공학박사학위논문

압축착화엔진에서 이종연료 연소모드 별 특성 분석과 예혼합 연소를 통한 운전영역 확장에 관한 실험적 연구

Experimental Study on the Characteristics of Dual-fuel Combustion Modes and Extension of Dual-fuel PCI Operating Range in a CI Engine

2016 년 2 월

서울대학교 대학원

기계항공공학부

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지도교수 민 경 덕

이 논문을 공학박사 학위논문으로 제출함

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서울대학교 대학원

기계항공공학부

이 정 우

이정우의 공학박사 학위논문을 인준함

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This thesis is dedicated to my parents.

Myungjae Lee

Kyungsook Park

Abstract

Experimental Study on the Characteristics of Dual-fuel Combustion Modes and Extension of Dual-fuel PCI Operating Range in a CI Engine

Jeongwoo Lee
Department of Mechanical and Aerospace Engineering
The Graduate School
Seoul National University

The thermal efficiency and the durability of diesel engines which are based on compression ignition system are superior to those of gasoline engines with a spark ignition system. However, diesel engines are suffering from the high level of particulate matter (PM) emission which is caused by locally rich regions in-cylinder, related to the heterogeneous air-fuel mixture combustion. Also, since diesel engine is based on auto-ignition with high compression ratio, the pressure rise rate which causes engine noise is higher than that of gasoline engine. In addition, a diesel engine needs to separate after the treatment systems to oxidize the unburned components and deoxidize the nitrogen oxides (NOx) emission, while a gasoline engine with theoretical air-fuel ratio can reduce emissions using only a three-way catalyst (TWC)

As a result, novel diesel combustion concepts, especially premixed compression ignition (PCI) which is a practical realization of homogeneous charge compression ignition (HCCI), have been studied. Most of the novel diesel combustion concepts are based on the higher exhaust gas recirculation (EGR) rate to suppress NOx emission by reducing oxygen concentration, combustion temperature and earlier diesel injection timing to implement the premixed air-fuel mixture condition to

reduce PM emission. Engine-out NOx and PM emissions could be effectively reduced by novel diesel combustion, but a higher in-cylinder pressure rises rare from the increasing premixed combustion and incomplete combustion due to low combustion temperature are inevitable. In particular, almost all novel diesel engines have a trouble with the extension of the operating range. Since novel diesel combustion concepts need enough time for air-fuel mixing, higher speeds and load conditions could not be realized.

In this circumstance, a reactivity controlled compression ignition (RCCI) which can be implemented by two different fuels (eg. gasoline and diesel) is one of good methods to extend the operating range with a clean combustion. However, in-cylinder pressure rise rate of RCCI is higher than that of conventional diesel combustion similar to HCCI combustion, because the major profit of RCCI combustion is higher thermal efficiency from rapid combustion. Especially, since the dual-fuel combustions including RCCI has unusual characteristics as the ratio between two fuels changes, the study of dual-fuel combustion characteristics must be conducted to suggest proper operating strategies under different load conditions.

Thus, in this work, the characteristics of dual-fuel combustion under various modes were studied and the investigations of the higher thermal efficiency and the load of dual-fuel premixed compression ignition (PCI) were verified. Especially, from this research, the definition of dual-fuel PCI was introduced and the potential for more practical applications of dual-fuel combustion for commercial CI engines can be confirmed.

The first experimental result was that dual-fuel combustion modes can be divided into three cases. The first one was dual-fuel combustion which was comprised of a premixed combustion of mainly diesel and some of gasoline which were entrained during the spray motion and the mixing controlled combustion of the residual air-fuel mixture. On the other hand, if the reactivity stratification was adjusted with a high portion of low reactivity fuels enough to combustible and the in-cylinder temperature and pressure reached a certain point, then split auto-ignition occurred, which can be verified by two peaks of HRR. The first peak of HRR came

from the diesel and some of the gasoline fuel, but the second peak of HRR came from faster auto-ignition of residual gasoline and some of the diesel fuels. The third dual-fuel combustion mode was entirely premixed combustion from two fuels simultaneously. As reactivity stratification of in-cylinder became smooth and gradual, then stratified auto-ignition occurred from diesel and gasoline fuels.

Therefore, in this work, the second and third dual-fuel combustion modes were selected to achieve a higher thermal efficiency with low NOx and PM emissions. Then these dual-fuel combustion modes based on the earlier diesel injection strategy whose ignition delay was longer than the diesel injection duration and a large amount of low reactivity fuel are called as 'dual-fuel PCI'.

The second objective was evaluating the relation between combustion index and dual-fuel combustion modes. As varying the total equivalence ratio, gasoline fraction, and diesel injection timings, LTHR (Low-Temperature Heat Release) region occurred from the third mode (single auto-ignition) and the tendency of MFB 50 location became opposite to the behavior of diesel injection timing. Therefore, although the second and third modes are based on the PCI region, the second mode was still related with diesel injection timing (spray motion) which means 'late injection PCI' and the third mode was seemed like an 'early injection PCI' which is based on the chemical reaction rather than combustion from spray.

Additional finding was the source of THC emission under dual-fuel combustion. From the results, too much leaner equivalent ratio condition is not suitable for the dual-fuel combustion. Also, there might be criteria for the maximum fraction of gasoline and diesel injection timings. In addition, in this work, split diesel injection strategy can be suitable for the improvement of combustion efficiency in dual-fuel combustion. Using split diesel injection strategy, the distribution of diesel, which has a role of ignition source mainly, was improved and wall-wetting can be reduced by shortened the diesel pray length. However, under high load condition, split diesel injection strategy was not recommended because of higher PRRmax.

The last experimental achievement of this work was the optimization of dual-fuel PCI under low and high loads respectively. Under a low load condition, since

PRRmax was low enough to satisfy the criteria (especially, 10 bar/deg), the third dual-fuel combustion mode which means simultaneous premixed combustion of diesel and gasoline fuels was favorable. In this research, 48 % of gross indicated thermal efficiency was achieved under 1,500 rpm/gIMEP 6.5 bar condition with low NOx (13 ppm) and near-zero PM emissions (Below 0.05 FSN).

On the other hand, under a higher load condition, PRRmax which is related with a knocking from gasoline fuel is the main challenge. Thus, the second dual-fuel combustion mode was applied to extend load. Using the split auto-ignition combustion strategy gIMEP 14 bar was successfully achieved without the knocking phenomenon under 2,000 rpm condition. This maximum value implied that there is a potential of dual-fuel PCI to cover the entire region of NEDC test mode.

This work includes the study of the characteristics of dual-fuel combustion modes and the investigation into the improved operating strategies of dual-fuel PCI. From the results of combustion characteristics, appropriate combustion modes were suggested as load conditions to achieve low NOx and PM emissions, higher thermal efficiency and especially the highest operating loads. Thus, this research can contribute the practical application of dual-fuel combustion in passenger cars with light-duty diesel engines

Keywords: diesel engine, diesel injection strategy, dual-fuel combustion, nitrogen oxides (NOx), particulate matter (PM), premixed compression ignition (PCI), reactivity controlled compression ignition (RCCI)

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Acronym

AFR	air-fuel ratio
ATDC	after top dead center
BMEP	brake mean effective pressure
BTDC	before top dead center
CA	crank angle
CO	carbon monoxide
CO_2	carbon dioxides
CoV	coefficient of variation
CVI	close valve injection
DI	direct injection
DPF	diesel particulate filter
EGR	exhaust gas recirculation
EOI	end of injection
EVC	exhaust valve close
EVO	exhaust valve open
FSN	filtered smoke number
gIMEP	gross indicated mean effective pressure
HCCI	homogeneous charge compression ignition
HFR	hydraulic flow rate
HSDI	high-speed direct injection
HTHR	high-temperature heat release
IVC	intake valve close
IVO	intake valve open
JP-8	jet propellant-8
LNT	lean NOx trap
LTC	low-temperature combustion
LTHR	low-temperature heat release

NOx	nitrogen oxides
NVO	negative valve overlap
PCI	premixed compression ignition
PFI	port fuel injection
PM	particulate matter
ppm	part per million
RCCI	reactivity controlled compression ignition
RoHR	rate of heat release
rpm	revolution per a minute
SCR	selective catalytic reduction
SOC	start of combustion
SOI	start of injection
THC	total hydrocarbon
uHC	unburned hydrocarbon

1. Introduction

1.1Background and Motivation

1.1.1 Emission regulations

Internal combustion (IC) engines, such as spark ignition (SI) and compression ignition (CI) engines, have been widely used in the global automobile industry. IC engines have the advantage of high-efficiency, affordable cost and good reliability compared to those of other power sources. Especially, CI engine with the high-speed direct injection (HSDI) system shows higher thermal efficiency than SI engine due to higher compression ratio (CR), which could affect the fuel economy of the vehicle. Also, since CI engine usually operates under the leaner air-fuel (AF) mixture condition rather than stoichiometric, nitrogen oxides (NOx) and unburned hydrocarbon (uHC) emissions from CI engines are lower than those of SI engine, which means more clean power source.

However, particulate matter (PM) emission from CI engine is a major problem, because diesel combustion under compression ignition system based on the diffusive flame from the heterogeneous air-fuel mixture [1]. Also, while SI engine could use a three-way catalyst (TWC) for the reduction of NOx emissions under the stoichiometric combustion, CI engine cannot use TWC for NOx reduction due to lean AF condition.

Moreover, EURO-6 regulation which is enforced in 2014 require a reduction of NOx emissions to half the level required by previous regulation, while the restrictions on PM emission levels remained the same in Figure 1.1. Especially, until

the early phase of EURO 6 regulation, new European driving cycle (NEDC) has been used for the emission regulation, but worldwide harmonized light vehicle test procedure (WLTP) which is harsher operating condition than NEDC will be used from 2017 as shown in Fig.1.2. Furthermore, after WLTP mode, real driving emission (RDE) procedure will be introduced. This restriction is challenging for diesel engines without emissions after-treatment because there is a trade-off relation between NOx and PM emissions under diesel diffusive combustion [2, 3].

Thus, recently diesel vehicles have been equipped emission after-treatment systems such as diesel particulate filter (DPF), lean NOx-trap (LNT) and selective catalytic reduction (SCR). DPF is widely used to reduce tail-pipe out PM emission. It could be classified as one of the physical filter rather than catalyst, unlike other after-treatment systems. DPF can usually eliminate PM emission up to 99 %, but it is needed to clean up periodically by the regeneration. Since fuel is injected during DPF regeneration to increase exhaust gas temperature, this action would make fuel economy worsened. Also, DPF cause back pressure increase.

As well as the usage of DPF, LNT or SCR are used for diesel vehicles recently to meet the EURO 6 regulation. Although LNT is cheaper and easier to be equipped than SCR, the filtration efficiency of LNT is remarkably lower than that of SCR and frequent regeneration of LNT would cause the deterioration of fuel economy. On the other hand, SCR is superior to LNT in the filtration sakes, but it is more expansive and it needs urea to operate. In summary, although after-treatment systems are effective to reduce engine-out emissions, it cannot avoid increasing the cost of diesel vehicles and there is an adverse effect on the fuel economy. Therefore, it needs to make the efforts on the reduction of engine-out emissions by improvement of the combustion system.

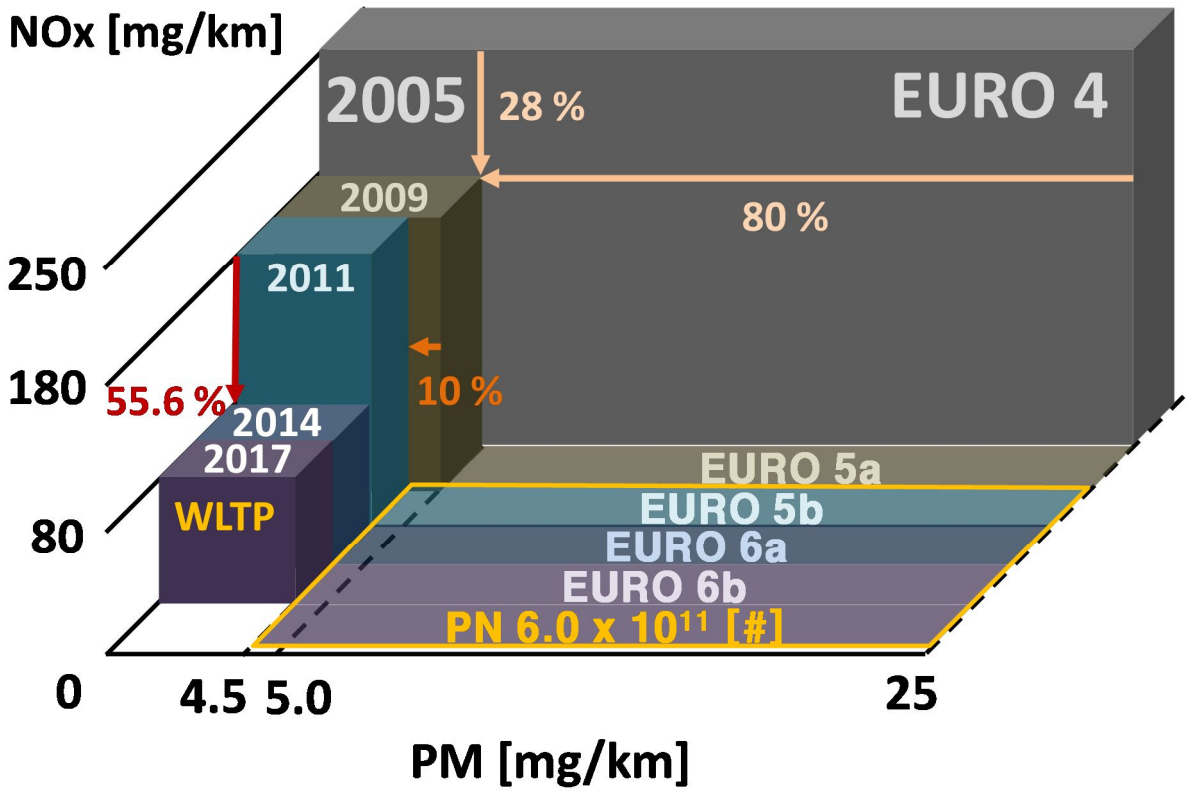
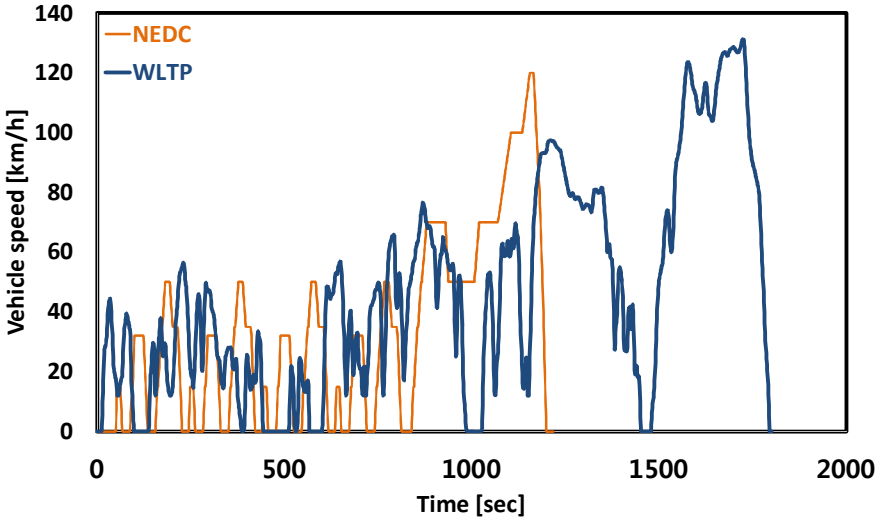


Figure 1.1 History of EURO emission regulations from EURO Ⅳ to Ⅵ



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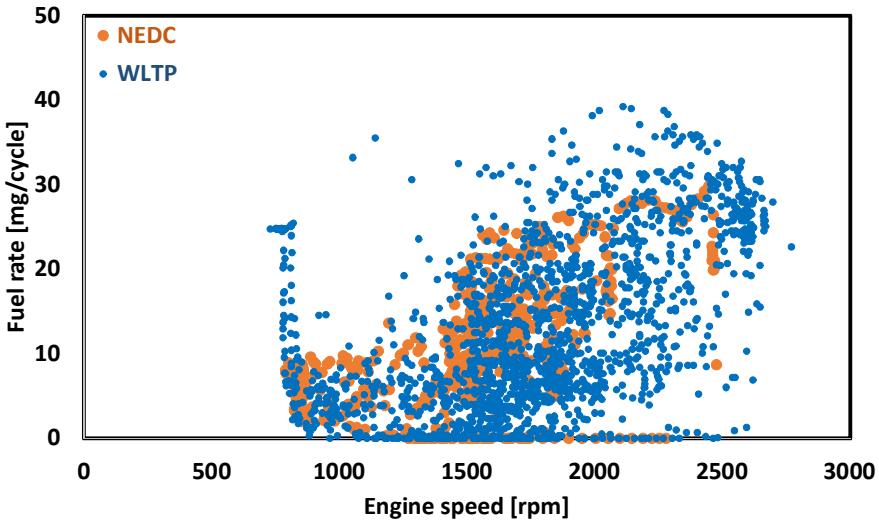


(b)

Figure 1.2 Comparison of NEDC and WLTP modes by driving pattern (a) and engine operating regions (b) in a conventional 1.6 L diesel engine

1.1.2 World transport energy

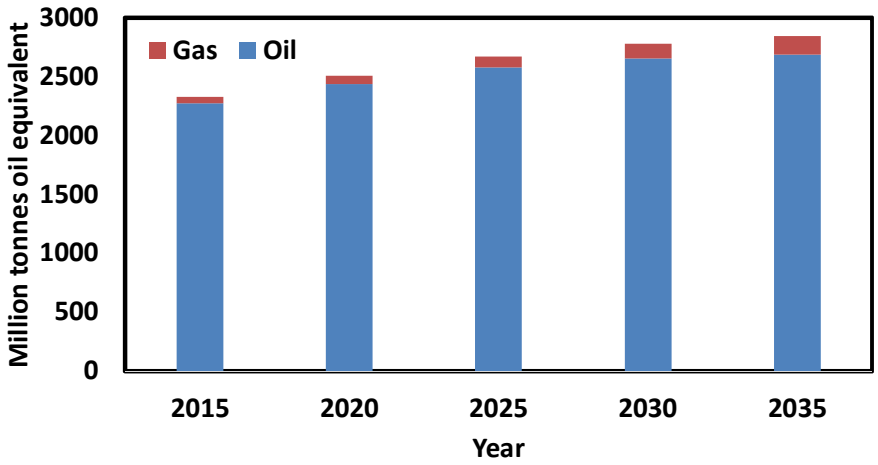
As well as an environmental problem related with emission regulations, diesel engine researchers encounter energy issue in these days. According to a report from the US Energy Information Administration (EIA), the usage of fossil fuel energy for transportation worldwide will increase by 35 % within 20 years in Figure 1.3 (a) [4]. In particular, in the future, the demand for diesel fuel may increase drastically compared to other fossil fuels, such as jet fuel and gasoline in Figure 1.3 (b). Therefore, a clean diesel combustion system with high thermal efficiency is needed.

As a result, researches of alternative fuels for CI engines have been studied. Biodiesel is the one of the representative alternative fuels. Since biodiesel contains oxygen atoms unlike general diesel fuel, it could promise low PM emission [5, 6]. However, since this fuel is already partly oxidized, low heating value (LHV) of biodiesel is usually lower than that of conventional diesel fuel.

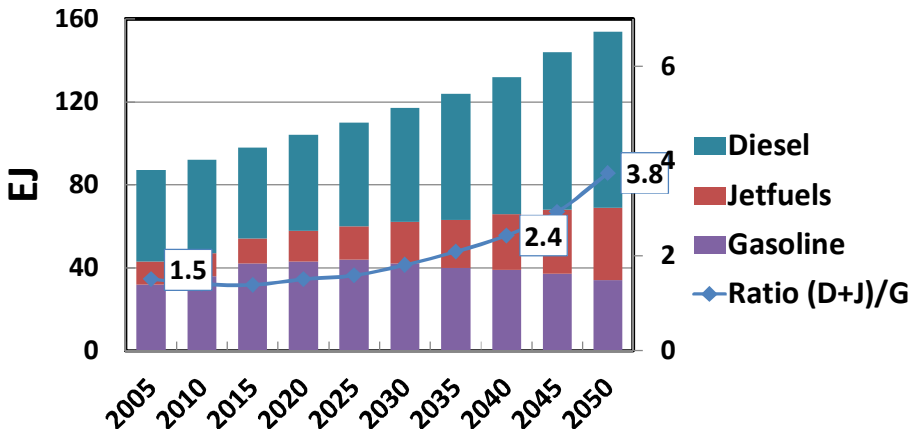
Jet propellant-8 (JP-8) is also one of alternative fuel for diesel. JP-8 is a type of jet fuel that is used in jet engines, particularly military aircraft. Recently, since the single fuel concept (SFC) policy was executed by NATO military, initiating the use of JP-8 in both jet and diesel engines, the application of JP-8 in diesel engines has become an important issue.

Pickett et al. [7] determined that the spray length of JP-8 is shorter than that of diesel due to its lower density and distillation temperature (i.e., higher volatility). In particular, a lower distillation temperature broadens the vapor phase. Lee et al. [8] also investigated the spray characteristics of JP-8 and found that the spray angle of JP-8 was wider than that of diesel. Thus, there is a possibility of spray overlap. This phenomenon could deteriorate the combustion efficiency and cause greater HC

emissions. Also, in the previous research of the author of this research, JP-8 was applied to a conventional 2.2 L diesel engine to reduce NOx and PM emissions simultaneously with maintaining equal thermal efficiency with that of diesel fueled conditions [9]. Then, by using prolonged ignition delay from low cetane number of JP-8, supplementation of more EGR with earlier diesel injection was possible, which is one kind of PCI combustion.



(a)



(b)

Figure 1.3 Prospect of the amount of fossil fuel usages in the world transport part during 25 years (a) and the each portion of fuel usages during 45 years (b) [4]

1.1.3 Advanced CI combustion

For conventional diesel combustion, there are two main combustion modes. The first mode is a premixed combustion phase achieved by the injected fuel prior to the start of combustion (SOC). A mixing controlled combustion phase occurred after the premixed combustion phase due to the heterogeneous air-fuel mixture during the combustion period [1]. Because most of the NOx and PM emissions were formed during the mixing controlled combustion phase, it is important to increase the premixed combustion phase by constructing a well-premixed air-fuel mixture before SOC.

Thus, homogeneous charge compression ignition (HCCI) is the best answer to improve thermal efficiency and achieve near-zero NOx and PM emissions simultaneously. HCCI concept is based on entire well-premixed air and fuel mixture [10~16]. In the DI system, early fuel injection is usually applied to make a homogeneous air-fuel mixture.

One kind of HCCI combustion, premixed lean diesel combustion (PREDIC) was introduced from the New ACE Institute in Japan [14]. The authors designed HCCI combustion by early diesel injection strategy with two colliding sprays. Especially three injectors, one is located on the center and the others are on the sides, were used in this study, so combustion efficiency and stability were improved compared to previous HCCI combustion. However, HC emission was still higher than that of conventional value.

Yanagihara et al. also performed HCCI combustion study, which is well known as ‘UNIBUS’ [15]. The author used piezo-actuator injectors with pintle-type injector nozzles to reduce wall-wetting by decreasing spray penetration length. Under the

overall lean condition, split diesel injection strategy was applied. The first injection was done at 50 °BTDC for the premixed mixture condition and the second one was injected at 13 °ATDC to take a role of an ignition source. Also, high EGR rate as 60 % was supplied to prolong ignition delay. This operating condition is a typical concept of 'early-injection' HCCI combustion with diesel fuel. However, fuel consumption penalty and load limit remained challenging problems.

For this reason, in Figure.4, to reduce the engine-out NOx and PM emissions simultaneously, PCI combustion with direct injection fuel, such as diesel has been the subject of most of the relevant research to date. Certainly, PCI combustion has the potential to reduce both NOx and PM emissions more effectively than conventional diesel combustion. In this work, PCI combustion is defined as combustion in which the ignition delay which is calculated as the duration between the start of injection (SOI) and SOC is longer than the diesel injection in Figure 1.5. Earlier diesel injection and the use of a higher exhaust gas recirculation (EGR) rate usually lead to an ignition delay exceeding the injection duration [15~17]. As a result, the premixed combustion phase increases, while the mixing controlled combustion phase decreases [18~20]. The enhanced air-fuel mixture resulting from the prolonged ignition delay produces low PM emission, and the higher EGR rate suppresses NOx emissions. PCI combustion path on the in-cylinder phi-T chart is shown in Figure 1.6.

However, because of early diesel injection, spray-wall impingement occurred under relatively low in-cylinder pressure conditions, which increased the spray penetration [21]. Thus, this wall impingement was problematic in terms of total hydrocarbons (THC), and oil dilution issues occurred. Moreover, the homogeneity of the air-fuel mixture from PCI combustion was still insufficient due to the low volatility of high-reactivity fuels. In addition, a shorter mixing time compared to that

of the port fuel injection was unavoidable when PCI combustion was achieved by a direct injection only.

