to about 13% with 27% EGR. With G50 and G100 at lower part load (1100 1/min - 213 Nm), the maximum cylinder pressure gradient increases when EGR is cooled more because this lowers the initial temperature. In addition, the mixing of air and fuel lasts longer because of the low speed

(1100 1/min). Thus, the mixing rate of air and fuel increases and the burned rate increases. Furthermore, GTL has very good ignition quality because of its very high cetane number. Consequently, the maximum cylinder pressure gradient increases at lower part load when GTL and its blend are used. Experimental results also confirm the maximum cylinder pressure gradient changes negligibly at high loads when the EGR temperature is reduced (see *Figure 5.15 c*).

5.1.1.9 Injection Pipe Pressure

Along with the properties of the fuel injected into the combustion chamber, injection pipe pressure (measured in the pipe) is another important factor that contributes to the quality of the combustion process because injection pressure significantly affects such fuel spray behavior as atomization, spray penetration, droplet size and thus spray evaporation. Many studies [1, 2, 20, 27] describe how droplet diameter diminishes significantly when injection pressure increases. This decrease in diameter improves the mixing process of fuel and the air charge in diesel fuel and thus an engine's combustion process.

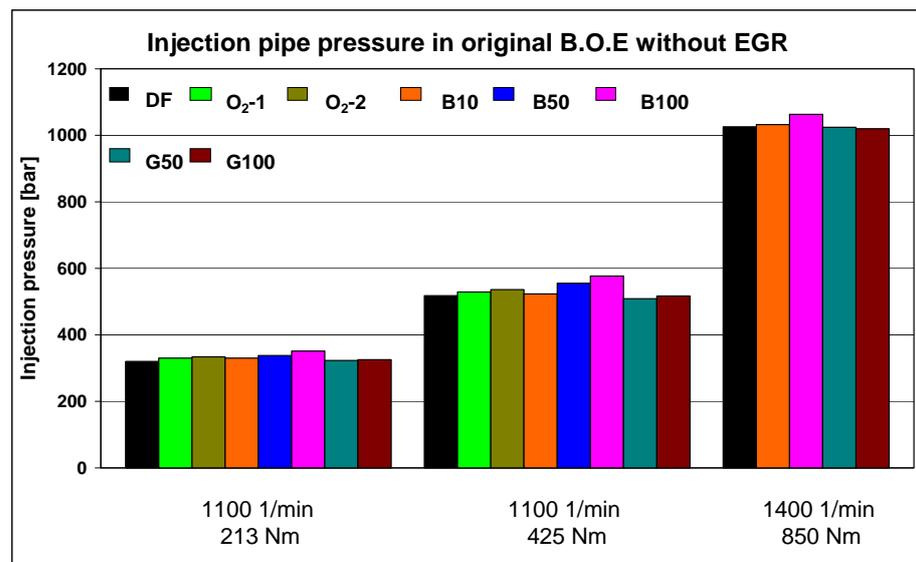


Figure 5.16: Injection pipe pressure with different fuels and at different operating points

Compared with conventional diesel fuel (see *Table 3.1 and 3.2*), the lower densities and higher viscosity of most alternative fuels, except GTL and its blend, necessitates increasing injection pressures to maintain constant operating conditions (see *Figure 5.16*). Since RME has the lowest density and the highest viscosity of alternative fuels, injection pressure is highest with B100, followed by B50, O₂-Diesel, B10, diesel fuel and GTL. The injection pressures of diesel fuel and GTL differ very slight because the viscosity of GTL and diesel fuel differ modestly. Compared with diesel fuel, the largest increased percentage of injection pressure is about 11% when B100 is used. The smallest changes are with GTL and its blend.

Compared with diesel fuel, the changing percentages of injection pressure of most alternative fuels increase when the speed remains the same (1100 1/min) and the engine load increases (from 213 Nm to 425 Nm).

5.1.1.10 Specific Fuel Consumption (b_e)

Specific fuel consumption is calculated with the following equation:

$$b_e = \frac{B_o}{P_e} \quad (5.3)$$

B_o : Gravimetric fuel consumption measured as a mass flow (g/h)

P_e : Engine power (kW)

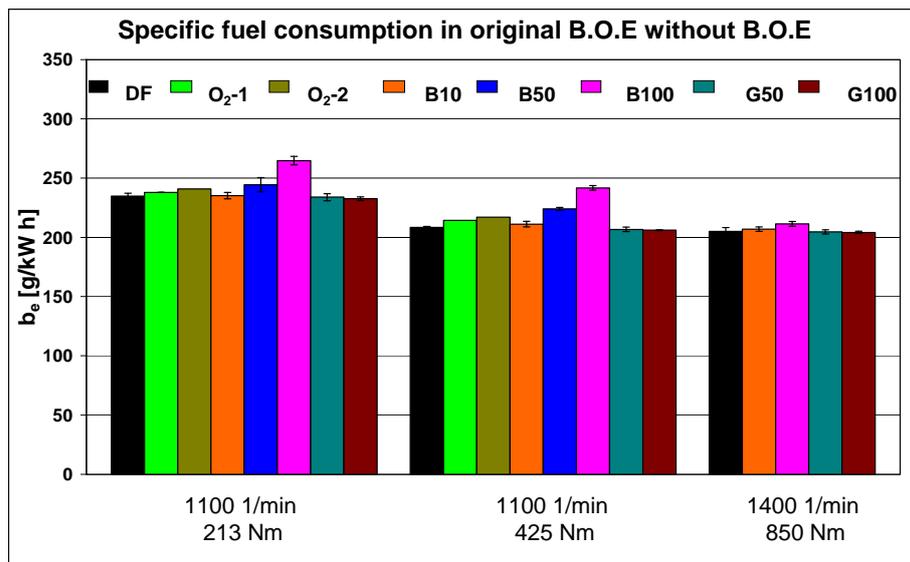


Figure 5.17: Variation of specific fuel consumption for each fuel (without EGR)

Since most alternative fuels have lower heat values and densities than diesel fuel, specific fuel consumption increases when alternative fuels are used instead of diesel fuel (see **Figure 5.17**). Specific fuel consumption is highest with B100, followed by B50, O₂-Diesel and B10 because RME has the lowest the heat value compared with diesel fuel and other alternative fuels and O₂-Diesel has a lower heat value and density than diesel fuel. The differences in specific fuel consumption between O₂-1 and O₂-2 are very small. Even though G100 has a lower density than diesel fuel, specific fuel consumption does not significantly change when GTL and its blend are used instead of diesel fuel because G100 has a slightly higher heat value than diesel fuel. Thus, G50 and G100 have a lower b_e than diesel fuel. With B 100, specific fuel consumption increases up to 29 g/kW h at lower part load, 33 g/kW h at higher

part load and 6.4 g/kW h at maximum torque. Thus, specific fuel consumption is highest at all operating points and conditions with B100. Compared with diesel fuel, the largest increased percentage of b_e is around 15% with B100 when exhaust gas does not recirculate into the intake manifold and the engine is operated at higher part load (1100 1/min - 425 Nm) (see **Figure 5.18 a**). The least increased percentage of specific fuel consumption is about 0.2 % with B10 when exhaust gas does not recirculate and the engine runs at lower part load (1100 1/min - 213 Nm).

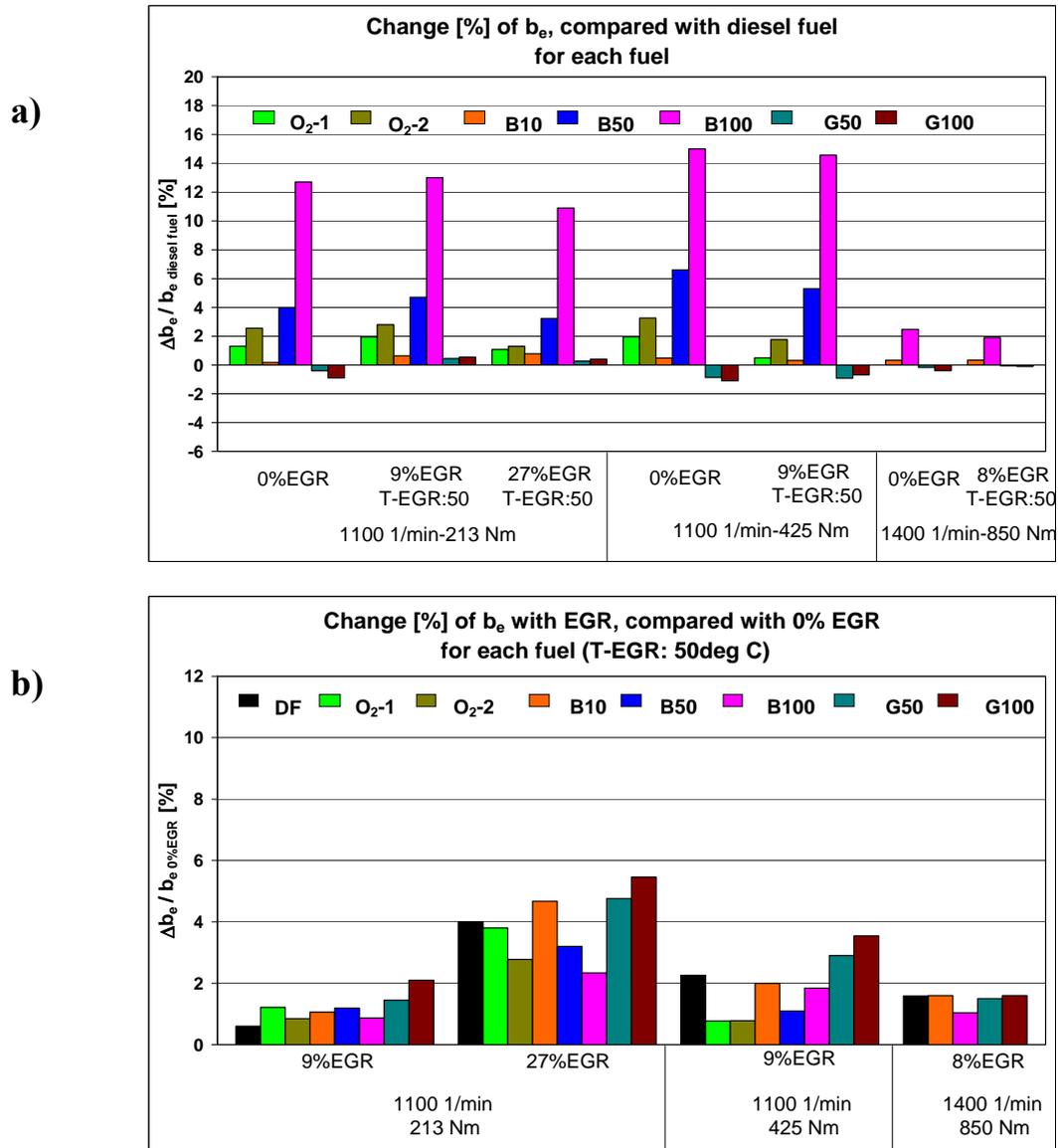


Figure 5.18: Percentage change of b_e : a) compared with diesel fuel; b) compared with 0% EGR

Compared with diesel fuel, increased percentages of b_e grow smaller with most alternative fuels when EGR increases (from 9% to 27%) (see **Figure 5.18 a**). They change from 13% to 11% with B100, from 3% to 1% with O_2 -2 and from 2% to 1% with O_2 -1. These changes are

negligible when B10 and GTL are used (see *Figure 5.18 a*). Compared with diesel fuel, the increased percentages of specific fuel consumption increase when the speed remains the same (1100 1/min) and the engine load increases (see *Figure 5.18 a*).

Compared with diesel fuel, the percentages of specific fuel consumption of all alternative fuels increase least when the engine is operated at maximum torque (1400 1/min - 850 Nm) (see *Figure 5.18 a*) because b_e usually decreases when the load is higher. This means that the ratio of work produced per cycle to the amount of fuel energy supplied per cycle that can be released in the combustion process increases. Therefore, efficiency is greater. Compared with diesel fuel, with the same EGR ratio (about 9%) and the same EGR temperature (50 °C), the increased percentage of b_e decreases from over 14 % (at higher part load) to about 2% (at maximum torque) with B100 (see *Figure 5.18 a*).

When exhaust gas recirculates, the oxygen concentration decreases. Thus, fuel consumption increases to maintain constant operating conditions. Specific fuel consumption increases when the EGR ratio increases. These drawbacks are acceptable though. Fuel consumption increases at most operating points and under most conditions and with all alternative fuels. Compared with 0% EGR, the increased percentage of b_e , is largest with GTL and its blend. It is over 5.5% and occurs at lower part load (1100 1/min - 213 Nm) with G100 and 27% EGR (see *Figure 5.18 b*). When the EGR ratio (around 9%) and EGR temperature (50 °C) are kept the same, compared with 0% EGR, increased percentages of b_e of all fuels are largest at higher part load (1100 1/min - 425 Nm). Furthermore, when the speed is kept constant (1100 1/min), they increase when the engine's load is higher, from about 0.5% to over 2% with diesel fuel and from around 2% to about 3.5% with G100 (see *Figure 5.18 b*). Experimental results also confirm that specific fuel consumption increases negligibly (a maximum of about 1.5%) when EGR is cooled more strongly.

5.1.1.11 Combustion Duration (C.D)

Combustion duration is a thermodynamic parameter used to evaluate an engine's combustion process. Crank angles of 5% and 90% that release heat serve as the basis for calculating combustion duration. These crank angles are determined from a heat release curve by the software Concerto (AVL). Combustion duration (C.D) is calculated with the following equations:

$$(C.D)_{C.A} = (C.A)_{90\%} - (C.A)_{5\%} \quad (5.4)$$

$(C.D)_{C.A}$: Combustion duration (based on crank angle)

$(C.A)_{5\%}$: Crank angle of 5% that releases heat

$(C.A)_{90\%}$: Crank angle of 90% that releases heat

Combustion duration is also calculated based on time (ms):

$$(C.D)_{\text{time}} (\text{ms}) = \frac{(C.D)_{C.A}}{0.006 \cdot n} \quad (5.5)$$

$(C.D)_{\text{time}}$ (ms): Combustion duration (based on time)

n : Engine speed (rpm)

In this study, combustion duration is calculated based on time (ms).

Experimental results demonstrate that combustion duration increases with most alternative fuel, except B50 and B100, under most operating conditions and with 27% EGR (see **Figure 5.19 a**). Compared with diesel fuel, the largest increased percentage is around 7% with G100 when the engine runs at higher part load (1100 1/min - 425 Nm). The largest decreased percentage is about 14% when B100 is used and the EGR ratio is around 27% (see **Figure 5.19 a**). With 27% EGR, the combustion duration of all alternative fuels decreases drastically. Significant increases of initial temperature and decreases of ignition delay lead to higher burned rates because the throttle is use in the intake manifold. As a result, compared with diesel fuel, combustion duration is reduced when EGR is 27%. Combustion duration does not change significantly with B10 at lower part load and without EGR or with B50 and B100 at higher part load and 9% EGR (shown by gaps in figure) (see **Figure 5.19 a**)

Flowing EGR reduces the concentration of oxygen and thus directly affects the composition of the fuel-air mixture in an engine. This causes ignition delay to increase (see **Figure 5.14 b**). The reduction of oxygen is expected to reduce the burn rate once diffusion combustion starts. As a result, the total combustion duration lasts longer (see **Figure 5.19 b**). The higher the EGR ratios are, the more the oxygen concentration decreases, the more incombustible component are aspired into the cylinder and the more the burn rate drops and the longer combustion duration lasts. The greatest change is with diesel fuel when the EGR ratio increases from 9% to 27%. Compared with 0% EGR, the increased percentage of combustion duration changes from around 1% to around 19% with diesel fuel. The increased percentage is from about 4% to 6% with B50. When engine load increases, small changes occur with most fuels (see **Figure 5.19 b**).

Figure 5.19 c) presents the percentage change of combustion duration when EGR temperature drops. When EGR temperature decreases from normal EGR temperatures (see *Page 48*) to 100°C at nominal power and 50 °C at the other operating points, combustion duration increases with all fuel and at all operating points and under all conditions (see **Figure 5.19 c**) because initial temperature and pressure drop, thus causing combustion to start later and the burned rate to decrease. The largest increase in combustion duration is 6% with

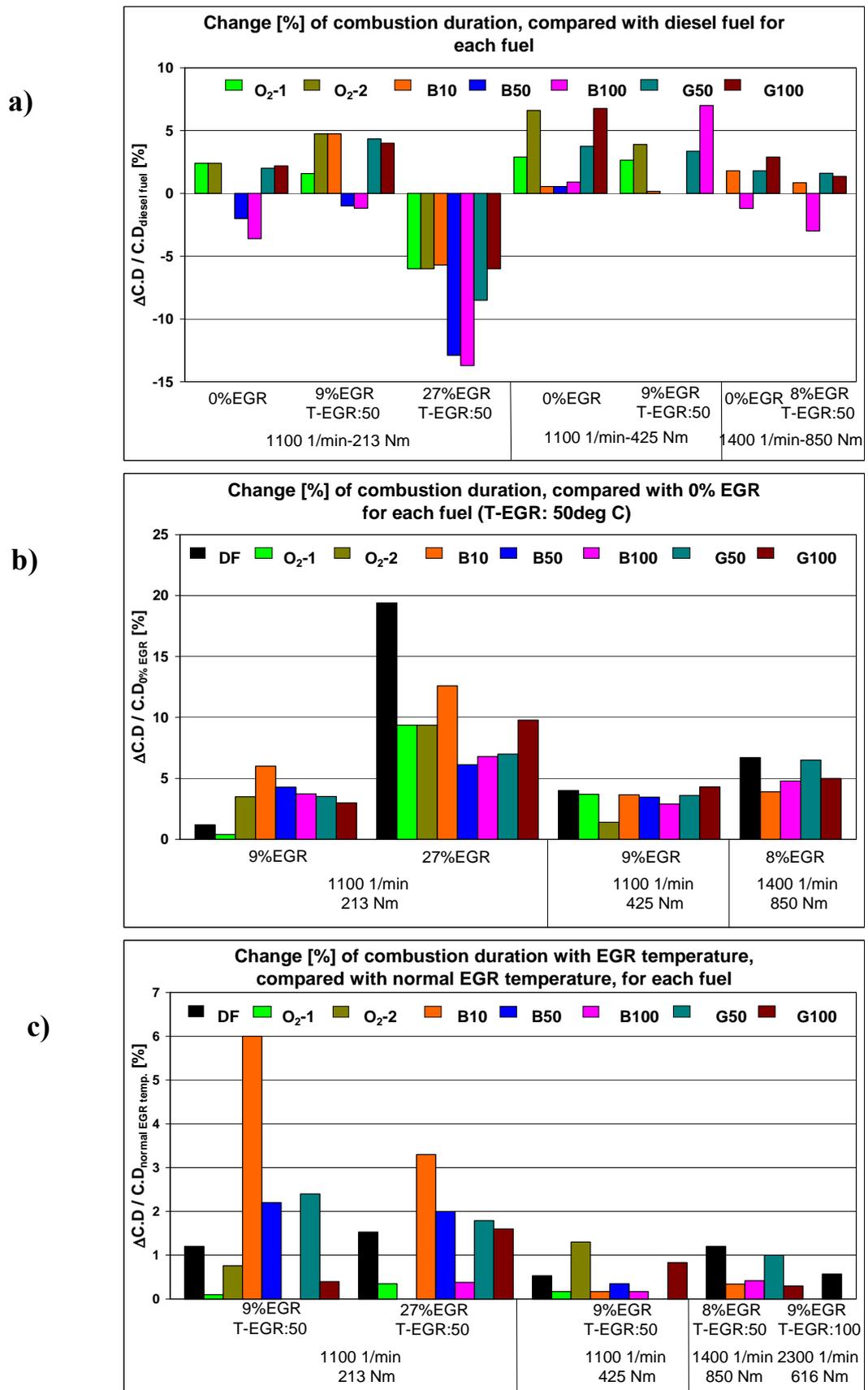


Figure 5.19: Percentage change of combustion duration: **a)** compared with diesel fuel; **b)** compared with 0% EGR; **c)** with decreases of EGR temperature (compared with normal EGR temperature)

B10 at lower part load (1100 1/min - 213 Nm) and with 9% EGR. Compared with normal EGR temperature, the smallest increased percentage is with B100.