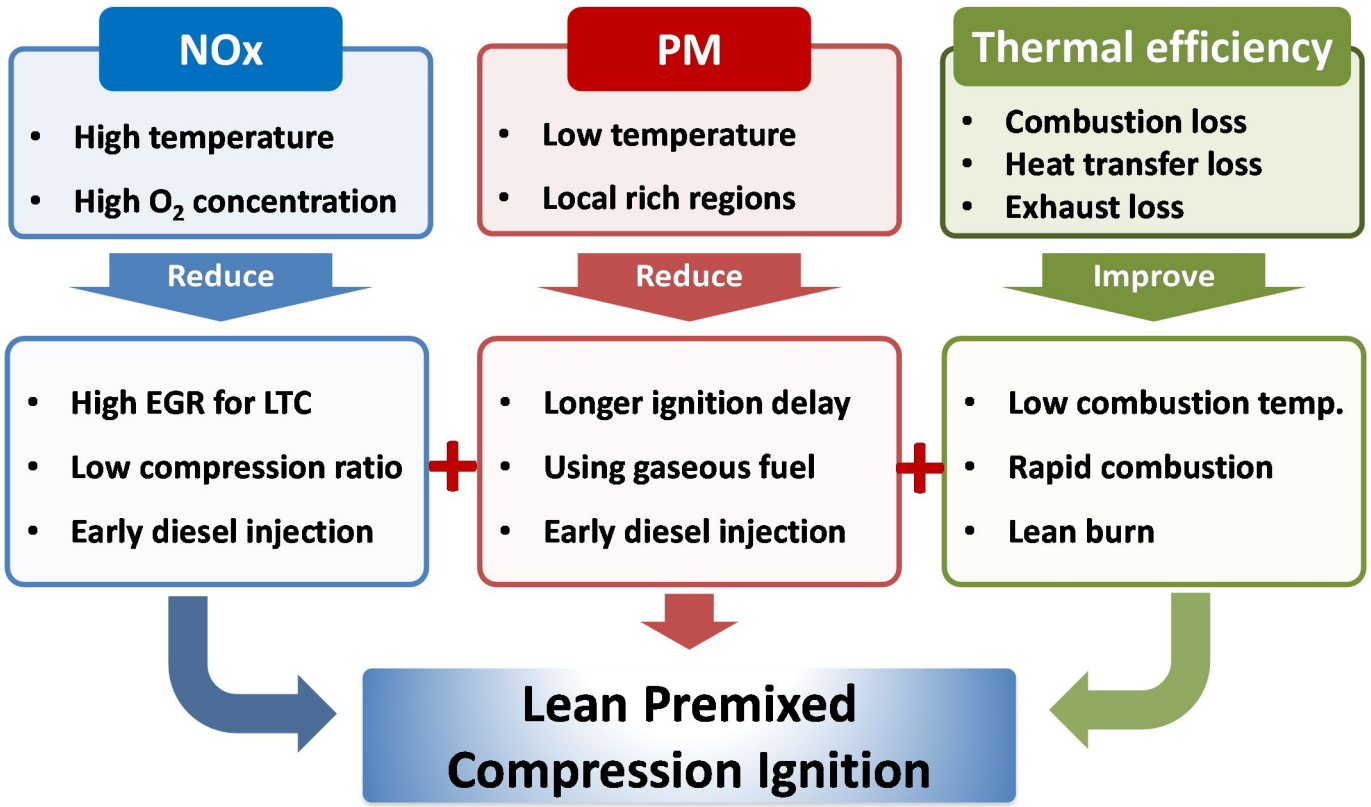


Figure 1.4 Diagrams of NOx and PM formation mechanisms and the ways to reduction two emissions and improve thermal efficiency

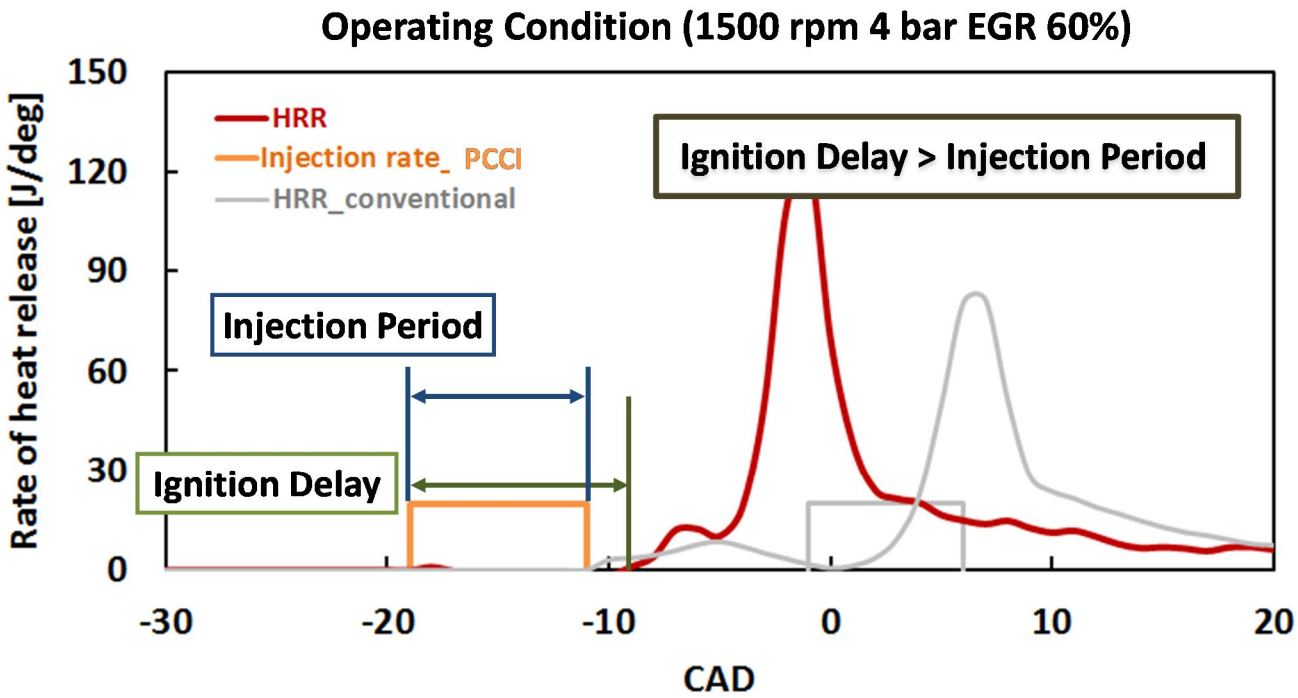


Figure 1.5 Graphs of heat release rate of diesel PCI combustion (red and bold line) and conventional diesel combustion (gray and narrow line) [22]

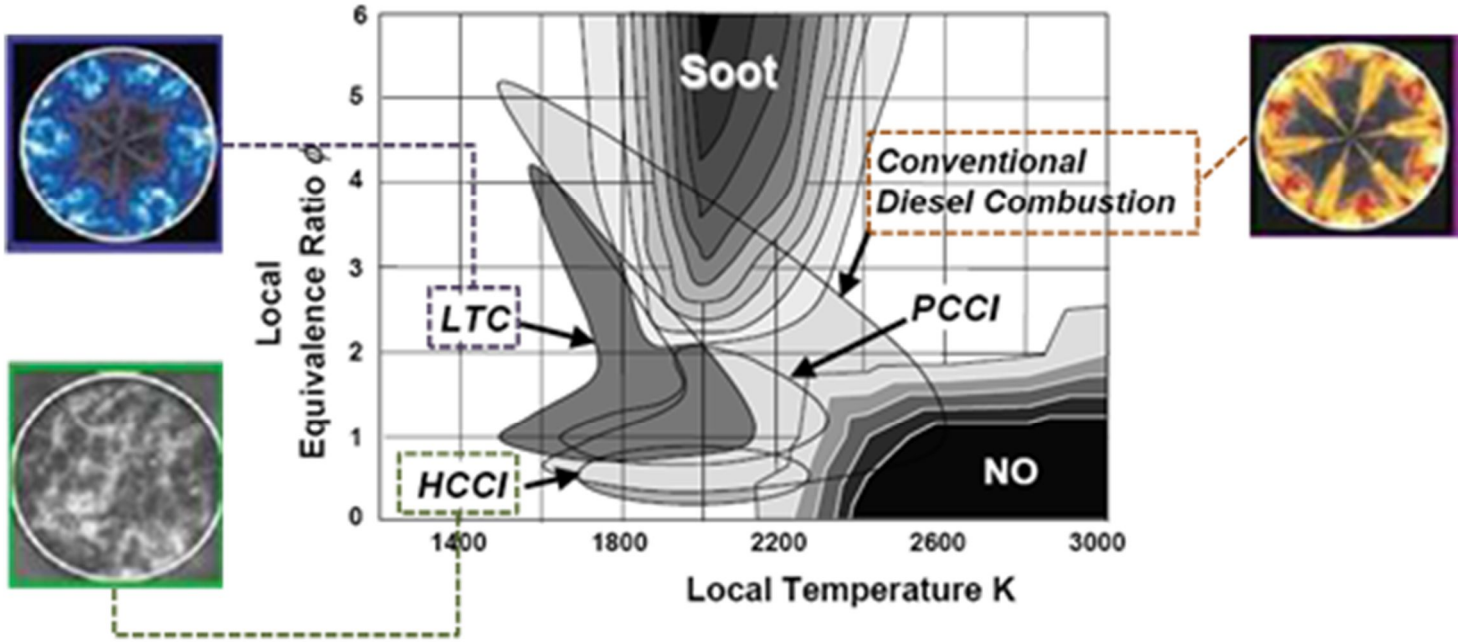


Figure 1.6 Operating traces of each combustion concepts on the in-cylinder phi-T chart [21]

1.1.4 Dual-fuel CI combustion

From the results from Chapter 1.1.3, it was concluded that although increasing premixed combustion by PCI combustion would be suitable for reducing NOx and PM emissions, PCI combustion by direct injection fuel only has limitations. Therefore, dual fuel combustion was introduced to the PCI combustion concept. Inagaki et al. first suggested the dual-fuel (premixed iso-octane and direct-injected diesel) premixed compression ignition (PCI) concept [23]. The purpose of this study was to reduce the EGR requirement while maintaining low NOx and PM emissions. In this research, premixed fuel was the main power source, and the high-reactivity fuel, i.e., diesel, enhances the overall in-cylinder reactivity. Because a high-volatility fuel, iso-octane, improved homogeneity, it caused near-zero PM emissions.

This concept had an influence on reactivity controlled compression ignition (RCCI) combustion. RCCI combustion could be implemented by supplying low-reactivity fuel, such as gasoline and gaseous fuels, through a port fuel injector (PFI) and high-reactivity fuel, usually diesel fuel, with direct injection (DI) [23-28]. The ratio of low-reactivity fuel was usually higher than that of high-reactivity fuel. High-reactivity fuel was injected by a split injection strategy. The first injection was performed at near 80 °BTDC for squish conditioning and the second one was injected at 30~45 °BTDC as an ignition source [24, 26]. The overall combustion process was dependent on the reactivity stratification in the cylinder. Because the low-reactivity fuel was well-distributed in-cylinder and the small amount of diesel fuel was used for ignition, this combustion was similar to homogeneous charge compression ignition (HCCI) combustion in Figure 1.7. The only difference between the two combustion concepts, RCCI, and HCCI, is whether high-reactivity fuel is supplied. RCCI combustion promises higher thermal efficiency due to a reduction of

heat loss from the shortened combustion duration (Figure 1.8), low combustion temperature and near-zero NOx and PM emissions from homogeneous mixture combustion in Figure 1.9 and 1.10.

Although RCCI combustion is the closest combustion to the HCCI combustion among advanced CI combustions until now, some challenges still exist. The first challenge was the high level of HC emission in Figure 1.11 [29~31]. Even if many of the advanced concepts of combustion were suffered from the HC emission due to wall-wetting of diesel spray, HC emissions from dual-fuel combustion concepts including RCCI combustion is also stemmed from the crevice effect. Since usually top-land of CI piston is larger than that of SI piston, crevice volume is larger. However, it does not know exactly the contribution of wall-wetting and crevice effect on HC emission from dual-fuel combustion. Since usually diesel fraction is low under RCCI combustion, the effect of wall-wetting may be lower than the PCI combustion condition. Not only the wall-wetting and the crevice effect, the low combustion temperature is also one of the main causes to increase HC emission, because combustion temperature is deeply related with the combustion efficiency. However, the low combustion temperature is important to reduce NOx emission, so this factor could not be controlled.

The second challenge is a high level of the maximum in-cylinder pressure rise rate (PRRmax) [32, 33, 35]. As it was mentioned above, since RCCI combustion is similar with HCCI combustion due to bulk combustion, simultaneous auto-ignition which makes a shorter combustion duration causes higher PRRmax than that of conventional diesel combustion. High PRRmax can bring out engine noise, so smoother combustion is needed [34].

Especially, the last challenge of RCCI combustion is related to the high PRRmax. Operating range of RCCI combustion should be extended (Figure 1.12) [34, 36, 37]. Although RCCI combustion with a heavy-duty diesel engine can be extended to near gross indicated mean effective pressure (gIMEP) 16 bar, light-duty engine with an RCCI combustion cannot extend to this value. Actually, knocking phenomenon of low reactivity fuel, i.e. gasoline, occurred under high load conditions of RCCI combustion. As a result, heavy EGR rate was used for the suppression of PRRmax under high load conditions, but it caused the deterioration of combustion efficiency. Therefore an improved dual-fuel combustion concept for the smooth and rapid combustion is needed.

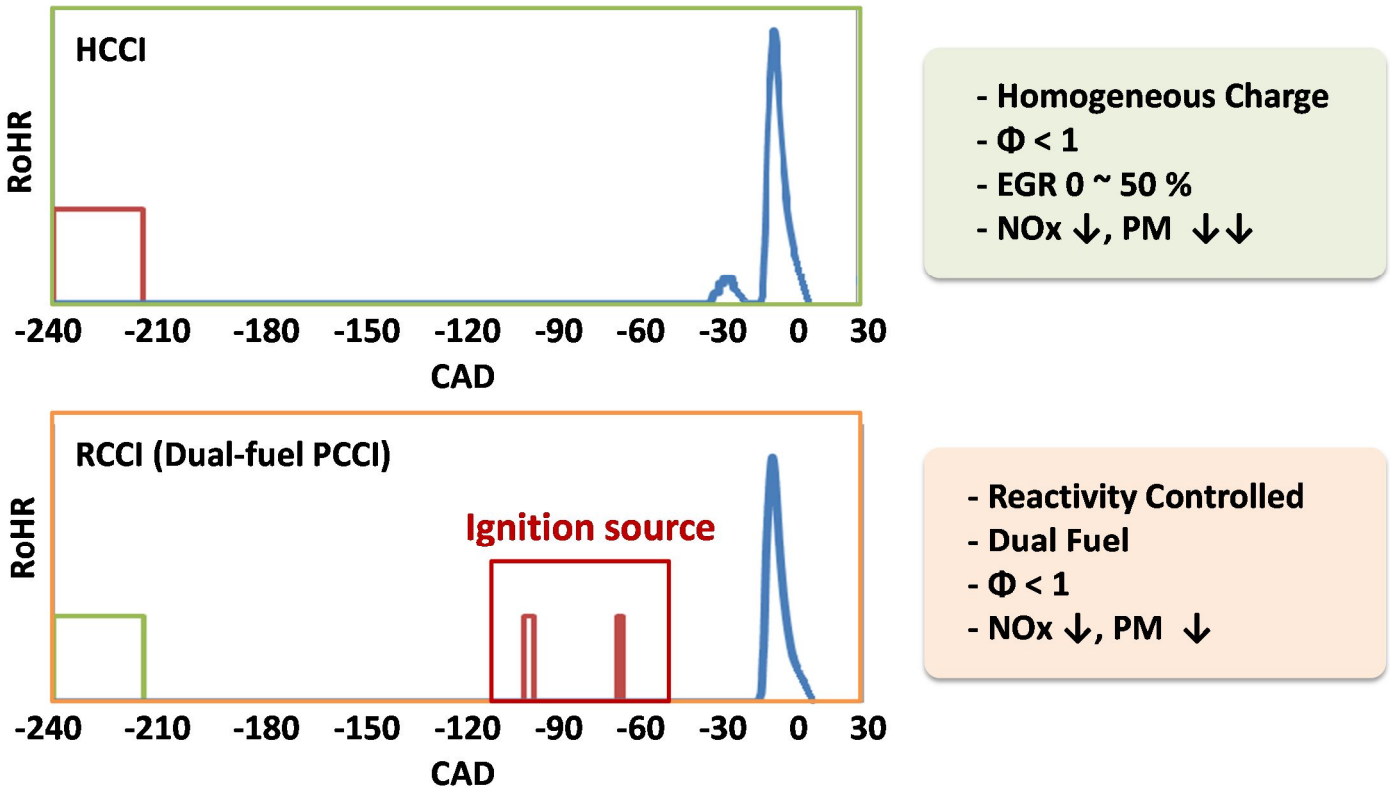


Figure 1.7 Conceptual diagrams of HCCI and RCCI combustion and injection strategies and simple explanations [22]

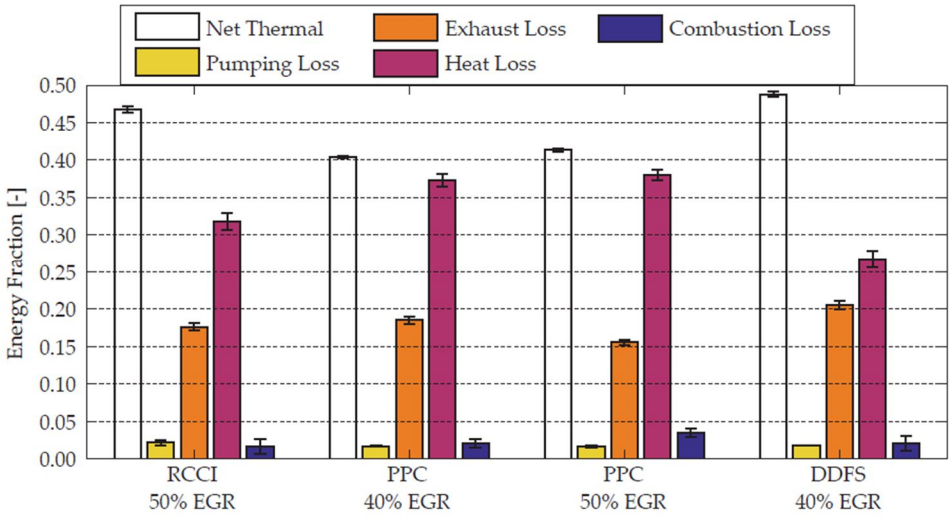


Figure 1.8 Indicated thermal efficiency and each loss of RCCI, PPC and DDFS (one of dual-fuel combustion) concepts [35]

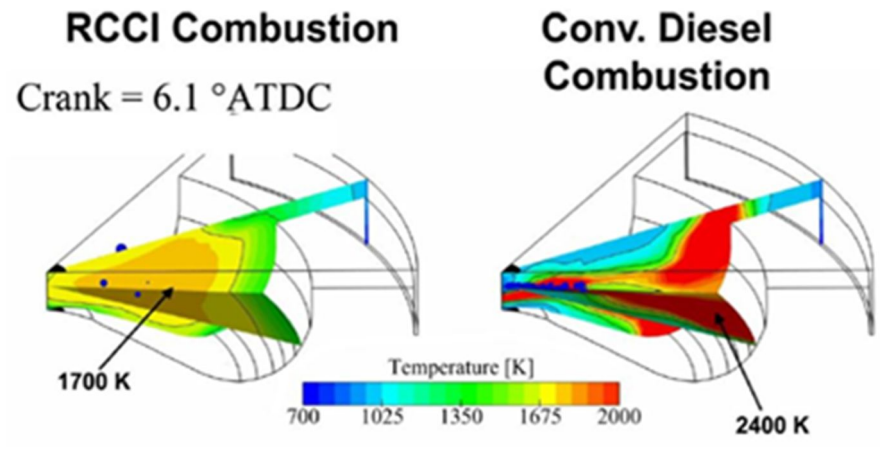
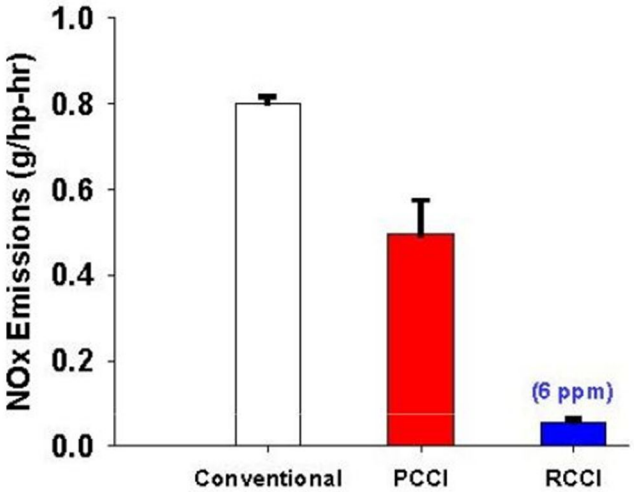


Figure 1.9 Calculated combustion temperature of RCCI and conventional diesel combustions at ATDC 6.1 CA from CFD analysis [30]



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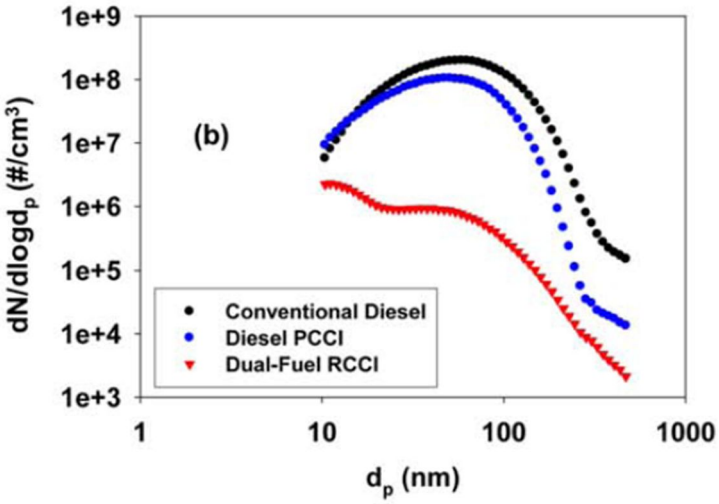


(b)

Figure 1.10 Comparisons of NOx emissions (a) and PM size distributions (b) among conventional diesel, diesel PCI, and RCCI combustion concepts [31]

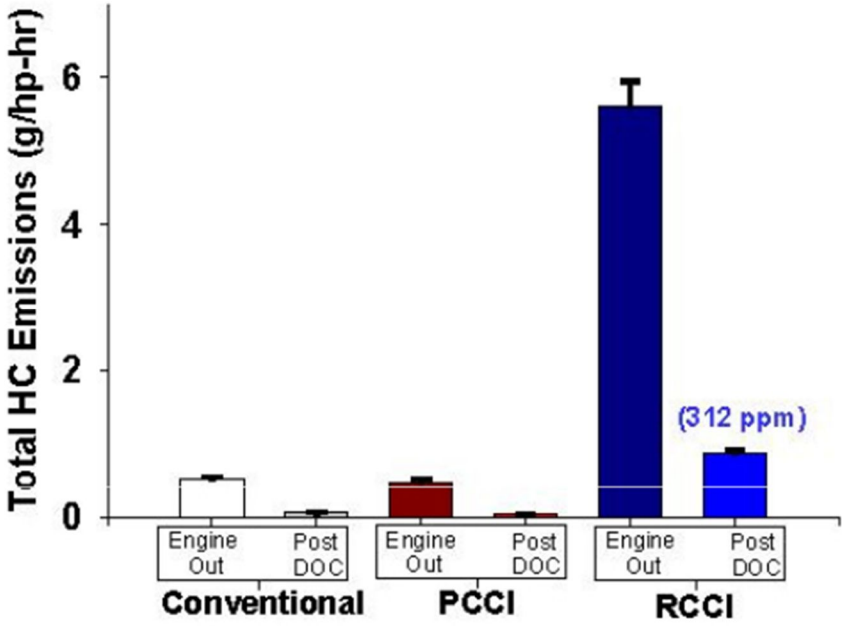


Figure 1.11 Comparisons of engine-out and post DOC_THC emissions among conventional diesel, diesel PCI, and RCCI combustion concepts [31]

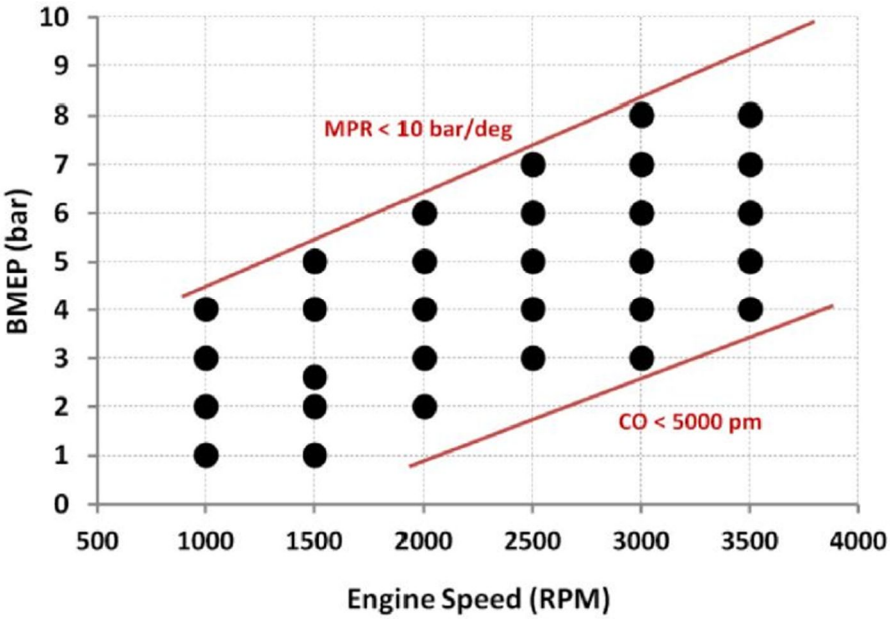


Figure 1.12 Current RCCI operating range of light-duty CI engine [34]

1.2 Motivation and Objective

In this study, the classification of dual-fuel combustion modes, especially dual-fuel premixed compression ignition (PCI), and the verification of effect of dual-fuel PCI under various operating ranges were mainly discussed. Also, from the results of the classification of dual-fuel combustion modes, proper dual-fuel PCI operating strategies for relatively low- and high- load conditions were suggested. Under low load conditions, higher thermal efficiency was mainly focused with low NOx and PM emissions which can satisfy EURO-6 regulation without any after-treatment systems. On the other hand, to extend high load condition, preventing higher PRRmax was the main target, also with maintaining low emissions level and high thermal efficiency.

Experiments were separated into two parts. The first experiment sequence was the investigation of dual-fuel combustion by using propane and diesel fuels. Propane was selected as a low reactivity fuel, because this fuel can be supplied as a gaseous state. Thus, extra injectors for the low reactivity fuel were not needed and side effect, especially liquid fuel film on the port, can be prevented with the base single cylinder research CI engine.

And the second experiment part was performed with the improved single cylinder research engine, which was optimized for dual-fuel combustion concept especially. Thus, in the second part, gasoline fuel was selected as a low reactivity fuel for the regular dual-fuel combustion experiments. Since gasoline and diesel fuels are widely used in internal combustion engine vehicles, gasoline and diesel dual-fuel test can be more helpful to the preparation for the mass production of dual-fuel engine systems.

The detailed objectives of this study are;

1. Classification of dual-fuel combustion modes by the analysis of heat release rate (HRR) shape: Investigation into the behavior of low reactivity fuel combustion

2. Investigation into the relation between combustion parameters and dual-fuel combustion modes

3. Definition and operating strategy of dual-fuel PCI

4. Investigation into the competitiveness of dual-fuel PCI combustion comparing to other advanced CI combustions (diesel-PCI)

5. Verifying the robustness of dual-fuel PCI under various engine operating conditions and the extension of high loads & thermal efficiency potential

2. Experimental Apparatus

2.1 Single cylinder diesel engine

2.1.1 Engine specifications

A single cylinder diesel engine (Figure 2.1) was used for the fundamental dual-fuel combustion of propane and diesel. This engine has fuel injection equipment (FIE) including a piezo injector, a common rail, and a high-pressure pump, which enables injection pressure up to 1,600 bars. FIE was controlled by Bosch ECU version of EDC 16, which allows maximum five injections (two pilots – main – two posts). Detailed specification of engine and fuel injection equipment are listed in Table 2.1 and 2.2.

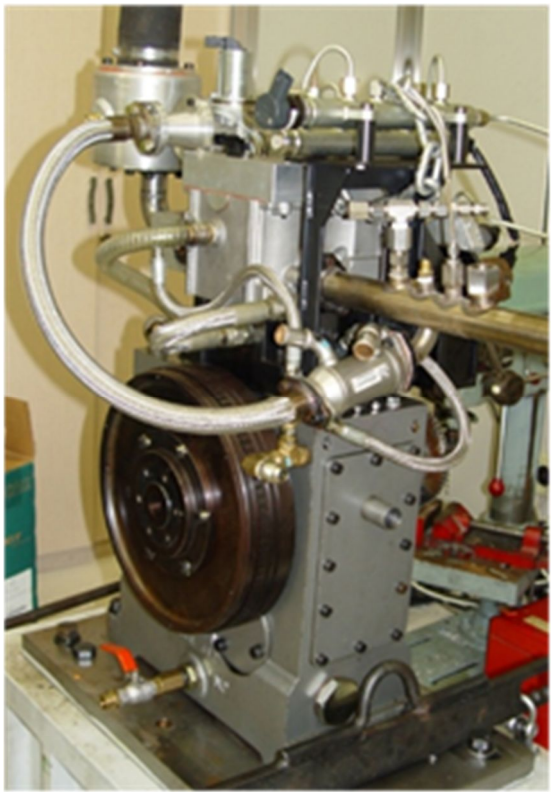


Figure 2.1 Single cylinder general research CI engine

Table 2.1 Engine Specification of the single cylinder general research CI engine

Description	Specification
Engine Type	4-stroke DI
Bore x Stroke (mm)	83 x 92
Displacement (cc)	497
Compression Ratio	15.5
Con. Rod Length (mm)	145.8
Valve Timing	IVO : 7° BTDC
	IVC : 43° ABDC
	EVO : 52° BBDC
	EVC : 6° ATDC

Table 2.2 FIE (Fuel injection equipment) Specification

Description	Specification
Manufacturer	Bosch
No. of Nozzle Holes	7
Spray Angle (°)	153
Hole Diameter (mm)	0.128
HFR (cc/100 bar/30 s)	380
High-Pressure Pump (Max. P)	CP3.2 (1,600 bars)
ECU Version	EDC 16 P373_V60

One of intake ports is a helical port (right side) for swirl and the other is a tangential port (left side). Swirl was controlled by SCV (swirl control valve) which is installed on the tangential port.

Conventional bowl (P-A) and two-step bowl type pistons were used shown in Figure 2.2. Conventional piston bowl was used for the test of the effect of diesel injection strategy on the dual-fuel combustion (Chapter 3.3), because this conventional bowl shape has suitable targeting for the diesel spray angle. On the other hand, since wider piston bowl is beneficial to premixed combustion condition such as dual-fuel combustion with supplying a small amount of diesel fuel, 2-step bowl was used for the experiment of the effect of diesel injection timings to verify the dual-fuel combustion modes (Chapter 3.1 and 3.2).

(a) (b)

Figure 2.2 Pictures of piston bowl shape (P-A) (a) and (P-B) (b)

2.1.2 Propane supplying system

In Chapter 3, propane was selected as the low-reactivity fuel to reduce the wall-wetting on the intake port [38]. The necessary amount of propane was fumigated into the intake port as gas. The quantity of propane is measured by a flowmeter (MK Precision Co., MFM (TSM-230)) (Table 2.3). Additionally, to maintain the propane gas rate throughout the experiment, the flow rate is controlled using sonic orifices by choke flow. The experimental setup of fuel supplying system is depicted in Figure 2.3. Additionally, the diesel fuel and propane gas properties are presented in Table 2.4.

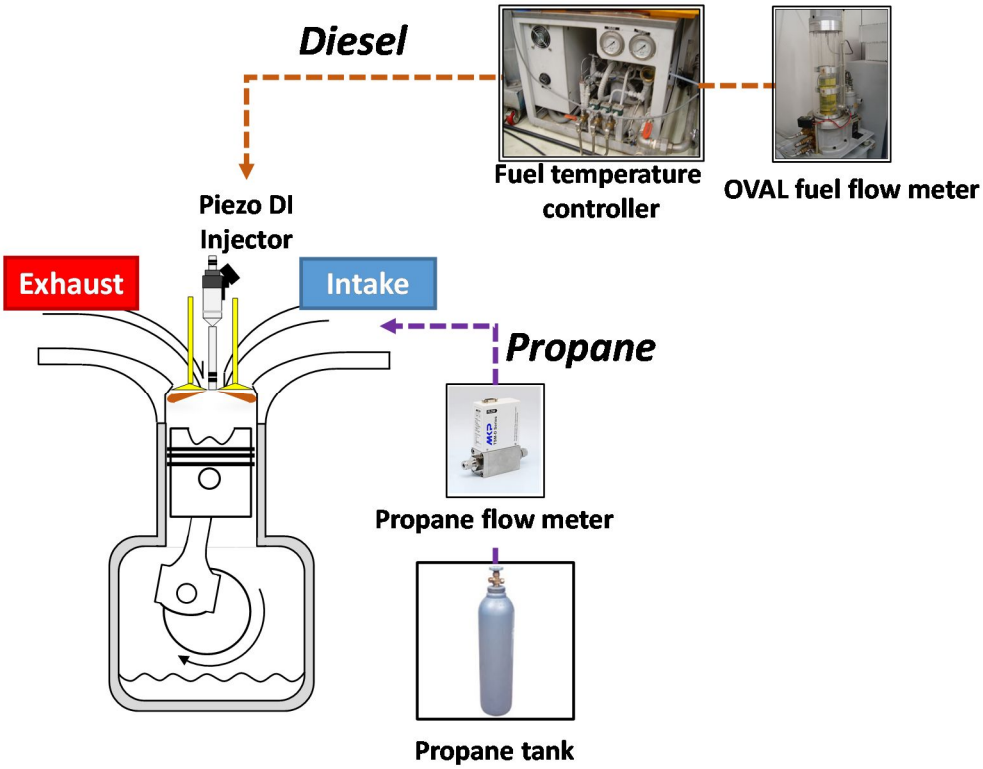


Figure 2.3 Schematic diagram of propane and diesel supplying system

Table 2.3 Specification of propane flow meter

Item	Specification
Manufacturer	MKP
Model	TSM-230
Flow range (SLM)	0 ~ 100
Accuracy (% FSO)	Lower than ±1
Linearity (% FSO)	Lower than ±0.2
Repeatability (% of reading)	Lower than ±0.2
Operating temperature (°C)	15 ~ 35

Table 2.4 Properties of diesel and propane

Properties	Diesel	Propane
Chemical formula	$C_xH_{1.8x}$	C_3H_8
Molecular weight [g/mol]	190-220	44.1
Density [g/cm^3]	0.831	21.7 (gaseous)
Low heating value [MJ/kg]	44	46.33
Auto-ignition temperature [K]	523	763
Stoichiometric ratio of AF [wt.%]	14.6	15.6

2.2 Single cylinder dual-fuel research CI engine

2.2.1 Engine specifications

A single cylinder dual-fuel research CI engine (Figure 2.4) was used for dual-fuel combustion with gasoline and diesel test. This engine was designed to be optimized for the dual-fuel combustion especially. FIE is including a DI solenoid injector, a common rail, and a high-pressure pump, which enables injection pressure up to 1,800 bars and PFI solenoid injector, whose injection pressure is 5 bar. DI system was controlled by Bosch ECU version of EDC 17, which allows maximum 5 injections (Two pilots– main – Two posts) and also PFI system was controlled by Bosch ECU. Detailed specifications of engine and fuel injection equipment are shown in Tables 2.5, 2.6 and 2.7.

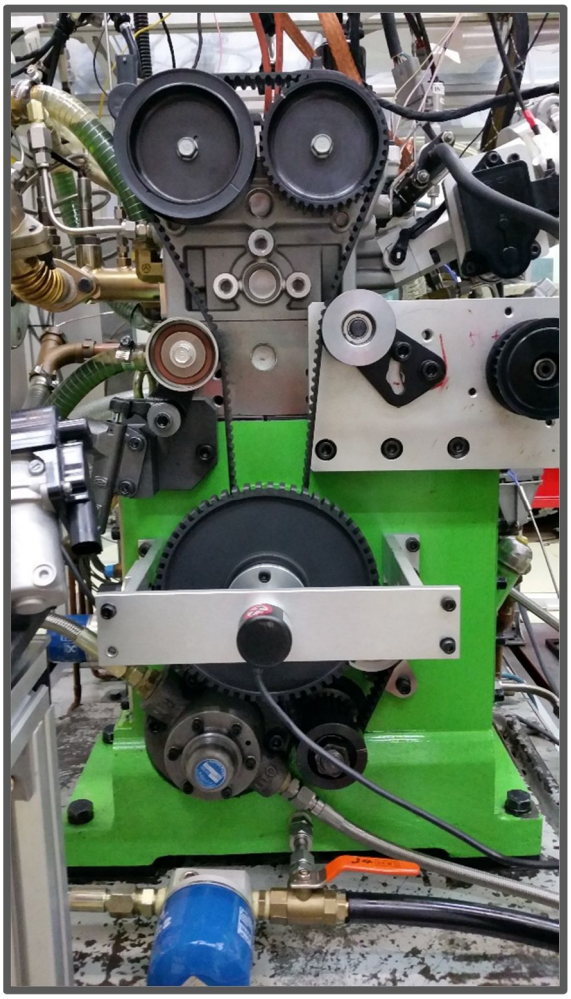


Figure 2.4 Picture of single cylinder dual-fuel research engine

Table 2.5 Engine Specification of the single cylinder dual-fuel research engine

Description	Specification
Engine Type	4-stroke DI
Bore x Stroke (mm)	77.2 x 84.5
Displacement (cc)	395.5
Compression Ratio	14.0
Con. Rod Length (mm)	140.0
Valve type	4-valves Intake CVVD cam
Valve Timing	IVO : 8° BTDC
	IVC : 36° ABDC
	EVO : 48° BBDC
	EVC : 4° ATDC

Table 2.6 Diesel FIE (fuel injection equipment) Specification

Description	Specification
Manufacturer	Bosch
No. of Nozzle Holes	6
Spray Angle (°)	130
Hole diameter (mm)	0.1
High-Pressure Pump (Max. P)	CP3.2 (1,800 bars)
ECU Version	EDC 17 P373_V60

Table 2.7 Gasoline FIE (fuel injection equipment) Specification

Description	Specification
Manufacturer	Denso
Operating type	Solenoid
Injection pressure (bar)	5
Gasoline injection timing for all the cases in this research (°ATDC)	-300

2.2.2 Introduction to improved systems for dual-fuel combustion

Single cylinder dual-fuel research CI engine was designed to improve dual-fuel combustion concept, as it was mentioned above. Mainly three systems were improved.

The first improved part is a PFI injection system. Actually, contrary to propane, since gasoline is usually injected as a liquid state, PFI system is essential for the experiment of gasoline and diesel dual-fuel combustion. Thus, two PFIs were equipped and the gasoline sprays from PFI aimed at the bottom area of intake valves. As a result, liquid fuel film effect can be reduced considerably. The conceptual figure of engine head and PFI system is described in Figure 2.5.

The second improved part is an LP-EGR system with a supercharger. To achieve low engine-out NOx emission as EURO-6 standard, higher EGR rate than conventional HP-EGR system is needed. Thus, in this experimental system, EGR and air are compressed simultaneously by using a supercharger system and compressed air and EGR mixture is cooled by an intercooler (Figure 2.6).

The last improved systems are narrower diesel cone angle and bathtub shaped piston bowl (Figure 2.7). Since bathtub shaped bowl is suitable for the premixed combustion system such as dual-fuel combustion, center edge of piston bowl was removed [39]. Also, in order to reduce diesel fuel wall-wetting, spray cone angle was reduced from 150 ° to 130 °.

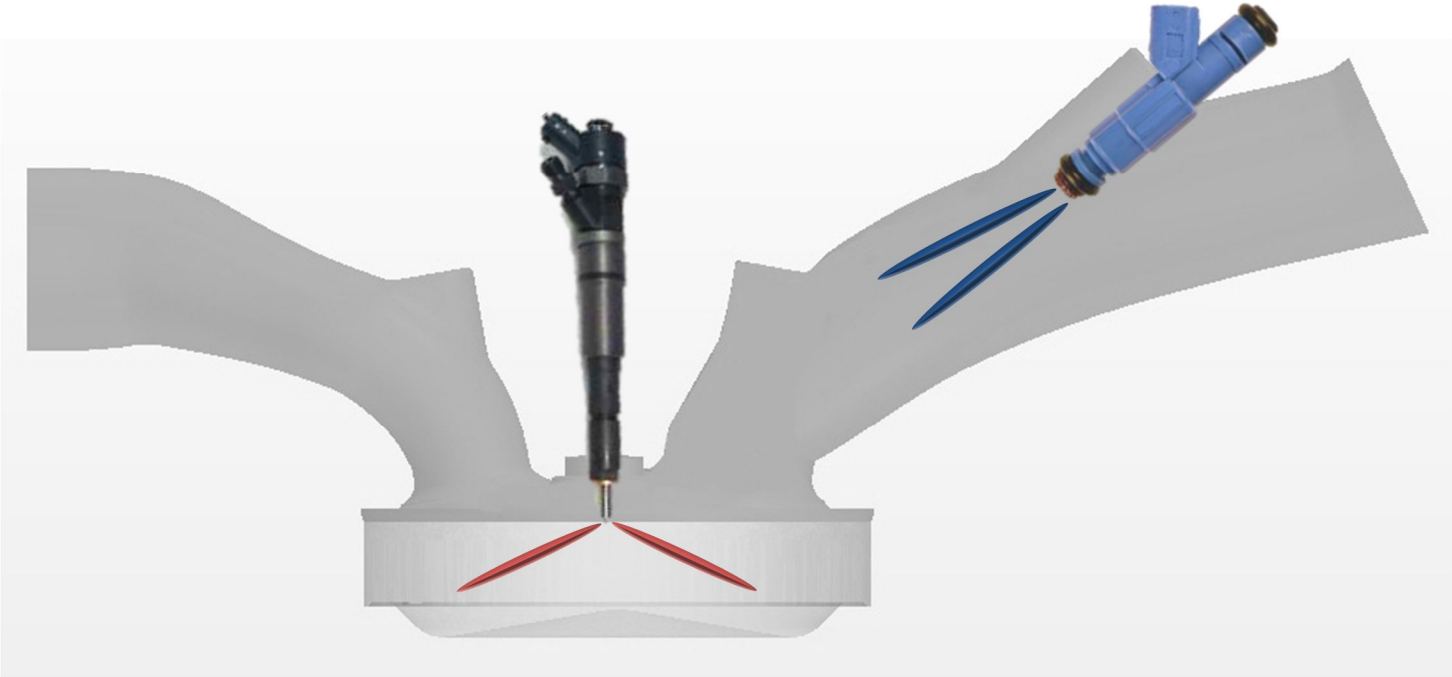


Figure 2.5 Schematic diagram of DI and PFI spray targeting

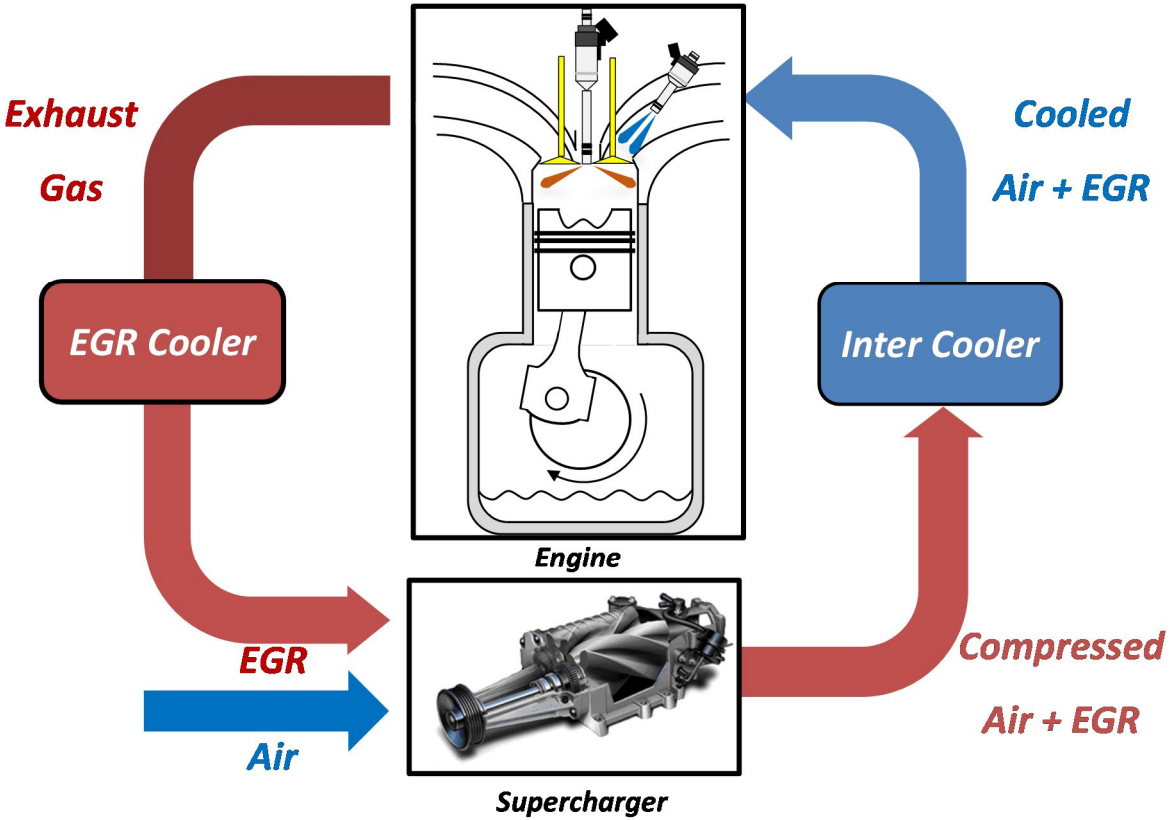


Figure 2.6 Schematic diagram of LP-EGR system

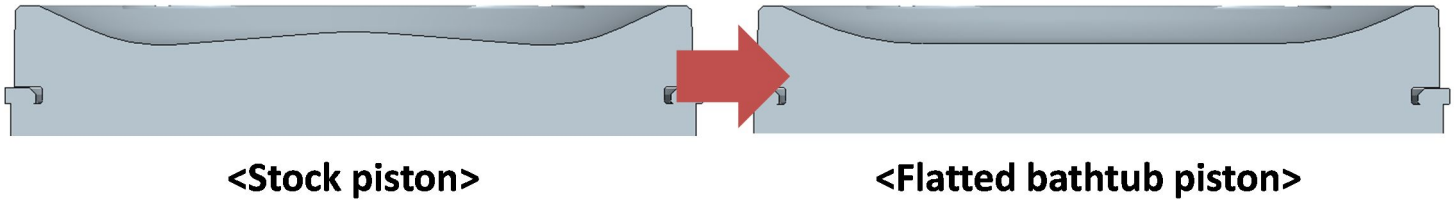


Figure 2.7 Schematic diagram of piston bowl shape

2.3 Engine test and measurement equipment

In this part, test and measurement equipment for the experiment will be introduced. The schematic diagrams of test and measurement equipment are shown in Figure 2.8 and 2.9.

2.3.1 Engine test equipment

The test engine is linked to a 37 kW DC dynamometer for engine speed and torque control. The specifications of the dynamometer are introduced in Table 2.8. Oil and coolant temperature were kept constant during the engine test using the PID controllers, equipped in dynamometer control system. The quality of diesel fuel was preserved from the large capacity fuel tank and fuel temperature was maintained as 40 °C using the fuel temperature controller (SAMBU, SFTC-140). The temperature of the engine test cell is controlled by an air-conditioning system.

To control injection parameters such as injection pressure, timing, number, and quantity, an emulation ECU was used. The ECU was connected to the control computer using the communication module (ETAS, ES-591) and engine parameters were monitored and controlled in the interface of INCA program.

Table 2.8 Specification of dynamometer

Item	Specification
Manufacturer	DAVID McClure LIMITED
Model	G-Cussions
Capacity	37 KW
Type	DC
Max. rpm	7,000
Cooling type	Air cooling
Weight	396 kg

2.3.2 Measurement of fuels flows and EGR rate

Diesel fuel mass flow rate was measured with a mass burette type flow meter (Table 2.9). Integrated fuel mass of 100 seconds was measured and the injection mass of one cycle was calculated from the integrated mass.

Gasoline fuel mass flow rate was measured by a test using the fuel mass flow meter (Table 2.10). Integrated fuel mass of 30 seconds was measured and the injection mass of one cycle was calculated from the integrated mass.

EGR rate was calculated with the CO_2 fractions in intake and exhaust gasses, which was measured by the gas fraction analyzer. The ambient CO_2 fraction was 0.043 % as the volume unit. Calculation equation is shown in Equation 2.1.

$$\text{EGR rate [\%]} = 100 * \frac{CO2)int - CO2)amb}{CO2)exh - CO2)amb} \tag{2.1}$$

Table 2.9 Specification of fuel mass flow meter for diesel

Item	Specification
Manufacturer	ONO SOKKI
Type	Mass burette type
Model	FX-203P
Range of flow amount (g/sec)	0 ~ 50
Max integration (g)	1,000
Resolution (g/sec)	0.01
Minimum unit of integration (g)	0.1
Withstand pressure (kg/cm^2)	2
Operating temperature (°C)	0 ~ 40

Table 2.10 Specification of fuel mass flow meter for gasoline

Item	Specification
Manufacturer	AVL
Measurement principle	Mass flow
Model	7030 flow meter
Range of flow amount (kg/h)	0 ~ 125
Measurement uncertainty (%)	<0.12
Measurement frequency (Hz)	20 (analog)
Response time (ms)	< 125
Operating temperature (°C)	0 ~ 80

2.3.3 Measurement of exhaust gas emissions

Exhaust gas composition of O_2 and CO_2 and emissions of NOx, THC, CO and PM were measured during the test. Exhaust gas was sampled to each device as shown in Figure 2.8 and 2.9. PM was measured in FSN (Filtered Smoke Number) by the smoke meter. Specifications of the smoke meter are shown in Table 2.11. To measure the air to fuel ratio during the test, AFR analyzer in Table 2.12 is installed. NOx, THC, CO, CO_2 and O_2 were measured by the exhaust gas analyzer (HORIBA, MEXA-7100DEGR). All species are measured in volume fraction (vol % / ppm) in wet condition. The measured principles of each emission are listed in Table 2.13.

Table 2.11 Specification of smoke meter

Item	Specification
Manufacturer	AVL
Model	AVL 415S
Measurement range	0 ~ 10 FSN / 0 ~ 32,000 mg/m³
Resolution	0.001 FSN / 0.01 mg/m³
Repeatability (as standard deviation)	$\sigma \leq \pm$(0.005 FSN + 3 % of measured value)
Reproducibility (as standard deviation)	$\sigma \leq \pm$(0.005 FSN + 6 % of measured value)

Table 2.12 Specification of AFR analyzer

Item	Specification
Manufacturer	HORIBA
Model	MEXA-110λ
Measurement range (H/C = 1.85)	A/F 10.0 ~ 30.0
Accuracy (H/C = 1.85)	±0.3 A/F when 12.5 A/F ±0.1 A/F when 14.7 A/F ±0.5 A/F when 23.0 A/F
Exhaust gas temperature (°C)	-7 ~ 900 (recommend 200 ~800)

Table 2.13 Measurement principle of emission analyzer (MEXA-7100DEGR)

Emissions	Measurement principle
NOx	Chemiluminecent Detector
THC	Flame Ionization Detector
O_2, CO_2, CO	Non Dispersive Infrared Rays

Air to fuel ratio was calculated using the measured O_2, CO_2, CO and THC by Spindt equation [1]. Equations are as follows.

$$A/F = \frac{(CO_2 \times 10000 + CO)}{(CO_2 \times 10000 + CO + THC)} \times (AA + BB) \tag{2.2}$$

$$AA = \frac{11.4919 \times FC \times \left(1 + \frac{CO/(10000 \times CO_2)}{2} + \frac{O_2}{CO_2}\right)}{\left(1 + \frac{CO}{(10000 \times CO_2)}\right)} \tag{2.2.1}$$

$$BB = \frac{119.8074 \times (1 - FC)}{\left(3.5 + \frac{CO}{(10000 \times CO_2)}\right)} \tag{2.2.2}$$

$$FC = \frac{12.011}{\left(12.011 + 1.008 \times HC\_ratio\right)} \tag{2.2.3}$$

2.3.4 Measurement of pressure and combustion analysis

Intake port and cylinder pressure were measured to analyze the intake flow and combustion phenomena. The cylinder pressure was measured by relative pressure transducer 6055Bsp (Table 2.14) through the glow plug type adapter and amplified by charge amplifier (Kistler, 5019 A). The intake port pressure was measured by absolute pressure transducer 4045A5 (Table 2.15) and amplified by a piezoresistive amplifier (Kistler, 4603). Signals from pressure transducers were recorded at every 1˚CA for 100 cycles by using a data acquisition system. Analog signals were converted to digital by data acquisition board (Table 2.16) and recorded by the in the data acquisition program.

From the measured cylinder pressure, the rate of heat release (ROHR), indicated mean effective pressure (IMEP) and coefficient of variation (COV) of IMEP were calculated to analyze the combustion phenomena. ROHR was calculated by the single zone model. The equations are as follows [1].

$$\frac{dQ_n}{dt} = \frac{\gamma}{\gamma - 1} p \frac{dV}{dt} + \frac{1}{\gamma - 1} V \frac{dp}{dt} \tag{2.3}$$

In the equations, dQ_n / dt means apparent net heat release rate. Specific heat ratio γ is assumed to 1.35. The heat release calculation is started from the injection timing.

Gross indicated thermal efficiency, combustion loss, and exhaust loss were calculated by below equations (Equation 2.4, 2.5 and 2.6). Heat loss was assumed the rest of total fuel energy excluding above three parts (Equation 2.7).

$$\text{Gross indicated thermal efficiency} = \frac{W_{gross}}{m_{total\ fuel} \times Q_{LHV\ of\ fuel}} \tag{2.4}$$

$$\text{Combustion loss} = \frac{m_{THC\ of\ each\ cycle} \times Q_{fuel} + m_{CO\ of\ each\ cycle} \times Q_{CO}}{m_{total\ fuel} \times Q_{LHV\ of\ fuel}} \tag{2.5}$$

$$\text{Exhaust loss} = \frac{m_{exh} \times h_{exh} - m_{int} \times h_{int}}{m_{total\ fuel} \times Q_{LHV\ of\ fuel}} \tag{2.6}$$

$$\text{Heat loss} = 1 - (\text{GIE} + \text{Combustion loss} + \text{Exhaust loss}) \tag{2.7}$$

Table 2.14 Specification of the cylinder pressure transducer

Item	Specification
Manufacturer	Kistler
Model	6055Bsp
Pressure sensor type	Relative pressure
Measuring range (bar)	0 ~ 250
Overload (bar)	300
Linearity (% FSO)	< ±0.4
Sensitivity (pC/bar)	-18.3
Natural frequency (kHz)	160
Operating temperature (°C)	-20 ~ 350

Table2.15 Specification of the intake port pressure transducer

Item	Specification
Manufacturer	Kistler
Model	4045A5
Pressure sensor type	Absolute pressure
Measuring range (bar)	0 ~ 5
Linearity (% FSO)	< ±0.3
Natural frequency (kHz)	> 80
Operating temperature (°C)	20 ~ 120 (compensated)

Table2.16 Specification of combustion analyzer

Item	Specification
Manufacturer	Kistler
Model	Kibox to go 2893
Channels	8
Sample rate (MHz)	2.5
The minimum pulse duration (us)	3.2
Uncertainty (ms)	Approximately 1(<<1 cycle)
Resolution of measurement data (kHz and °CA)	312.5/ 0.1

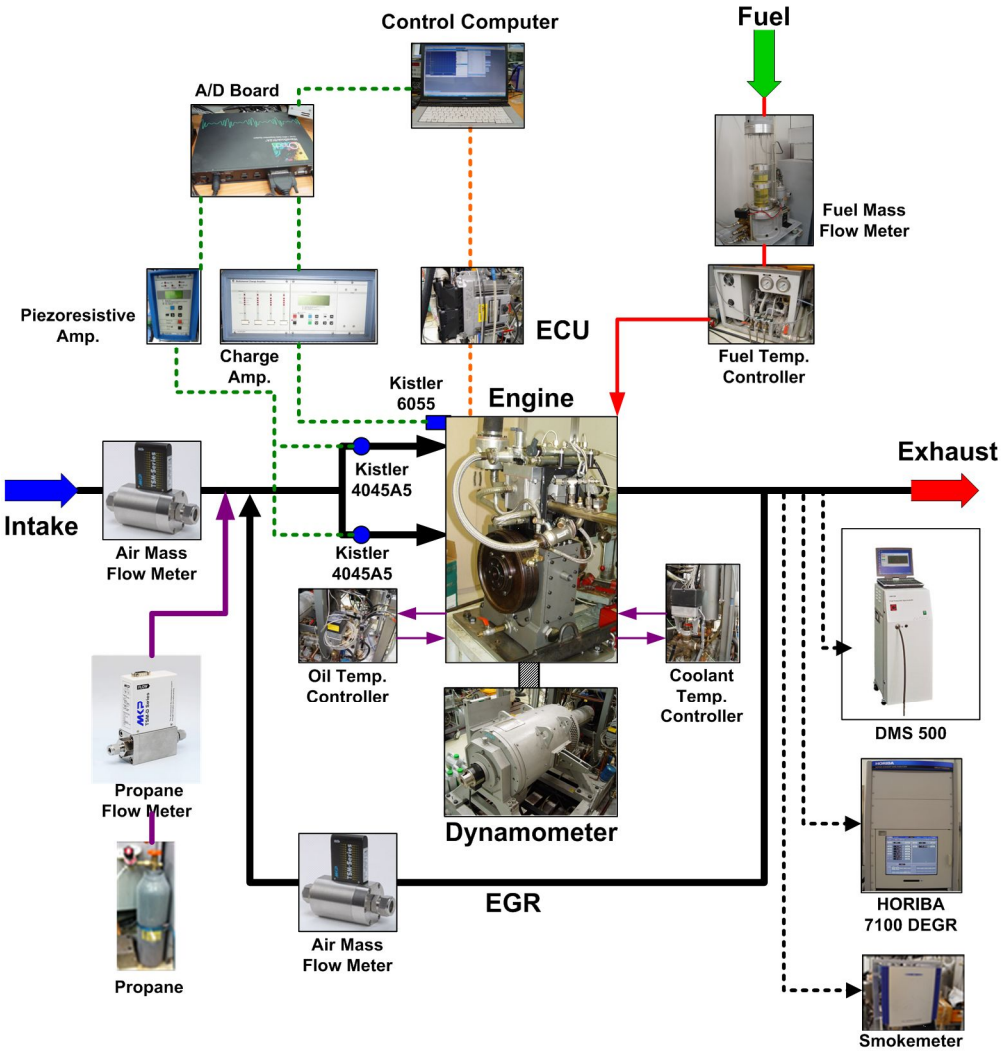


Figure 2.8 Schematic diagram of single cylinder diesel engine test and measurement equipment

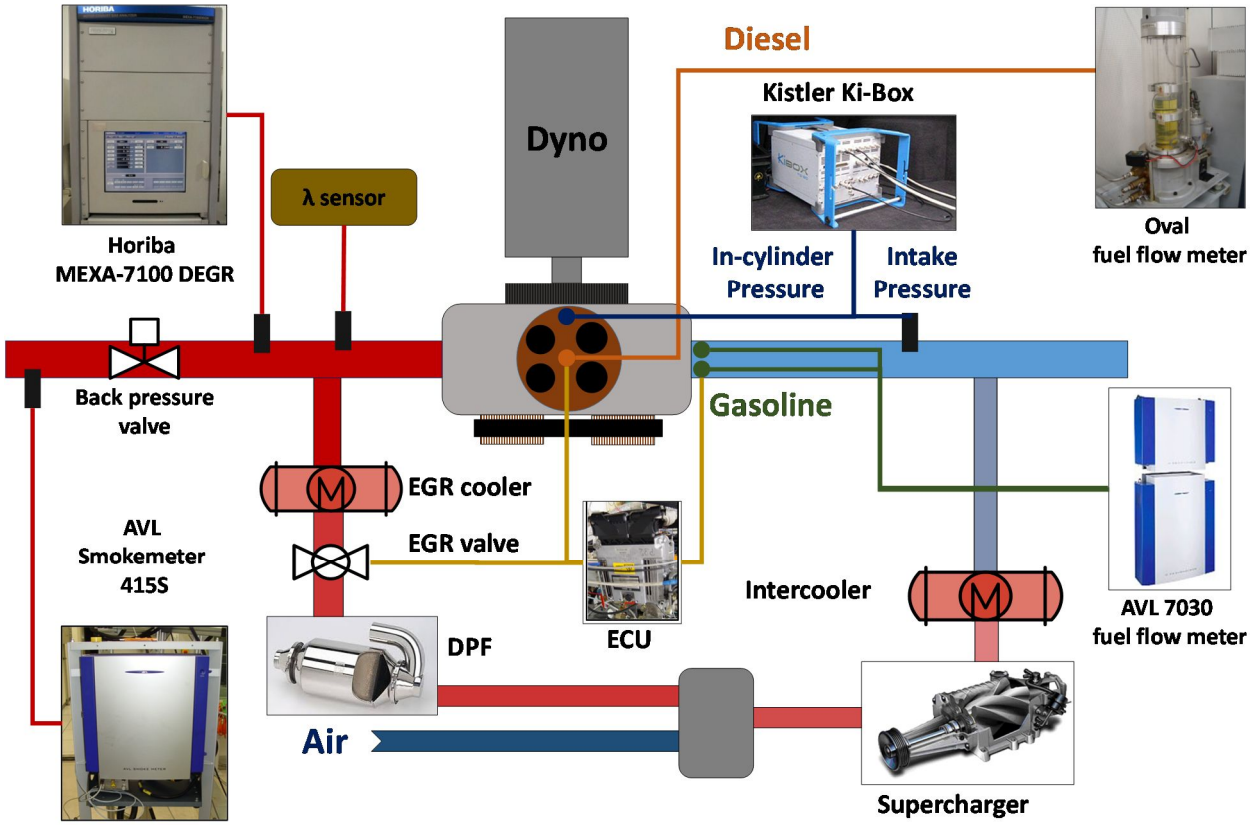


Figure 2.9 Schematic diagram of single cylinder dual-fuel research engine test and measurement equipment

3. Experimental Results of Dual-fuel Combustion with propane and diesel

First of all, basic dual-fuel combustion experiment was performed with the single cylinder general research engine. Since this engine did not optimize for the dual-fuel combustion, especially this engine was not equipped with PFI injectors, propane gas was selected as a low reactivity fuel. Propane was supplied into intake port by just a fumigation. Then, most experiments were performed under the relatively low load and low speed condition.