When the EGR remains the same (9%), the changing percentage decreases when the engine load increases (see *Figure 5.19 c*). At normal EGR temperature, combustion duration does not change significantly with B100 at lower part load and 9% EGR, with O₂-2 at lower part load and 27% EGR or with G50 at higher part load and 9% EGR (indicated by the gaps in *Figure 5.19 c*).

5.1.2 Spectrum of Cylinder Pressure

- **Noise Emissions**

Anybody who has ever experienced a motor vehicle with a rigidly mounted engine or intake noise with no muffler - not to mention “naked” exhaust noise - will be in no doubt that of the many subdivisions of vehicle acoustics, engine acoustics was the first, and for a long time remained the most important [2]. Passengers’ demands for greater comfort and the concerns of the general public - represented by the law - drove developments in the engine acoustics field, which has now achieved an extremely high level of sophistication despite the enormous increases in engine output. A noise signal is generally evaluated on the basis of two criteria. These are the most precise possible evaluations in terms of subjective human perception and in terms of the most efficient possible extraction of information on noise generation and the transmission route [20]. With respect to subjective perception, the human ear is capable of registering acoustic pressures across a range of orders of magnitude of several powers of ten. For this reason, the depiction of sound levels on a logarithmic scale (dB) has become established in acoustics. Along with the many other sources such as crankshaft, flow noise (intake or exhaust system), valve seating noise, etc, combustion noise is the one of the most significant source of noise from an internal combustion engine. The essential initial basis for analyzing noises is their spectrum, i.e., the classification of a signal into its frequency components. Thus, the spectrum of cylinder pressure is analyzed and evaluated in this study.

The shape of a noise source’s frequency spectrum is subjectively important because it determines the difference between the unweighted value of L_p and the value in dB(A) [83]. It is also important to a noise control engineer because it provides clues about the noise generating mechanism and the impacts of speed change. The selection of an appropriate noise control system is also intimately related to the noise spectrum.

• Spectrum of Cylinder Pressure

Spectrums of cylinder pressure are determined from digital signals using the FFT (Fast Fourier Transformation), a highly efficient variant of the digital Fourier transformation. In this study, it is based on an FFT analysis model created with the software Concerto (AVL). The amplitudes of the spectrum of cylinder pressure are represented as levels by the following equation [83]:

$$L_p \text{ (dB)} = 20 \log_{10} \frac{P}{P_{\text{ref}}} \quad (5.6)$$

$$P_{\text{ref}} = 2 \cdot 10^{-10} \text{ (bar)}$$

With different fuels, cylinder pressure levels develop similarly to cylinder pressure. Thus, compared with diesel fuel, cylinder pressure levels of alternative fuels increase and cylinder pressure level is highest with B100, followed by B50, O₂-Diesel and B10. The difference between O₂-1 and O₂-2 is small. With G100 and G50, cylinder pressure levels decrease and drop more sharply than with diesel fuel in the high frequency range (above 400 Hz) and at low loads, (see **Figure 5.20 a, b and c**). Compared with diesel fuel, the maximum increase of cylinder pressure level is approximately 3-4 dB when B100 is used and approximately 20 dB (maximum case) at about 720 Hz and higher part load when G100 is used (see **Figure 5.20 a**). At low frequency (below 300 Hz), the higher cylinder pressure levels indicate higher engine loads because, in this stage, cylinder pressure levels essentially depend on maximum cylinder pressure. At higher frequency (above 300 Hz), cylinder pressure levels develop conversely because cylinder pressure levels are influenced by the cylinder pressure gradient. Significant decreases of cylinder pressure level occur with GTL and its blend at this stage.

When exhaust gas is recirculated into the intake manifold, the changes in cylinder pressure levels are modest (see **Figure 5.21 a, b and c**). The cylinder pressure levels change similarly to the different fuels employed in the experiments. The maximum difference of cylinder pressure levels is about 2 dB when the EGR is 27%. The levels of cylinder pressure with GTL with engine load change far more than with B100 and diesel fuel (see **Figure 5.21 a, b and c**). In the lower frequency range (below 350 Hz), cylinder pressure levels decrease as the EGR ratios increase because, in this stage, maximum cylinder pressure also decreases when the EGR ratio increases. In the higher frequency range (above 350 Hz), cylinder pressure levels increase modestly when exhaust gas is recirculated. These trends appear with all the fuels employed in the experiments. Experimental results also confirm that cylinder pressure levels change negligibly when the EGR is cooled more strongly (see **Figure 5.21 a, b and c**). Similar tendencies are also found with virtually all alternative fuels used instead of conventional diesel fuel.

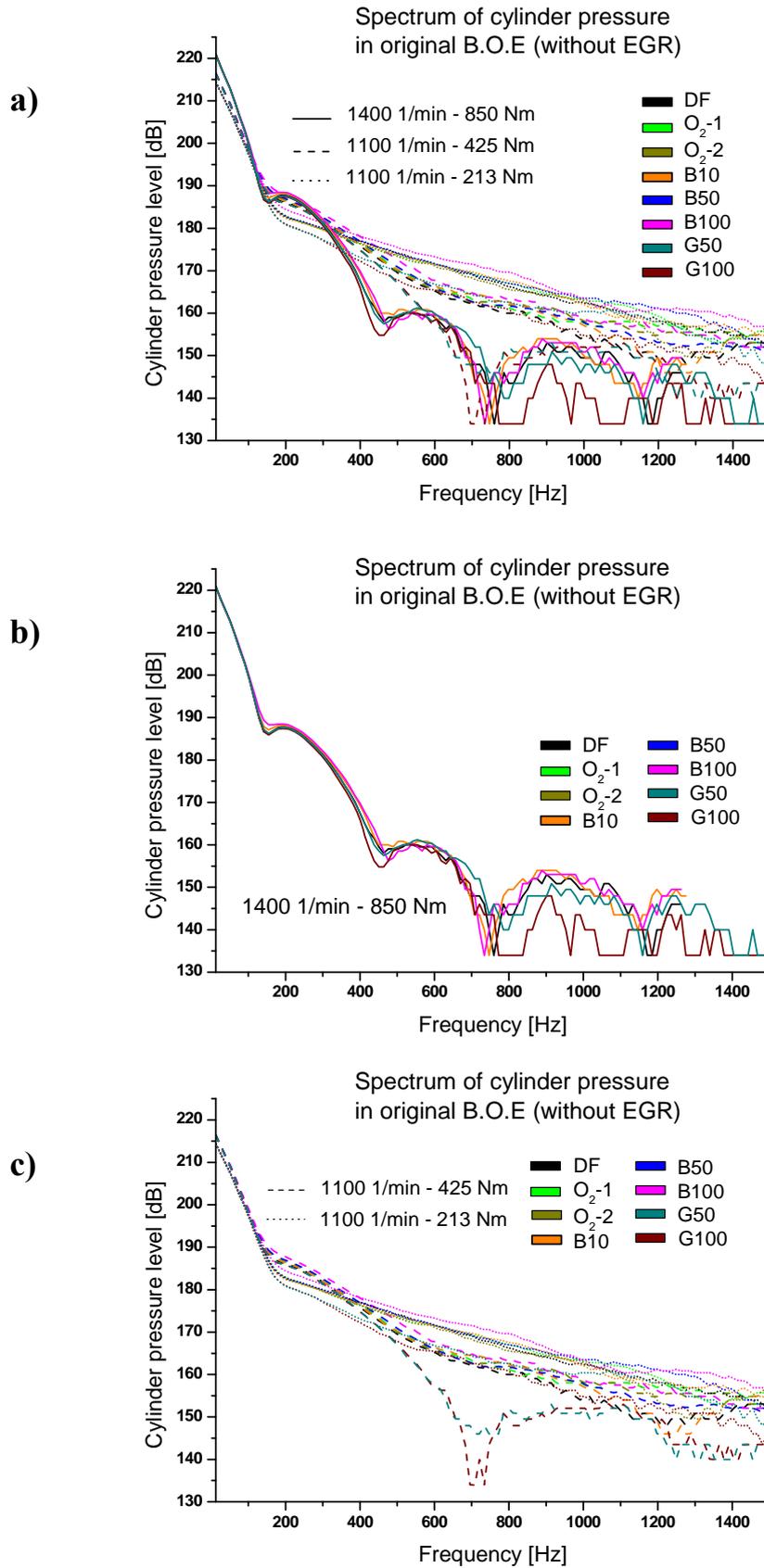


Figure 5.20: Spectrum of cylinder pressure with different fuels and at different operating points: **a)** at different operating points; **b)** at maximum torque; **c)** at two part load

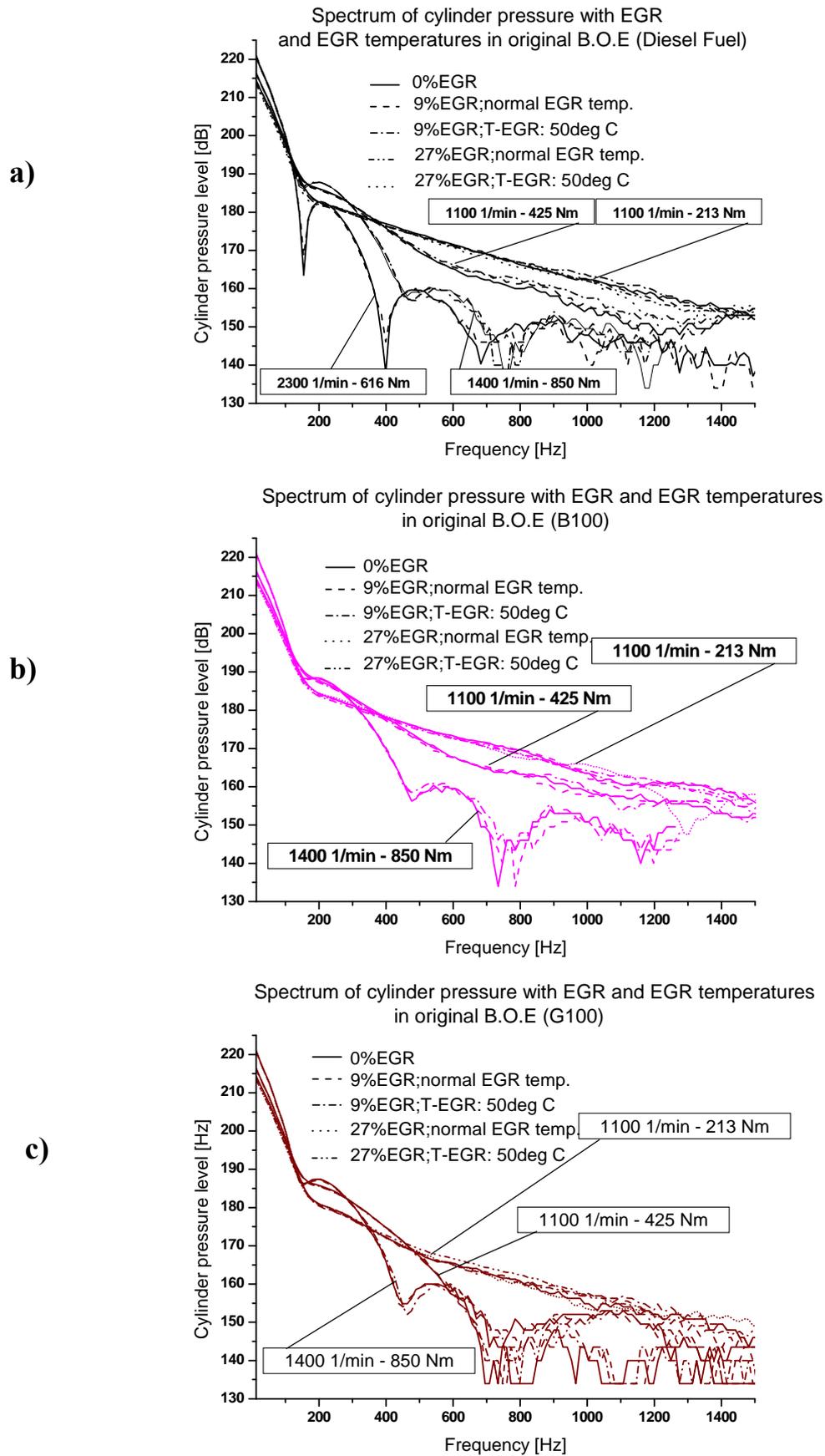


Figure 5.21: Spectrum of cylinder pressure: **a)** with diesel fuel; **b)** with B100; **c)** with G100

5.1.3 Emissions

5.1.3.1 NO_x

The kinetic mechanisms for NO and NO₂ formation are described in section 4.2.1.1. NO formation is a function of the gas temperature and air fuel ratio in postflame gases. Thus, by virtue of its chemical and physical properties, e.g. higher viscosity leads to higher injection pressure, lower heat values and lower density lead to a higher injection mass rate of fuel, boiling temperature and oxygen content, RME causes “harsher” and “faster” combustion than conventional diesel fuel. This results in increases in gases temperature and increases of NO_x emission at all operating points and under all conditions (see *Figure 5.22 a*).

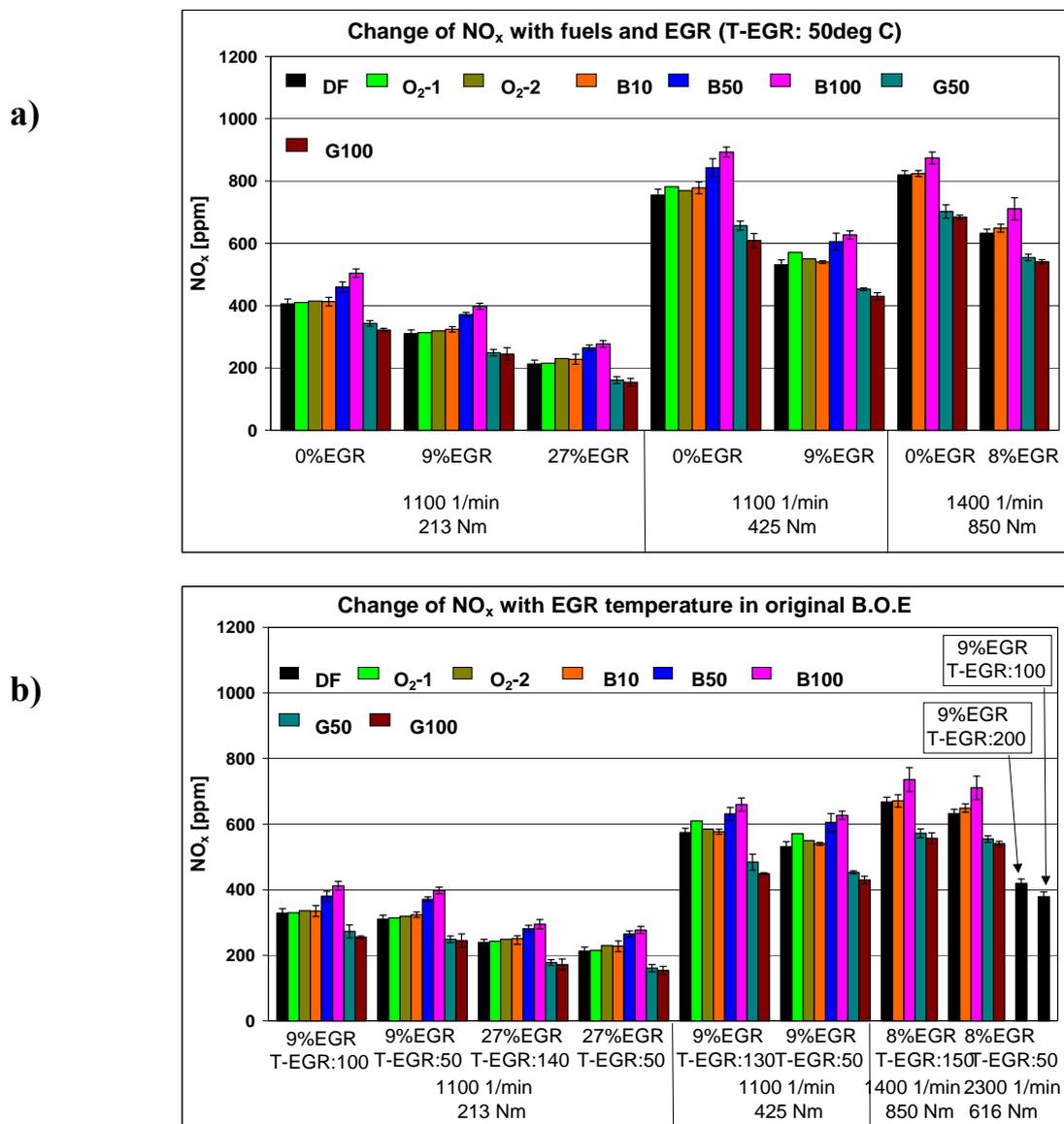


Figure 5.22: Changes of NO_x emission: **a)** with fuels and EGR; **b)** with EGR temp.

NO_x emission is greatest with B100, followed by B50, B10 and O₂-Diesel. By contrast, NO_x emission decreases when GTL and its blend are used instead of conventional diesel fuel. Since the ignition delay period is short with GTL, combustion starts earlier (lower initial temperature and pressure) and lasts longer. This leads to decreases of temperature peaks as well as decreases of the maximum cylinder pressure, heat release rate and maximum cylinder pressure gradient. NO levels drop with GTL and its blend.

The NO_x emissions of both types of O₂-Diesel differ slightly. Since ethanol evaporates more easily in O₂-Diesel, combustion starts earlier and the ignition delay period is shorter (see *Figure 5.12*). Thus the burned rate increases. Consequently, sharper temperature peaks occur and NO_x emission is higher than with diesel fuel (see *Figure 5.22 a*).

Figure 5.22 b shows decreases of NO_x emission at all operating points and conditions when EGR temperature are reduced from normal EGR temperatures (see *Page 48*) to 100 °C at nominal power and 50 °C at the other operating points. These trends are caused by drops of mixture (air charge and EGR) temperature and initial temperature in cylinder, thereby the temperature peaks in cylinder decrease.

Compared with diesel fuel, the largest increased percentages are always found with B100 and the strongest decreased percentages usually occur when G100 is used (see *Figure 5.23 a*) because RME has the lowest heat value, the highest density, highest injection mass and highest injection pressure not only compared with diesel fuel but also other alternative fuels. Therefore, the combustion rate and combustion temperature peak are highest with RME. Compared with diesel fuel, the largest increased percentage is over 30% with B100 when EGR is 27%. The strongest decreased percentage is over 20% with G100 under the same conditions (see *Figure 5.23 a*). RME blends have lower changing percentages than neat RME and G50 has lower changing percentages than G100. Changing percentages of NO_x emission of different fuels are smaller when engine load increases (see *Figure 5.23 a*). When the speed is maintained (1100 1/min), there is no EGR and the engine load increases from 213 Nm to 425 Nm, the changing percentage of NO_x with B100 drops from 24% to 18% and is around 6.7% when the engine runs at maximum torque (1400 1/min - 850 Nm).

EGR reduces the NO_x emission of diesel engines. The literature [1, 2, 22] cites three mechanisms by which EGR acts on combustion and, hence, NO_x formation and reduction:

- Dilution mechanism: The potentially prolonged mixing time and duration of burning facilitated by the effects of EGR dilution lead to a lowered flame temperature.
- Thermal mechanism: The increased heat capacity of an EGR-laced mixture produces lower flame temperature.
- Chemical mechanism: The increased dissociation from more complex EGR molecules (such as CO₂ and H₂O) results in a lowered flame temperature.

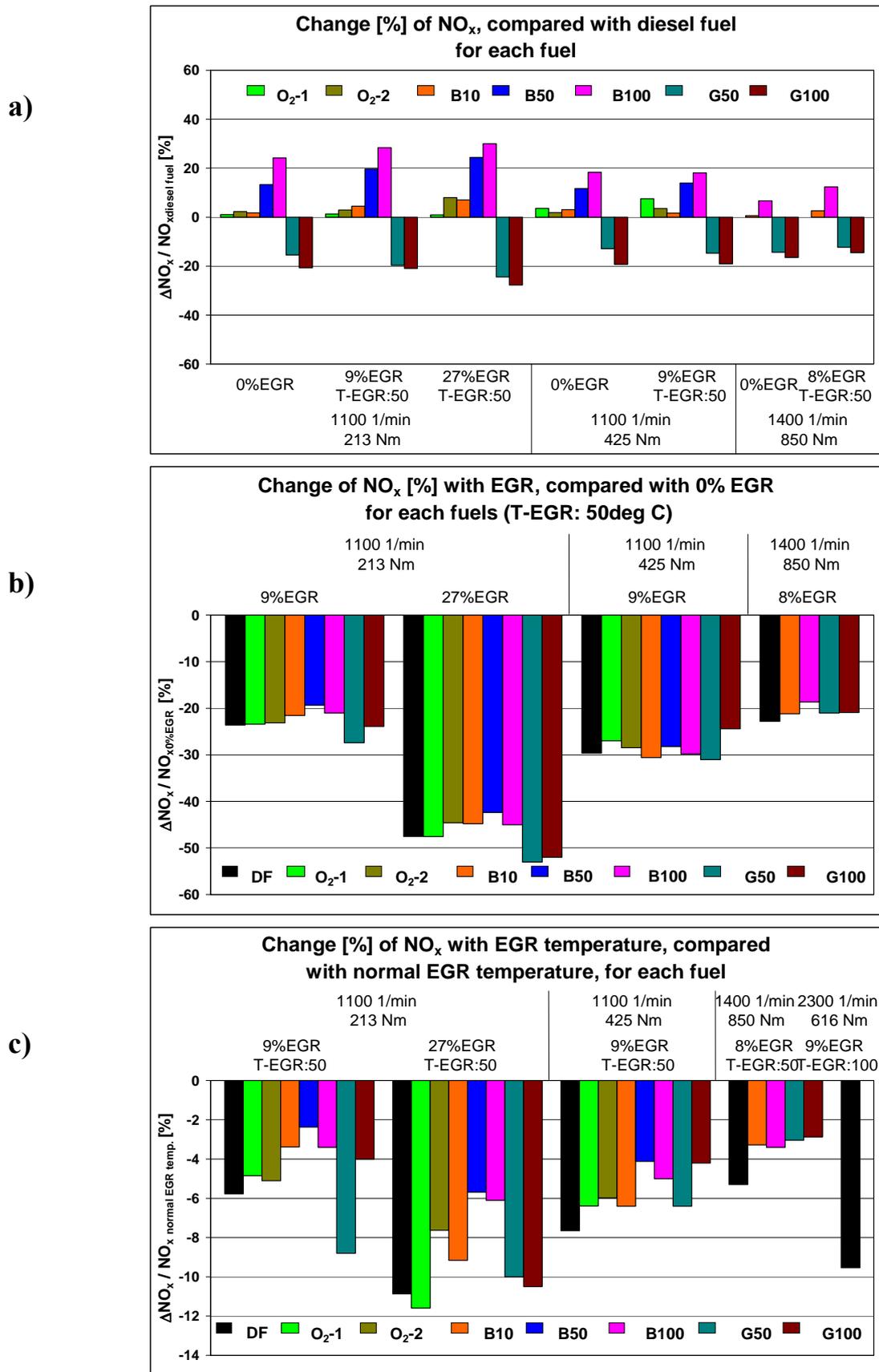


Figure 5.23: Percentage change of NO_x: a) compared with diesel fuel; b) compared with 0% EGR; c) with decreases of EGR temperature (compared with normal EGR temperature)

The chemical effect stemming from the dissociation of CO₂ and H₂O forms free radicals and has only a minor impact on combustion and emissions [2]. The principal influence of the EGR process on emissions associated with combustion is the dilution of the oxygen fraction with CO₂ and H₂O in the inlet charge. As a consequence of the reduced oxygen fraction, the start of the combustion process is delayed until the expansion stroke has begun. This delay results in less NO_x being formed because the products of combustion are exposed to high temperature for shorter periods. Thermal throttling generated by the increased intake charge temperature caused by the addition of hot EGR negates some of the NO_x reduction.

When the EGR ratio increases, NO_x emission decreases with all fuels and at all operating points and under all conditions (see **Figure 5.22 a** and **5.23 b**). Excessive dilution by the inert gases of EGR produces poor combustion quality, part burning and eventually misfire. As a result, the flame temperature peak and the NO_x level drop. Experimental results demonstrate that the changing percentage is largest with GTL and smallest with B50, compared between other fuels (see **Figure 5.23 b**). The largest changing percentage, compared between other operating points, is over 50% with G50 when the EGR is 27%. The lowest changing percentage is about 19% with B100 when the engine runs at maximum torque (1100 1/min - 850 Nm). Changing percentage decreases at higher engine load (see **Figure 5.23 b**) because smaller decreased percentages of air-fuel ratios with EGR have been found at higher engine load (see **Figure 5.11**). When the points are the same (lower part load) and EGR changes from 9% to 27%, the decreased percentage of NO_x emission, compared with 0% EGR, leaps from 23% to 47% with diesel fuel and from 24% to 52% with G100 (see **Figure 5.23 b**).

As experiments that EGR temperatures are reduced from normal EGR temperatures (see *Page 48*) to 100 °C at nominal power and 50°C at the other operating points have confirmed, temperature plays an important role in NO_x formation and reduction (see **Figure 5.22 b** and **5.23 c**). Lower initial temperature caused by more strongly cooled EGR decrease flame temperature peaks. Consequently, NO_x emission is reduced. The changing percentage with EGR temperature also depends on EGR ratios. It is higher if the EGR ratio increases. When the EGR shifts from 9% to 27%, the changing percentage increases from approximately 6% to about 11% with diesel fuel, from 5% to around 12% with O₂-Diesel, from over 3% to 6% with B100 and from 4% to over 10% with G100 (see **Figure 5.23 c**). NO_x emission changes modestly with EGR temperature when the engine load increases.

5.1.3.2 Soot

Diesel particulates principally consist of carbonaceous material (soot) generated in combustion, which has absorbed some organic compounds. Soot is a significant pollutant in diesel exhaust gas. The conditions for forming NO_x, soot (illustrated by FSN) and HC conflict. Some conditions for reducing FSN and HC, e.g. small drop size, high injection

pressure, small injection hole, etc., produce higher temperature peaks and, hence, are increasing NO_x emission. As opposed to NO_x emission, FSN decreases at most operating points and under most conditions when alternative fuels are used instead of conventional diesel fuel (see **Figure 5.24**), with the exception of G50 and G100 at lower part load and 27% EGR.

FSN emission is the lowest with B100 (see **Figure 5.24**). When the EGR is 27%, G100 and G50 have a higher FSN than diesel fuel. The FSN of G100 and its blend lies between the FSN of diesel fuel and the FSN of other alternative fuels. The FSN of O₂-1 and O₂-2 differ little. In addition, the FSN differences between diesel fuel and B10 are negligible.

When exhaust gas is recirculated into the intake manifold, more inert components flow back, more fuel is injected into the cylinder to maintain constant working conditions and flame temperature peaks drop. As a result, soot concentrations increase. With EGR, levels of FSN increase more at higher loads. When the EGR ratio increases from 0% to around 9%, the FSN of diesel fuel changes from 0.65 to 0.7 at lower part load (1100 1/min - 213 Nm) and from around 1 to around 1.55 at maximum torque (1400 1/min - 850 Nm) (see **Figure 5.24**).

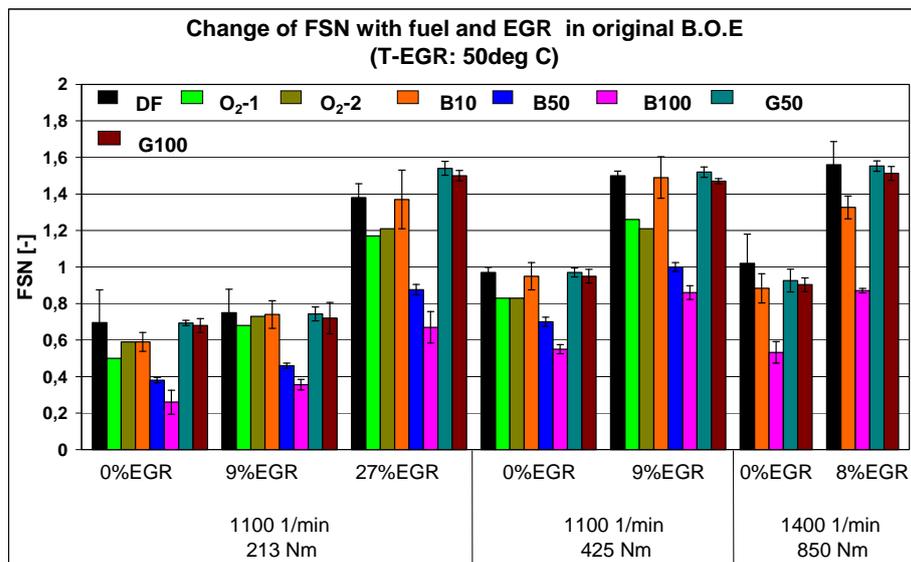


Figure 5.24: Change of FSN with fuels and EGR at different operating points

Figure 5.25 a presents decreasing trends of FSN for all alternative fuels, compared with diesel fuel, except when the EGR is 27% and except for G100 and G50. The changing percentages are largest with B100 and lowest with GTL, its blend and B10. The maximum changing percentage is over 60% with B100 at lower part load (1100 1/min - 213 Nm). When the speed is the same (1100 1/min), EGR is 9% and engine load increases from 213 Nm to 425 Nm, the changing percentage decreases from over 50% to over 40% with B100 and increases modestly with O₂-Diesel.

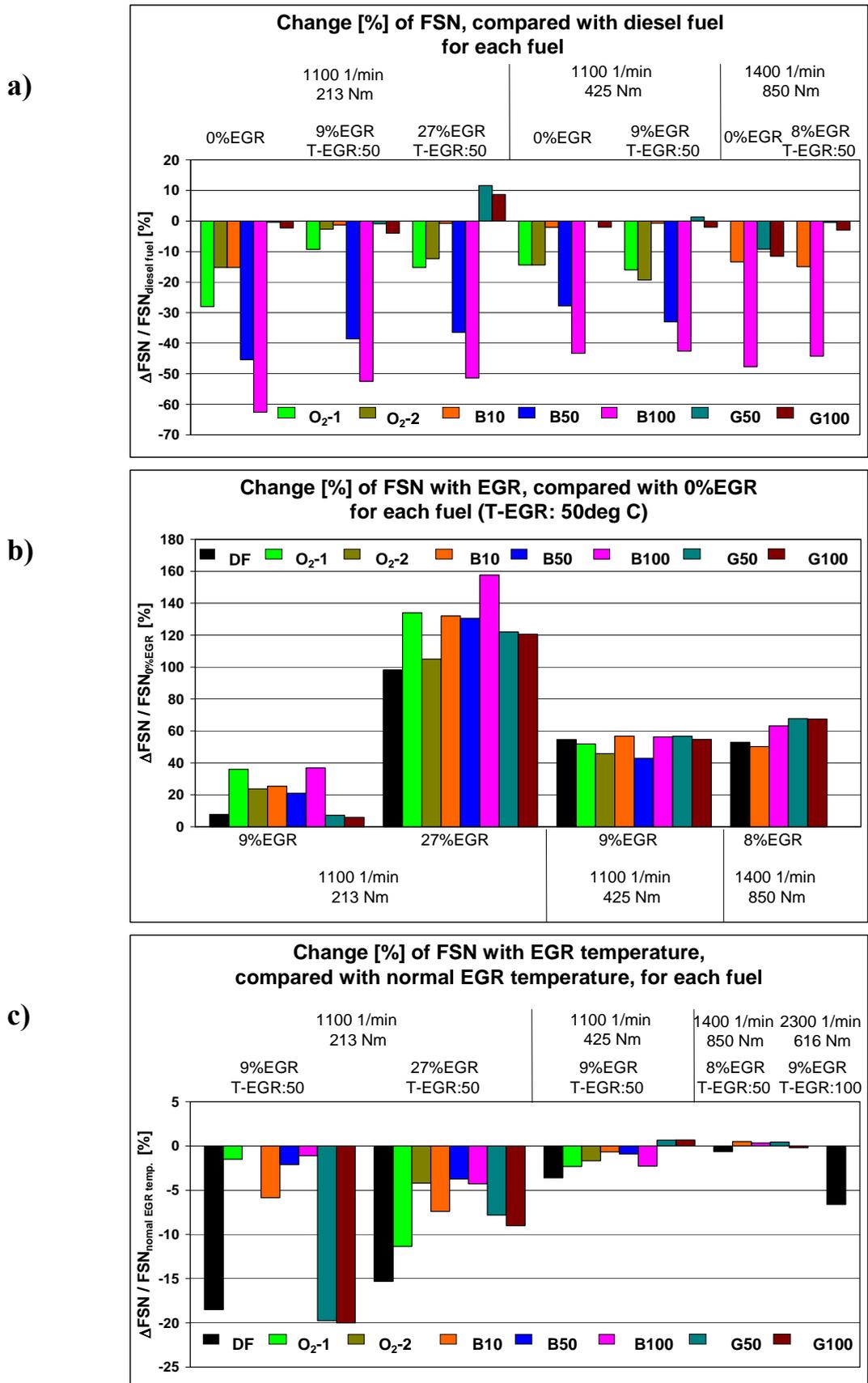


Figure 5.25: Percentage change of FSN: **a)** compared with diesel fuel; **b)** compared with 0% EGR; **c)** with decreases of EGR temperature (compared with normal EGR temperature)

The FSN increases strongly when the EGR ratio increases (see **Figure 5.25 b**). This occurs with all fuels and under all operating conditions. When the EGR ratio increases from 9% to 27%, changing percentages of FSN increase from about 7% to 100% with diesel fuel, from 20% to over 130% with B10 and from 5% to 120% with G100. Changing percentages with EGR increase when engine load increases when EGR is around 9% because the inert components aspirated into the intake manifold increase (see **Figure 5.25 b**).

Since the C-O ratio critical for soot formation increases as the temperature increases [2], the FSN decreases with all fuels when the EGR is cooled more strongly (see **Figure 5.25 c**). Changing percentage of FSN change greatly with EGR temperature at low engine load. The maximum changing percentage is around 20% with GTL and its blend. These percentages are very small when the engine runs at maximum torque. At low EGR temperature and thus low inlet port temperature, only aromatics or highly unsaturated aliphatic compounds of high molecular weight very effectively form solid carbon through pyrolysis. Thus, decreases in FSN have been found under all operating conditions when the EGR temperatures drop from normal EGR temperatures (see *Page 48*) to 100 °C at nominal power and 50 °C at the other operating points (see **Figure 5.25 c**). Furthermore, at low load, the initial temperature drops in conjunction with a decrease of EGR temperature. Consequently, the initial temperature is lower than at higher load. Changing percentage of FSN with EGR temperature falls when engine run at higher load (see **Figure 5.25 c**).

5.1.3.3 Hydrocarbon

Hydrocarbon (HC) emission arises from unburned and partly burned hydrocarbons and related products of thermal cracking. The origin of these components can be both the fuel and the lubricants. Incomplete hydrocarbon combustion occurs because of part ignition of the overall combustion chamber volume and fuel deposits on the wall. Other reasons are residual fuel in the dead spaces such as gaps in the cylinder head seal, valve seats, piston rings and squish areas. Hydrocarbons from lubricant and absorption of fuel molecules in the lubricant film of the cylinder barrel and at sites with impurities also lead to increases of HC emission. Important source of HC emission is fuel in sac hole of nozzle. At the end of the fuel injection process, the injector sac volume is left filled with fuel. As the combustion and expansion processes proceed, this fuel is heated and vaporizes, and enters the cylinder at low velocity through the nozzle holes. This fuel vapor (possibly large drops of fuel also) will mix relatively slowly with air and may escape the primary combustion process. The formation of partly combusted hydrocarbons essentially depends on the temperature and oxygen content and to a lesser extent on the molecular structure.

With the exception of O₂-Diesel and B10, most alternative fuels produce lower HC emission trends than diesel fuel (see **Figure 5.26**). HC emission increases more modestly with B10

than diesel fuel because the low RME content in the blend causes combustion similar to diesel fuel. In addition, residuals deposit on the wall or the bowl of the piston because a long life engine was used and because of EGR effects. As a result, misfiring and quenching phenomena can occur in the wall region in the cylinder. Ignition quality decreases when O₂-Diesel is used because it has a lower energy content and lower cetane number than diesel fuel. Moreover, the experimental fuel temperature of O₂-Diesel is low. This causes a reduction of the flame temperature peak. Thus, HC emissions increase moderately with O₂-Diesel (see **Figure 5.26**). Increased temperature peaks occur for the same reasons, i.e. the cetane number very high and the ignitability better. Consequently, HC concentrations decrease significantly when GTL and its blend are used. The HC emissions of O₂-1 and O₂-2 differ negligibly. HC emission levels vary widely depending on the operating conditions and different HC formation mechanisms are of varying importance for different operating modes. Engine idling and light load operation produces significantly higher HC emissions than full load operation because the initial temperature and pressure are higher at full load. Hence, the combustion process is stronger. The physical and chemical properties of RME cause “harsher” and “faster” combustion and increases of flame temperature. HC concentrations decrease when RME and its blends, except for B10, are used (see **Figure 5.26**).