In this chapter, the basic characteristics of dual-fuel combustion were mainly discussed such as diesel injection timing, intake/injection pressures of diesel and the diesel injection strategy on dual-fuel combustion. Thus, the characteristics of dual-fuel combustion known from this chapter became the base data for the dual-fuel combustion experiment with the improved hardware system in the next chapter.

3.1 Characteristics of dual-fuel combustion as varying diesel injection timing: Introduction to the early diesel injection dual-fuel combustion

3.1.1Experimental conditions

As the first experiment, diesel injection timing was varied from the conventional timing (5 ° BTDC) to earlier state (30 ° BTDC) under propane 75 % and diesel 25 % conditions, which was calculated based on LHV values. Thus, from this experiment, the behavior of combustion phase under dual-fuel combustion was investigated as different diesel SOI, which means the stratification grade of diesel fuel. Detailed experimental conditions were introduced in Table 3.1.

Table 3.1 Base engine operating conditions for the experiments of diesel injection timing effect on the dual-fuel combustion

Description	Value
Engine speed [rpm]	1,500
gIMEP [bar]	5.2~5.5
Coolant & oil temperature [K]	358
Total LHV for each stroke [J/str]	680
Diesel injection pressure [bar]	750
Propane supplying pressure [bar]	2.5
Intake pressure [bar]	1.13
EGR rate [%]	0
Propane ratio (LHV base) [%]	75
Diesel SOI [°BTDC]	5/(10)/20/30

3.1.2 Experimental results and analysis

In Figure 3.1, heat release rate and in-cylinder pressure traces are depicted as varying diesel SOI conditions. When diesel SOI was at 5 °BTDC, combustion information from heat release rate graph was very similar with those of conventional diesel combustions as usual. At first, premixed combustion occurred from premixed diesel and propane fuels which were entrained into the diesel spray. It was a natural phenomenon, but following combustion state would be different from the conventional diesel combustion.

Usually, most of the conventional diesel combustion is constructed by major two parts. The first one is a premixed combustion from the early injected diesel fuel between the start of injection (SOI) and the start of combustion (SOC). And the second phase is a diffusion flame phase form the later injected diesel fuel after the SOC [1]. Since diesel fuel which was injected after the SOC could not mix well with air, most of the PM emission was emitted during this period. From this point of view, there was rarely diffusive flame in late combustion phase, since PM emission was lower than 0.2 FSN which is a tenth of PM emissions from the conventional diesel combustion.

When diesel injection timing was advanced at the 20 °BTDC, the peak of heat release rate became lower and the main combustion duration was prolonged. Since more diesel fuels were premixed compared to the first case, local rich regions might be reduced. Thus, combustion became slower as reactivity stratification was smaller. Also, at the last part of the main combustion, the second peak of HRR can be shown and then, combustion dropped sharply. This phenomenon can be regarded as the auto-ignition of end gas, i.e. residual propane, due to surrounding temperature and pressure were increased due to the first combustion from diesel and propane fuels

[40]. Therefore, it was different from the first case that late combustion phase was diminished.

For the last case, when diesel injection timing was advanced at the 30 °BTDC, ignition delay which is the duration between the end of injection (EOI) and SOC became distinctly increased. Also, the most remarkable characteristic was that combustion phase became very smooth like a flame propagation of SI combustion. However, this combustion might be not a flame propagation, because overall-equivalent ratio was too lean to propagate. Even though there was the possibility to the propagation of propane, the turbulent intensity was not enough to make faster combustion speed like this. Therefore, this combustion was simply the stratified auto-ignition of two fuels gradually.

Especially, SOC underwent the transient sequence as diesel injection timing was advanced in Figure 3.2. This phenomenon was related the peak equivalent ratio prior to the main combustion from the auto-ignition. Since diesel fuel stratification became weakened as diesel injection timing was advanced [41, 42]. Therefore, this combustion mode could be considered as one of 'PCI combustion', because this combustion has enough long ignition delay and is based on the chemical kinetics reaction.

NOx emission and other combustion parameters such as the peak HRR, the peak of in-cylinder pressure and ignition dwell as the varying diesel injection timings under the dual-fuel combustion condition are depicted in Figure 3.2. For all cases, PM emission was not over 0.2 FSN which can satisfy the EURO-6 emission regulation without DPF system. Then, NOx emission had the good relation with these three parameters. As the peak in-cylinder pressure was higher, NOx emission was also high because the high in-cylinder pressure was originated from high in-

cylinder temperature, especially. Also, shorter ignition dwell, in this case, it was same with ignition delay because injection duration was fixed, brought a high level of NOx emission [43, 44]. It was related with the local equivalent ratio. Since there was not enough time to mixing well, high local equivalent ratio brought high-temperature regions which are a prompt condition for high thermal NOx emission.

Last, the peak of HRR was related with NOx emission in this experiment especially. As usual, higher the peak HRR did not mean a high level of NOx emission always, for example, HCCI combustion. However, in the dual-fuel combustion, the peak of HRR is needed to be lowered, because it means two fuels were undergone stratified auto-ignition gradually. Unlike neat diesel premixed combustion, premixed combustion phase of dual-fuel combustion was slow due to the low reactivity fuel [40].

Therefore, under high propane ratio condition, early diesel injection was recommended to achieve smooth and low emissions simultaneously. Especially, these results can be expected due to the stratified auto-ignition by two opposite characterized fuels. In chapter 4, differences in dual-fuel combustion with advanced hardware system were more discussed, especially aspect of thermal efficiency and emissions.

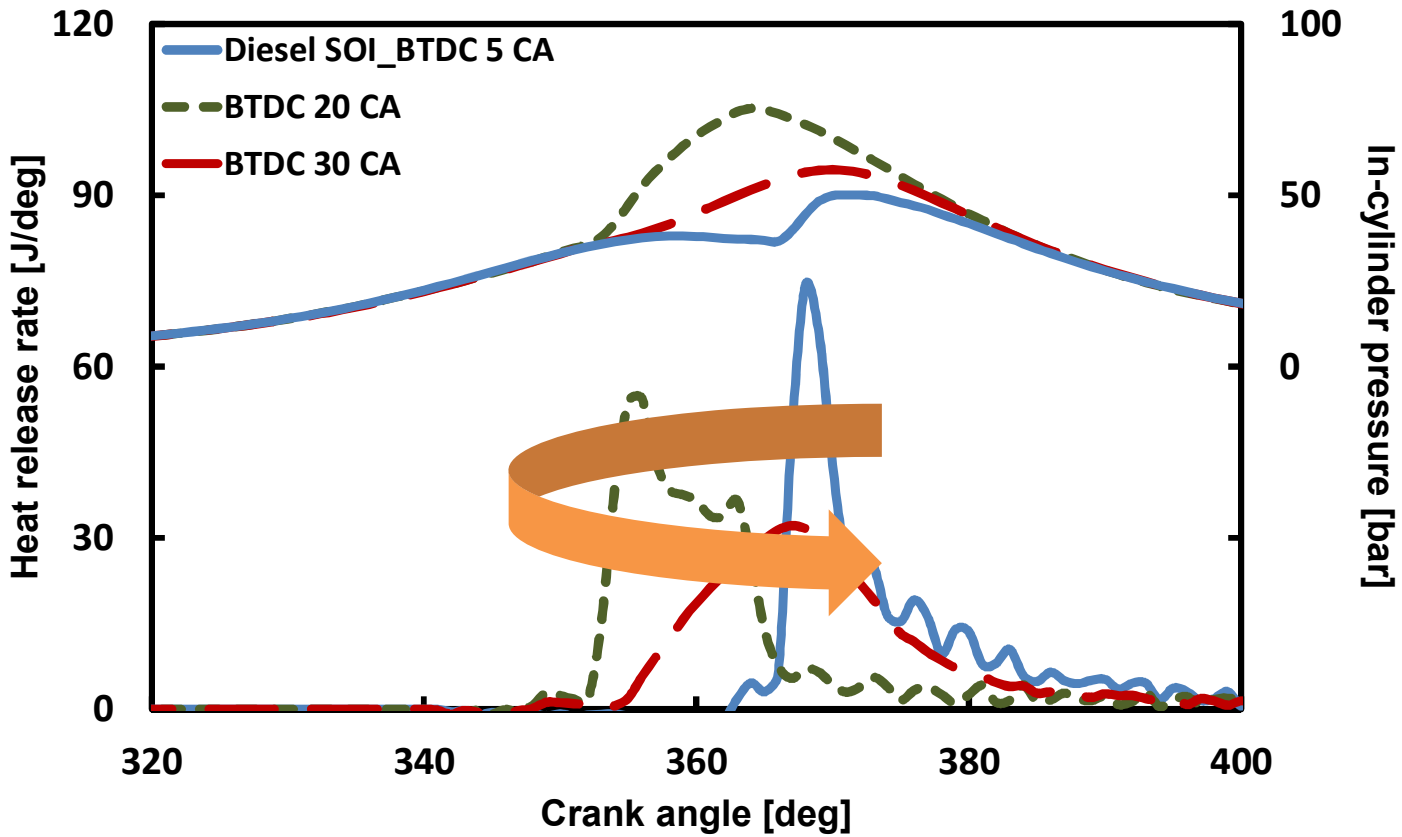


Figure 3.1 Comparisons of heat release rate and in-cylinder pressures as various diesel SOI under dual-fuel combustion

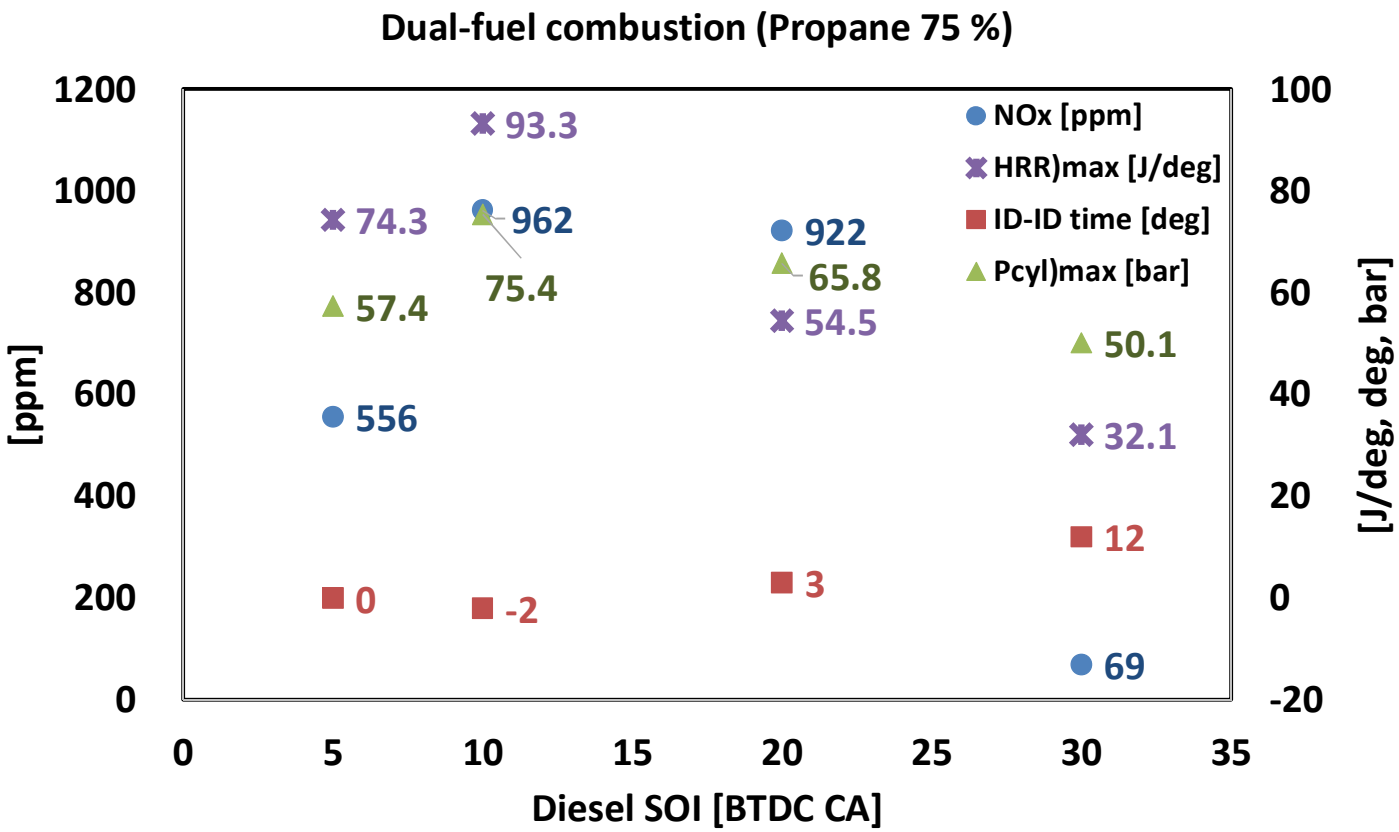


Figure 3.2 NOx and combustion parameters as various diesel injection timings under dual-fuel combustion

3.2 Effect of engine operating parameters on the early diesel injection dual-fuel combustion

Standing on the results from chapter 3.1, the effect of engine operating parameters on the early diesel injected dual-fuel PCI was studied. In addition to early Diesel injection PCI, the use of propane at conventional engine operating conditions was investigated. The possibilities for reducing engine emissions with two different methods are described in this paper. One method is the simple substitution of propane for Diesel fuel under conventional conditions without changing the boost pressure. The result supports the practical application of propane gas in conventional CI engines. The other method is dual-fuel PCI combustion without supplemental EGR. Because emissions are reduced without the use of EGR, a loss of combustion efficiency can be avoided. This result indicates stable, controllable combustion in dual-fuel combustion regimes.

The two suggested uses of propane in a CI engine yielded low PM emission (under 0.2 FSN versus 1.5 FSN for the conventional case) while maintaining low NOx emissions (40~50 ppm). Furthermore, the number of large particles (> 100 nm) was drastically reduced by the addition of propane, and the total number of particles was also reduced.

3.2.1 Experimental conditions

The experiment consisted of three parts. Part 1 was focused on dual-fuel combustion without early diesel injection and under conventional operating conditions. This experiment was performed with two different engine speed/load conditions, one at 1500 rpm/5.5 bar IMEP and the other at 2000 rpm/7.6 bar IMEP.

The propane ratio was varied from zero to the burning limit condition. This test indicates the robustness of the dual-fuel combustion emissions reduction with respect to the engine operating conditions. Although the propane ratio changed, the combustion 50% angle (CA 50) was fixed at 10° ATDC by adjusting the diesel injection timing, and the AFR was approximately 20. The AFR was calculated with the C/H ratio for the various propane ratios.

Part 2 tested the dual-fuel combustion with early diesel injection, i.e., dual-fuel PCI combustion, without any EGR. When this experiment was performed, the engine speed and the AFR were fixed at 1500 rpm and 36, respectively, using the ratio between the diesel fuel and the propane, but the IMEP was varied. Then, the Diesel injection and boost pressures were varied. The Diesel injection timing was set at 30° BTDC, and the ratio of the propane to the total fuel was maintained at 75%. The purpose of this experiment was to confirm the effectiveness of dual-fuel PCI without EGR for emissions reduction.

Part 3 compared conventional diesel combustion, dual-fuel combustion without early diesel injection and dual-fuel PCI combustion at the 1500 rpm/5.5 bar IMEP (BMEP = 4 bar) condition.

First, to examine the practical application and the robustness of propane gas addition to conventional diesel combustion, the propane ratio was varied under two different engine speed/load conditions in Part 1. The diesel injection timing was fixed at 30° CA regardless of the propane ratio, and the AFR was fixed at approximately 20 (Φ~0.74). The AFR was lower than that of previous RCCI experiments (usually Φ= 0.3~0.5) to prevent increasing the boost pressure.

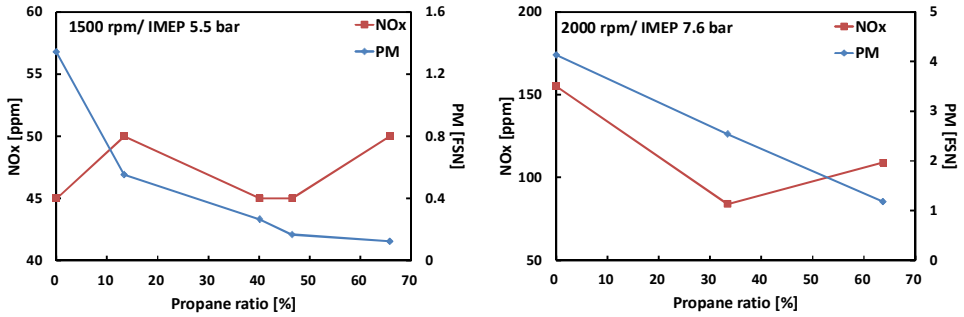
3.2.2 Experimental results and analysis

In Figure 3.3-(a), the NOx emission was essentially constant at 45~50 ppm, even though the propane ratio was increased to 68% at the 1500 rpm/ 5.5 bar IMEP condition. Because the NOx emission is affected by the local adiabatic temperature [2, 3], this result indicates that there was no significant change in the combustion temperature, even though a significant portion of the diesel fuel was substituted with propane. In Figure 3.3-(c), the ignition dwell at the 1500 rpm/ 5.5 bar IMEP condition was already long without the addition of propane. Thus, the addition of propane did not affect the ignition dwell or the combustion temperature significantly [45].

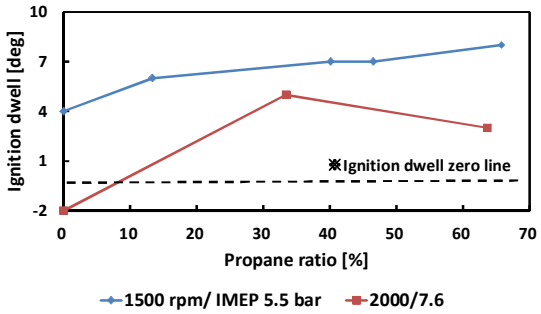
However, the NOx emissions drastically decreased as the percentage of propane increased at the 2000 rpm/ 7.6 bar IMEP condition, as can be observed in Figure 3.3-(b). Unlike the result of the 1500 rpm/5.5 bar IMEP condition, the ignition dwell was considerably prolonged as 8 degrees. Partially premixed regions were created, and the local combustion temperature was reduced [3, 45]. The two contradictory results are related to the EGR rate and the amount of diesel fuel. Because propane has dilution and thermal effects similar to EGR by itself, these effects were amplified by the amount of propane substitution and the lower EGR condition [45~47]. Therefore, the trends of the ignition dwell were closely related with the results of the NOx emissions under the two operating conditions.

PM emission was reduced as the proportion of propane increased, regardless of the engine speed/load condition. This result came from the improved pre-mixing with the gaseous fuel added at the intake port. Because the propane was supplied in the gaseous state, wall-wetting on the intake port was prevented.

PM emission was reduced significantly for the two operating conditions, as can be observed in Figures 3.4-(a) and (b). For conventional diesel combustion, the level of PM emissions was 10^8~10^9 particles/cc, but it was reduced by a factor of 10 with the addition of a large proportion of propane. There are two reasons for this result. The first reason is the decrease of the diesel fuel, which provides polycyclic aromatic hydrocarbons (PAHs) that lead to the formation of PM [31, 46]. The other reason is the improvement of the pre-mixing process between the air and the fuel mixture. Because the propane was supplied in the gaseous state into the intake port, a better air-fuel mixture is formed.

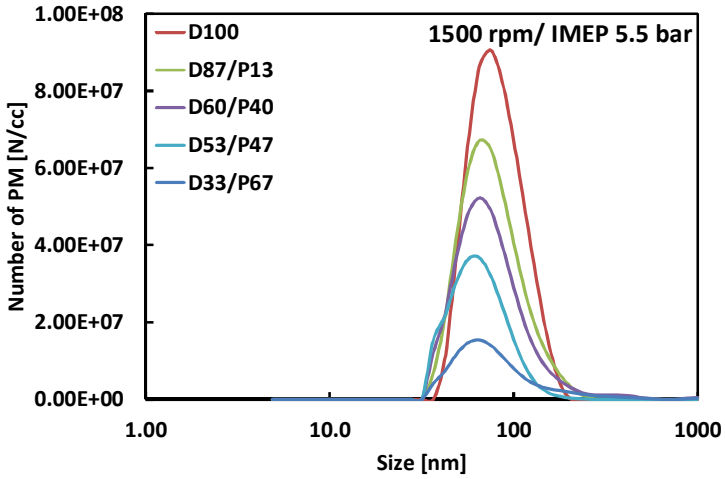


(a) (b)

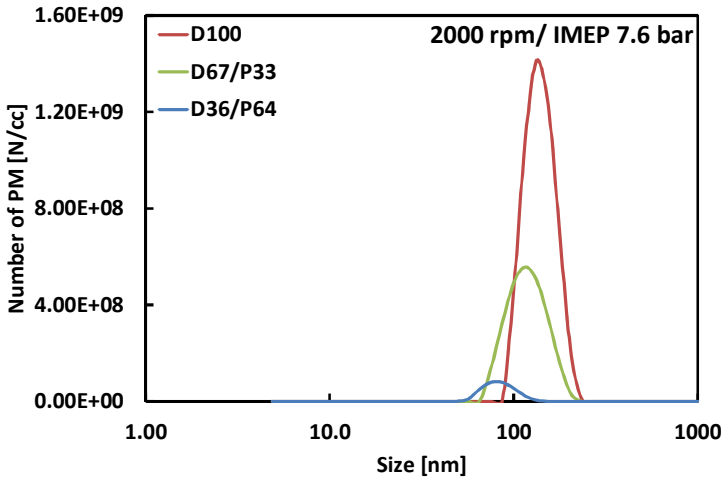


(c)

Figure 3.3. NOx and PM emissions for various propane ratios at 1500 rpm/5.5 bar IMEP (a), 2000 rpm/7.6 bar IMEP (b) and the ignition dwell (c) [48]



(a)



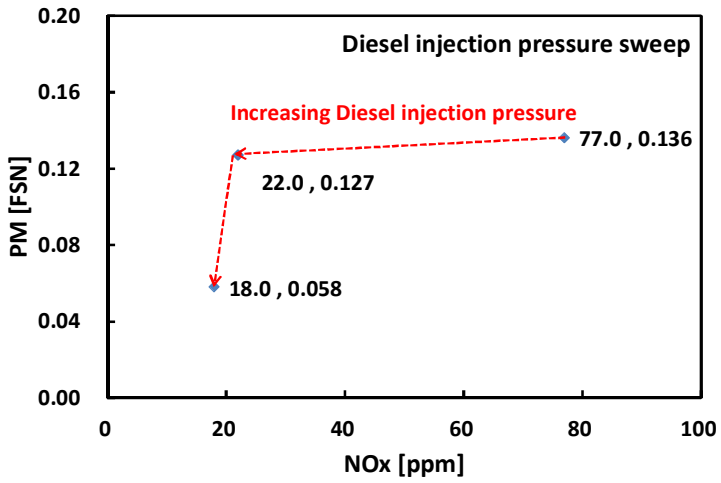
(b)

Figure 3.4. PM size distribution for various propane ratios at 1500 rpm/5.5 bar IMEP (a) and 2000 rpm/ 7.6 bar (b) (D: Diesel ratio/P: propane ratio)[48]

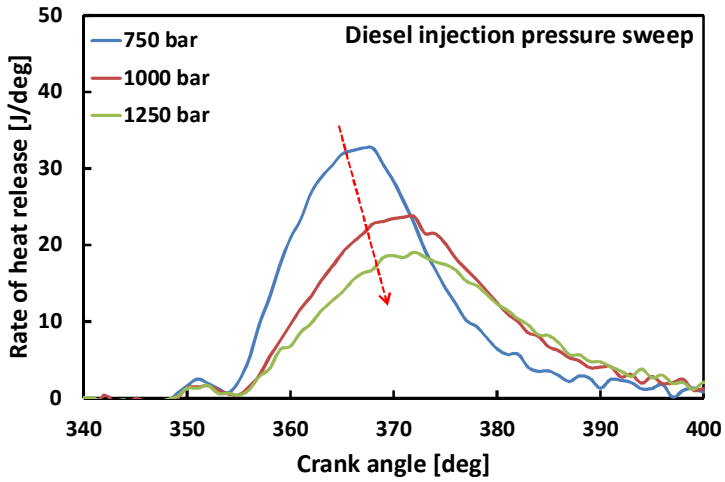
The CO and THC emissions increased as the propane percentage increased for the two operating conditions, by 3~5-fold. However, if an efficient diesel oxide catalyst (DOC) had been used, these emissions could be reduced [26, 31].

In Part 1, the practicality of adding propane in conventional diesel combustion was evaluated. In Part 2, the characteristics of dual-fuel combustion with PCI was investigated. The engine speed was fixed at 1500 rpm, and the amount of diesel fuel was 4 mg/str. The diesel injection timing was 30° BTDC with a single injection. EGR was not employed, and the diesel injection and boost pressures were varied. In addition, the ratio of low-reactivity fuel, i.e., propane, was 75% of the total amount of fuel.

At first, as the diesel injection pressure was increased, the NOx and PM emissions decreased, as can be observed in Figure 3.5-(a).This tendency of the NOx emissions was quite different from that of conventional diesel combustion. This resulted in incomplete combustion, and the local combustion temperature was lower. This phenomenon is reflected in the RoHR graph in Figure 3.5-(b). The combustion phase was retarded, and the peak point was lowered. The IMEP decreased from 5.5 bar to 4.1 bar as the diesel injection pressure increased [48].



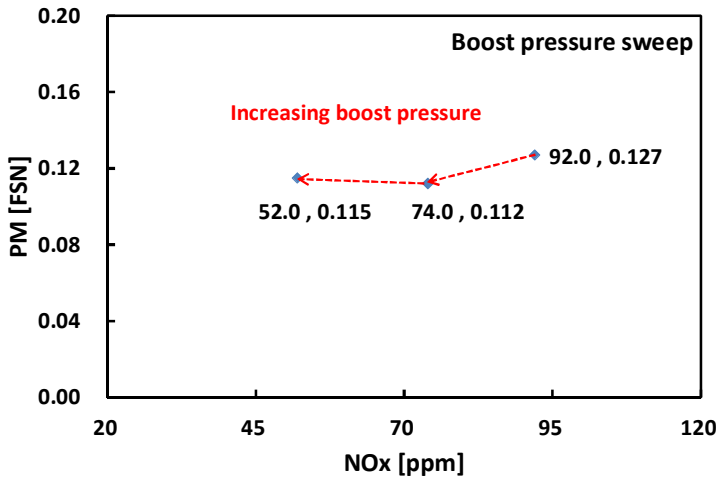
(a)



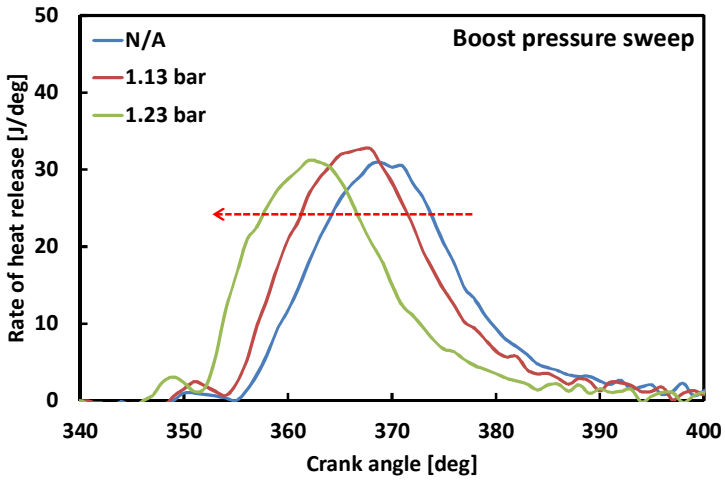
(b)

Figure 3.5. NOx versus PM emissions (a) and RoHR versus Crank Angle (b) for various Diesel injection pressures [48]

The IMEP was not varied during the boost pressure sweep. Although the diesel spray length shortened as the boost pressure increased, the high density of the air and the in-cylinder temperature provokes detonation in the combustion process [49]. Figure 3.6-(a) shows that the NOx emissions decreased with increasing boost pressure for the same reason as with increasing diesel injection pressure. PM emission remained at a low level of 0.1 FSN. Unlike the diesel injection pressure sweep cases, the combustion phase advanced significantly with increasing boost pressure. It may be that the effect of a higher in-cylinder temperature at the end of the compression stroke is more effective than a leaner in-cylinder equivalence ratio.



(a)

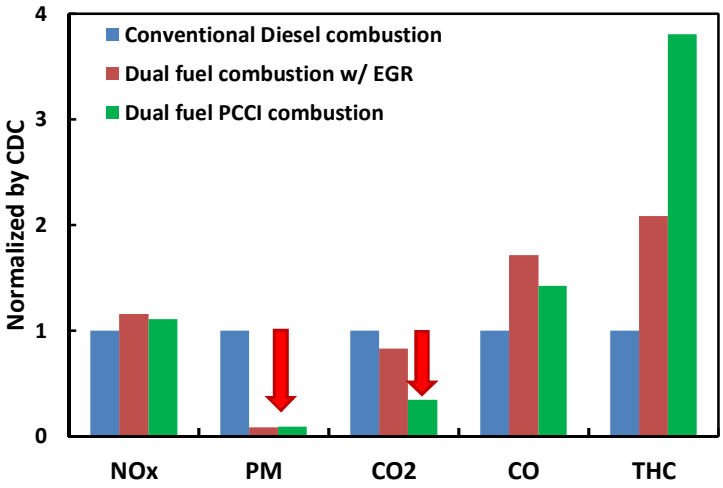


(b)

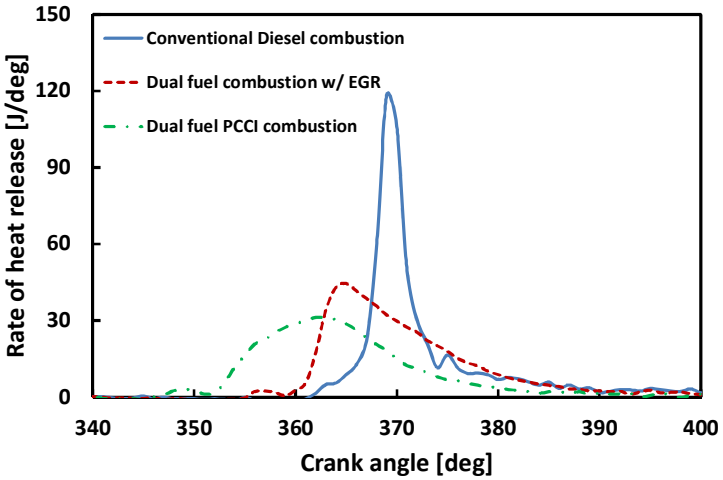
Figure 3.6. NOx versus PM emission (a) and RoHR versus Crank Angle (b) for various boost pressures [48]

Therefore, it is preferable in dual-fuel PCI combustion to use a higher boost pressure but a lower diesel injection pressure to avoid incomplete combustion and maintain the IMEP.

To assess the potential of dual-fuel combustion for reducing NOx and PM emissions, dual-fuel combustion in conventional operation and with early diesel injection were compared with conventional diesel combustion. One type of dual-fuel combustion is simply a propane mixture without early diesel injection and using EGR. The other type is with early diesel injection at 30° BTDC and no EGR.



(a)



(b)

Figure 3.7. Engine emissions (Normalized by conventional Diesel combustion values) (a) and RoHR (b) for three combustion cases [48]

Table3.2 Engine emissions for conventional Diesel combustion at 1500 rpm/ 5.5 bar IMEP

Emission	Value
NOx [ppm]	45
PM [FSN]	1.344
CO2 [%]	10.87
CO [ppm]	1847

In Figure 3.7, comparisons of engine emissions and RoHR for the three conditions are given. In Figure 3.7-(a), each value was normalized by the value of the conventional diesel combustion case. The emission values of the conventional diesel combustion are listed in Table 3.2.

NOx emission increased slightly but remained at less than 50 ppm for the dual-fuel combustion cases. Although the dual-fuel PCI combustion did not use EGR, NOx emission were very low. In Figure 3.7-(b), for the dual-fuel PCI combustion case, the sharp peak from the diffusion flame of the diesel spray was diminished, thus helping to reduce the combustion temperature [50, 51]. In addition, the reason for the prolonged combustion duration could be the simultaneous local auto-ignition spots [52]. Therefore, dual-fuel PCI combustion has the potential to reduce NOx emissions without any EGR.

The total mass and the count of PM were drastically reduced as propane was added for both dual-fuel combustion cases, as can be observed. This effect is related to the improved mixing process between the air and the fuels. In particular, the large size of PM particle, the so-called accumulation mode, were distinctly reduced because the PAHs from the diesel fuel decreased [31, 46].

However, CO and THC emissions rapidly increased 2~4 fold as propane was added. CO emissions were higher for the dual-fuel combustion with EGR, but the opposite was true for the THC emissions. The reason is that CO emission was strongly affected by the combustion temperature, whereas THC emission is related to crevice effects [49]. CO emission was higher with the EGR, and THC emissions increased with early diesel injection.

CO_2 emission was much lower with dual-fuel PCI combustion. Because propane has less carbon than diesel fuel for the same LHV, dual-fuel PCI with propane has advantages for the reduction of CO_2 emission.

In this research, the addition of propane to a CI engine for dual-fuel combustion was systemically evaluated. Two methods of dual-fuel combustion were studied. One method was with conventional CI operation without increasing the boost pressure, and the other method was dual-fuel PCI combustion without supplemental EGR. The results of this research can be summarized as follows:

In conventional operation, the addition of propane in a CI engine can drastically reduce PM emissions because of the improved mixing process between the air and the fuels. In addition, there is a potential to reduce NOx emission under relatively high load conditions because of the decrease in the diffusion flame, which locally increases the combustion temperature.

Dual-fuel PCI combustion with propane significantly reduces the combustion temperature, which results in low NOx emission without EGR. In addition, the total mass and the number of PM emissions were lower using this combustion concept. The behavior of dual-fuel PCI combustion with various diesel injection/boost pressures was different from that of conventional diesel combustion because of the differences in the combustion mechanism. Thus, high boost and low Diesel injection pressures are recommended for dual-fuel PCI combustion to maintain stable combustion and power output. Compared with conventional Diesel combustion, CO and THC emissions increased for both dual-fuel combustion cases, but it was possible to drastically reduce PM and CO_2 emissions while achieving low levels of NOx emissions in dual-fuel (diesel+propane) combustion.

3.3 Effect of single or split diesel injection strategy on the early diesel injection dual-fuel combustion under various propane ratios

Since the potential of emission reduction of early diesel injection dual-fuel PCI from the Chapter 3.2, different operating strategies including control of EGR rate, diesel injection timing, and strategy for dual fuel PCI combustion were assessed for various premixed fuel ratios. Since early diesel SOI strategy and usage of EGR have the potential to reduce NOx and PM emissions effectively, in this chapter, improved operating strategy would be introduced. The premixed fuel ratio (propane ratio in this research) was increased within the range under which stable combustion can be maintained (from 30 to 70 %). Stable combustion was assessed by the coefficient of variation (CoV) of the gross indicated mean effective pressure (gIMEP); in this study, the CoV of the gIMEP was not to exceed 5 % [34, 53]. While the combustion remained stable, the restrictions on NOx and PM emissions were set at under 50 ppm and 3 mg/m^3, respectively [53]. The EGR rate was also varied with the propane ratio to ensure combustion stability and low NOx emission levels. The results of this work emphasized that there was an appropriate diesel injection strategy for each propane ratio to stabilize the dual fuel PCI combustion and achieve low NOx and PM emissions simultaneously while maintaining the thermal efficiency at same level as in the conventional diesel combustion

3.3.1 Experimental conditions

At low speed and low load engine operating conditions, 1,500 rpm/gIMEP 0.55 MPa, propane ratios of 30, 50, 70 % to the total LHV of the supplied fuels (propane and diesel) were tested. The LHV of the total amount of fuel was maintained during the dual fuel combustion experiments. Additionally, the air-fuel ratio was maintained at the same level under the same EGR rate conditions. The EGR rate for each propane ratio condition was adjusted to keep the NOx emissions under 50 ppm. The dual fuel combustion characteristics were studied starting with the conventional EGR rate (32 %) and 50 % propane. In particular, the effectiveness of early diesel injection strategy for dual fuel PCI was evaluated. For the next step, approaches to operating strategies for dual fuel PCI combustion under a higher EGR rate (35 %) and higher propane ratio (70 %) were followed. The base engine operating conditions are given in Table 3.3. More detailed operating conditions, such as diesel injection strategy and the diesel/propane ratio will be introduced in the following section

Table 3.3 Base engine operating conditions for the experiments of diesel injection strategy on the dual-fuel combustion

Description	Value
Engine speed [rpm]	1,500
gIMEP [bar]	5.2~5.5
Total LHV for each stroke [J/str]	680
Diesel injection pressure [bar]	750
Intake pressure [bar]	1.13
Coolant & oil temperature [K]	358

3.3.2Effect of diesel injection strategy under conventional EGR rate condition

The investigation of the dual fuel combustion began with dual fuel combustion under conventional diesel combustion conditions (EGR rate 32 %, overall equivalent ratio 0.65). The propane ratio was fixed at 50 %, which was the maximum value under which a stable combustion state could be maintained. There were two different diesel injection strategies: the multiple injection strategies, which is the same as the conventional diesel operating condition, two pilots, and one main injection; and the early diesel single injection strategy. Table 3.4 shows the details of the diesel injection strategy used in each case.

In Figure. 3.8, cumulative heat release under three different combustion modes was introduced into each combustion phase. For 40 CA after firing TDC, the total burn rate of the dual fuel combustion was higher than that of conventional diesel combustion, which indicated that the majority of the fuel was burned during the earlier and shorter expansion stroke. Thus, the power density could be improved. The premixed combustion phase of conventional diesel operating condition was the shortest among the three combustion phases. In this analysis, the premixed combustion duration was verified from the SOC to the point where the gradient of the slope from the cumulative heat release in Figure.3.8 become gradual. It was related to the fuel reactivity in the cylinder. Under 50% propane with dual fuel combustion, the amount of high-reactivity fuel, i.e., diesel, was reduced such that the overall reactivity was decreased despite the overall equivalent ratio being similar in each case [47, 52]. Thus, the premixed combustion phase under dual fuel combustion was longer than that under conventional diesel combustion.

On the other hand, the mixing controlled combustion phase of conventional diesel combustion was the longest among the three combustions. There were two main reasons for this result: a reduction in the amount of premixed fuel and the longer duration of diesel injection relative to the ignition delay. The second reason can be explained in terms of the heat release rates. By contrast, the mixing controlled combustion phase of dual fuel combustions was distinctly shortened, while the premixed combustion duration was prolonged. This result provides a basis for the PM reduction.

Therefore, the introduction of low-reactivity fuel to the combustion initially increased the premixed combustion phase compared to that of diesel combustion. Thus, low-reactivity fuel caused the rapid combustion region to increase. Then, if an early diesel injection strategy was applied, the rapid combustion region, i.e., the premixed combustion phase, could be expanded due to the improvement of the premixing conditions of the ignition source.

The heat release rates and in-cylinder pressure are shown in Figure. 3.9. For the same multiple injection strategies, the ignition delay for dual fuel combustion was longer than that for conventional diesel combustion. Meanwhile, for the conventional diesel combustion case, SOC occurred before EOI and SOC occurred after EOI for a propane ratio of 50 %. The decreased reactivity in-cylinder prior to auto-ignition and shortened injection duration due to the reduced diesel fuel were the main causes of this phenomenon [23, 49, 52].

Comparing conventional diesel and dual fuel combustion with early single diesel injection, the premixed combustion was greater, but the mixing controlled combustion period decreased markedly for dual fuel combustion. Additionally, comparing two dual fuel combustion modes with different diesel injection strategies,

the main combustion duration and post-oxidation, i.e., the late combustion phase, were decreased by early diesel injection. This result could reflect the characteristics of PCI combustion. Under the conventional diesel injection strategy (in this work, two pilots and main injection closer to TDC), diffusive flame and premixed combustion coexisted. If the early diesel injection was adjusted, premixed combustion became dominant, which is one advantage of PCI combustion.

Thus, there were combined effects of the overall equivalent ratio and fuel reactivity distribution in the cylinder. Meanwhile, conventional diesel combustion was affected by the equivalent ratio only. Dual fuel PCI by early single diesel injection could be regarded as the combustion strategy with the most improved homogeneity and greatest fuel reactivity stratification.

Table 3.4 Diesel injection strategy for dual fuel combustion (EGR 32 % & propane 50 %)

	Multiple injection strategies		Early single injection strategy	
	Fuel rate [mg/cycle]	SOI [° BTDC]	Fuel rate [mg/cycle]	SOI [° BTDC]
Pilot 1	0.8	30.1	-	-
Pilot 2	1.0	20.3	-	-
Main	6.2 for dual fuel 14.2 for diesel only	5.1	8.0	30.1

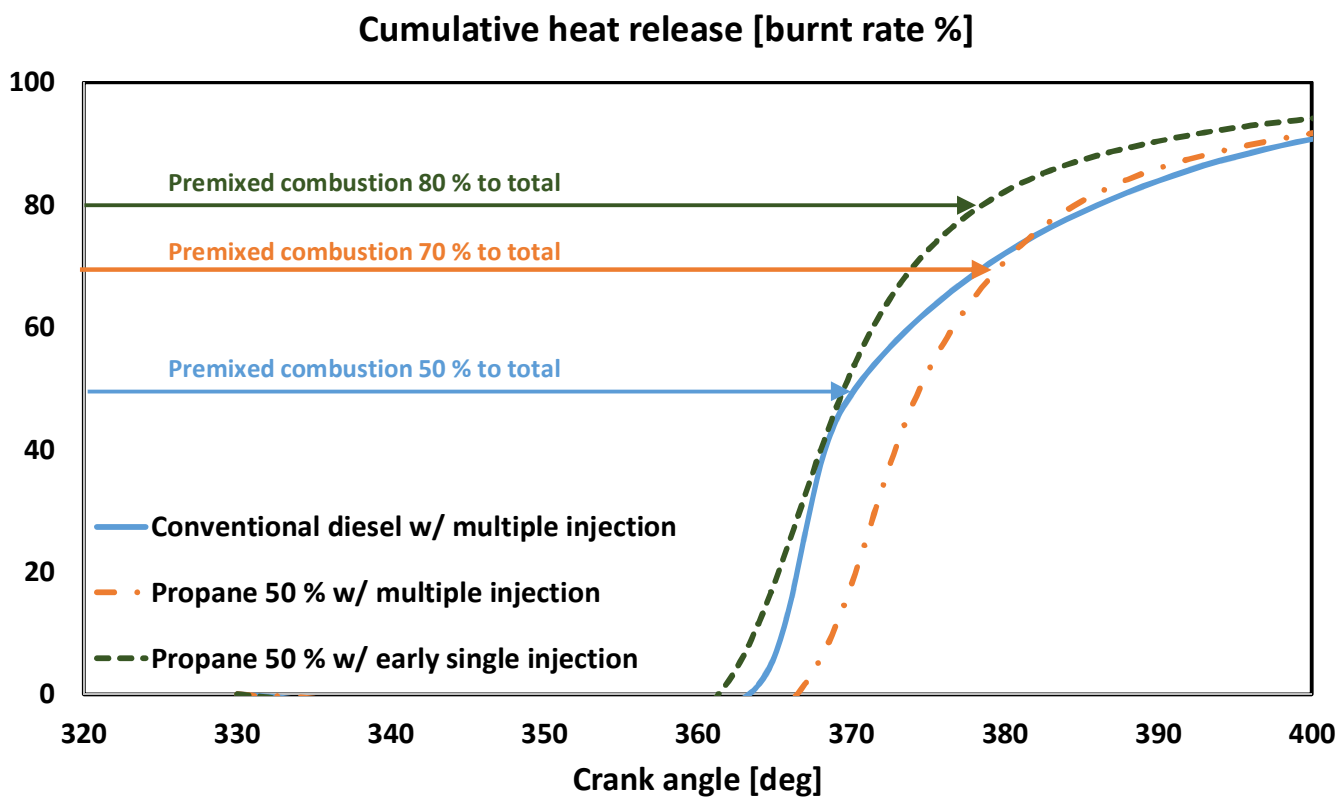


Figure 3.8 Cumulative heat release for different injection strategies (EGR 32 % & propane 50 %) [54]

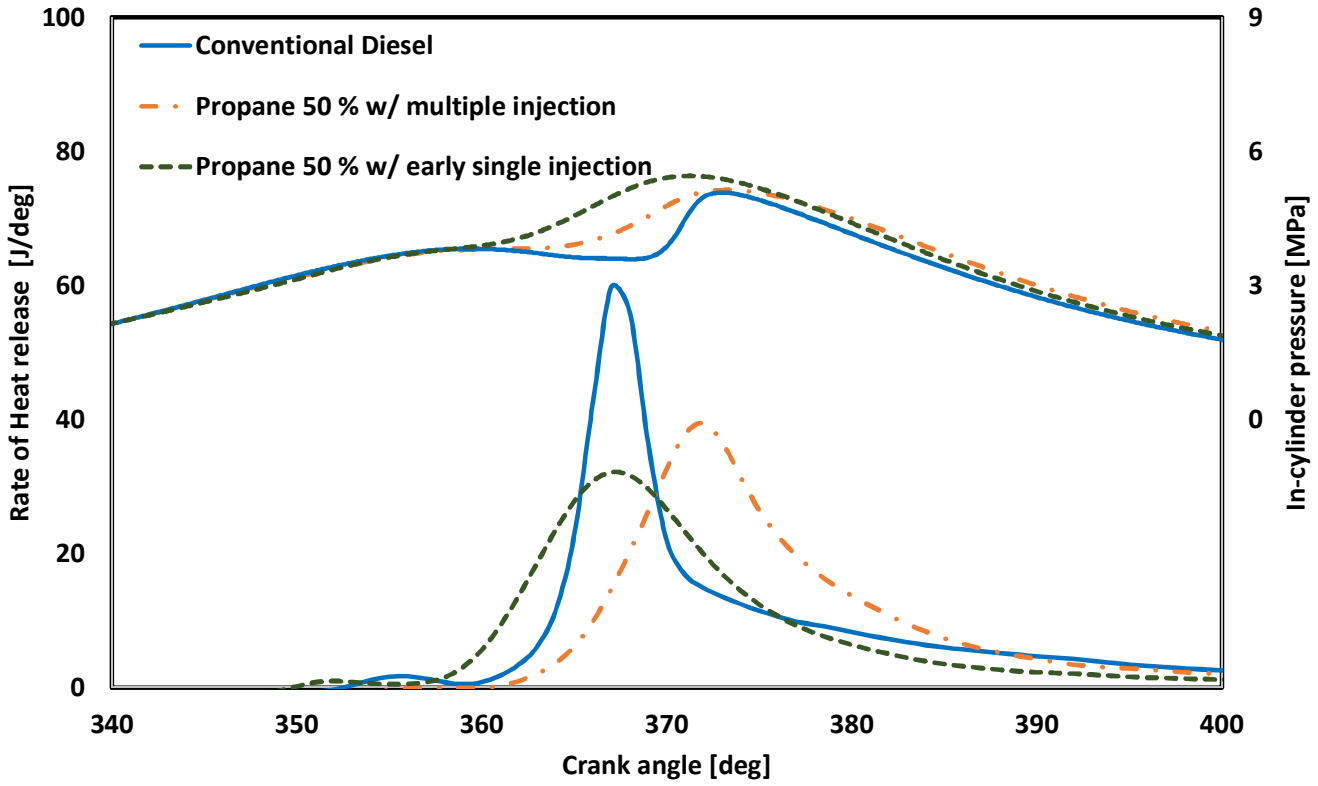


Figure 3.9 Rate of heat release and in-cylinder pressure graphs for different injection strategies (EGR 32 % & propane 50 %) [54]

Next, the combustion parameters, such as a combustion duration, the maximum pressure rise rate and CoV of gIMEP and emissions results, are addressed in this chapter for the above three combustion modes.

The combustion duration of the conventional diesel combustion was longer than those of the other dual fuel combustion modes, as shown in Figure 3.10. The combustion duration was calculated from MFB 5 to 90. This result was due to the longer mixing controlled combustion duration due to the diffusive flame, while the premixed duration was longer for higher propane ratios. In addition, early diesel injection shortened the combustion duration. Because the diesel fuel was injected earlier than in the multiple injection cases, the reactivity gradient in the cylinder could be reduced. In other words, because early injection provided a long enough ignition delay for the diesel fuel to be distributed more homogeneously, auto ignition sites could be enlarged by the early diesel injection strategy.

The maximum cylinder pressure rise rate could also be reduced by dual fuel combustion. This result would also be evidence of 'in-cylinder fuel stratification' [55, 56, 57]. As mentioned above, reduction of high-reactivity fuel could be the main cause. Because previous dual fuel PCI studies were conducted using a higher pressure rise rate because of spontaneous autoignition, appropriate fuel reactivity stratification of this combustion could help moderate the combustion phase.

The CoV of gIMEP inevitably increased as propane was supplied. It was also related with the reduced amount of diesel fuel, which acted as an ignition source. In other words, the combustibility was worsened by the low-reactivity fuel under the compression ignition system for the same diesel SOI. However, the early single diesel injection strategy was relatively effective at stabilizing the CoV of gIMEP. An increase in the number of auto ignition sites could improve the combustion stability.

The emissions results showed distinct differences among the three combustion modes in Figure 3.11. NOx emissions slightly increased as propane was supplied under the same diesel injection strategy. Clearly, the amount of diesel changed, but dual fuel combustion with early diesel injection led to lower NOx emissions, as shown in Figure 3.11-(a). The reduction of NOx emissions was related to the overall lean premixed condition of the air-fuel (propane and diesel) mixture, which decreases the local high burnt gas temperature [2, 23].

The PM emission trend as propane was supplied was clear. The reduction of PM emissions was caused by the improved homogeneity and reduction of the direct injection fuel, i.e., diesel. In particular, early single diesel injection was advantageous for achieving near-zero PM emissions. From this result, although PM emissions could be reduced by propane ratio adjustment only, early diesel injection PCI could achieve a greater reduction of PM emissions by improving the premixed combustion. In other words, under dual fuel combustion with the conventional diesel injection (multiple injection) strategies, the diffusive flame from diesel injection remained near the TDC.

The indicated thermal efficiency was higher under dual fuel combustion with the early diesel injection strategy than under conventional diesel combustion. Because the combustion duration decreased as the early diesel injection was adjusted under the dual fuel combustion, the heat transfer loss could be reduced [26, 31]. Although dual fuel combustion with the multiple injection strategies also provided a shorter combustion duration than conventional diesel combustion, the combustion loss might be increased due to the retardation of the combustion phase. Thus, the effects of the shorter combustion duration and the retardation of the combustion phase canceled one another out.

As shown in Figurer 3.11-(b), the CO and THC emissions, which are representative of unburned emissions, increased as propane was supplied and early diesel injection was adjusted. Because the high-reactivity fuel was decreased, CO emissions resulted from the incomplete autoignition. Although the main diesel SOI was varied from BTDC 5 to 31 CA, CO emission did not change markedly. THC emission increased by a factor of nearly ten under the dual fuel combustion mode. Crevice effects of gaseous fuel are the major cause of THC emission [39]. In addition, there was a possibility of wall-impingement from diesel spray under the early single diesel injection case. Thus, although dual fuel combustion with early single diesel injection was superior in terms of the simultaneous reduction of NOx and PM emissions, emissions from incomplete combustion increased. However, deterioration of the combustion efficiency did not mean low thermal efficiency in this research. Because the combustion duration of dual fuel PCI combustion was shorter than that of conventional diesel combustion, it was possible to reduce heat loss, which would, in turn, improve the thermal efficiency.