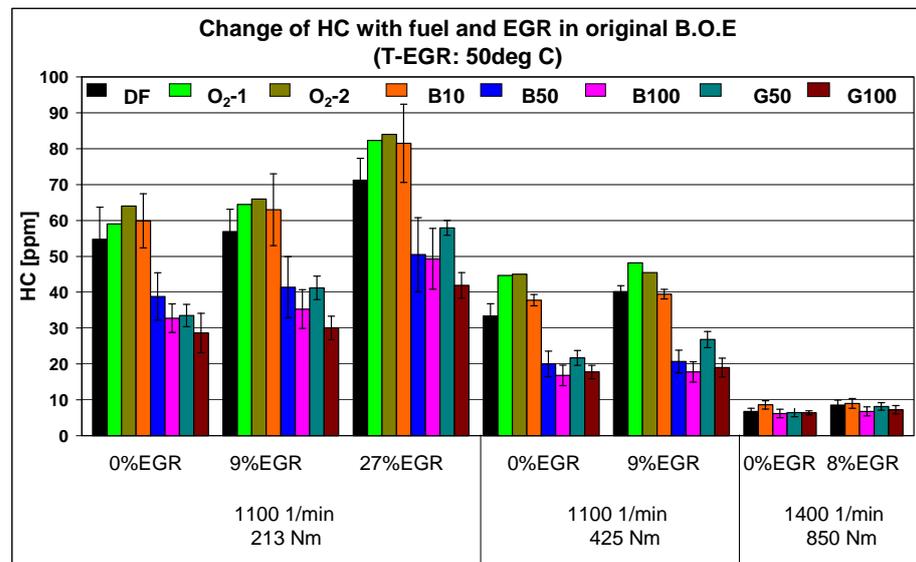


Figure 5.26: Change of HC with fuel and EGR at different operating points

When exhaust gas is recirculated into the inlet manifold, flame temperature peaks and oxygen concentrations decrease, resulting in too little oxygen for complete combustion (especially with throttling in the intake manifold and 27% EGR). Furthermore, fuel consumption increases as EGR ratios increase. This can lead to undermixing (a local overrich mixture) in some regions of the cylinder. Consequently, HC concentrations increase sharply as EGR ratios increase (see **Figure 5.26**). This holds true for all fuels and at all operating points and under all conditions.

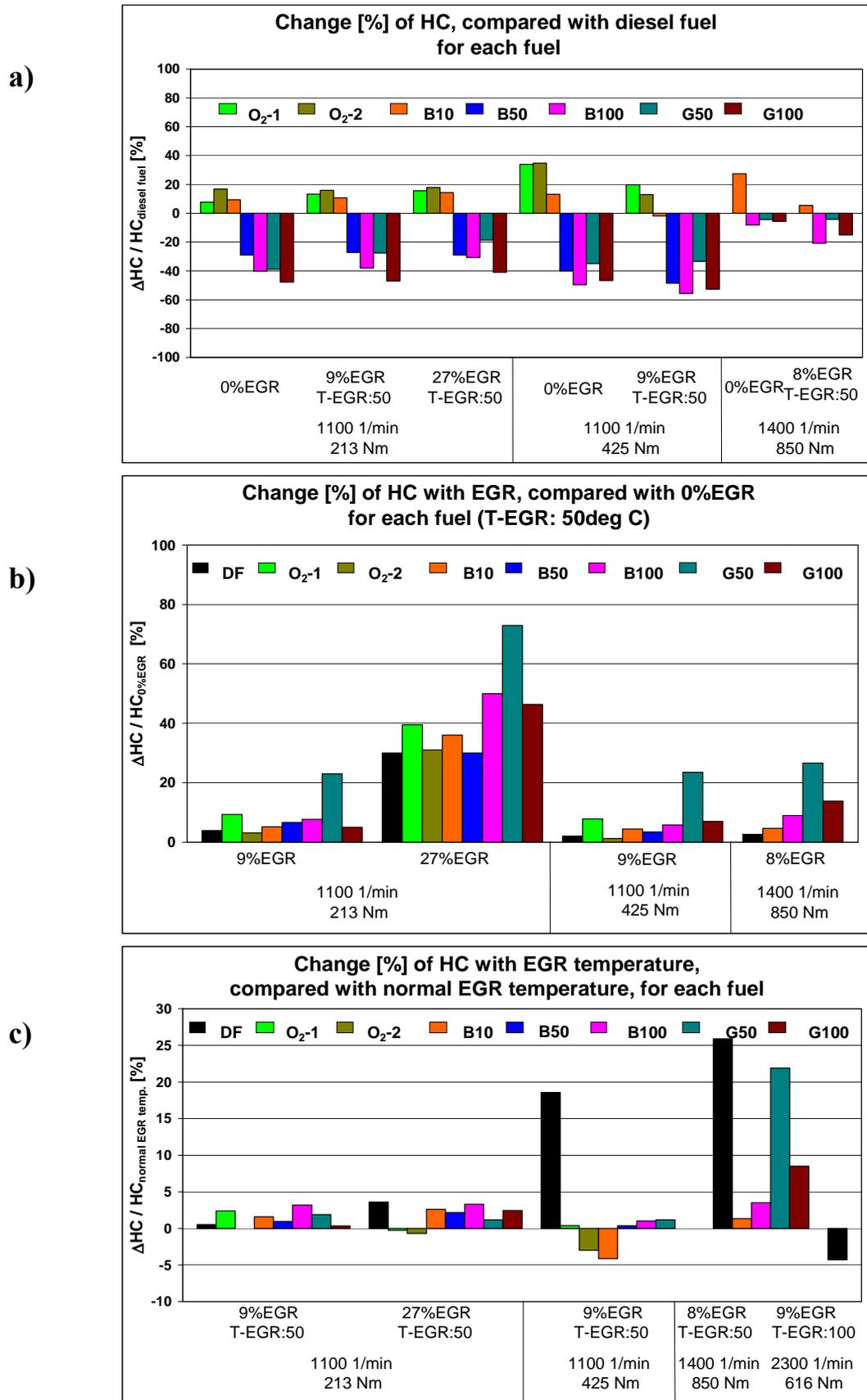


Figure 5.27: Percentage change of HC: **a)** compared with diesel fuel; **b)** compared with 0% EGR; **c)** with decreases of EGR temperature (compared with normal EGR temperature)

HC emission changes significantly with B100, G100 and G50 (see **Figure 5.27**). Maximum changing percentages (over 50%) occur with B100 and G100 when the EGR ratio is about 9% and the engine runs at higher part load (see **Figure 5.27 a**). Compared with diesel fuel, the maximum increased percentages are over 20% (about 8 ppm) with B10 at maximum torque and without EGR and over 30% (over 30 ppm) with O₂-Diesel at higher part load and without EGR (see **Figure 5.27 a**).

HC emission also changes significantly when B100, G100 and G50 are used instead of diesel fuel when exhaust gas is recirculated into intake manifold (see **Figure 5.27 b**). Changing percentages increase when EGR ratios increase. Thus, the maximum changing percentage (over 70%) appears when EGR is 27% and G50 is used. With the same EGR ratio (around 9%), engine load modestly influences changing percentages (compared with 0% EGR) of HC emission.

With the exception of a few points, experimental results also confirm HC emission increases when EGR is cooled more strongly (EGR temperatures drop from normal EGR temperatures (see *Page 48*) to 100 °C at nominal power and 50 °C at the other operating points). Diesel fuel has the maximum changing percentage (over 25%, 10 ppm) at maximum torque. Moreover, changing percentages also increase greatly with G100 and G50 at maximum torque (about 8% and over 20% respectively) (see **Figure 5.27 c**). They change negligibly (less than 4%) at other operating points and under other conditions. Clearly, different EGR temperatures do not change HC emissions significantly.

5.1.3.4 Emissions at Idle

Idle is a special operating point of an engine. The flame temperature is particularly low and the speed is low. Thus, emissions, especially HC, are detectable when an engine idles.

Just like emissions trends at other operating points, NO_x emission increases with all alternative fuels except G100 and G50, compared with diesel fuel, for the reasons already set forth. B100 has the highest NO_x concentration, followed by O₂-Diesel and B10 (see **Figure 5.28 a**). NO_x emission decreases significantly when GTL and its blend are used instead of diesel fuel because of the ignition and combustion characteristics of GTL and its blends.

With the exception of HC of B10, the FSN and HC emissions of alternative fuels in the experimental results exhibit positive trends compared with diesel fuel (see **Figure 5.28 b** and **c**). As at other operating points, the FSN drops with all alternative fuels. B100 has the lowest FSN, followed by O₂-Diesel, B10, G100 and G50. GTL and its blends have an FSN between that of diesel fuel and RME and its blends (see **Figure 5.28 b**). Section 5.1.3.2 provides explanations of the FSN trends with different alternative fuels.

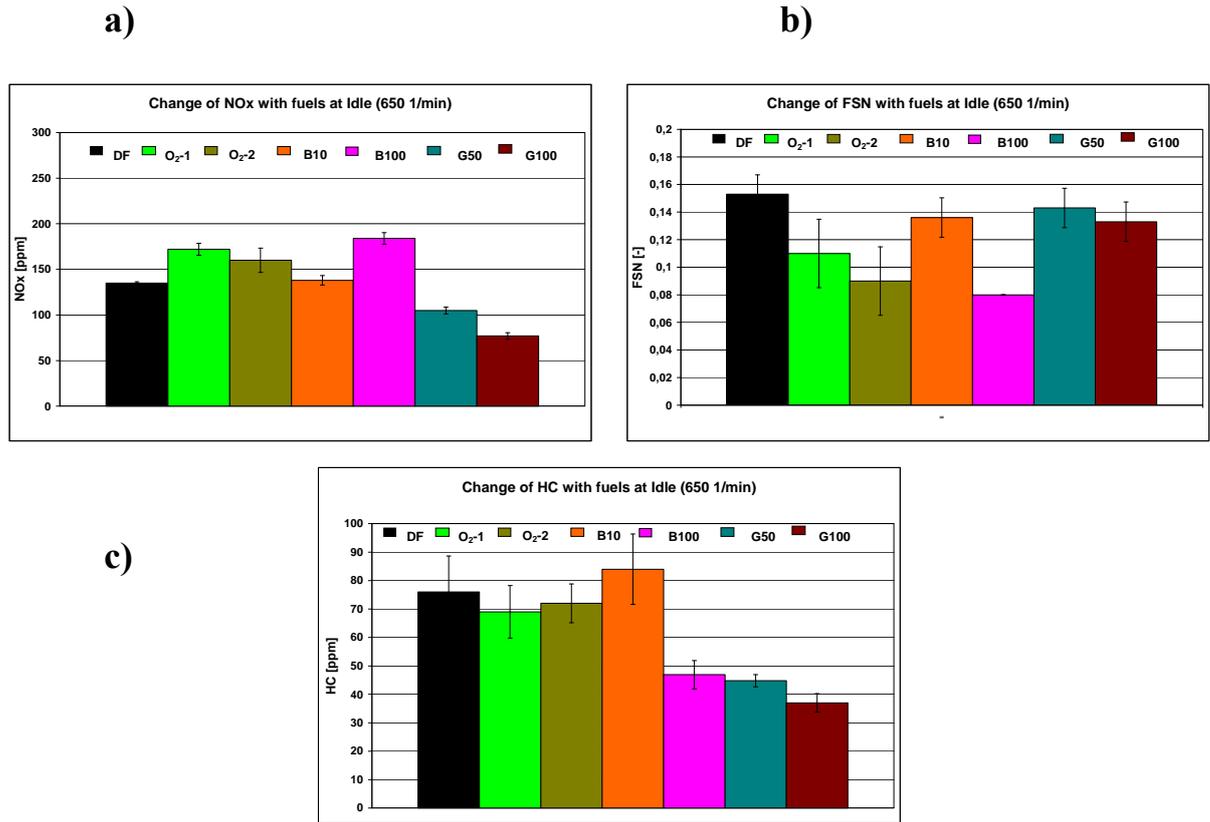


Figure 5.28: Emissions at Idle: **a)** NO_x emission; **b)** FSN; **c)** HC emission

Similar to FSN and HC trends at other operating points, HC concentration decreases with most alternative fuels, except for B10, for the reasons outlined in section 5.1.3.3 (see *Figure 5.28 c*). G100 has the lowest HC concentration, followed by G50, B100, O₂-Diesel, diesel fuel and B10. B10 has a higher HC concentration than diesel fuel due to misfiring caused by residuals deposited on the cylinder and quenching phenomenon at the wall. In the experimental results, GTL and its blend exhibited decreased emission trends compared with diesel fuel and other alternative fuels (see *Figure 5.28 b* and *c*).

5.2 Influences of Different Injection Timing (B.O.E)

Injection system characteristics are some of the most important factors that influence the emissions and performance of DI diesel engines. Injection systems for modern, low emission diesel engines are designed with the following objectives in mind:

- Potential for high injection pressure
- Capability to shape injection rate
- Flexibility controlling injection parameters (pressure level, pressure rise, start of injection, end of injection, etc.)
- Capability of pilot injection and post-injection with flexibility of control (selecting start and end, pressure level, etc.)

Along with such injection characteristics as injection spray characteristics and the shape of the rate of injection over the time curve, the start of injection strongly affects mixture formation and consequently DI diesel combustion. Changing injection timing affects an engine's period of ignition delay. Under normal engine conditions (fully warmed engine at low to medium speed), the minimum delay occurs with the start of injection at about 10 to 15 deg CA BTDC [2]. Increases in the delay with earlier or later injection timing occur because the air temperature and pressure change significantly close to TC. If injection starts earlier, the initial air temperature and pressure are lower, thus increasing the delay. If injection starts later (closer to TDC) the temperature and pressure are initially slightly higher but then decrease as the delay proceeds [2]. Changes in the period of ignition delay related to injection timing significantly affect mixture formation and thus the engine's process.

5.2.1 Cylinder Pressure and Heat Release Rate

- **Cylinder Pressure**

If injection starts later (closer to TDC) and especially very late, temperature and pressure are initially slightly higher but then decrease. Consequently, maximum cylinder pressure decreases (see *Figure 5.29*).

Maximum cylinder pressure drops drastically when injection timing is retarded. Experimental results demonstrate that, at maximum torque, maximum cylinder pressure increases from 105 bar to 114 bar (8.5 %) with diesel fuel, from 106 bar to 115 bar (8.5%) with B10 and from 107 bar to 117.8 bar (10%) with B100 when injection starts about 4 deg CA earlier than the original injection timing (see *Figure 5.29 a*). At lower part load, maximum cylinder pressure increases from 60.5 bar to 65.6 bar (8.4 bar) with diesel fuel, from 61 bar to 67 bar (9.8%) with B10, from 62 bar to 67 bar (8%) with B50, from 63.7 bar to 68.7 bar (8%) with B100, from 61.8 bar to 68.5 bar (10.8%) with O₂-1 and from 63bar to 68.6 bar (8.8%) with O₂-2 when B.O.E changes from -12 deg CA BTDC to -7.7 deg CA BTDC (original B.O.E) (see *Figure 5.29 b*). Changing percentages of maximum cylinder pressure with all fuels reach up to 8%-10% (near equal ratios) at most operating points when injection starts about 4 deg.CA earlier than the original injection timing. By contrast, the maximum cylinder pressure decreases significantly from 105 bar to 94 bar (10.5%) with diesel fuel, from 106 bar to 93.8 bar (11.5%) with B10 and from 107 bar to 92.5 bar (13%) with B100 at maximum torque when injection start about 6 deg CA later than the original injection timing (see *Figure 5.29 a*). The maximum cylinder pressure decreases from 60.5 bar to 53.5 bar (11.5%) with diesel fuel, from 61 bar to 54 bar (11.4%) with B10, from 62 bar to 55 bar (11.3%) with B50, from 63.7 bar to 56 bar (12%) with B100, from 61.8 bar to 54.6 bar (11.6%) with O₂-1 and from 63 bar to 54.8 bar (13%) with O₂-2 at lower part load when injection timing changes from the original angle to -2 deg CA BTDC (see *Figure 5.29 b*). Changing percentages of maximum

cylinder pressure of all fuels are approximate (11%-13%) when B.O.E starts 6 deg CA later than the original B.O.E. Since the period of ignition delay increases, combustion starts later and the initial temperature and pressure drop because the piston descends during the expansion stroke. As a result, maximum cylinder pressure decreases when injection is retarded. The trends of maximum cylinder pressure are the same with different fuels and different injection timing (earlier, original or later). For the reasons expounded in section 5.1.1.2, B100 has the highest maximum cylinder pressure, followed by B50, O₂-Diesel, B10 and diesel fuel. The experimental results exhibit decreased trends for equal ratios of maximum cylinder pressure with all fuels and at all operating points when injection timing starts later.

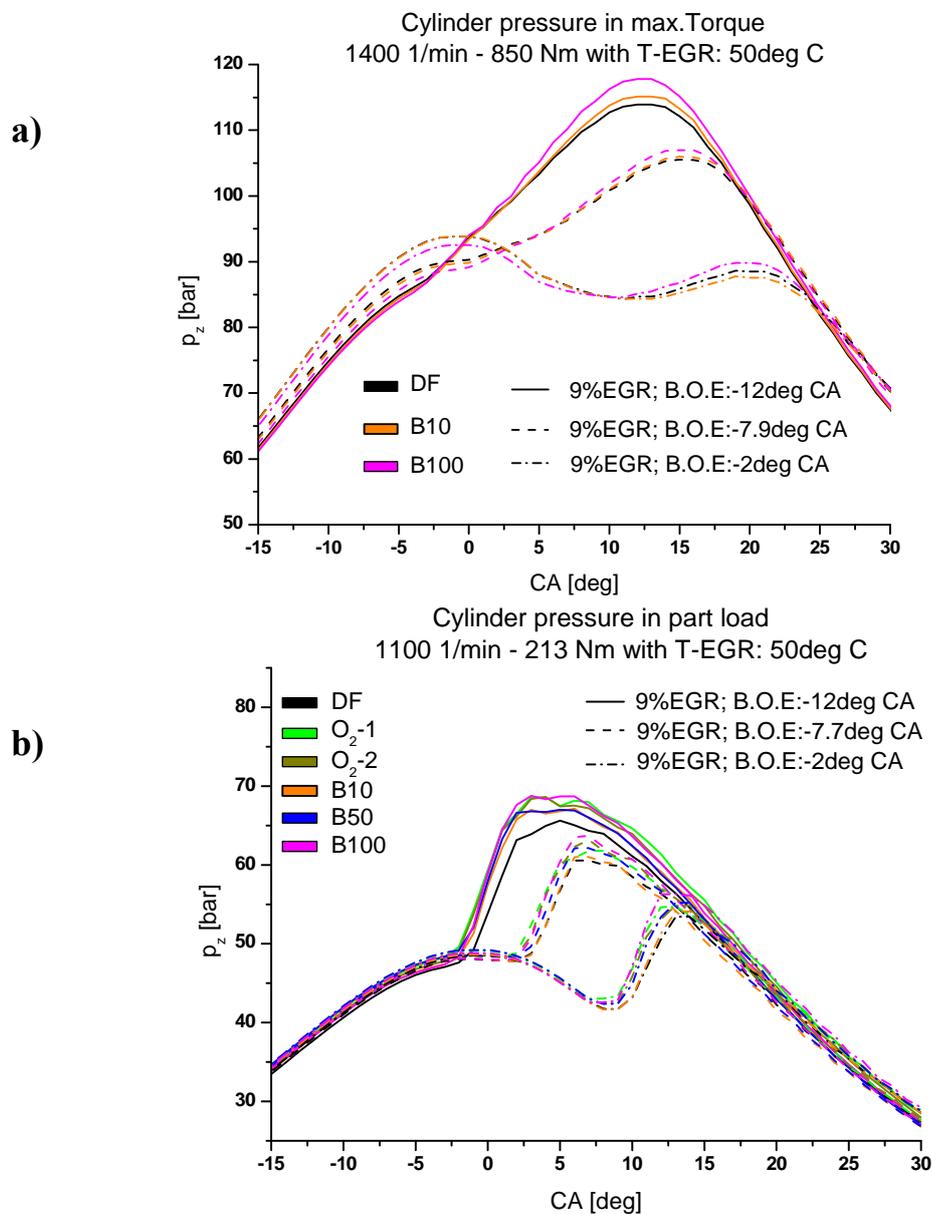


Figure 5.29: Cylinder pressure over CA at variable B.O.E: **a)** at maximum torque (1400 1/min - 850 Nm); **b)** at lower part load (1100 1/min - 213 Nm)

• Heat Release Rate

As opposed to cylinder pressure trends, the maximum heat release rate increases when injection timing starts later. The developments of the maximum heat release rate at two operating points are presented in **Figure 5.30 a** and **b**. The behavior of the maximum heat release rate in a premixed combustion phase essentially depends on the periods of ignition delay. When injection starts later, ignition delay periods increase at most operating points and under most conditions (see **Figure 5.31**), which leads to more mixing of fuel with air for combustion. In addition, when B.O.E is retarded, the initial temperature and pressure in the cylinder are very high because of it is closer to TDC and, hence, the combustion rates increase. Consequently, the maximum heat release rate increases. Changing percentages of maximum heat release are higher at low loads. When injection starts earlier (4 deg CA earlier than original B.O.E), maximum heat release rates drop from 117 kJ/m³ deg CA to 104.5 kJ/m³ deg CA (10.7%) with diesel fuel, from 118 kJ/m³ deg CA to 107 kJ/m³ deg CA (9.3%) with B10 and from 119 kJ/m³ deg CA to 109 kJ/m³ deg CA (8.4%) with B100 at maximum torque (see **Figure 5.30 a**). At lower part load, they decrease from 61 kJ/m³ deg CA to 56.8 kJ/m³ deg CA (7%) with diesel fuel, from 64 kJ/m³ deg CA to 57.4 kJ/m³ deg CA (10%) with B10, from 64.6 kJ/m³ deg CA to 59.4 kJ/m³ deg CA (8%) with B50, from 67 kJ/m³ deg CA to 62 kJ/m³ deg CA (7.5%) with B100 and from 65.7 kJ/m³ deg CA to 62.3 kJ/m³ deg CA (5%) with O₂-2 (see **Figure 5.30 b**). The maximum heat release rates drop significantly with DF and RME and its blends (7%-10%) and decrease modestly with O₂-Diesel (4%) when B.O.E starts earlier (4 deg CA earlier than original B.O.E). They increase from 117 kJ/m³ deg CA to 126.5 kJ/m³ deg CA (8%) with diesel fuel, from 118 kJ/m³ deg CA to 126.7 kJ/m³ deg CA (7.3%) with B10 and from 119 kJ/m³ deg CA to 128 kJ/m³ deg CA (7.5%) with B100 at maximum torque when injection starts later (6 deg CA later than original B.O.E) (see **Figure 5.30 a**).