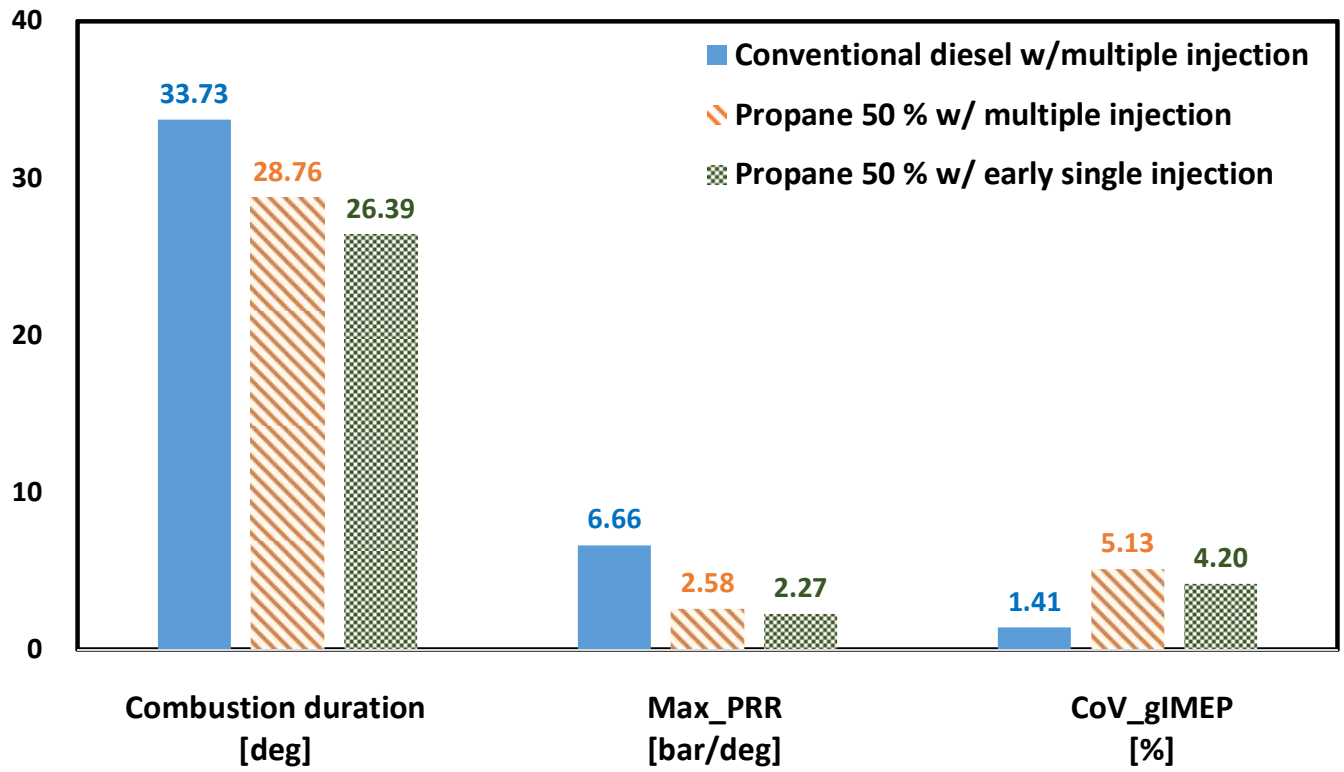
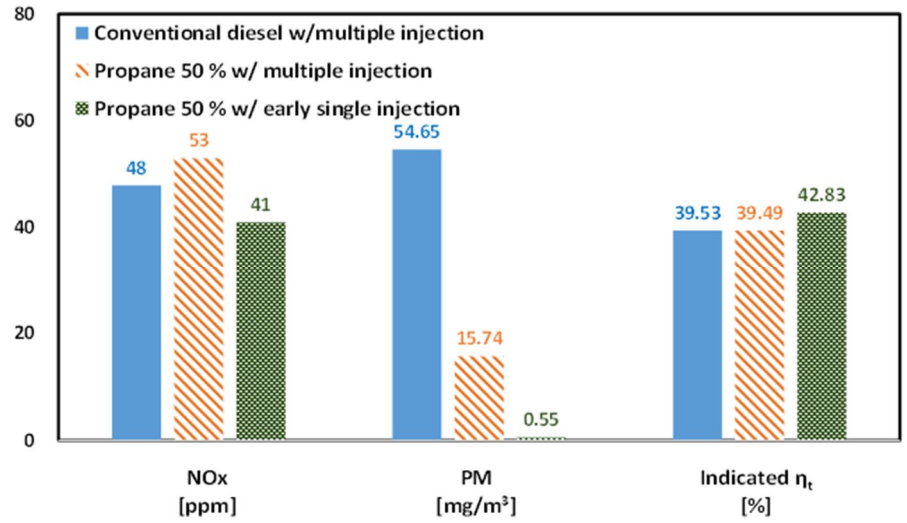


Figure 3.10 Combustion parameters as different injection strategies under conventional diesel and dual fuel combustions (EGR 32 % & propane 50 %) [54]



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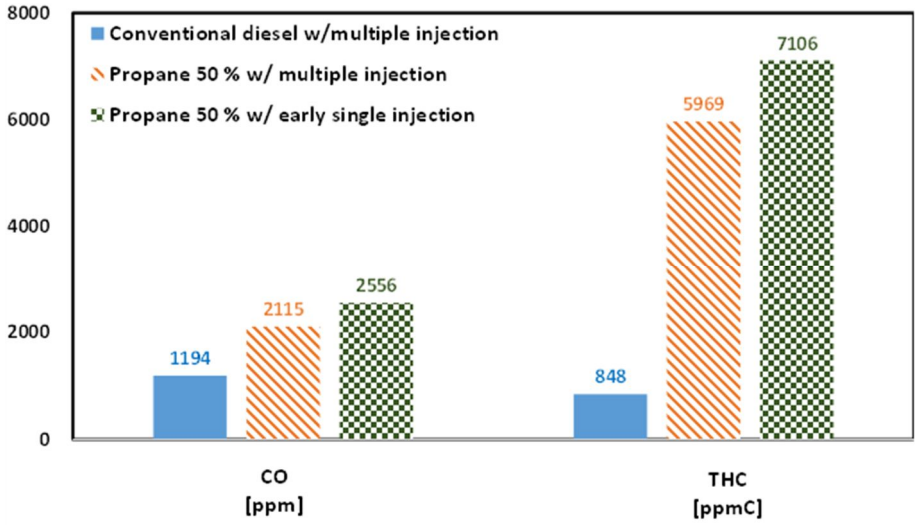


(b)

Figure 3.11 NOx/PM emissions and indicated thermal efficiency (a) and CO/THC emissions (b) (EGR 32 % & propane 50 %) [54]

3.3.3 Effect of diesel injection strategy under high-EGR rate condition

Under higher EGR rates than used in conventional diesel operating conditions, dual fuel combustion mode was studied by adjusting the diesel injection strategy and propane ratio. Because the oxygen rate in the cylinder was decreased, the propane ratio should be decreased to 30 % to stabilize combustion. Similarly to the above result, early diesel injection was the main focus. Additionally, to magnify the effect of stratification of fuel reactivity, the diesel injection was split into two injections. The injection strategies are introduced in Table 3.5.

In Figure 3.12, the heat release rates and in-cylinder pressure of conventional diesel and dual fuel combustions with propane 30 % condition are shown. It is clear that the premixed combustion region increased and the mixing-controlled combustion diminished under dual fuel combustion conditions. When early split diesel injection was applied, the SOC was advanced, but the rise rate of heat release became smoother than that of dual fuel combustion with early single diesel injection. This result might be derived from the improvement of the reactivity stratification. While the overall equivalent ratio was equally distributed, high-reactivity fuel, i.e., diesel, would be layered in the cylinder in the case of the split injection strategy [23, 57, 58].

Table 3.5 Diesel injection strategy for dual fuel combustion (EGR 35 % & propane 30 %)

	Early single injection strategy		Early split injection strategy	
	Fuel rate [mg/cycle]	SOI [° BTDC]	Fuel rate [mg/cycle]	SOI [° BTDC]
Pilot 1	-	-	-	-
Pilot 2	-	-	3.5	67.1
Main	8.0	30.1	4.5	30.1

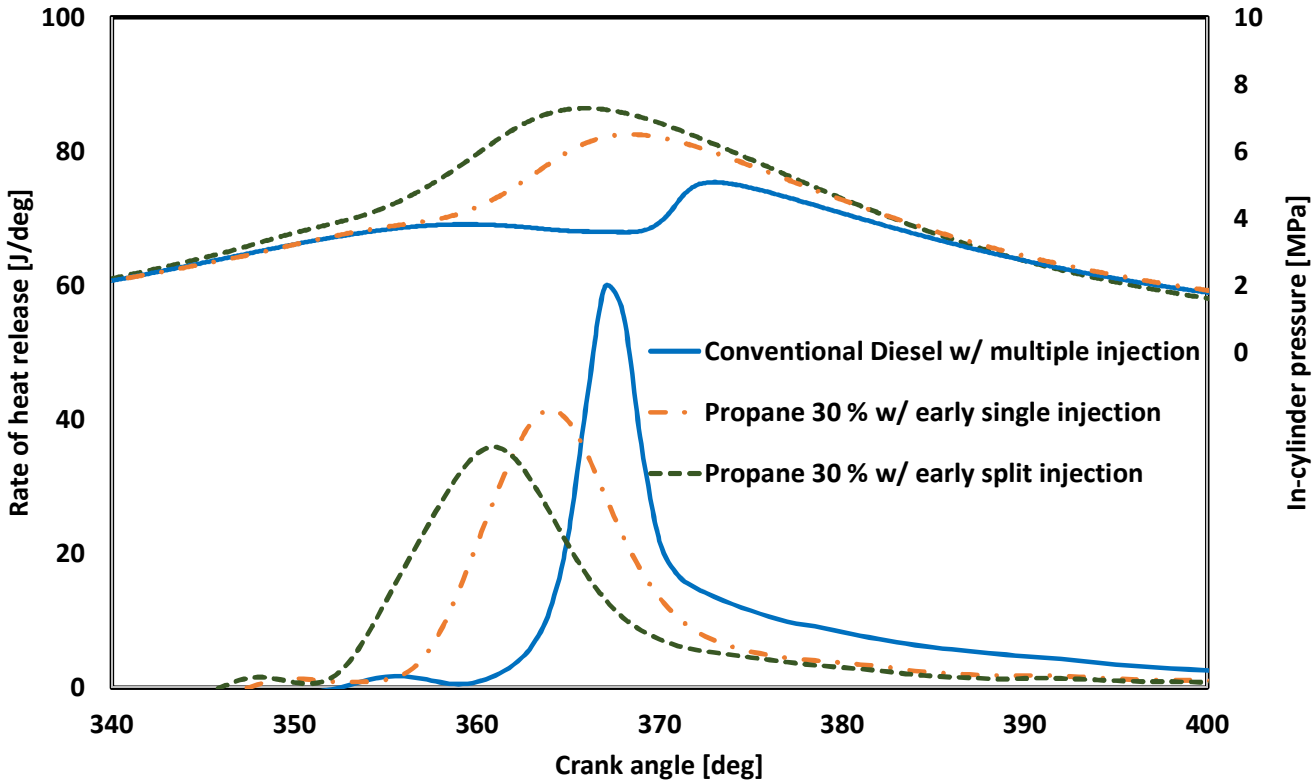


Figure 3.12 Rate of heat release and in-cylinder pressure graphs for different injection strategies (EGR 35 % & propane 30 %) [54]

In Figure 3.13, the combustion duration of dual fuel combustion with the early split injection strategy was more prolonged than that with the early single diesel injection. This difference was related to the improved high-reactivity fuel stratification, which led to layered combustion. For this reason, the CoV of gIMEP increased, but it remained under 5 %. Under dual fuel combustion with early single diesel injection, the maximum pressure rise rate exceeded the result found for 50 % propane. Because the amount of diesel fuel was increased in this condition, auto-ignitibility might be improved.

NOx emission could be drastically reduced by increasing the EGR rate to 35 %, as shown in Figure 3.14-(a). PM emission was slightly higher in this case than in the case of 50 % propane due to the increased EGR rate. Additionally, the emission for dual fuel combustion with early split injection strategy was 2 mg/m^3. Wall-impingement from diesel injection at 67 BTDC could be one reason for the increment of PM emission. Thus, THC emission increased when early split diesel injection was adjusted in Figure 3.14-(b). However, CO emission still remained at the same level regardless of whether split or single injection was used. From this result, it is clear that the increment of THC and PM emissions resulted from wall-impingement rather than locally rich regions. The indicated thermal efficiency did not change under dual fuel combustions compared to that of conventional diesel combustion.

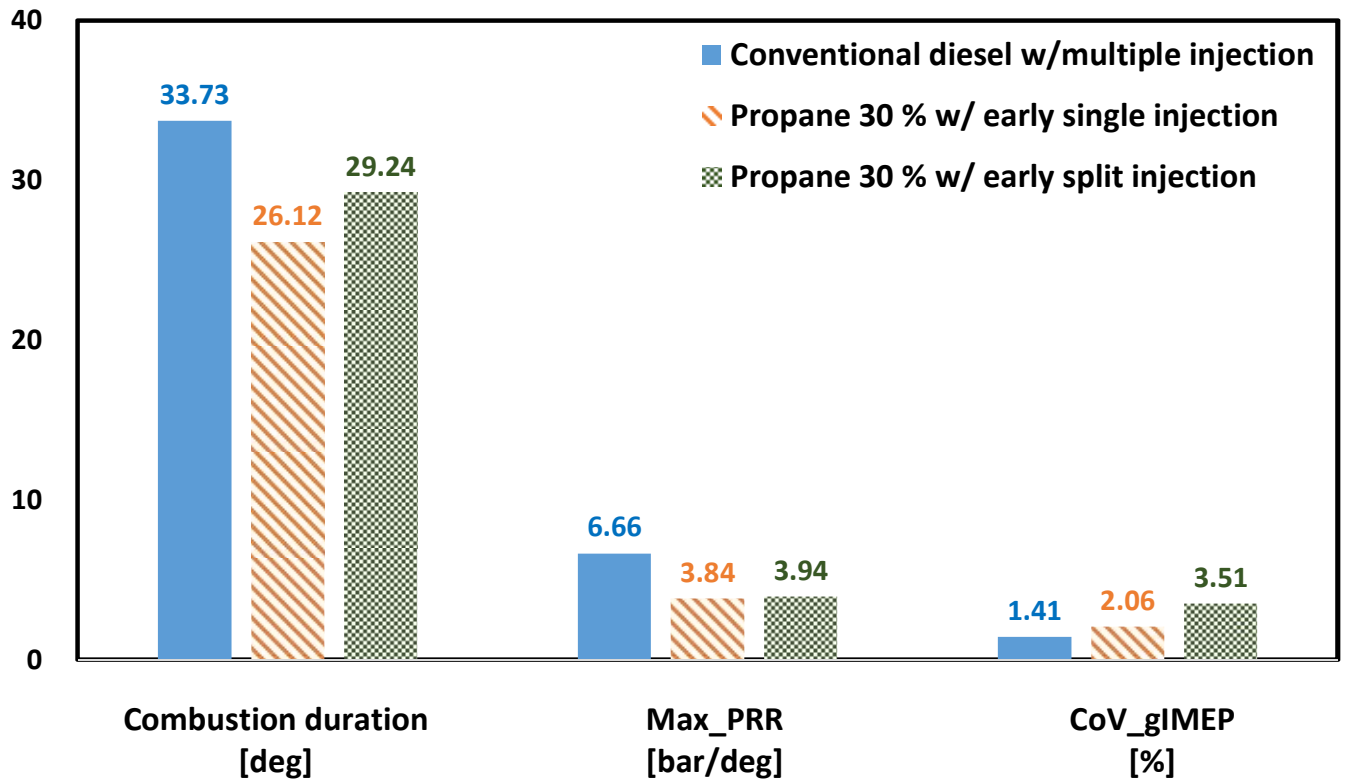
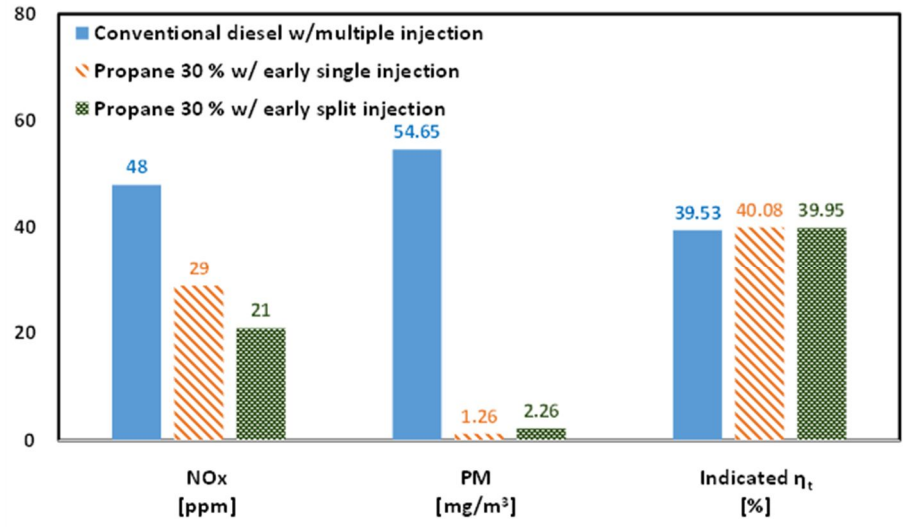


Figure 3.13 Combustion parameters for different injection strategies (EGR 35 % & propane 30 %) [54]



(a)

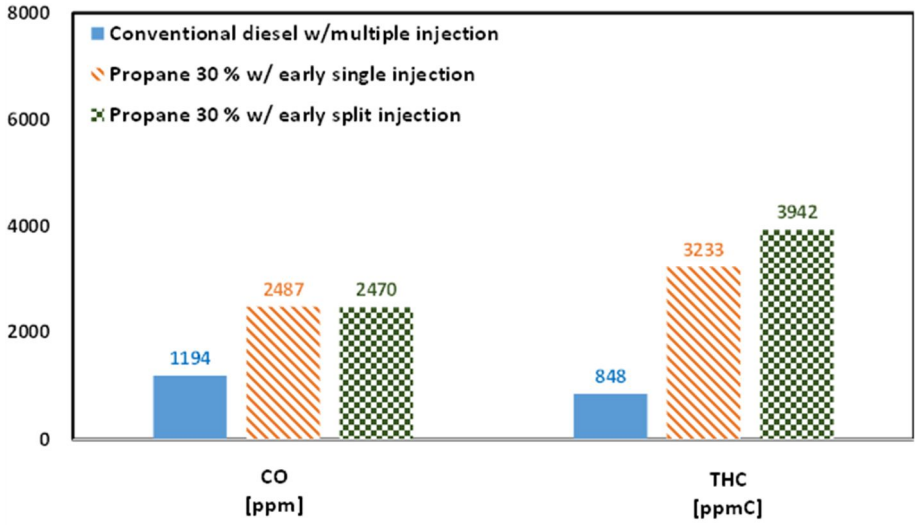


Figure 3.14 NOx/PM emissions and indicated thermal efficiency (a) and CO/THC emissions (b) as different injection strategies (EGR 35 % & propane 30 %) [54]

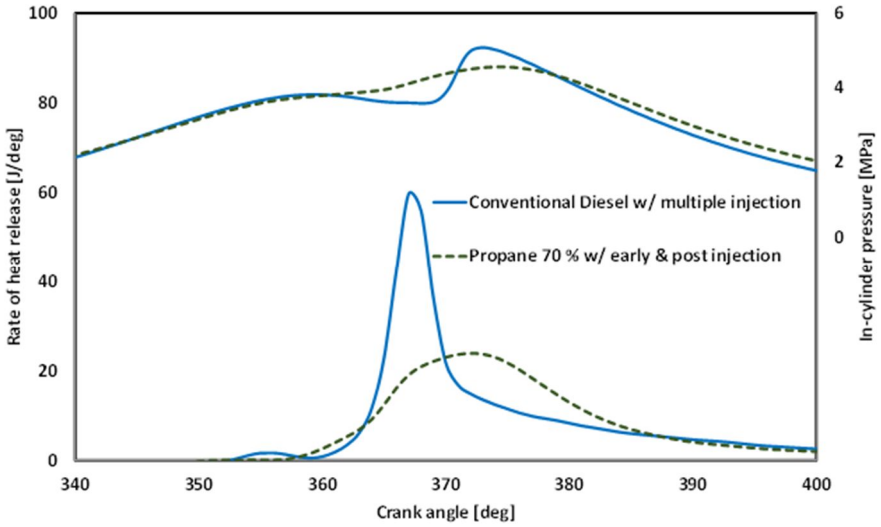
3.3.4 Effect of diesel injection strategy under low-EGR rate condition

Under a higher propane ratio condition, dual fuel combustion mode was studied by the adjustment of the diesel injection strategy and EGR rate. Because the ratio of low-reactivity fuel, i.e., propane, was increased, the EGR rate should be decreased to 22 % to ensure a sufficient oxygen concentration for the combustion stabilization. Similarly to the above result, the early diesel injection was the main focus. However, for a 70 % propane ratio, unstable combustion was observed only in the earlier diesel injection strategy due to the lack of an equivalent ratio prior to auto-ignition. Thus, a small amount of diesel fuel was injected at TDC as a triggering source. The early and post-injection strategy is introduced in Table 3.6.

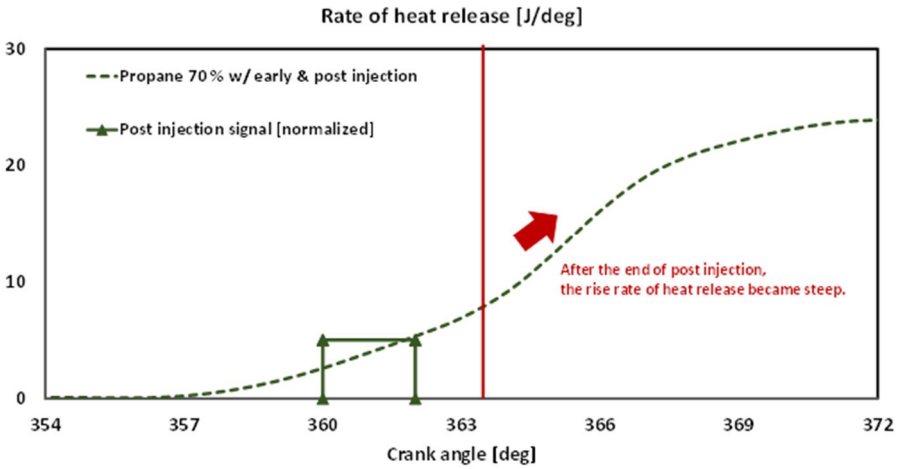
As the propane ratio was increased to 70 %, the peak rate of heat release was markedly reduced and the combustion phase became lengthened, as shown in Figure 3.15-(a). Most of the combustion were involved in premixed combustion phase, and the boundary between premixed combustion and late combustion (oxidation) was ambiguous. Although combustion occurred prior to the start of post-injection, the main combustion could be activated after the end of post-injection, as shown in Figure 3.15-(b).

Table 3.6 Diesel injection strategy for dual fuel combustion (EGR 22 % & propane 70 %)

	Early and post-injection strategy	
	Fuel rate [mg/cycle]	SOI [° BTDC]
Pilot 1	-	-
Pilot 2	-	-
Main	3.8	28.9
Post	1.0	0 (TDC)



(a)



(b)

Figure 3.15 Rate of heat release and in-cylinder pressure graphs (a) and magnified window from the start to end of post-injection (b) (EGR 22 % & propane 70 %) [54]

In Figure 3.16, the combustion duration of dual fuel combustion with a propane ratio of 70 % was longer than that of diesel. Because the decrease in the amount of high-reactivity fuel was concurrent with an increase in the amount of low-reactivity fuel, the premixed and late combustion phases were combined. Because of a lack of ignition source, the CoV of gIMEP worsened but still remained under 5 %. Additionally, as the combustion duration was prolonged, the maximum pressure rise rate could be decreased substantially.

Low NOx emission could be achieved despite the EGR rate being lower than under conventional diesel operating conditions (22 % vs. 32 %) in Figure 3.17. It could be concluded that the diffusion flame from diesel injection could be reduced, in turn diminishing the local high burned gas temperature sites. Additionally, PM emission could be reduced to the 1 mg/m^3 level. Some of the diffusion flame from diesel post-injection could influence the PM emission.

However, deterioration of the indicated thermal efficiency was unavoidable when a large amount of low-reactivity fuel was used due to the reduction of auto-ignition sites. The combustion efficiency was also worsened due to the high THC emission levels. In contrast with the result of dual fuel combustion with a high EGR rate (35 %), the major contributor to the THC emission may be the crevice volume, as the diesel spray length would be sufficiently shortened to avoid wall impingement.

For three dual fuel combustion conditions differing by EGR rate and propane ratio, CO emission remained at the 2,000 ppm level. Therefore, while THC emission came from the wall-impingement of early diesel injection and gaseous fuel from the crevice volume, CO emission was dominated by the overall air-fuel mixture condition.

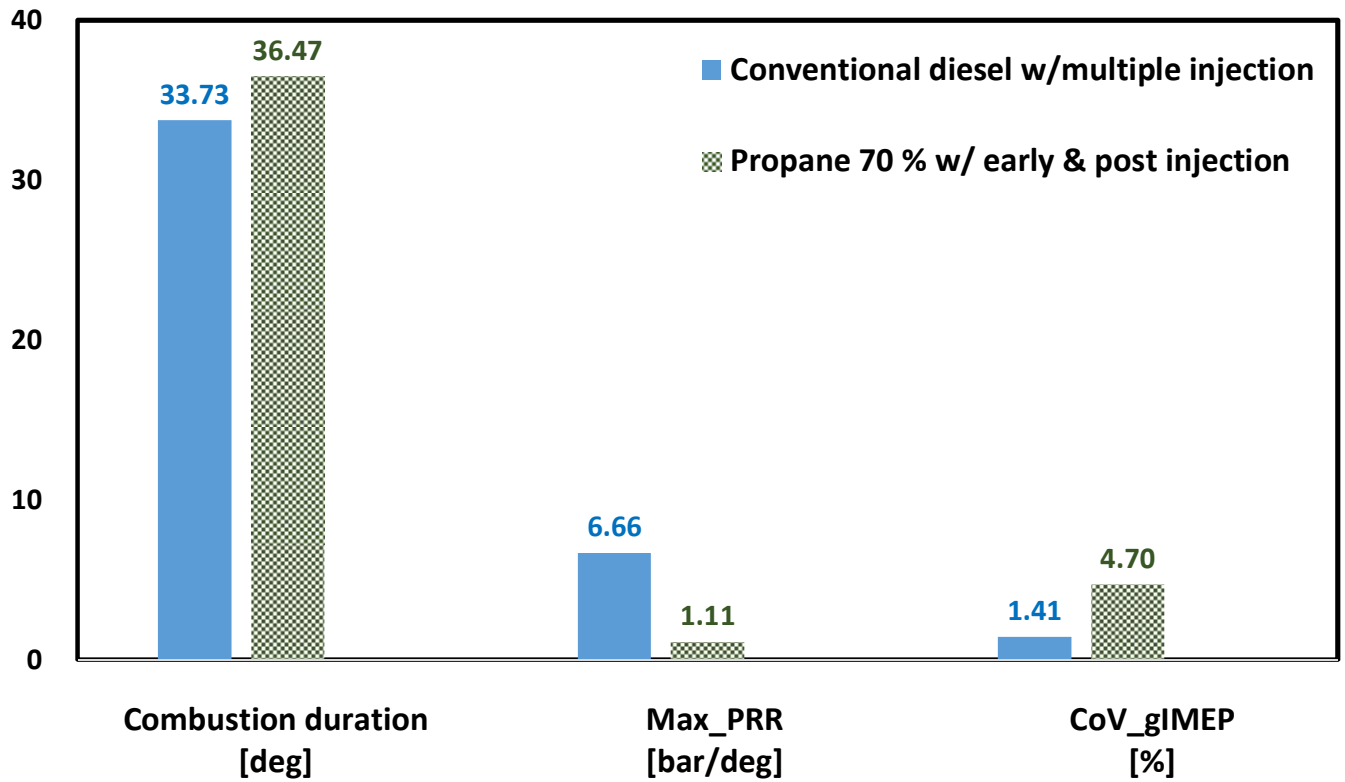
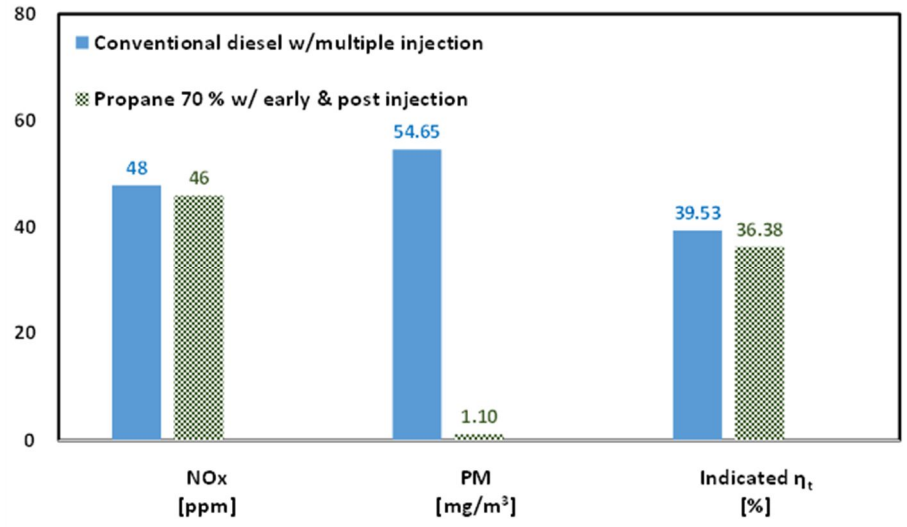
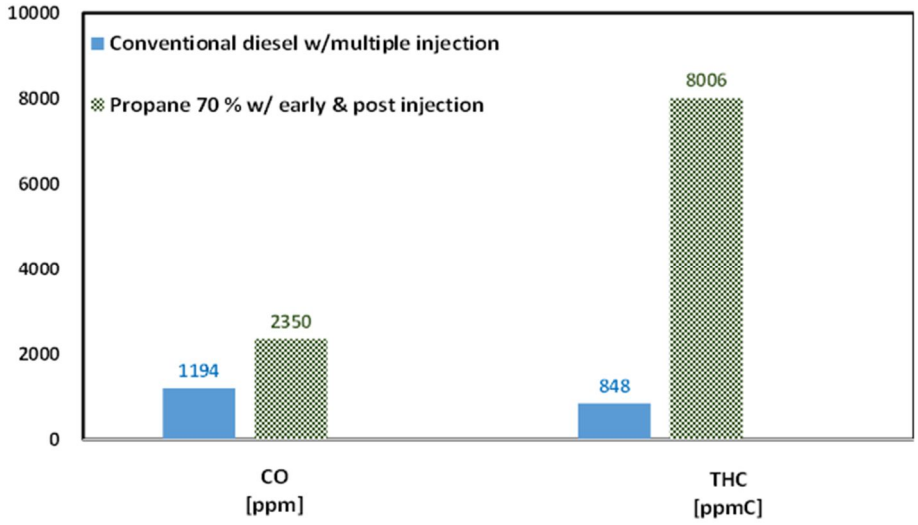


Figure 3.16 Combustion parameters for different injection strategies (EGR 22 % & propane 70 %) [54]



(a)



(b)

Figure 3.17 NOx/PM emissions and indicated thermal efficiency (a) and CO/THC emissions (b) for different injection strategies (EGR 22 % & propane 70 %) [54]

In this Chapter, the characteristics of dual fuel combustion under different EGR rates and propane ratios with adjusted diesel injection strategies were evaluated. Low NOx and PM emission could be achieved by the early diesel injection strategy, while there was no penalty in terms of the indicated thermal efficiency. However, using a high amount of low-reactivity fuel caused a deterioration of the thermal efficiency. Below is a summary of each operating strategy and the general conclusions of this work. Comparisons of the emissions and thermal efficiency results are shown in Table 3.7.