At lower part load, heat release rates increase from 61 kJ/m³ deg CA to 84 kJ/m³ deg CA (37%) with diesel fuel, from 64 kJ/m³ deg CA to 85.7 kJ/m³ deg CA (34%) with B10, from 64.6 kJ/m³ deg CA to 88.7 kJ/m³ deg CA (37%) with B50, from 67 kJ/m³ deg CA to 92 kJ/m³ deg CA (37%) with B100, from 60.7 kJ/m³ deg CA to 84.7 kJ/m³ deg CA (39%) with O₂-1 and from 65.7 kJ/m³ deg CA to 88 kJ/m³ deg CA (34%) with O₂-2 when injection starts later (6 deg CA later than original B.O.E) (see **Figure 5.30 b**). The maximum heat release rates increase with all fuels and equal percentages (34% - 37%) and at all operating points when B.O.E changes about 6 deg CA later than original B.O.E

The same trends of the maximum heat release rate are exhibited with all fuels in different B.O.E. B100 has the highest maximum heat release rate, followed by B50, B10, O₂-2, O₂-1 and diesel fuel in all different B.O.E. Section 5.1.1.3 explains why the maximum heat release rate changes with different fuels.

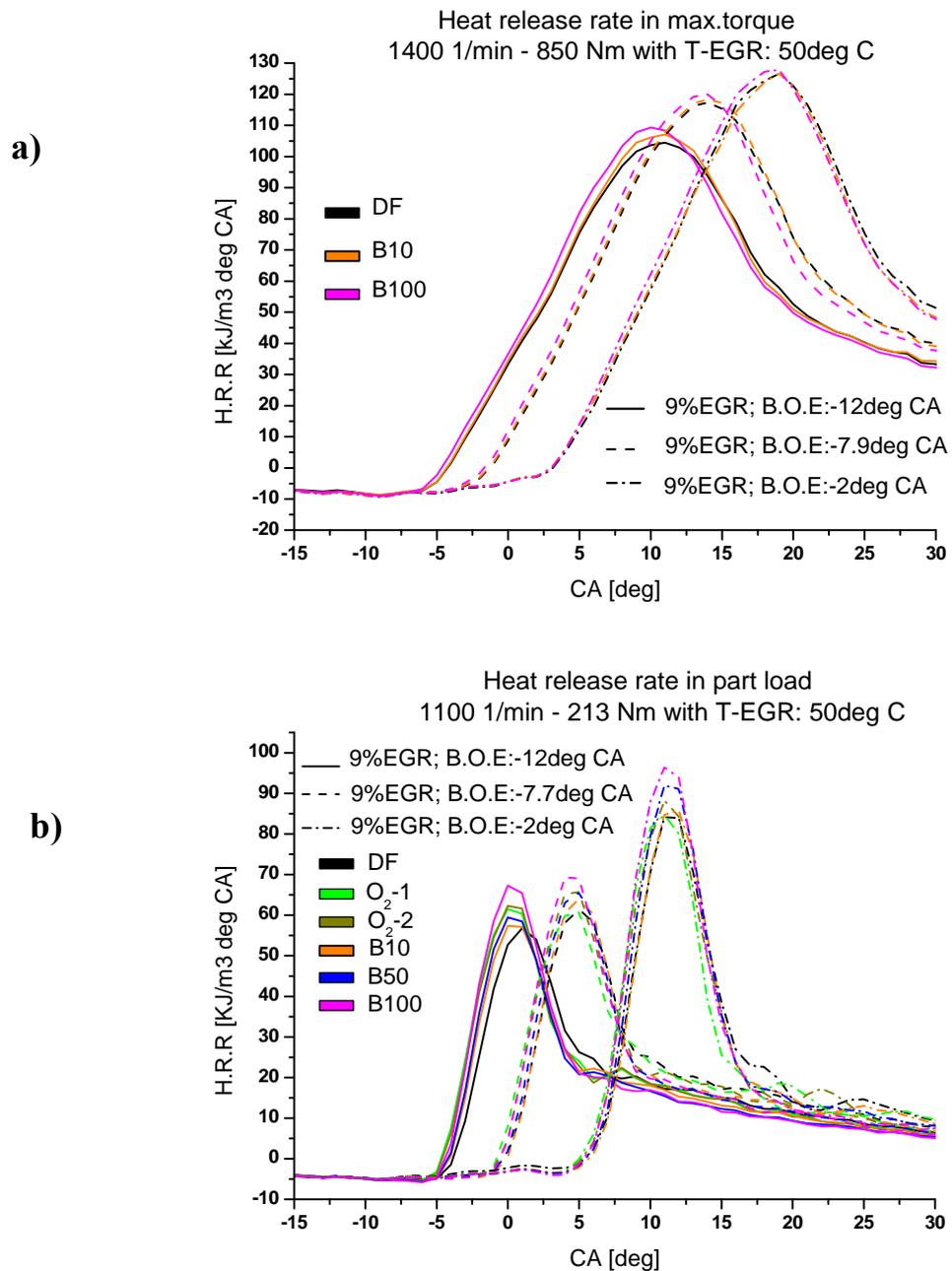


Figure 5.30: Heat release rate over crank angle at variable B.O.E: **a)** at maximum torque (1400 1/min - 850 Nm); **b)** at lower part load (1100 1/min - 213 Nm)

5.2.2 Ignition Delay

Figure 5.31 plots how, at two part load points when B.O.E is retarded (-2 deg CA BTDC at lower part load and 0 deg CA BTDC at higher part load - C point in the graph), injection starts at a point very near TDC or in the expansion stroke. Thus, ignition delay is initially slightly lower but then increases modestly (see **Figure 5.31**). At maximum torque when

B.O.E is also retarded to -2 deg CA BTDC, ignition delay decreases (see **Figure 5.31**) because the speed (1400 1/min), initial temperature and pressure are higher than at part load. This is why the changes in the maximum heat release rate decrease at higher loads (see **Figure 5.30**). The trends of the period of ignition delay are similar without EGR. Similar tendencies of ignition delay of most alternative fuels with different B.O.E have been also found due to the reasons presented in section 5.1.1.7.

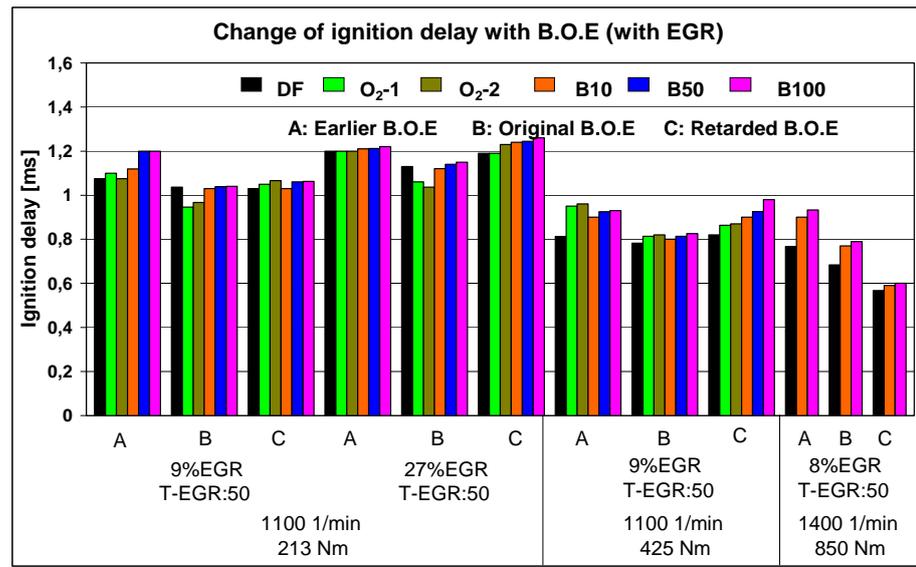


Figure 5.31: Change of ignition delay with different B.O.E (with EGR)

As presented in **Figure 5.31**, ignition delay increases about 0.1ms with all fuels except at lower part load with 9% EGR and 50°C of EGR temperature when injection starts earlier at all operating points and increases about 0.1 ms with all fuels when injection starts later except at maximum torque. At maximum torque, ignition delay decreases about 0.1 ms when injection starts later. At lower part load, 9% EGR and 50°C of EGR temperature, the ignition delay changes very little for all fuels when the original injection timing is changed to -2 deg CA BTDC (see **Figure 5.31**). Ignition delay is the longest with B100 (about 0.2 ms longer than diesel fuel), followed by B50, B10, O₂-2, O₂-1 and diesel fuel (see **Figure 5.31**). These trends are also exhibited in different B.O.E. Section 5.1.1 explains the changes with different fuels.

5.2.3 Specific Fuel Consumption

Specific fuel consumption increases moderately when injection timing is retarded (see **Figure 5.32**). Specific fuel consumption increases around 2.5% with diesel fuel, 3.5% with O₂-1 Diesel and 2% with B100 at lower part load (1100 1/min - 213 Nm) when injection starts later. Compared with original B.O.E, a higher increased percentage (about 4.5% in the maximum case) has been found with B100 when the engine runs at maximum torque (1400 1/min - 850 Nm). With maximum increased percentage (about 4.5%) of specific fuel

consumption, drawbacks of specific fuel consumption are acceptable through when injection starts later (about 6 deg CA later than original B.O.E).

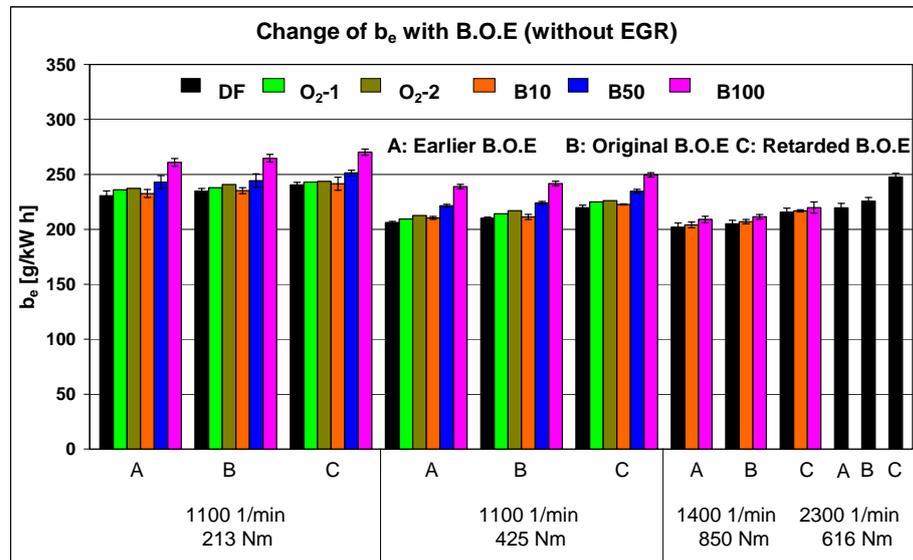


Figure 5.32: Change of b_e with different B.O.E (without EGR)

Trends increase similarly with all fuels and at all operating points and under all conditions (see **Figure 5.32**) and with EGR. Compared with diesel fuel, the changing percentages of b_e of all alternative fuels are similar in different B.O.E. Section 5.1.1.10 explains the reasons ratios change when specific fuel consumption changes with different fuels.

5.2.4 Spectrum of Cylinder Pressure

Just as the trends of cylinder pressure, cylinder pressure levels decrease significantly when injection starts later. The trends are presented in **Figure 5.33**.

In low frequency ranges (below 210 Hz), cylinder pressure levels drop when injection timing is retarded because their trends essentially depend on maximum cylinder pressure. Trends of cylinder pressure levels reverse in higher frequency ranges (above 210 Hz). Cylinder pressure levels increase when injection starts later because, at this stage, cylinder pressure levels essentially depend on the cylinder pressure gradient. Compared with original injection timing at maximum torque (see **Figure 5.33 a**), cylinder pressure levels at low frequency (below 210 Hz) drop approximately 9 dB with diesel fuel and approximately 14 dB with B100, approximately 10 dB with diesel fuel and approximately 7 dB with B100 at lower part load (see **Figure 5.33 b**). In high frequency ranges (above 210 Hz), cylinder pressure levels differ less when injection starts later because the changes in the cylinder pressure gradient are small.

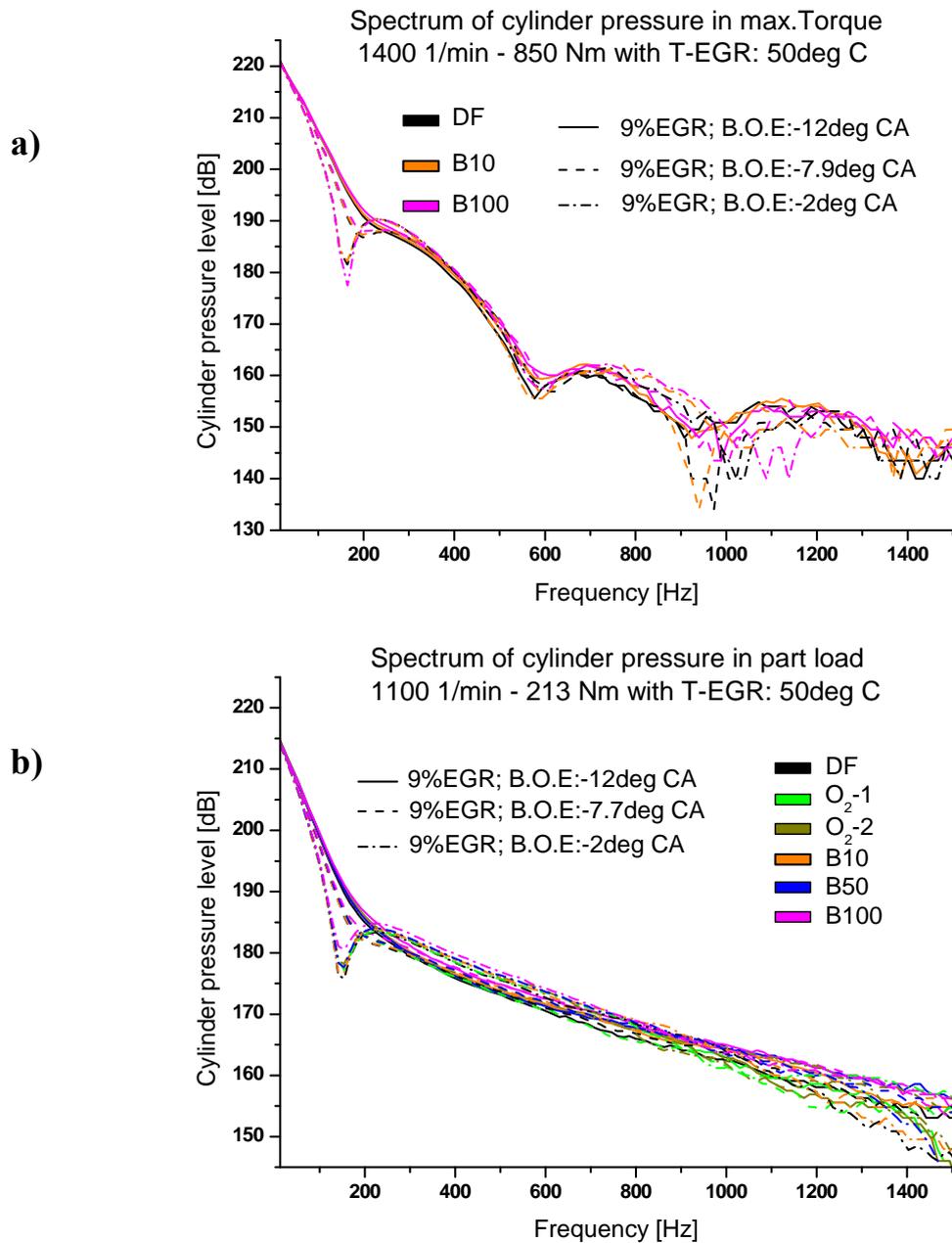


Figure 5.33: Spectrum of cylinder pressure with different B.O.E: **a)** at maximum torque (1400 1/min - 850 Nm); **b)** at lower part load (1100 1/min - 213 Nm)

In all different B.O.E, B100 has the highest cylinder pressure level, followed by B50, B10, O₂-Diesel and diesel fuel (see *Figure 5.33 a and b*). Section 5.1.2 explains these trends.

5.2.5 Emissions

Injection timing strongly influences engine emissions, especially NO_x emissions, because the combustion temperature and pressure change significantly when the injection timing changes. These changes cause combustion in the cylinder to change strongly, thus causing emissions to change.

- NO_x

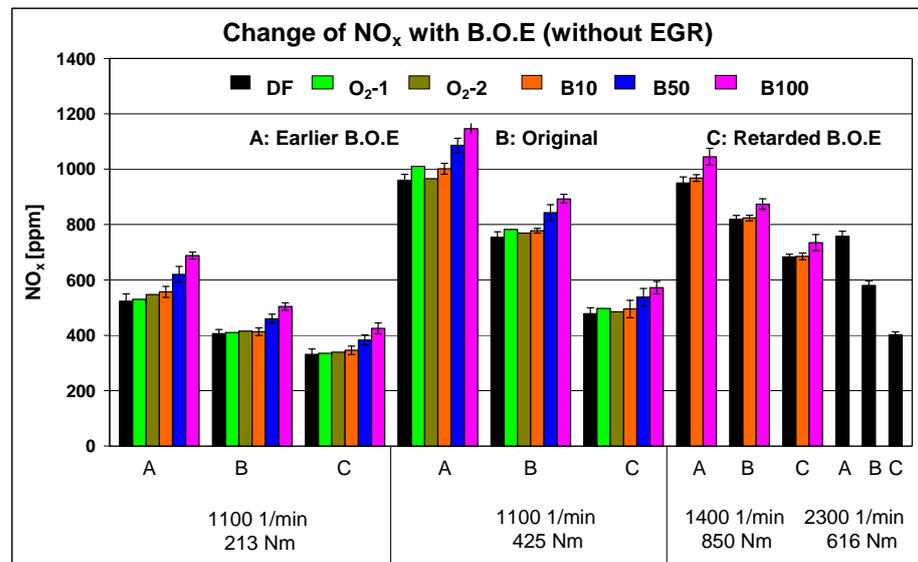
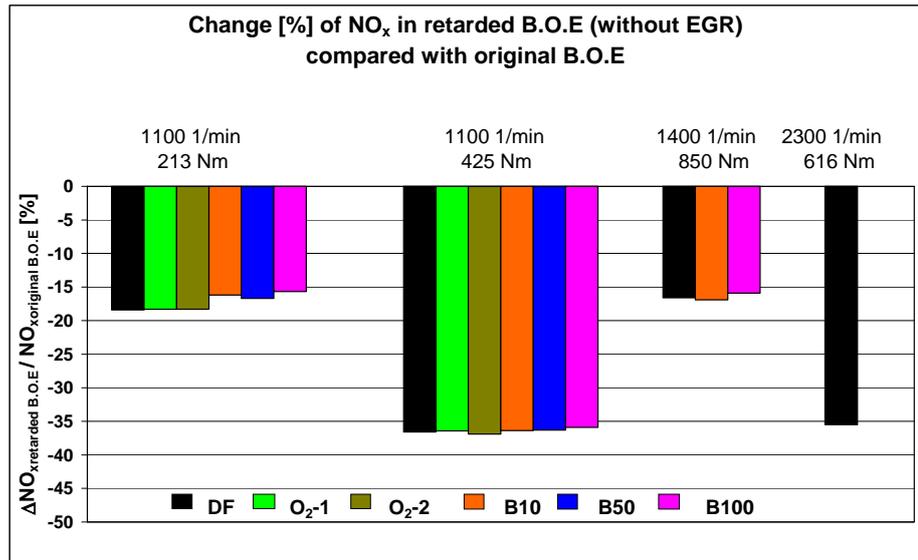


Figure 5.34: Change of NO_x with different B.O.E (without EGR)

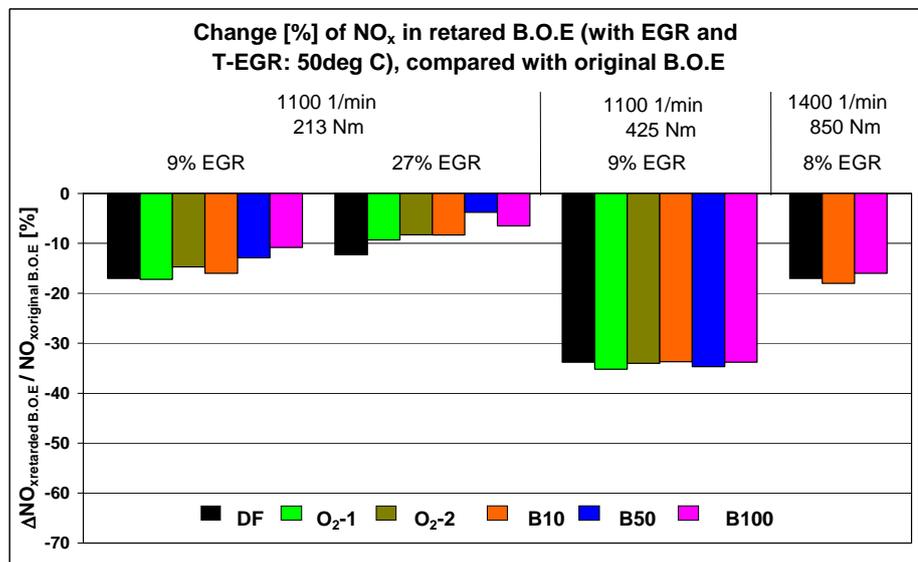
Retarding injection timing is another method for reducing NO_x emissions. Although the retarded injected fuel increases the ignition delay, it causes a greater fraction of combustion to occur later in the expansion stroke. Since the majority of NO_x is formed in premixed combustion and at high temperatures, the net amount of NO_x formed during retarded fuel injection decreases.

Compared with original B.O.E, at higher part load and nominal power (see *Figure 5.34* and *Figure 5.35 a*), experimental results display maximum decreases of NO_x emission (over 35%) when injection starts later. At lower part load and maximum torque, the decreased percentage reaches up to over 15%. As injection timing is retarded, the combustion process is retarded. NO_x formation occurs later and concentrations are lower since peak temperatures are lower. Experimental results exhibit the same percentage changes with all fuels.

a)



b)



c)

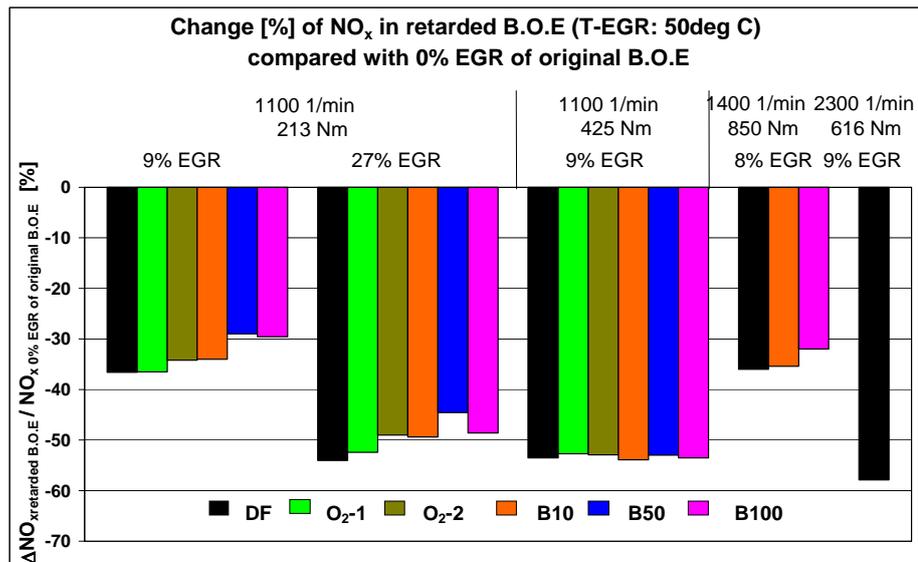


Figure 5.35: Percentage change of NO_x: a) compared with original B.O.E (without EGR); b) compared with original B.O.E (with EGR); c) compared with 0% EGR of original B.O.E

When exhaust gas recirculates into the intake manifold, NO_x emission decreases with later injection timing (see **Figure 5.35 b**). Changing percentages of NO_x of most fuels are similar at all operating points and under all conditions with and without EGR (see **Figure 5.35 a** and **b**). When EGR increases from 9% to 27% and injection starts later, changing percentages of NO_x also decrease (see **Figure 5.35 b**). These trends occur with all fuels.

NO_x emissions also drop when EGR is combined with variable retarded injection timing (see **Figure 5.35 c**). Changing percentages decrease more at higher part load and nominal power (over 55%). With the same EGR ratio (around 9%), the changing percentage of NO_x reaches up to over 35 % at lower part load and maximum torque (see **Figure 5.35 c**). When EGR increases from 9% to 27% and injection timing is retarded, changing percentages of NO_x increase approximately 20% with all fuels (see **Figure 5.35 c**). This demonstrates that combining EGR and retarded injection timing can significantly reduce NO_x formation. The changes of NO_x emission with different fuels in retarded B.O.E are the same as in other B.O.E.

- Soot

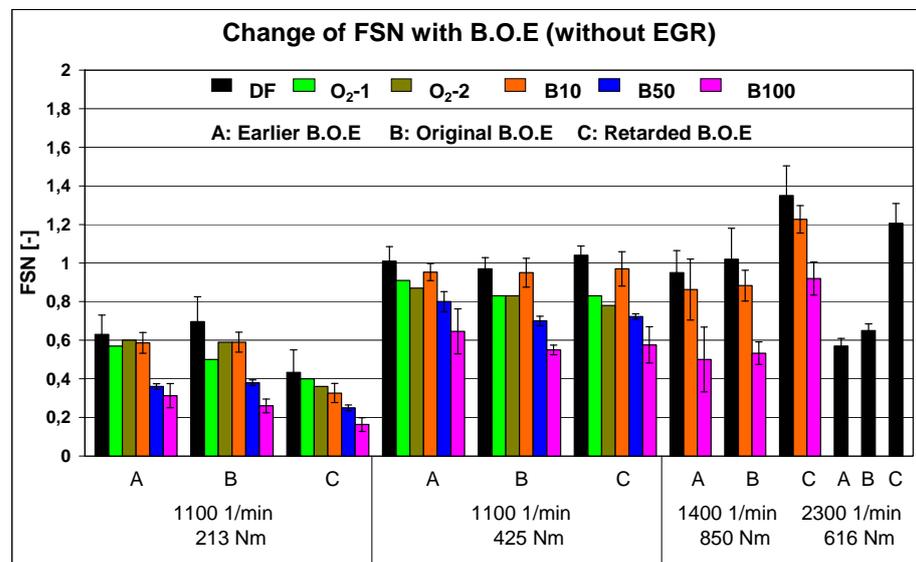


Figure 5.36: Change of FSN with different B.O.E (without EGR)

Soot are formed in the diesel combustion environment at temperatures between 1000 and 2800 K, at pressures of 50 to 100 bar and with sufficient air overall to completely burn all the fuel [2]. The C:O ratio critical for soot formation increases as the temperature increases but is only weakly dependent on pressure. The conditions of soot and NO_x formation conflict. Conditions such as high injection pressure and injection droplet parameters, which cause NO_x and thus flame temperature peaks to increase, also cause soot or hydrocarbon to decrease.

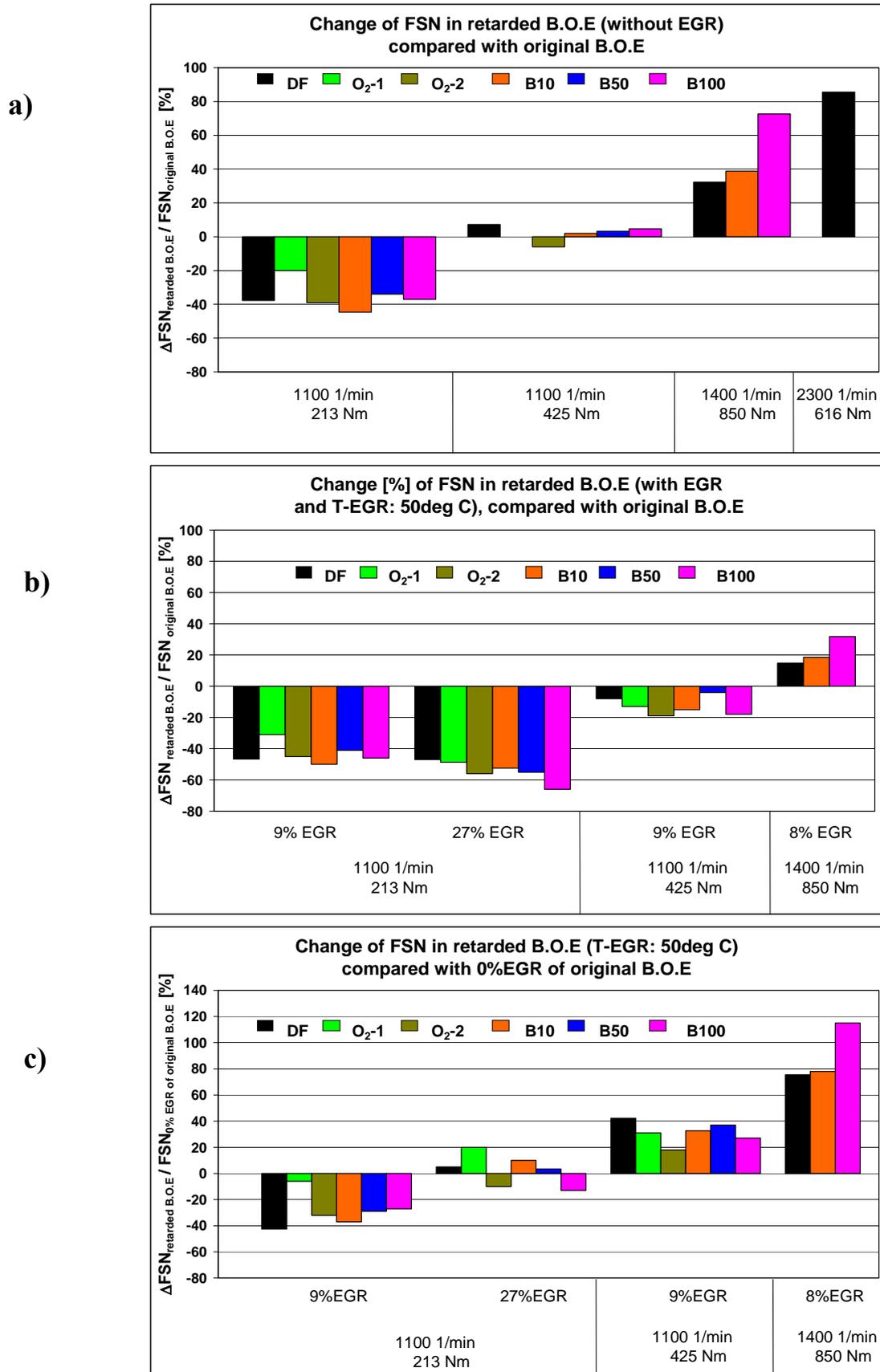


Figure 5.37: Percentage change of FSN: **a)** compared with original B.O.E (without EGR); **b)** compared with original B.O.E (with EGR); **c)** compared with 0% EGR of original B.O.E

The experimental results indicate soot concentration (indicated by FSN) increases when injection is retarded, except at lower part load. At lower part load, low speed, longer cycle duration and thus a better mixture process and low injection mass rate, decreases in the critical C:O ratio cause FSN to decrease (see *Figure 5.36*). Changing percentages of FSN increase when the engine runs at higher loads. All fuels had similar trends in the experiments. Compared with diesel fuel, the FSN trends of alternative fuels in retarded injection timing resemble the trends in other injection timing.

When injection is retarded, the FSN decreases at low speed and load and increases at high load (see *Figure 5.37 a and b*). Compared with original B.O.E, the maximum decreased percentage of FSN at lower part load is over 40% (B10) and the maximum increased percentage of FSN at nominal power is over 80% (diesel fuel) (see *Figure 5.37 a and b*). FSN trends are similar with and without EGR (see *Figure 5.37 b*). When EGR increases from 9% to 27%, changing percentages of FSN also increase (see *Figure 5.37 b*).

When EGR and B.O.E are combined, FSN trends decrease at low part load and increase at higher loads with later injection (see *Figure 5.37 c*). The maximum decreased percentage of FSN is around 40% with most fuels and over 110 % with B100 (see *Figure 5.37 c*). Except at lower part load, the FSN increases with retarded injection timing when EGR is combined with variable injection timing (see *Figure 5.37 c*). Changing percentages increase as engine loads increase. This is true for all fuels. Thus, when injection timing is retarded, NO_x emission trends are positive and FSN trends are negative.

• Hydrocarbon

Hydrocarbon emission levels from diesel vary widely depending on operating conditions. A correlation of unburned HC emissions with the length of the ignition delay would be expected [2]. The magnitude of the unburned HC from overlean regions (close to the spray boundary) depends on the amount of fuel injected during the ignition delay and the mixing rate with air during this period. When injection occurs closer to TDC, cylinder temperature and pressure are high initially and then lower because the piston descends. Consequently, conditions for a complete combustion process do not materialize and HC emissions increase.

When injection timing is later, HC increases with most fuels and at most operating points and under most conditions, except when HC decreases at nominal power (see *Figure 5.38 and 5.39 a*). When exhaust gas recirculates, HC increases with later injection timing at all operating points (see *Figure 5.39 b*). Changing percentages of HC with later injection also increase when EGR increases from 9% to 27%. The maximum increase of HC with later injection is over 70% with 27% EGR and at lower part load. Changing percentages of HC are reduced when the engine runs at higher loads (see *Figure 5.39 a and b*).

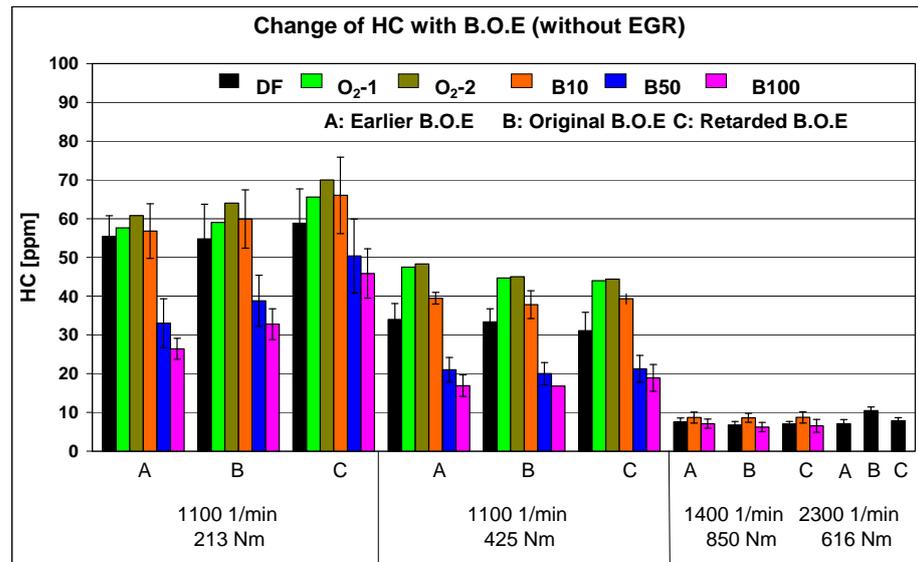


Figure 5.38: Change of HC with different B.O.E (without EGR)

Combining EGR and retarded injection timing causes HC to increase with all fuels and at all operating points and under all conditions (see *Figure 5.39 c*). The highest increased percentages always occur with B100. They are over 155% when B100 is used instead of diesel fuel and the EGR ratio is 27% at lower part load. Changing percentages decrease at higher loads. They increase with higher EGR ratios. These trends hold true for all fuels (see *Figure 5.39 c*).