1) Propane 30 % & EGR 35 %: Because the quantity of low-reactivity fuel was reduced, a higher EGR rate could be used. Thus, effective reduction of NOx emission could be implemented. However, the PM emission increased slightly due to the low combustion temperature, which led to lower oxidation. Although the combustion duration was shortened, corresponding to a reduction of heat loss, the thermal efficiency was lower than that achieved using 50 % propane because of the higher EGR rate. The early split diesel injection strategy was particularly preferable, similarly to the case of RCCI combustion.

2) Propane 50 % & EGR 32 %: In this case, the early single diesel injection strategy was recommended because the early split diesel injection strategy led to incomplete combustion and a small post-injection at near TDC led to higher PM emission due to diffusive flame. Because the propane condition was improved and the combustion duration was shortened, a much higher thermal efficiency could be achieved with low NOx and PM emissions simultaneously. However, high THC emission could not be avoided due to the crevice effect and wall-wetting.

3) Propane 70 % & EGR 22 %: When a small amount of diesel fuel was used, early diesel injection with a small post-injection near the TDC for the triggering source was appropriate, while a lower EGR rate was used. Because a large quantity of low-reactivity fuel was supplied, the amount of inert gas used should be reduced to stabilize the combustion phase. The highest level of THC emission among three dual fuel PCI combustions resulted from the crevice regions rather than wall-wetting because a small amount of diesel fuel was present. In this case, although low NOx and PM emissions could be achieved despite the low EGR rate (~22 %), deterioration of the thermal efficiency could not be avoided due to a large amount of low-reactivity fuels.

Basically, early diesel injection to enhance the premixed phase under dual fuel combustion was effectively able to reduce NOx and PM emissions while maintaining the same indicated thermal efficiency as conventional diesel combustion. High-reactivity fuel stratification by the split injection strategy for low propane ratios and the addition of a triggering source by post-injection for high propane ratios worked efficiently. However, supplying a large amount of low-reactivity fuel disturbed stable combustion and thermal efficiency. Therefore, appropriate diesel injection strategies for dual fuel PCI combustion were identified, and this approach promises low emissions while sustaining a high thermal efficiency.

Table 3.7 Comparisons of the emissions and thermal efficiency results among conventional diesel combustion and dual fuel PCI combustions for various propane ratios

	Conventional diesel	Propane 30 % [EGR 35 %]	Propane 50 % [EGR 32 %]	Propane 70 % [EGR 22 %]
NOx [ppm]	48	21	41	46
PM [mg/m^3]	54.65	2.26	0.55	1.10
CO [ppm]	1,194	2,470	2,556	2,350
THC [ppmC1]	848	3,942	7,106	8,006
Indicated thermal efficiency [%]	39.53	39.95	42.83	36.38

4. Experimental Results of Dual-fuel Combustion by Gasoline and Diesel with Improved Hardware System

In this chapter, the relation between combustion parameters and dual-fuel combustion modes, which are classified, was investigated (Chapter 4.2) with the improved single cylinder dual-fuel research engine. These experiments aimed at the implementation of dual-fuel PCI which is introduced in Chapter 4.3 and comparison of dual-fuel PCI with diesel PCI concept was evaluated to verify the competitiveness of its performance in Chapter 4.4.

4.1 Research on the classification of dual-fuel combustion modes as varying diesel mixing rates by HRR characteristics

4.1.1 Motivations

Dual-fuel combustion has distinct characteristics from classical combustion concepts such as an auto-ignition from CI engines and flame propagation from SI engines, because it was constructed by two fuels which have opposite characteristics. For example, high reactivity fuel such as diesel fuel, tends to auto-ignite well by itself under high-pressure conditions, while low reactivity fuel such as gasoline and propane in this research, is well-distributed due to high volatility but it is hard to auto-ignite. Therefore, there is a possibility of the convergence of two different combustion characteristics.

Especially, experimental results from Chapter 3.1 showed that the combustion characteristics of dual-fuel condition could be deferred by just mixing the rate of

diesel fuel, that is, different diesel injection timings. This information was reflected in the heat release rates. However, single cylinder diesel engine research about engine systems was not suitable for the dual-fuel combustion, because diesel spray angle was wider as 153° and omega-shaped bowl. Thus, with such a system, it was hard to exclude the effect of inappropriate diesel spray targeting diesel injection timing was advanced.

As a result, in this chapter, the differences in dual-fuel combustions as various diesel mixing-rate conditions were investigated deeply by using gasoline and diesel fuel with the improved dual-fuel single cylinder engine system. Diesel mixing-rate was controlled by varying diesel injection timings.

The comprehension of dual-fuel combustion characteristics and classification of its modes are important to improve the performance of the dual-fuel combustion. As dual-fuel combustion modes are classified, a proper combustion mode can be suggested to the different engine operating conditions.

4.1.2 Experimental conditions

At the engine speed of 1,500 rpm and total LHV of fuels 580 J condition, diesel injection timing was varied from 6 to 46 °BTDC. The gasoline fraction to the total fuel amount was also varied from zero to 70 % based on the mass standard. More details are introduced in Table 4.1.

Table 4.1 Engine operating conditions for the classification of dual-fuel combustion modes

Description	Value
Engine speed [rpm]	1,500
Total amount of LHV [J/cycle]	580
Coolant & oil temperature [K]	358
Diesel injection pressure [bar]	450
Intake pressure [bar]	1.10
EGR rate [%]	0
Overall-equivalent ratio	0.44
Gasoline ratio [%]	0~70
Diesel SOI [°BTDC]	6~46

4.1.3 Experimental results: Classification of three dual-fuel combustion modes

In Figure 4.1., the gross indicated thermal efficiency (GIE) is depicted as various diesel injection timings and gasoline ratios under total equivalence ratio 0.44. Then, there were two high GIE peaks when the gasoline ratio was 0.7. Thus, these two points were focused to more researches, because these two points were useful due to their high GIE. In addition to these two points, the additional case whose diesel injection timing was at 6 °BTDC, similar with conventional diesel operating condition was also selected to classify the dual-fuel combustion modes. As a result, three experimental cases with the highest gasoline ratio (0.7) were chosen as representative dual-fuel combustion modes. To clarify the characteristics of each dual-fuel combustion, neat-diesel combustion cases which had the same diesel injection timings with above three dual-fuel cases are used as the comparison group.

For the first dual-fuel combustion mode, in Figure 4.2., the shape of heat release rates between neat diesel and dual-fuel cases were similar to each other. At the first, the highest peak of premixed combustion was shown. Next, mixing controlled shape combustion was followed. Since the amount of diesel fuel was reduced under the dual-fuel combustion case, the amount of heat release from the premixed combustion phase was reduced to 45 J, but most of this energy were emitted during the late combustion phase. Then, there was an issue that the late combustion is whether a mixing controlled combustion is like conventional diesel combustion or just slow flame propagation.

For this reason, to investigate the behavior of diesel fuel during dual-fuel combustion, the comparison of heat release graphs between dual-fuel combustion

and only diesel fuel combustion (gasoline was removed.) was introduced in Figure 4.3. Although only the diesel fuel combustion phase was different from the behavior of diesel fuel during dual-fuel combustion, because it was hard to separate the each behavior of diesel and gasoline under in-cylinder mixed combustion. Then, the mixing controlled distribution from diesel fuel was small, compared to the entire late combustion phase of dual-fuel combustion. Thus, it can be regarded that the left heat release came from the other combustion phase which is different from mixing controlled combustion.

Thus, in Figure 4.4., the different combustion formation was suggested as a conceptual diagram. First of all, this late combustion phase of dual-fuel combustion mode 1 in Figure 4.2. was not a simultaneous bulky combustion which means fast auto-ignition. Also, there is no possibility that this combustion is a kind of flame propagation, because the overall equivalence ratio was too lean to propagate normally. Therefore, this combustion phase might be a kind of 'propagation of auto-ignition' due to heat transfer. At first, when premixed combustion occurred, this heat energy made surrounding air and fuel mixture, especially premixed gasoline, hotter due to the turbulent flow. So the air and fuel mixture nearby the flame reach the certain temperature and pressure, then it is also ignited. Due to this combustion mechanism, relatively slow propagation of auto-ignition can be implemented. Therefore, this combustion was different from the conventional mixing controlled combustion which is usually inhomogeneous.

The heat release rate of the second dual-fuel combustion mode which has high GIE over 45 % is shown in Figure 4.5. Although the neat diesel combustion phase did not change and just the SOC was advanced as diesel injection timing became earlier, the shape of heat release rate under dual-fuel combustion was changed. Additional the peak of HRR appeared following the first premixed combustion.

Thus, during the same combustion period, more heat release energy was emitted from dual-fuel combustion. Therefore, GIE of dual-fuel combustion was higher than that of neat diesel combustion when diesel injection timing was the same as 13 °BTDC. The second peak came from the simultaneous bulky auto-ignition of residual fuel, especially premixed gasoline. Since combustion started before TDC which means combustion occurred during the compression stroke, in-cylinder temperature and pressure became higher than those of dual-fuel combustion mode 1. Thus, the higher temperature and pressure made that residual fuel easy to self-ignite.

The last (third) dual-fuel combustion shape is shown in Figure 4.6. In this case, the diesel injection timing was earlier as 46 °BTDC. Then, under the neat diesel case, the duration of the main combustion was just 7.5 degree. It means the most of the fuel was well premixed due to enough ignition delay and ready to ignite simultaneously, even though diesel fuel is not easy to be premixed. Therefore, under dual-fuel combustion with the earliest diesel injection case, almost fuels might be well premixed. Thus, the third dual-fuel combustion mode had the single HRR peak and it was smoother than neat diesel case. Also, this combustion showed the lowest NOx emission level and acceptable PRRmax value in Table 4.2. It was a kind of 'PCI' or 'RCCI' combustion due to enough ignition delay to give more mixing time.

Therefore, dual-fuel combustion modes can be classified as three different cases in Figure 4.7. Although all of the dual-fuel combustion begins with the ignition of the diesel (high reactivity) fuel, dual-fuel combustion modes can be divided as the behavior of residual fuel (most of the premixed gasoline and some residual diesel). Among these three cases, mode 2 and 3 had longer ignition delay than injection duration. Also, these two cases had high GIE due to low combustion and exhaust losses in Figure 4.8. Therefore, these two cases can be called as 'PCI' combustion whose meaning is 'premixed compression ignition'.

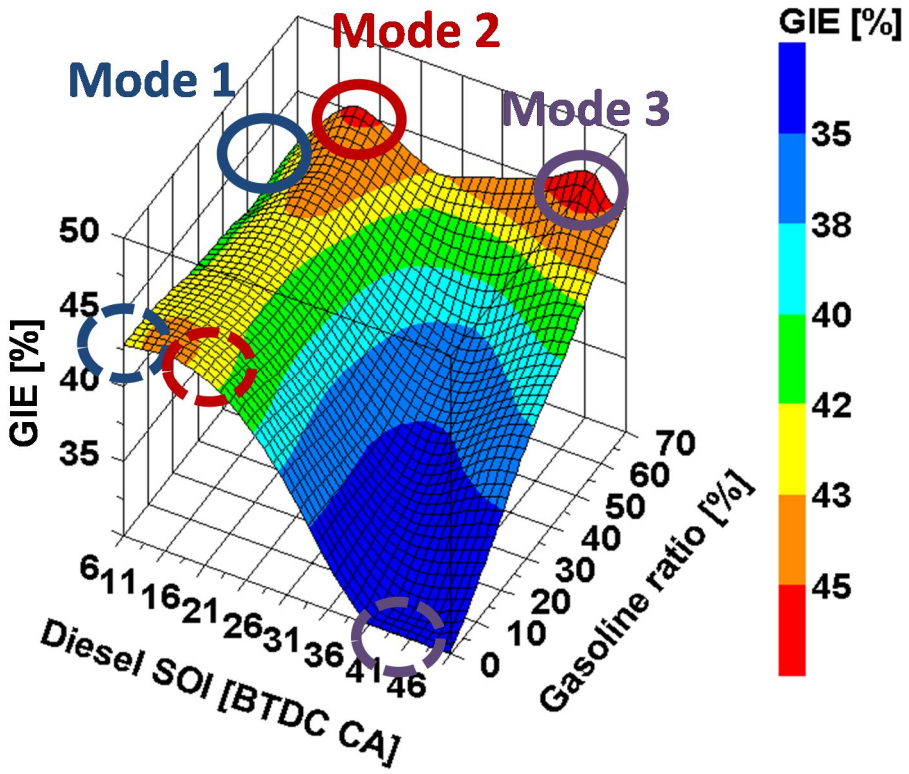


Figure 4.1 Gross indicated thermal efficiency as various diesel injection timings and gasoline fraction under overall equivalence ratio 0.44

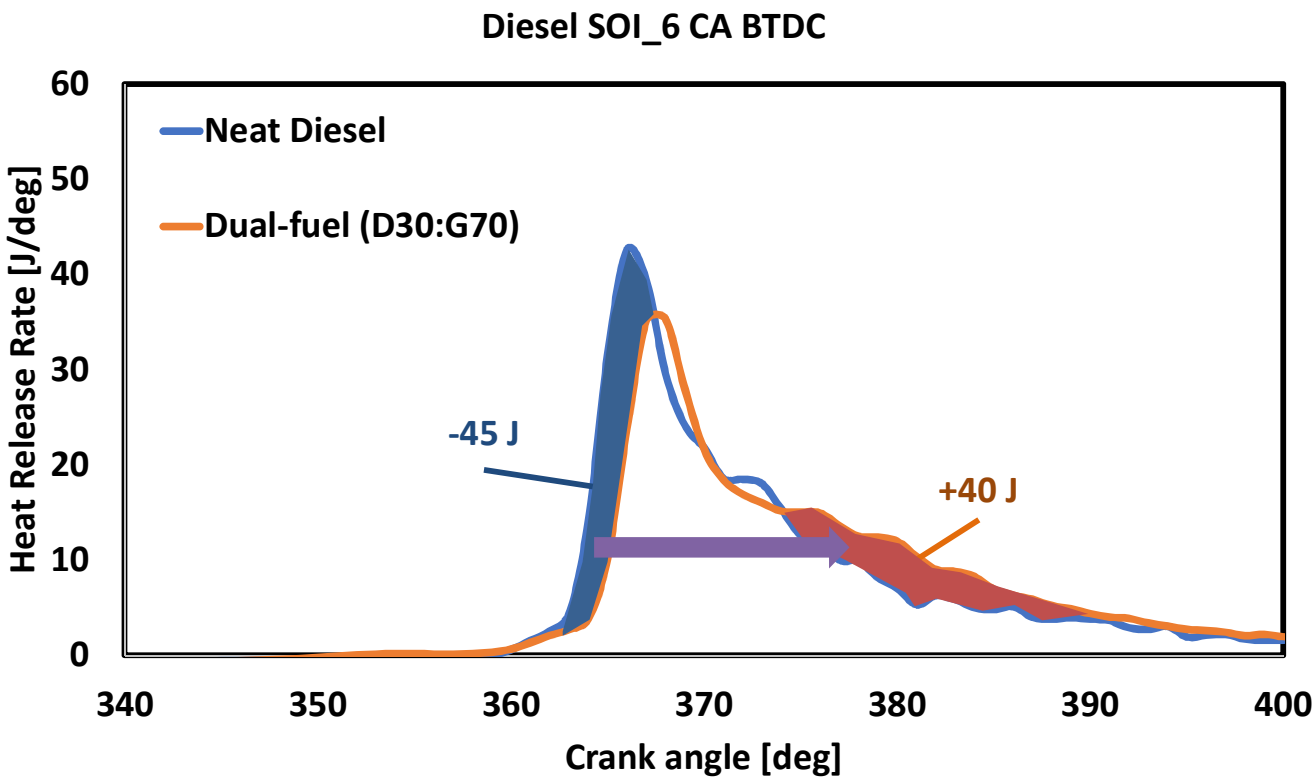


Figure 4.2 Heat release rate graphs of neat diesel and dual-fuel cases under diesel SOI at 6 °BTDC

(Dual-fuel mode 1: Premixed combustion and slow late combustion)

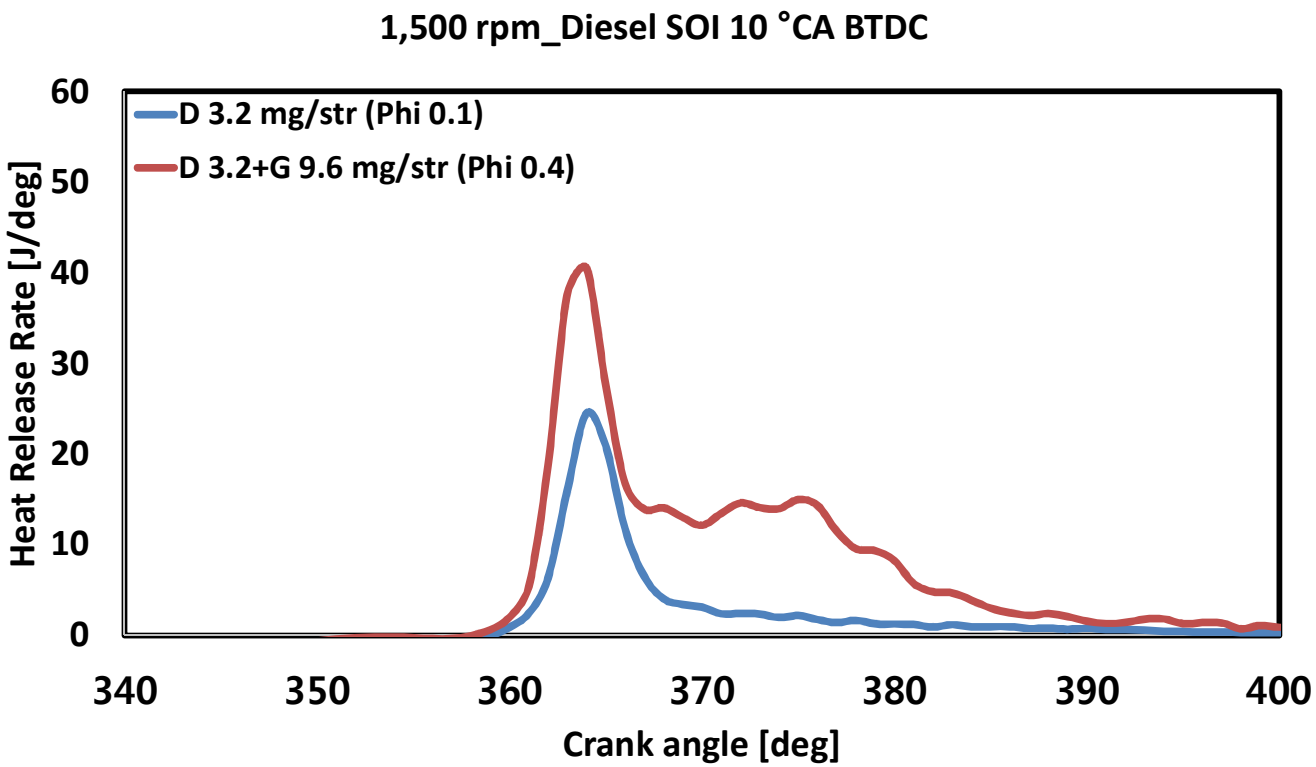


Figure 4.3 Heat release graphs of dual-fuel combustion and diesel fuel only (gasoline removed) cases under diesel SOI at 10 °BTDC

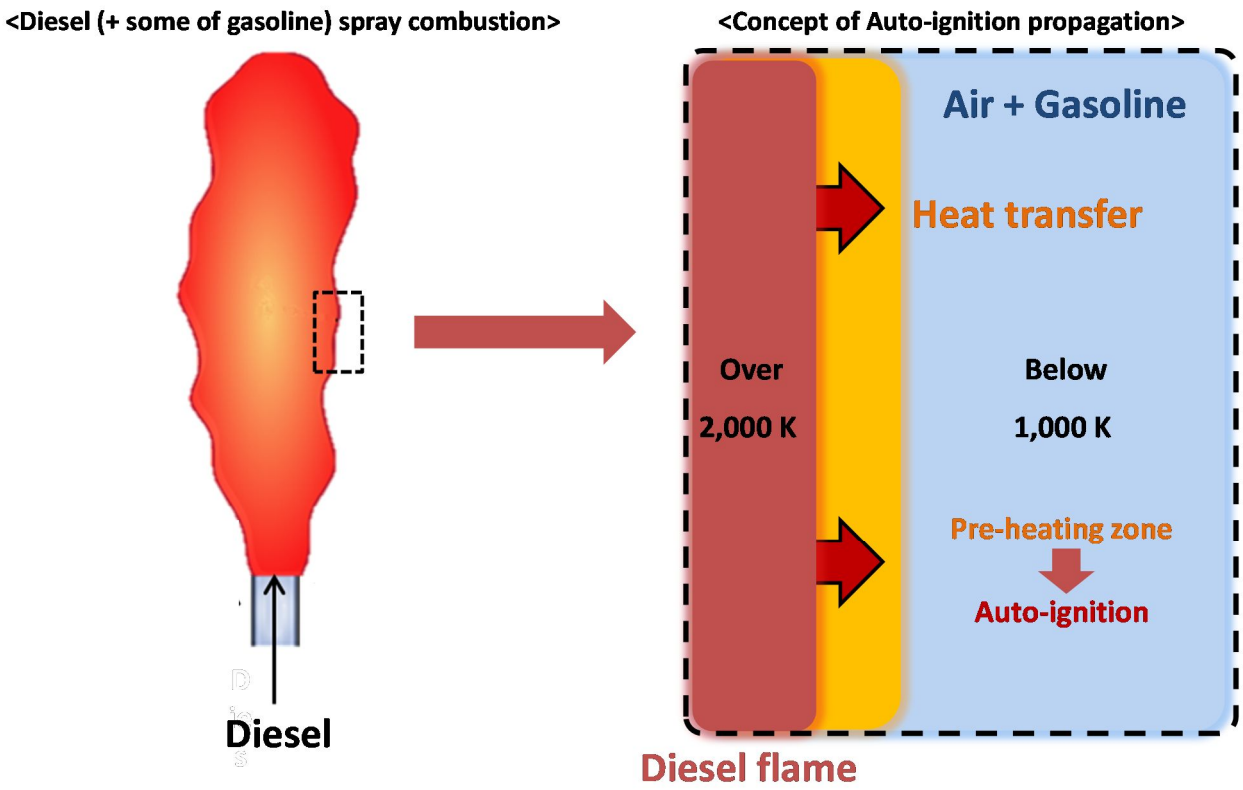


Figure 4.4 Schematic diagram for the explain of 'Auto-ignition propagation' concept

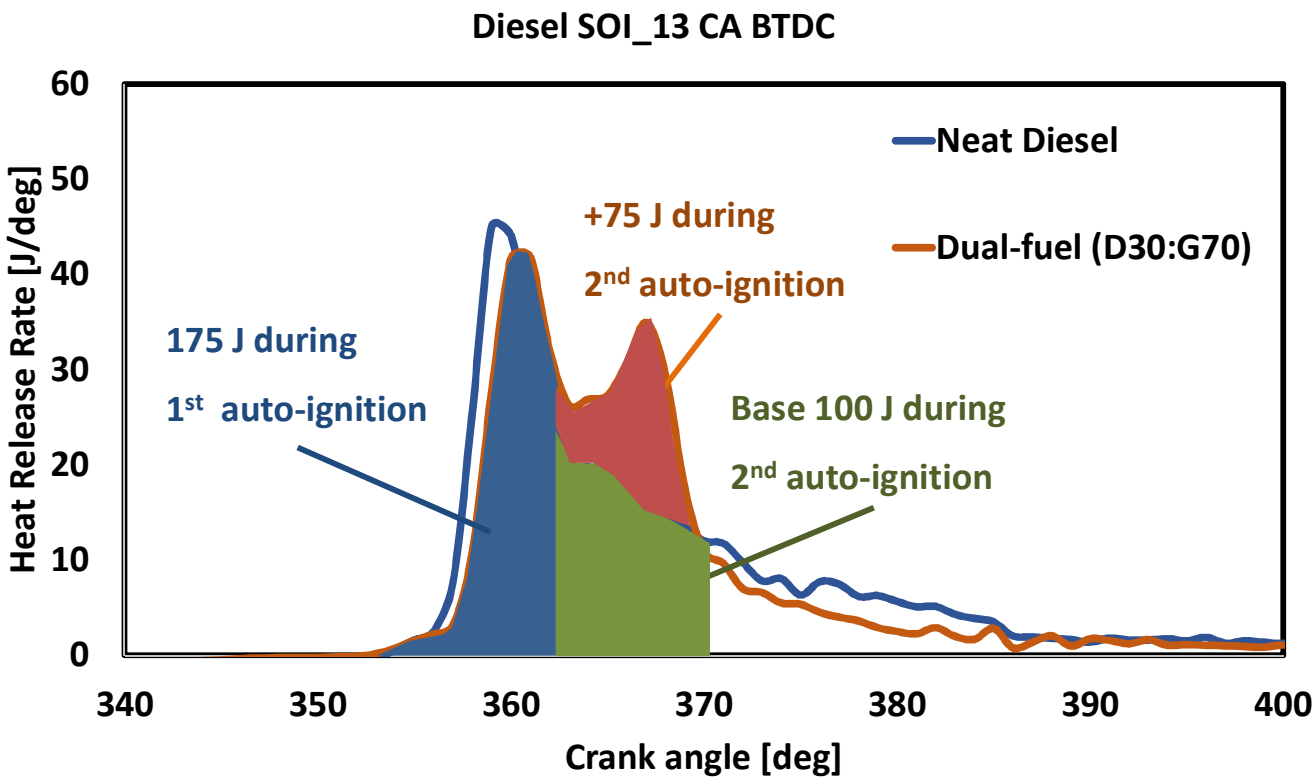


Figure 4.5 Heat release rate graphs of neat diesel and dual-fuel cases under diesel SOI at 13 °BTDC

(Dual-fuel mode 2: Split auto-ignition)

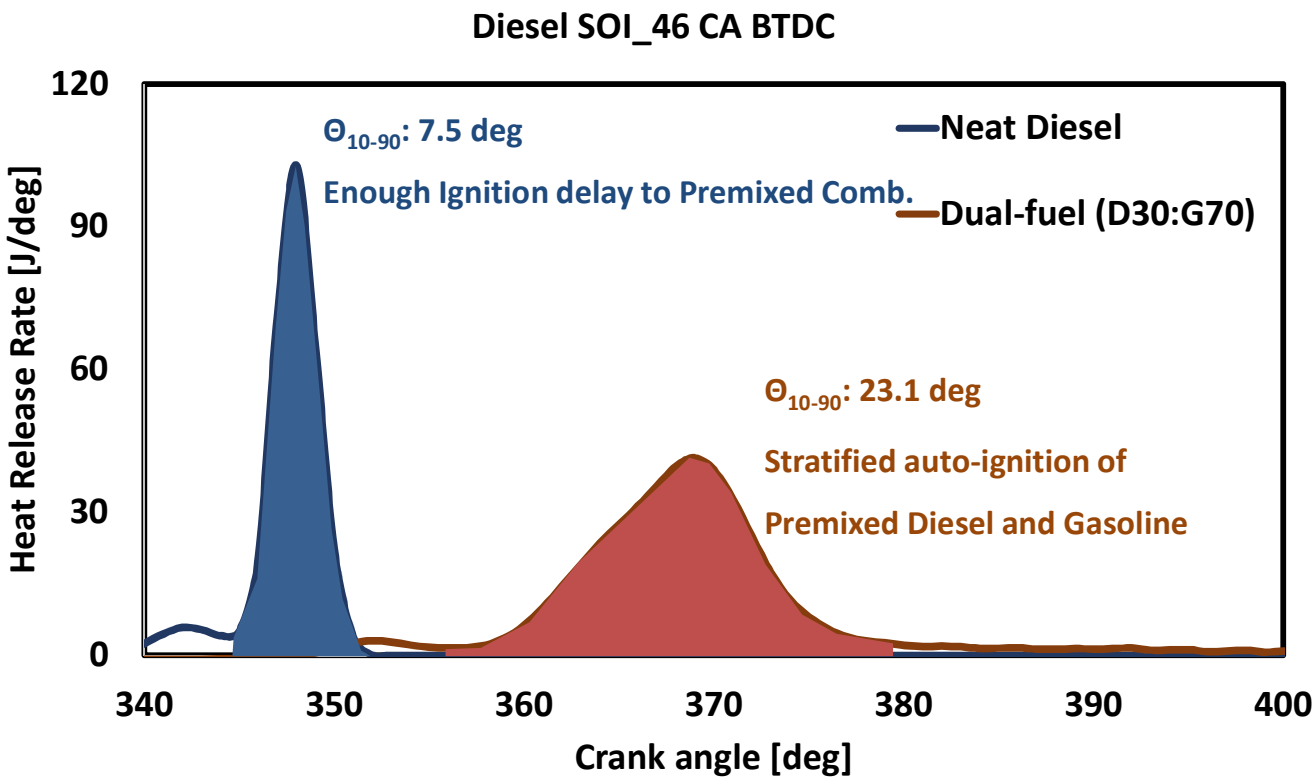


Figure 4.6 Heat release rate graphs of neat diesel and dual-fuel cases under diesel SOI at 46 °BTDC