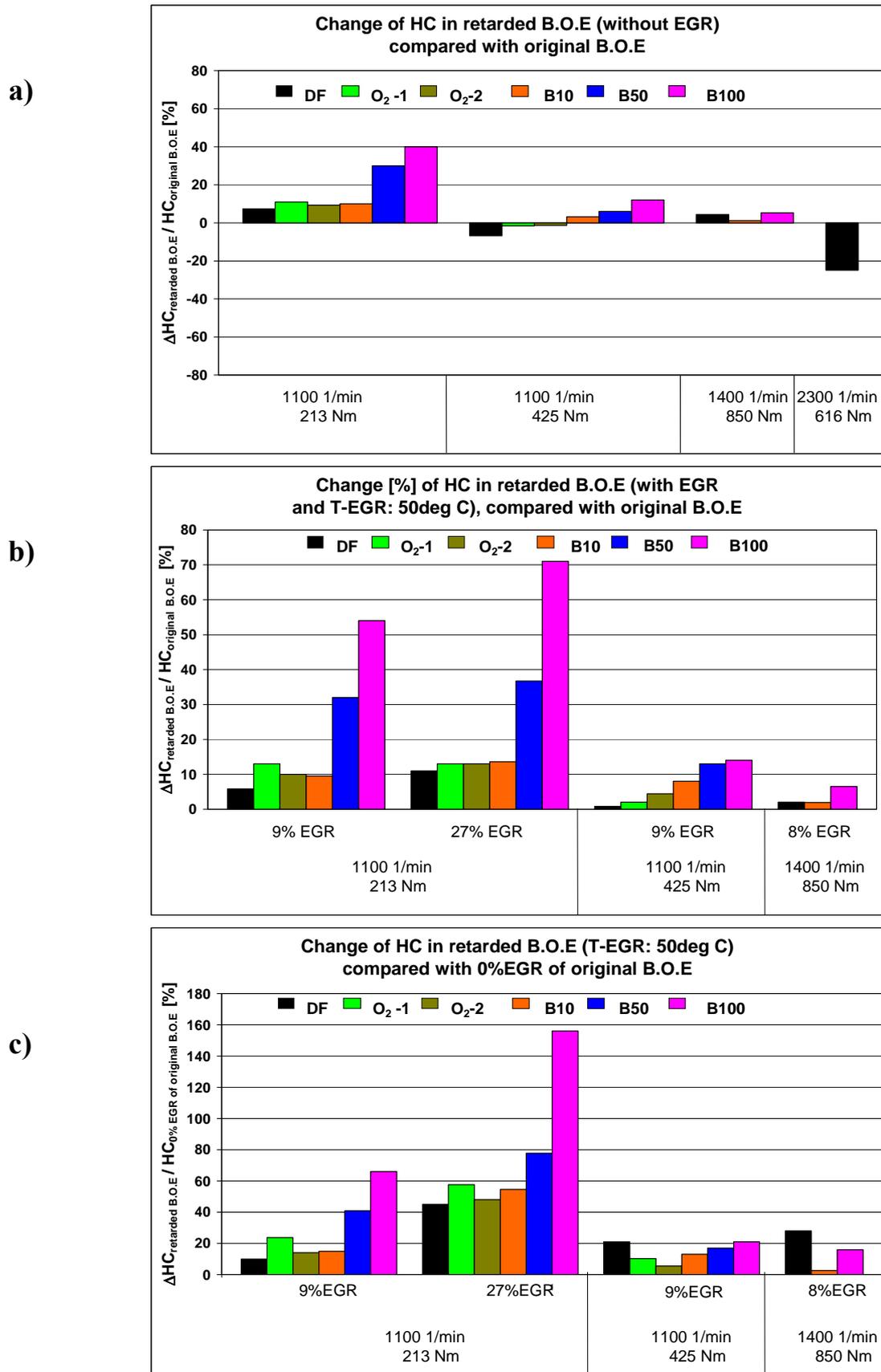


Figure 5.39: Percentage change of HC: **a)** compared with original B.O.E (without EGR); **b)** compared with original B.O.E (with EGR); **c)** compared with 0% EGR of original B.O.E

6 Results of Optimization

Optimizations of operating points with different fuels are based on lower NO_x concentration, a lower FSN and low as possible specific fuel consumption, compared with conventional diesel fuel. Optimization was carried out at three operating points: maximum torque, higher part load and lower part load.

At most operating points, optimized points with different fuels are selected on the basis of lower NO_x emissions, a lower FSN and acceptable higher specific fuel consumption. Of two fuels of O_2 -Diesel (O_2 -1 and O_2 -2), O_2 -1 was only optimized. With B10 at maximum torque, the operating point (8% EGR and EGR temperature: 150 °C) is chosen because NO_x concentration decreases more strongly, than with diesel fuel, although the FSN of the optimized operating point is higher than with diesel fuel (1.32 compared with 1.02) (see **Figure 6.1 a**). With B100 at maximum torque, the operating point (8% EGR and EGR temperature: 150 °C) is also optimized point because its FSN differs modestly from point with lower EGR temperature and b_e is lower. At maximum torque and the other operating points, the optimized operating points for G100 and G50 are without EGR because, although their NO_x concentration are higher (see **Figure 6.1 a, b** and **c**), their FSN decrease significantly compared with diesel fuel and other operating points with EGR.

Similar situations also occur when B10, B100 and O_2 -Diesel are used for optimizations at higher part load (see **Figure 6.1 b**). Optimized operating points with B10 and O_2 -Diesel are 9% EGR and an EGR temperature of 130 °C because, although their NO_x concentration and FSN are higher (see **Figure 6.1 b**), b_e is less than with 9% EGR and in the case of EGR of temperature 50 °C. The optimized operating point with B50 is 9% EGR and an EGR temperature of 50 °C because, although its b_e is higher than in the operating point (9% EGR and 130°C of EGR temperature) (see **Figure 6.1 b**), its NO_x concentration decreases more than with diesel fuel.

Although NO_x emission are reduced strongly with 27% EGR specifically at lower part load, most optimized operating points are chosen without 27% EGR because the FSN and specific fuel consumption are very high (see **Figure 6.1 c**). The optimized operating point with B10 and B100 is 9% EGR and an EGR of temperature 50°C because its NO_x concentration is reduced more than with diesel fuel and another point (9% EGR and an EGR temperature of 100 °C). The optimized operating points with B50 and O_2 -Diesel are 9% EGR and an EGR temperature of 100 °C because, although their FSN increase modestly, the specific fuel consumption is lower than with the operating point (9% EGR and 50 °C of EGR temperature). Just as at maximum torque and higher part load, the optimized operating points of G50 and G100 are without EGR (see **Figure 6.1 c**).

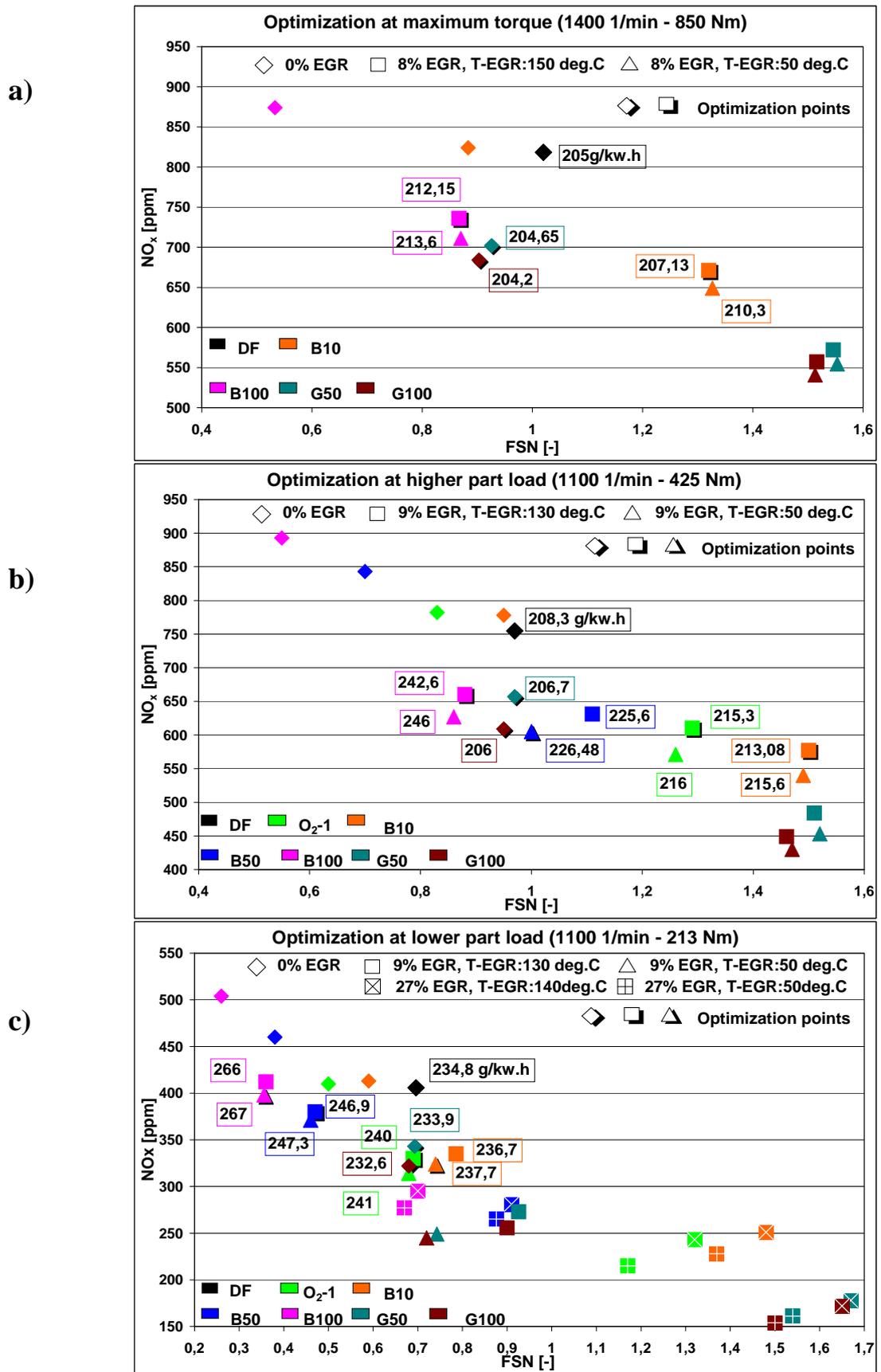


Figure 6.1: Results of optimization with different fuels: a) at maximum torque; b) at higher part load; c) at lower part load

Optimizations are also based on minimum differences of specific energy consumption ($b_e \cdot H_u$, MJ/kWh) with different fuels compared with conventional diesel fuel. The results of the optimizations and the percentage changes of NO_x emission, FSN, specific fuel consumption and specific energy consumption compared with diesel fuel are presented in **Table 6.1**.

Table 6.1: Percentage change of optimization parameters, compared with diesel fuel

	Fuels	EGR ratio [%]	EGR temp. [°C]	NO _x [%]	FSN [%]	b _e [%]	Specific Energy Consumption [%]
Maximum torque 1400 1/min 850 Nm	B10	8	150	-17.9	+29	+1	+1.9
	B100	8	150	-10	-15	+3.5	-8
	G50	0	-	-14.2	-9.2	-0.2	+0.43
	G100	0	-	-16.4	-11.5	-0.4	+1.15
Higher part Load 1100 1/min 425 Nm	O₂ fuel	9	130	-19	+24	+3.3	+0.14
	B10	9	130	-23.5	+35	+2.3	+1.5
	B50	9	50	-19.8	+3	+8.7	+0.96
	B100	9	130	-12.5	-9.2	+16	+2.5
	G50	0	-	-12.9	0	-0.74	+0.16
	G100	0	-	-19	-2	-1.1	+0.43
Lower part Load 1100 1/min 213 Nm	O₂ fuel	9	100	-18.7	-0.9	+2.2	-0.96
	B10	9	50	-20	+6	+1.2	+0.5
	B50	9	100	-6.3	-32	+5	-2.35
	B100	9	50	-2	-48	+13	+0.15
	G50	0	-	-15	-0.4	-0.4	+0.216
	G100	0	-	-20.6	-2.3	-0.9	+0.6

With most alternative fuels, optimized operating points indicate positively optimized parameters such as lower NO_x emission, lower FSN and acceptable increases of specific fuel consumption, except for B10 with FSN at three operating points (see **Table 6.1**). Compared with diesel fuel, most alternative fuels present ideal differences in energy consumption, except for B100 with an acceptable difference of about 8%.

Results of optimization also indicate that combination of uses of most alternative fuels with low EGR ratios exhibits positive trends of both working process and emissions of modern heavy-duty engine. This combination also proves more suitability to meet stringent emission requirements. GTL and its blend always display potentials for optimizing working process of modern diesel engine.

7 Summaries and Conclusions

This experimental study presents the results of research on the influences of exhaust gas recirculation in combination with different alternative fuels and their diesel fuel blends on a heavy-duty engine's working process and emissions. Accordingly, it focuses on changes in thermodynamic parameters, e.g. cylinder pressure, heat release rate, ignition delay period, etc. The spectrum of cylinder pressure and exhaust gas emissions are also analyzed. The results obtained serve as the basis for evaluating and optimizing developments of the combustion process and cylinder pressure levels and the emissions behavior for such different alternative fuels and their blends as RME, O₂-Diesel fuel and Gas-to-Liquid that were combined with varying EGR ratios, EGR temperatures and injection timing.

Research was performed on a heavy-duty DEUTZ 1013EC engine with EURO II emission standards. It is a six cylinder in-line, turbocharged, direct injection diesel engine. The engine was retrofitted with an external exhaust gas recirculation system and uses water from the engine as its coolant. EGR ratios were varied by varying the opening stroke of the EGR valve by changing the pressure of the compressed air and adjusting the throttle in the intake manifold. A cylinder pressure sensor, an injection pipe pressure sensor, emission measuring equipment and data processing software were used to measure the working process, exhaust gas emissions and differences among the alternative fuels and their blends. The behavior of cylinder pressure levels were analyzed on the basis of the development of cylinder pressure and the FFT method (Fast Fourier Transformation).

Experiments were conducted with 0%, 9% and 27% EGR in conjunction with two EGR temperatures: Normal temperatures at which water from the engine was the EGR coolant and at 50 °C. Normal EGR temperatures differed at different operating points: 150 °C at maximum torque, 200 °C at nominal power, 130 °C at higher part load, 100 °C (9% EGR) and 140 °C (27% EGR) at lower part load. The reference fuel used in the experiments was diesel fuel. The alternative fuels used in the experiments were 100% RME (B100), RME with 50% diesel (B50) and 90% diesel (B10), O₂-Diesel fuel (O₂-1: diesel fuel without RME and O₂-2: diesel fuel blended with 5% RME), 100% GTL (G100) and 50% GTL (G50).

The lower heat values, higher viscosities and lower densities of most alternative fuels and their blends (except for Gas-to-Liquid) necessitated adjusting the injection parameters so that the injection pressure and specific fuel consumption were higher and average effective efficiencies were lower than for conventional diesel fuel. The engine's electric control unit controlled injection parameters to compensate for caloric differences among fuels (only at part load). Since Gas-to-Liquid and its blend only have a slightly higher heat value than conventional diesel fuel, their specific fuel consumption changed insignificantly. Since they have lower caloric values and densities than conventional diesel fuel, all of the alternative

fuels caused specific fuel consumption to increase. As a result, effective efficiencies decreased with all alternative fuels and especially with O₂-Diesel (the torque of O₂-Diesel at full load is the lowest of all the fuels). Differences in torque at full load were minimal between gas-to-liquid and conventional diesel fuel.

In response to increases in incombustible components aspirated into the cylinder when EGR ratios were higher and oxygen content was lacking, specific fuel consumption was increased to keep operating condition constant when exhaust gas was recirculated and EGR was cooled more strongly. The drawbacks of specific fuel consumption were acceptable. Furthermore, when injection was retarded, thus causing combustion to start later, injection parameters such as injection pressure and injection mass rate were adjusted to be higher. So that, specific fuel consumption increased when injection was retarded.

The developments of cylinder pressure exhibited higher cylinder pressure gradients when RME and its blend and O₂-Diesel were used instead of conventional diesel fuel. The physical and chemical properties of RME and O₂-Diesel, e.g. lower densities, higher viscosities, higher injection pressure and evaporation of ethanol in the O₂-Diesel fuel, made the combustion process “harsher” and “stronger”. Developments of cylinder pressure exhibited lower cylinder pressure gradients and maximum cylinder pressures when Gas-to-Liquid and its blend were used in the engine because their very high cetane numbers and better ignition quality cause combustion to start earlier and initial temperature and pressure in the cylinder to be lower. When exhaust gas was recirculated and cooled more, the cylinder pressure exhibited decreases in cylinder pressure gradients and maximum cylinder pressures. The decreases were greater when the EGR ratios increased. When EGR ratios were around 9%, the cylinder pressure gradient and maximum cylinder pressure deviated negligibly. All of the alternative fuels and their blends produced similar trends.

One of the most important thermodynamic parameters is the ignition delay period because it significantly influences an engine’s combustion process, in particular the mixing process of the air charge and injected fuels, and thus also significantly influences the heat release rate. RME and its blends produced moderate increases in the ignition delay period because RME has poorer ignition qualities than conventional diesel fuel (indicated by its cetane number). These trends appeared both at low loads and full loads. The combustion mass rate increased conspicuously when RME and its blends were used because a portion of the mixture prepared during the longer ignition delay period ignited. O₂-Diesel and Gas-to-Liquid produced decreases in the ignition delay period because more ethanol evaporated in O₂-Diesel and Gas-to-Liquid has a very high cetane number. When the alternative fuels used instead of conventional diesel fuel were combined with exhaust gas recirculation, ignition delay periods were prolonged and combustion delays occurred. When many more inert components are in the cylinder and oxygen deficiencies increase because the EGR ratios increase, ignition points

appear later. Initial conditions such as temperature and pressure greatly affect the emergence of the first ignition point. With most of the fuels, ignition delays increased when the recirculated exhaust gas was cooled more strongly. Longer ignition delay periods and the higher combustion rate of RME and its blends caused the duration of combustion to be shorter than with diesel fuel. Trends of combustion duration reversed when O₂-Diesel and gas-to-liquid are used. Combustion duration increased when exhaust gas was recirculated. This led to a drop in the cylinder pressure gradient.

The duration of ignition delay periods significantly influences heat release in the premixed combustion phase. Just as the developments of cylinder pressure, the higher cylinder pressure gradients and better mixture preparation process brought about by the longer ignition delay period when RME and its blends and O₂-Diesel were used led to a significant heat release from the mixture formed during the ignition delay period. RME and its blends and O₂-Diesel had higher maximum heat release rates than conventional diesel fuel. The maximum heat release rates decreased when Gas-to-Liquid and its blend were used because they have shorter ignition delay periods than diesel fuel and other alternative fuels. With all fuels, combustion started later when exhaust gas was recirculated and the combustion rate was low due to the lack of oxygen. Consequently, maximum heat release rates decreased. Especially when the throttle was used to increase EGR ratios, maximum heat release rates rose modestly because the initial temperature was higher even though the oxygen concentration diminished substantially. The trends of maximum heat release rates also increased when recirculated exhaust gas had lower temperatures because the ignition delay periods were slightly longer. Like most other thermodynamic parameters, the ratios of maximum heat release rates changed far more when the engine ran at low loads. Although the temperature and pressure were initially slightly higher but then decreased as the delay proceeded with later injection timing, maximum heat release rates increased sharply (as opposed to the trends of maximum cylinder pressure).

A data processing software and the FFT method were employed to analyze and evaluate the spectrums of cylinder pressure in different frequency ranges with all fuels and at different operating points and under different conditions such as varied ratios and temperatures of recirculated exhaust gas. Since cylinder pressure levels essentially depend on maximum cylinder pressure and particularly on cylinder pressure gradients, the spectrums of cylinder pressure developed similarly to the trends of cylinder pressure. Cylinder pressure levels also increased with most alternative fuels because the maximum cylinder pressure gradients increased with all alternative fuels (except for G100 and G50) as opposed to diesel fuel and decreased when exhaust gas was recirculated and cooled more strongly. Differences in cylinder pressure levels were modest with different fuels, EGR ratios and EGR temperatures (around 3-4 dB in maximum cases). Significant decreases occurred in the low frequency range (below 120 Hz) when injection was retarded. These sharp decreases were influenced by

the trends of the maximum cylinder pressure gradient. They increased modestly when injection timing was retarded in higher frequency ranges.

In terms of emission behavior, most of the alternative fuels had inferior FSN and hydrocarbon (HC) emissions and superior NO_x emissions to conventional diesel fuel. For the most part, Gas-to-Liquid produces lower NO_x , FSN (except for the case of 27% EGR at lower part load) and HC emissions than conventional diesel fuel and other alternative fuels. The stronger combustion process and evaporation of ethanol caused flame temperature peaks to increase when RME and its blends and O_2 -Diesel were used. Therefore, NO_x emission increased compared with diesel fuel. As opposed to RME and O_2 -Diesel's NO_x emission behavior, Gas-to-Liquid and its blend had lower NO_x emission than conventional diesel fuels and other alternative fuels. This can be explained by the lower flame temperature peaks (see the trends of cylinder pressure and heat release rate). With lower temperature peaks and the lack of oxygen, NO_x emissions declined when exhaust gas was recirculated and cooled more strongly. Combining exhaust gas recirculation with lowered EGR temperatures produced better operating conditions (lower or similar NO_x emission). Combining alternative fuels with exhaust gas recirculation substantially reduced NO_x emission (about 50% with most alternative fuels in maximum cases and over 60% with GTL and its blend). However, noticeable increases of HC emission and FSN always accompany recirculated exhaust gas and more cooled EGR. NO_x emissions decrease and HC emission and FSN increase when injection was retarded.

Indicated by the FSN, soot emission were also improved when alternative fuels are used instead of conventional diesel fuel. The FSN was lowest with neat RME. O_2 -Diesel had a higher FSN than RME and its blends (with the exception of B10). Gas-to-Liquid and its blend had an FSN between that of conventional diesel fuel and other alternative fuels. Compared with diesel fuel, the maximum reduction of FSN of over 60% occurred with B100 at lower part load (1100 1/min - 213 Nm) and without EGR. Compared with diesel fuel, the minimum reduction of FSN of around 0.5% occurs with GTL and its blend at lower part load and without EGR. At higher part load and without EGR, the FSN of G100 was the same as that of diesel fuel. Since the experimental results also show that the FSN of B100 increase over 150% with 27% EGR, the EGR ratios selected when an exhaust gas recirculation system is used should be small so that increases in the FSN and HC emission are acceptable. The FSN increases with later injection timing, and it decreases with lower EGR temperatures.

B100, B50 and Gas-to-Liquid and its blend exhibit better HC emissions behavior than conventional diesel fuel. The maximum decreased percentage was over 50% with B100 at higher part load and 9% EGR. O_2 -Diesel and B10 produce poorer HC emission than diesel fuel. The maximum increased percentage was over 30% with O_2 -1 at higher part load and

without EGR. Just as the FSN, the HC concentrations also increase with higher EGR ratios and at lower recirculated exhaust gas temperatures and with retarded injection timing.

Experimental results also confirm that neither the thermodynamic parameters (maximum cylinder pressure differs around 1%, maximum heat release rate differs around 5%, cylinder pressure gradient differs around 2% and specific fuel consumption differs around 1.5% in maximum cases) nor the emissions (NO_x concentration differs around 4%, HC concentration around 7%, FSN around 15% in maximum cases) of O_2 -1 and O_2 -2 Diesel differ significantly. Thus, whenever diesel fuel with 5% RME was used in the blend of O_2 Diesel, any changes were insubstantial.

The results of the investigations indicate there is practical potential to enhance the percentage of RME (up to 10%) in conventional diesel fuel blend because the changes in thermodynamic parameters and emissions are acceptable compared with diesel fuel. However, a formal conclusion about increasing the percentage of RME (up to 10%) in the diesel fuel blend at a real filling station can only be drawn after more research has been done, e.g. on impacts on the lifetime or corrosion of injection system components. When the percentage of RME in the conventional diesel fuel blend approaches 10%, the chemical properties of RME in the blend cause thermodynamic parameters to increase modestly (maximum heat release rate around 1.6%, maximum cylinder pressure around 0.6%, specific fuel consumption around 0.7% and cylinder pressure gradient around 8% in maximum cases). B10 fuel reduces NO_x and FSN, but, compared with diesel fuel, the HC concentration increases as much as 8% in the maximum case.

While FSN and HC emission trends were positive, NO_x emissions increase more with B50 and B100 than with conventional diesel fuel. Recirculated exhaust gas or retarded injection timing have proven to be the solutions for reducing NO_x emission.

When Gas-to-Liquid and its blend were used instead of conventional diesel fuel, they show potential to reduce emissions. In addition, since Gas-to-Liquid has a higher caloric value, it produces less specific fuel consumption than diesel fuel. This is an advantage of Gas-to-Liquid.

The experimental results serve as the basis for optimizing operating points with the different alternative fuels. Optimizations were based on similar or lower NO_x concentrations, FSN and acceptable changes in specific fuel consumption and energy consumption. Optimizations were also carried out with low EGR ratios. Since injection timing was not varied much in the experiments, optimizations were not based on injection timing. Optimized operating points for RME and its blends with low EGR ratios were selected because of the reductions of NO_x emission always found with recirculated exhaust gas and the acceptable increases of FSN

(only with B10) and the specific fuel consumption. With other alternative fuels such as O₂-Diesel and particularly Gas-to-Liquid and its blend, the operating points optimized exhibit positive parameter trends which were the basis for optimizations. Gas-to-Liquid and its blends display potentials for optimization because they do not have to be combined with any other methods. With the exception of B100 at maximum torque (an acceptable difference of about 8%), the optimized operating points of most different alternative fuels exhibit ideal differences in energy consumption.

This study has expanded basic understanding of the differences in performance and behavior that result from the use of alternative fuels and their blends in conjunction with recirculated exhaust gas in a diesel heavy-duty engine. The experimental results obtained provide a basis to optimize engine operating points with different alternative fuels and a means to control the engine process by using alternative fuels and their blends.

Blends of conventional diesel fuel and RME with more than 5% RME are absolutely feasible, although more research is needed, e.g. effects on the life of components, wear and erosions and on competitive pricing. The absence of significant changes in its thermodynamic parameters coupled with its excellent emissions trends are evidence that speak for the feasibility of using B10 in automobiles. Most of the blends of RME and conventional diesel fuel proved more suitable for use in vehicles and better suited to meet stringent emissions requirements when they were combined with low EGR ratios. In view of their potentials for positively influencing the engine process and emissions, GTL and its blend can be used realistically in commercial vehicles.

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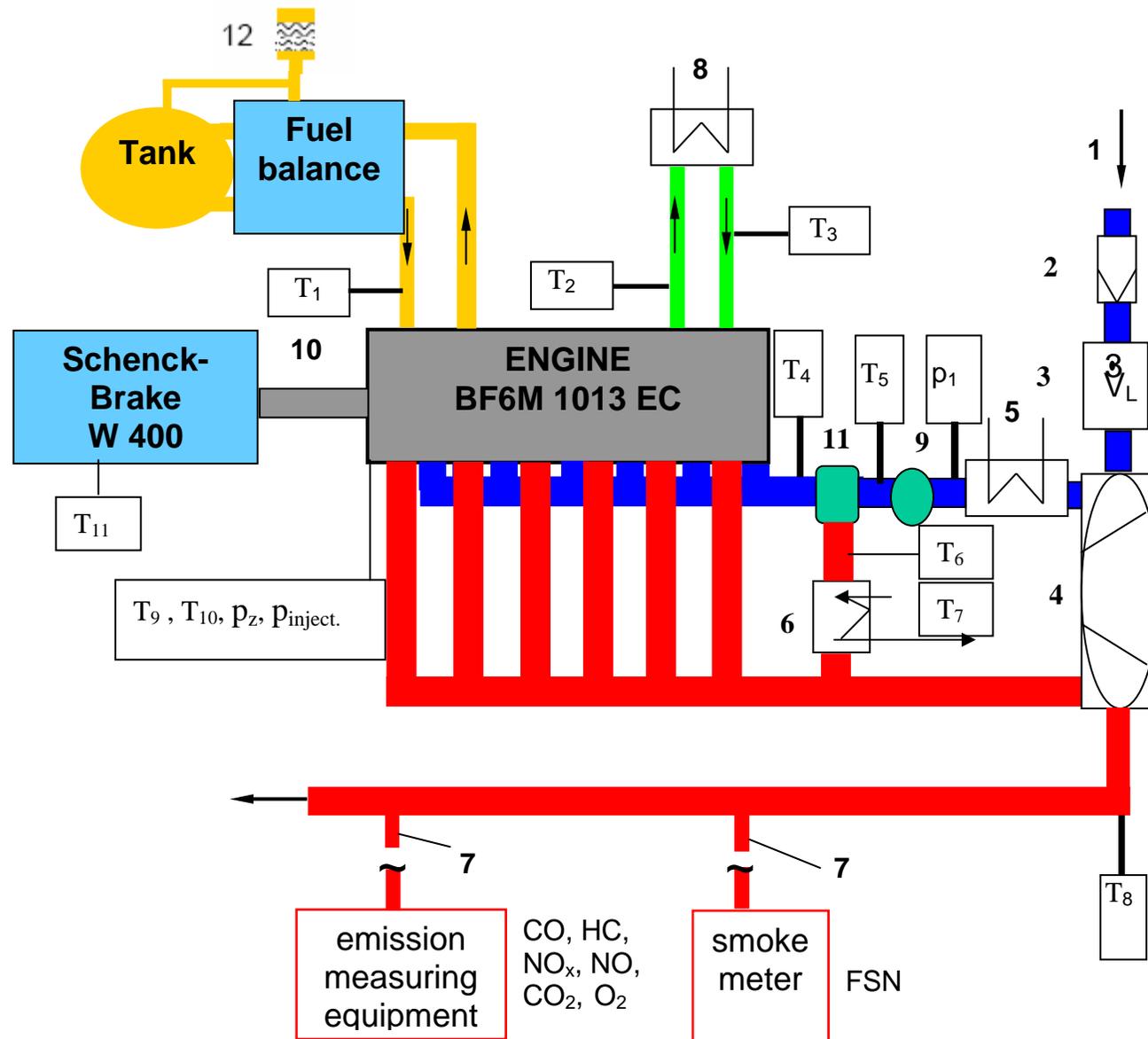
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Appendix

Appendix



- 1 Air entrance
- 2 Air filter
- 3 Air mass meter
- 4 Turbocharger
- 5 Charge cooler
- 6 EGR cooler
- 7 Exhaust gas probe
- 8 Water engine cooler
- 9 Throttle
- 10 Clutch
- 11 EGR valve
- 12 Flame trap valve

- █ Air charge
- █ Exhaust gas
- █ Cooling water
- █ Fuel

- T₁: Fuel temp. before engine
- T₂: Water temp. after engine
- T₃: Water temp. before engine
- T₄: Inlet charge temp.
- T₅: Air temp. before EGR valve
- T₆: EGR temp. before EGR valve
- T₇: EGR coolant temp.
- T₈: Exhaust gas temp. after turbo
- T₉: Engine coolant temp.
- T₁₀: Lubricant temp.
- T₁₁: Brake temp.
- p₁: Charge pressure
- p_z: Cylinder pressure
- p_{inject}: Injection pipe pressure

Figure A.1: Schematic of test bench

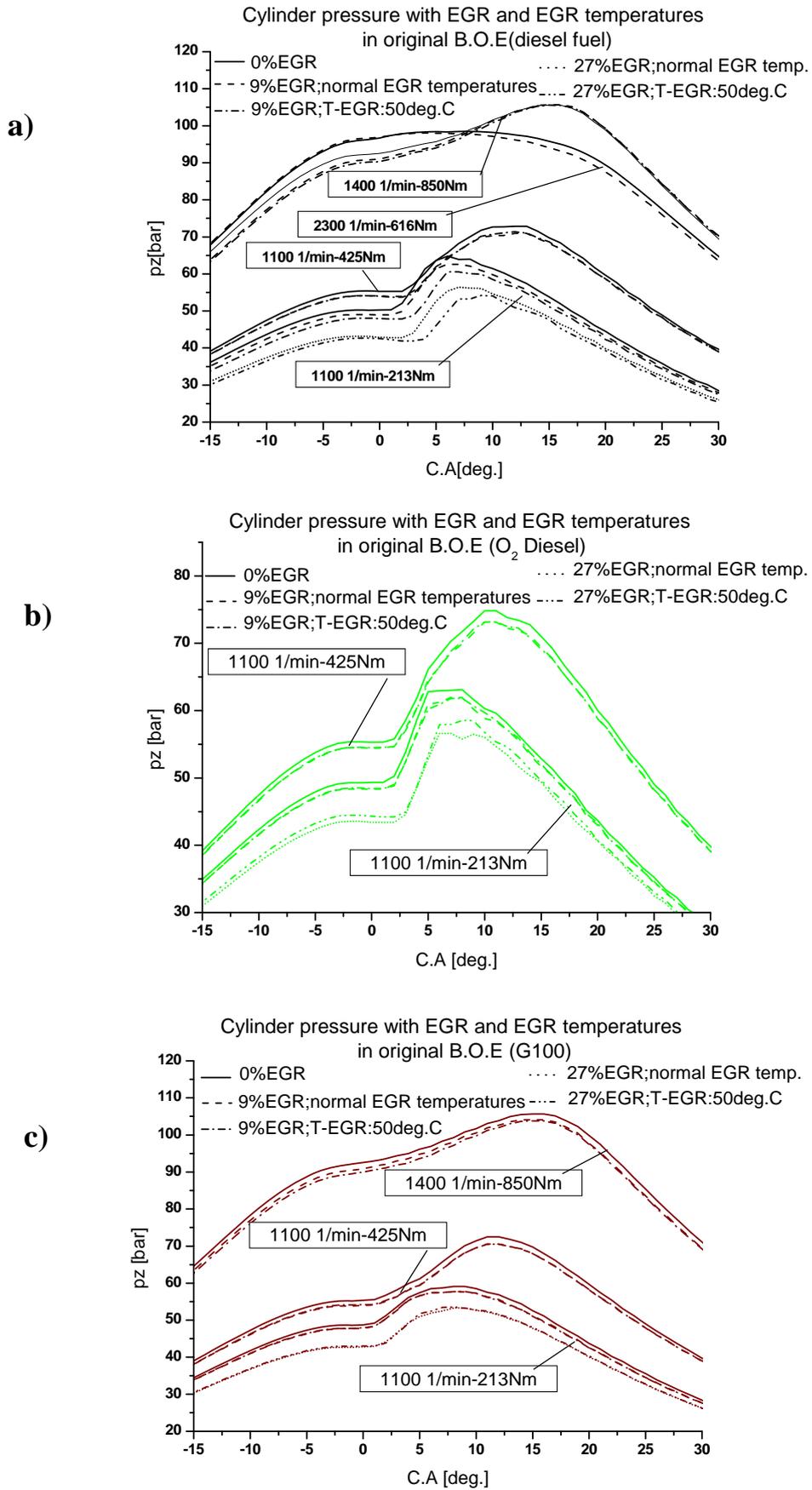


Figure A.2: Cylinder pressure with different EGR ratios and EGR temp.: a) with diesel fuel; b) with O₂-Diesel; c) with G100

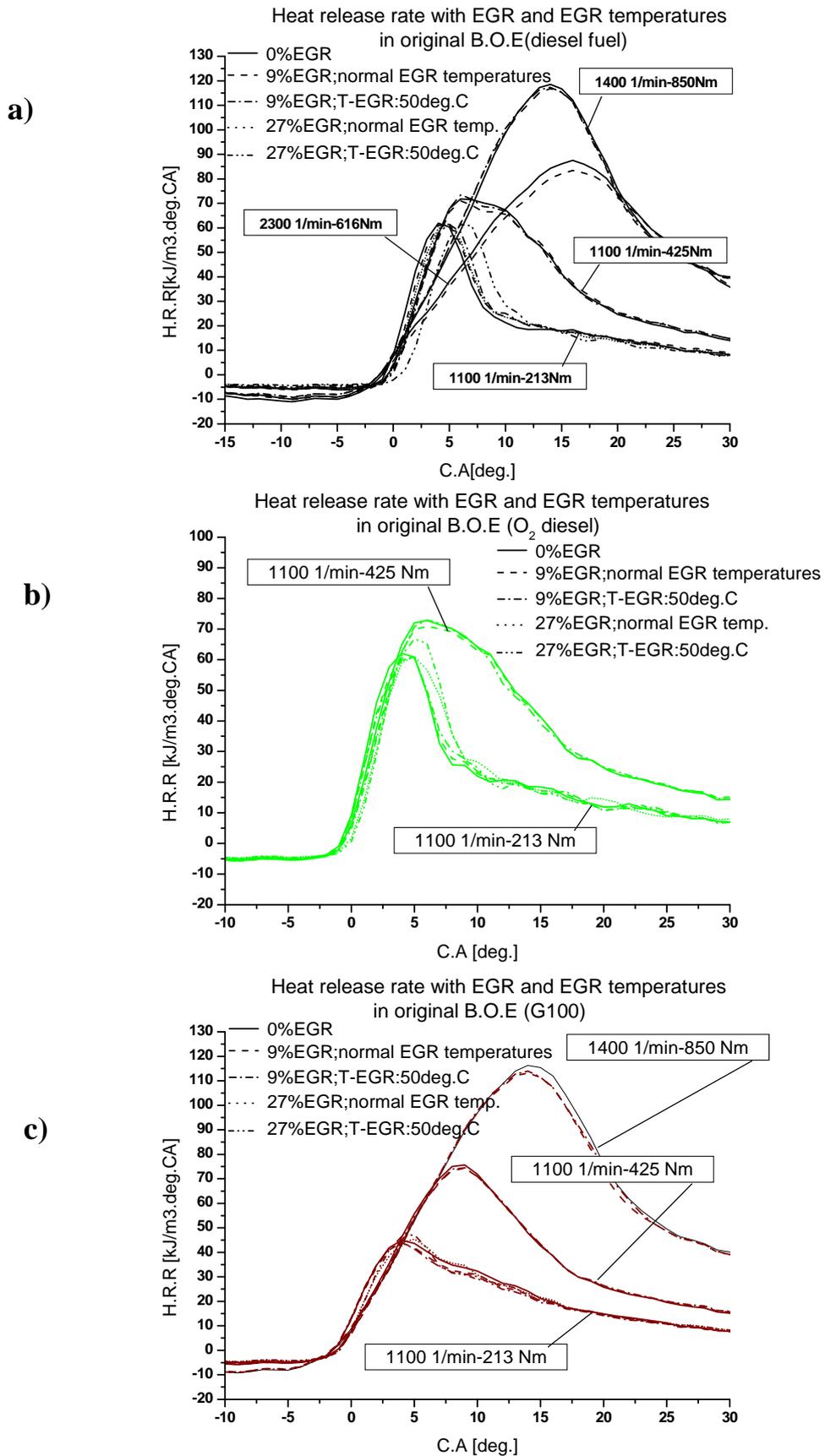


Figure A.3: H.R.R with different EGR ratios and EGR temperatures: **a)** with diesel fuel; **b)** with O₂-Diesel; **c)** with G100

Lebenslauf

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