(Dual-fuel mode 3: Single auto-ignition)

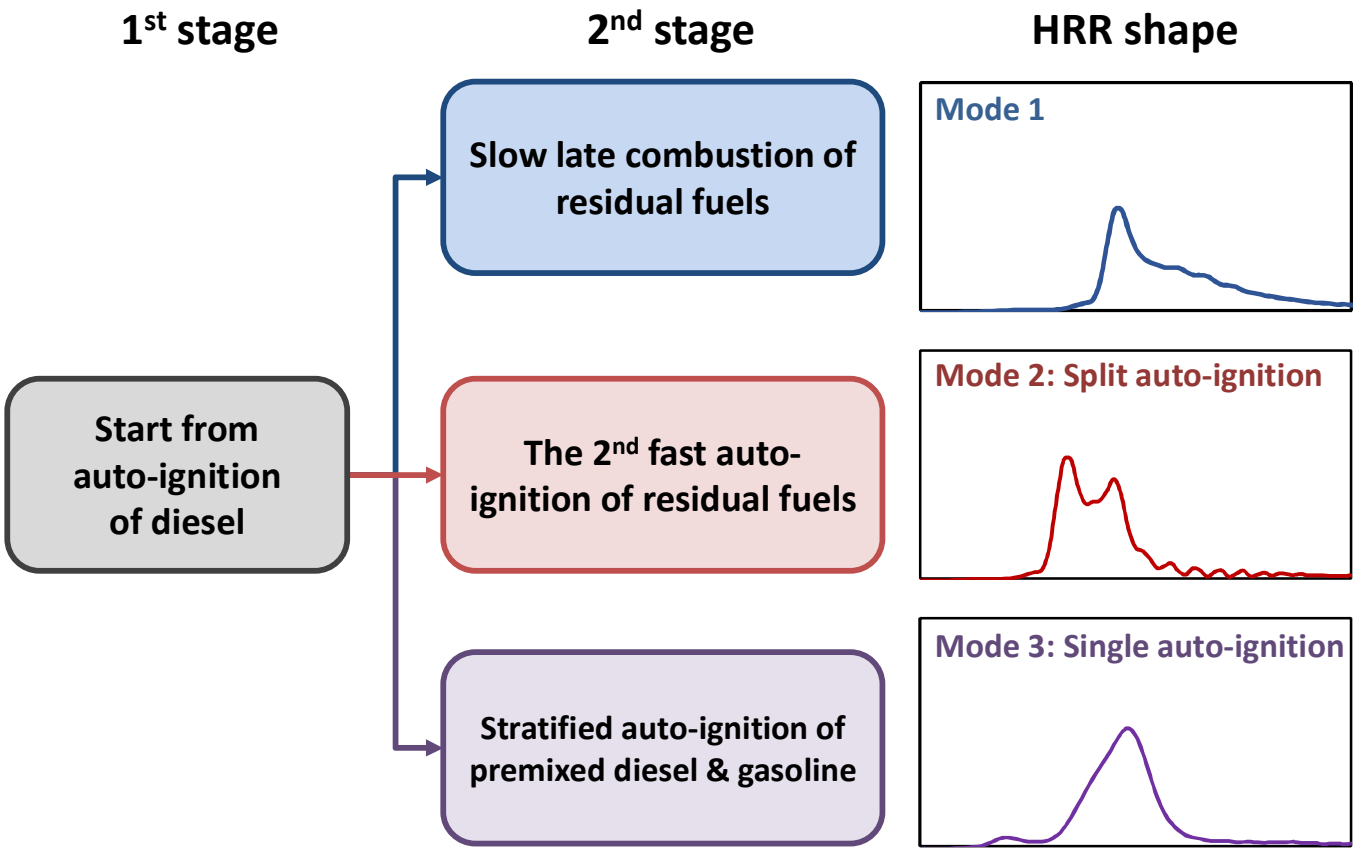


Figure 4.7 Classification of three different dual-fuel combustion modes

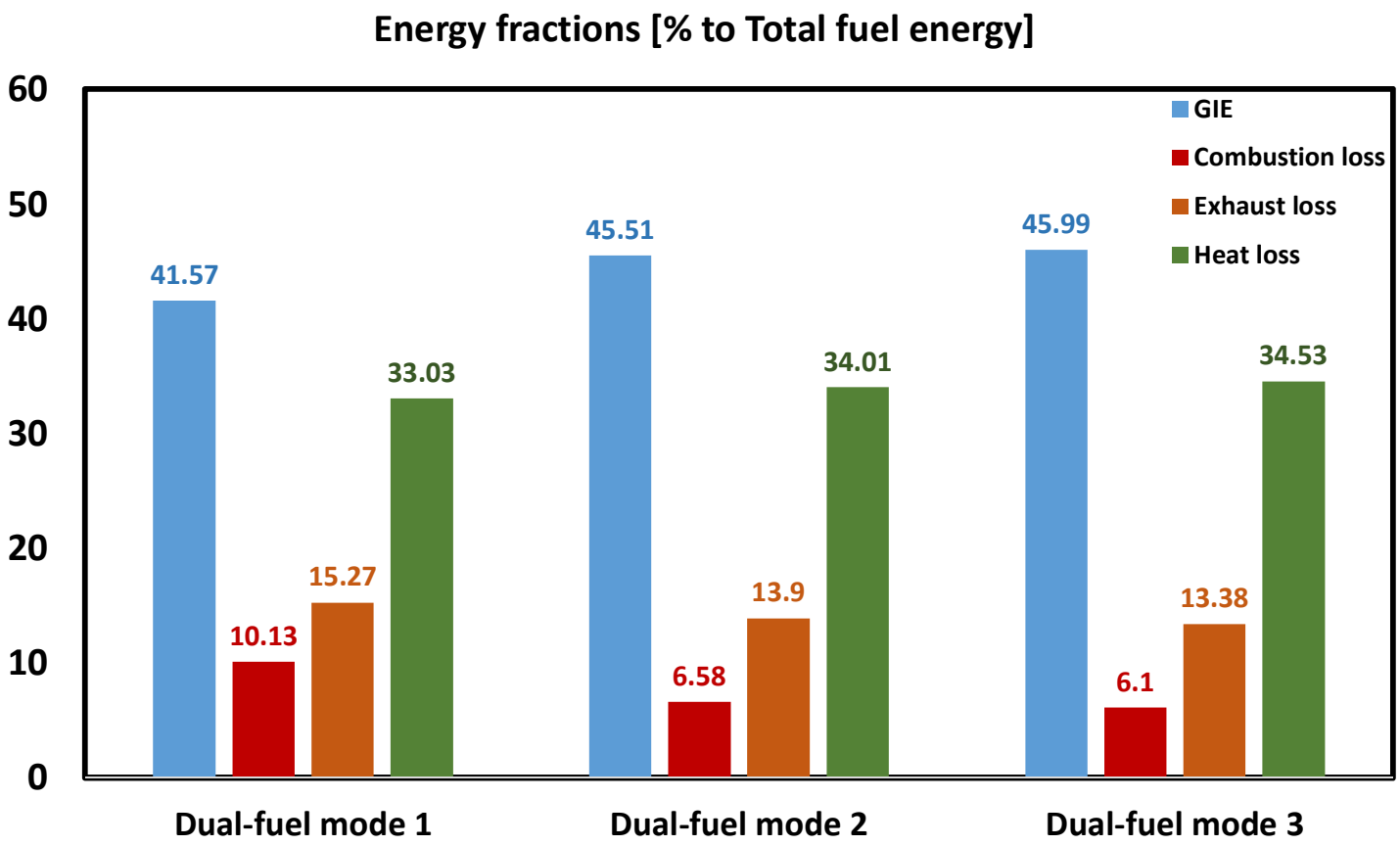


Figure 4.8 Energy budget as various dual-fuel combustion modes

Table 4.2 Emissions and combustion parameter results of three different dual-fuel combustion modes

Description	Values		
Mode number	1	2	3
gIMEP [bar]	6.04	6.61	6.68
NOx [ppm]	402	916	53
PM [FSN]	0.04	0.03	0.04
CO [ppm]	2828	1225	1007
THC [ppmC]	6317	4507	4186
CO_2 in Exh. [%]	5.57	6.01	5.85
PRRmax [bar/deg]	5.70	9.71	4.65
GIE [%]	41.57	45.51	45.99
Exh. Temp. [K]	592	579	583

4.2 Investigation into the relation between combustion parameters and dual-fuel combustion modes

In this chapter, the relation between combustion parameters (index) and classified dual-fuel combustion modes was mainly investigated. It was important to find the major factors of each dual-fuel combustion modes, because it made the control of the appropriate combustion modes. Also, in this chapter, the source of THC emission under dual-fuel combustion was studied using the relation mentioned above. Under dual-fuel combustion, not only wall-wetting from early diesel injection, but also the crevice effect due to gaseous fuel from intake mixture was one of the source to THC emission. Therefore, if it can be estimated which one more contributed to THC emission, this result can suggest the way to improve the combustion efficiency of dual-fuel combustion. More detailed experimental conditions were introduced in Table 4.3.

Table 4.3 Engine operating conditions for the investigation of relation between combustion parameters and each mode under dual-fuel combustion

Description	Value
Engine speed [rpm]	1,500
Coolant & oil temperature [K]	358
Diesel injection pressure [bar]	450
Intake pressure [bar]	1.10
EGR rate [%]	0
Overall-equivalent ratio	0.36/0.44/0.58
Gasoline ratio [%]	0~70
Diesel SOI [°BTDC]	6~46

In Figure 4.9., the distributions of each dual-fuel combustion mode as various diesel injection timings and gasoline ratios was introduced under three different overall equivalence ratios. When the overall equivalence ratio was the highest as 0.58, these were no mode 2 shapes. Since the amount of fuel which cause a drastic increase in the in-cylinder temperature and pressure was larger than other equivalence ratio cases, two combustion phases tended to be merged rapidly, rather than the separation of two peaks. Also, under the leanest equivalence ratio condition (0.36), mode 2 was rarely shown, compared to the middle equivalence ratio condition. For the gasoline fuel during dual-fuel combustion, this equivalence ratio was too lean to self-ignite [52]. Thus, the second peak of HRR in the leaner or richer cases were rarely emerged.

The obvious combustion parameters which can take the role of standard to divide combustion modes were the existence of LTHR (low-temperature heat release) region and the tendency of MFB 50 in Figure 4.10. LTHR region usually appeared under dual-fuel combustion mode 3, which is a kind of PCI combustion in Figure 4.10-(a). Also, the tendency of MFB 50 was changed in the opposite direction to the diesel injection timing under dual-fuel combustion mode 3 in Figure 4.10-(b). Since dual-fuel combustion mode 3 is not dependent on the diesel spray combustion and rather related to the stratification of high and low reactivity fuels in the cylinder, these two clearer evidence existed.

Especially, there were two high GIE cases for the overall equivalence ratio 0.44 in Figure 4.11-(a), because there were two suitable MFB 50 location for the higher GIE due to the changed combustion phase. This combustion change has significant meaning. Based on this turning point, dual-fuel combustion mode 2 which has the same tendency between diesel injection timing and combustion phase, i.e. MFB 50, is ‘late injection PCI’ and dual-fuel combustion mode 3 which has the opposite

tendency between above two parameters can be called ‘Early injection PCI’. Although NOx emission from dual-fuel combustion mode 3 was lower than that of dual-fuel combustion mode 2 in Figure.4.11-(b), there is a possibility to control NOx emission by using a high level of EGR under dual-fuel combustion mode 2. Thus, dual-fuel combustion mode 2 can be used to ‘high EGR LTC’ and mode 3 was a base condition for the ‘early injection PCI (=PCI)’. These applications were introduced in Chapter 4.4.

As shown in Figure 4.12-(a), relatively high load condition (overall-equivalent ratio 0.58), THC emission was sensitive to the gasoline fraction rather than diesel injection timings. Especially, under the neat diesel combustion condition, which means gasoline ratio was zero, early diesel injection caused high THC emission level due to wall-wetting effect. In this condition, the amount of diesel fuel was 16.6 mg/str. Since the in-cylinder pressure was lower as diesel injection timing was advanced, diesel spray penetration length became longer. Thus, the wall-wetting effect was the main source under higher load conditions with low gasoline fractions. As a result, THC emission became rather decreased as gasoline fraction was increased, in other words, diesel fraction reduced (= diesel spray penetration length shortened) [36, 54]. Although combustion loss was decreased due to the reduction of THC emission as increasing gasoline fraction under the earliest diesel injection timing, GIE became lower unlike tendency of other cases, because the most of the combustion occurred during the compression stroke in Figure 4.13. Therefore, it made more negative work during compression stroke.

On the other hand, when diesel injection timing was closer to TDC, THC emission was increased as a increasing gasoline fraction. There are two reasons for the phenomenon. The first one is increasing gaseous fuel in the crevice volume as a gasoline fraction increased. Also, the second reason is that higher low reactivity fuel

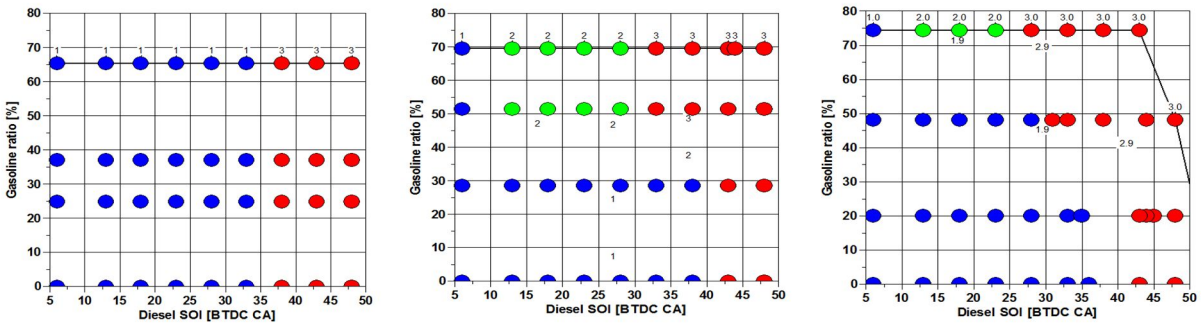
made combustion phase retarded, so it brought the low combustion temperature which is not enough temperature to oxidize THC emission well. Therefore, under the relatively high load condition which means the larger amount of fuels was supplied, crevice effect was dominant when diesel injection timing was near TDC. On the other hand, the effect of wall-wetting effect from the diesel spray was stronger than crevice effect when diesel early injection was applied.

However, as the overall equivalent ratio became leaner as 0.44 and 0.36 (Figure 4.12-(b) and (c)), crevice effect was more dominant, compared to the effect of a wall-wetting problem. Since the total amount of fuels became smaller, the influence of diesel spray penetration length became small. Also, THC emission became larger as the overall equivalent ratio was reduced, which means lower load and leaner condition. There are also two main reasons. One of them is that there is not enough combustion temperature and heat energy to oxidize THC emissions from diesel fuel. The second reason is that there are locally over-mixing regions, which means lean pockets.

Although crevice effect from low reactivity fuel is more dominant to bring a high level of THC emission under dual-fuel combustion, it should avoid early diesel injection timing when the diesel amount was larger, especially in this experiment, the overall-equivalent ratio was higher than 0.5.

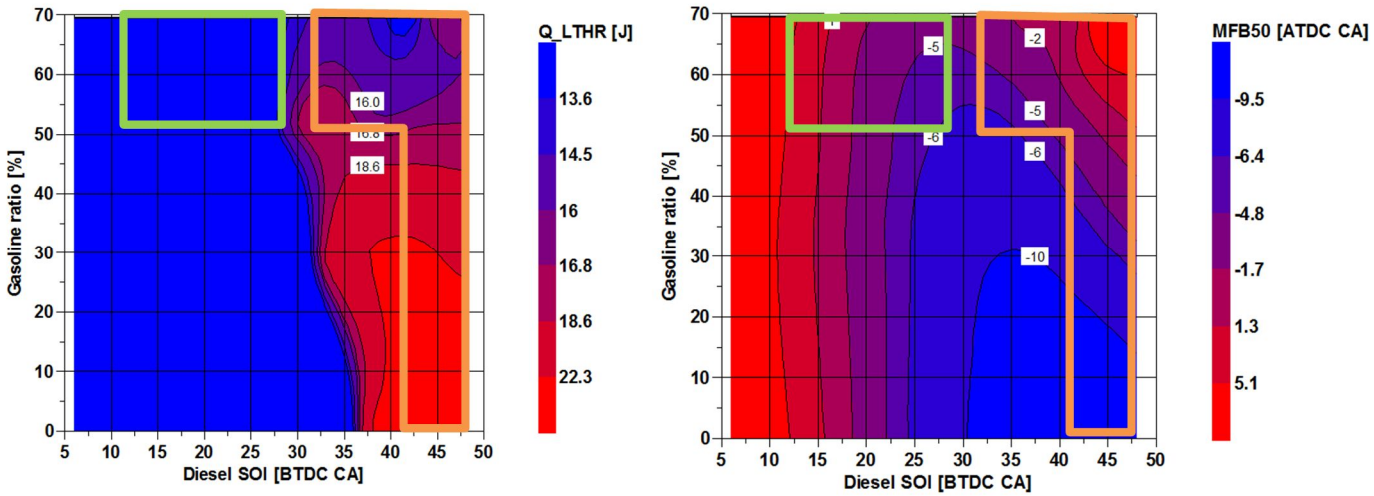
Therefore, from the results of this chapter, it can be concluded that too much leaner equivalent ratio is not suitable for the dual-fuel combustion. Also, it can be found that there is a trade-off relation between the amount of diesel and gasoline fractions. If the amount of diesel fuel was reduced to prevent the wall-wetting condition, gasoline fraction should be increased, but crevice effect was increased. On the other hand, if the gasoline fraction was reduced to decrease the crevice effect,

wall-wetting from diesel spray could occur with a earlier injection strategy. As a result, there is an optimal value of diesel and gasoline fractions for better dual-fuel combustion.



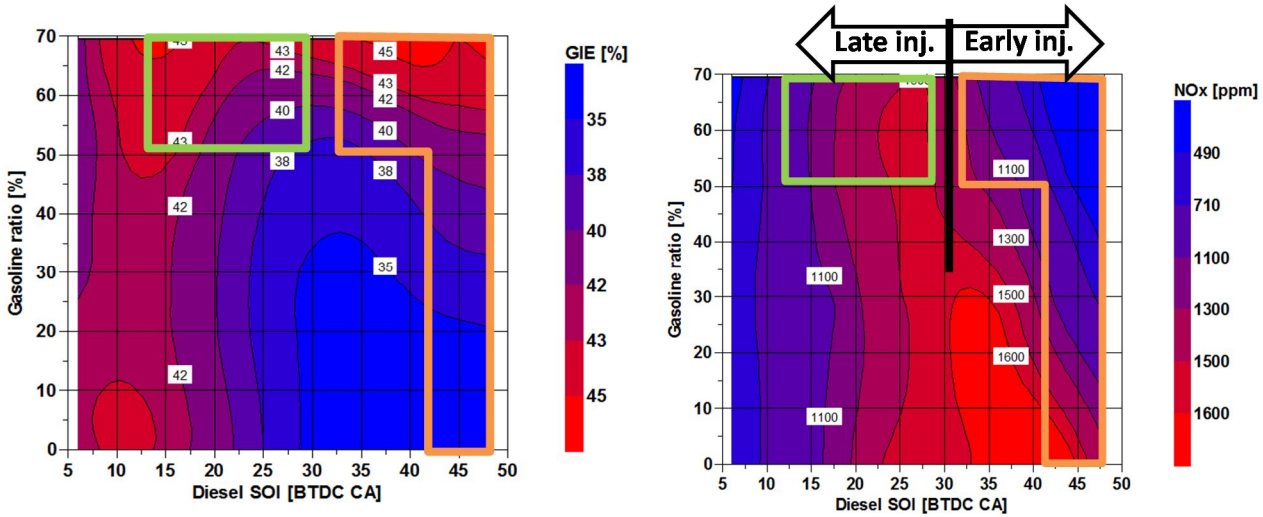
(a) (b) (c)

Figure 4.9 Distribution of dual-fuel combustion modes as diesel injection timings and gasoline ratios under overall equivalence ratio 0.58 (a), 0.44 (b) and 0.36 (c) (Blue: mode 1/ Green: mode 2/ Red: mode 3)



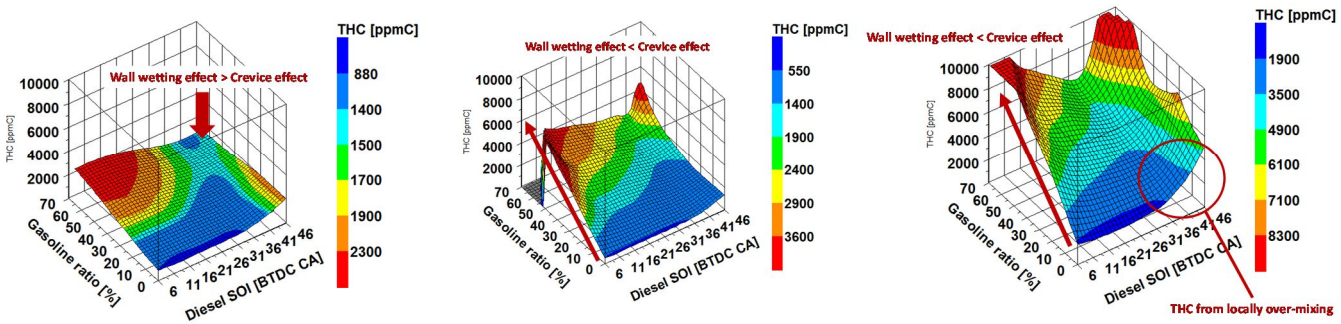
(a) (b)

Figure 4.10 Heat from LTHR (a) and MFB50 (b) of dual-fuel combustion as various diesel injection timings and gasoline ratios under overall equivalence ratio 0.44



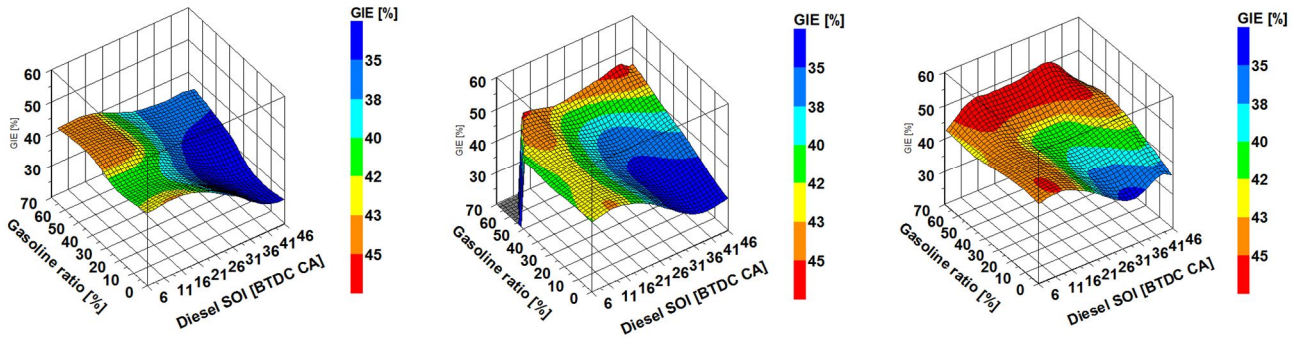
(a) (b)

Figure 4.11 Gross indicated thermal efficiency (GIE) (a) and NOx emission (b) of dual-fuel combustion as various diesel injection timings and gasoline ratios under overall equivalence ratio 0.44



(a) (b) (c)

Figure 4.12 THC emission of dual-fuel combustion as various diesel injection timings and gasoline ratios under overall equivalence ratio 0.58 (a), 0.44 (b) and 0.36 (c)



(a) (b) (c)

Figure 4.13 Gross indicated thermal efficiency of dual-fuel combustion as various diesel injection timings and gasoline ratios under overall equivalence ratio 0.58 (a), 0.44 (b) and 0.36 (c)

4.3 Definition and operating strategy of dual-fuel PCI

From the results of previous experiments, it can be shown that early diesel injection with high and low reactivity fuel is preferable for the low NOx and PM emissions with the high thermal efficiency. Therefore, as it was mentioned in Chapter 4.1, this combustion is named 'dual-fuel PCI', because most of the high and low reactivity fuels are burned as a bulky premixed combustion. Especially, in Figure 4.10-(a), low-temperature heat release regions occurred in the area of a tendency to NOx reduction, which is depicted as the black lined area in Figure 4.11-(b). Since prolonged ignition delay by two-staged ignition, brought the overall lean premixed mixture condition, it is favorable to reduce NOx and PM emissions.

From Figure 4.14. to 4.19, each stage for the implementation of dual-fuel PCI is introduced with its HRR graphs. During this experiment sequence, diesel injection pressure was fixed to 450 bar. Coolant and oil temperatures were also fixed at 85 ℃. At first, low reactivity fuel was supplied as possible, not exceeding the amount to spoil the appropriate combustion efficiency. Then, it can be shown that NOx and PM emissions can be reduced just by substituting the diesel fuel by gasoline fuel, because local rich regions can be eliminated.

Then, if diesel injection timing was advanced earlier from 6 to 46 °BTDC, the late combustion phase became diminished and only one sharp curved occurred. This means premixed diesel and gasoline fuels burned simultaneously, but stratified auto-ignition existed. Thus, smoother combustion can be achieved by the phenomenon similar with 'auto-ignition propagation'. As a result, low NOx (53 ppm) and PM emission (under 0.05 FSN) can be achieved without using EGR.

Next, EGR was supplied into this condition for more NOx reduction. EGR was still a strong method to reduce NOx emission. Since there is a limitation to reduce NOx emission by advancing diesel injection timing to achieve more lean and homogeneous mixture with maintaining high thermal efficiency, the supplementation of EGR was not inevitable. However, because of the high EGR rate, gross thermal efficiency became lower. One of the reasons is lowered oxygen concentration by the EGR addition which made richer combustion condition. The other reason is that lowered combustion temperature by thermal and dilution effect of EGR. Therefore, there was also deterioration of combustion efficiency.

To overcome low combustion efficiency, the split diesel injection strategy was applied. Since split diesel injection made shortened diesel spray penetration length compared to single diesel injection condition, wall-wetting can be reduced as introduced in Chapter 4.2. Also, the improvement of high reactivity stratification by split injection helped to increase combustion efficiency. Thus, gross thermal efficiency became a little higher by recovering the combustion loss.

At last, increasing the boost pressure with higher EGR rate contributed to improve thermal efficiency by solving the problem of richer combustion. Especially, more EGR addition was possible due to increasing the oxygen rate with the high boost pressure. Therefore, more reduction in NOx emission with near zero PM emission can be achieved. Gross thermal efficiency was also recovered up to 45 % which is higher than that of conventional diesel combustion condition, even though the compression ratio was lower than conventional in Figure 4.20. (From 17.3 to 14).

Actually, in Figure 4.21, the optimized dual-fuel PCI combustion was constructed by most of the premixed combustion compared to the conventional

diesel combustion. It was clear that there was a rarely diffusive flame stage in the dual-fuel PCI.

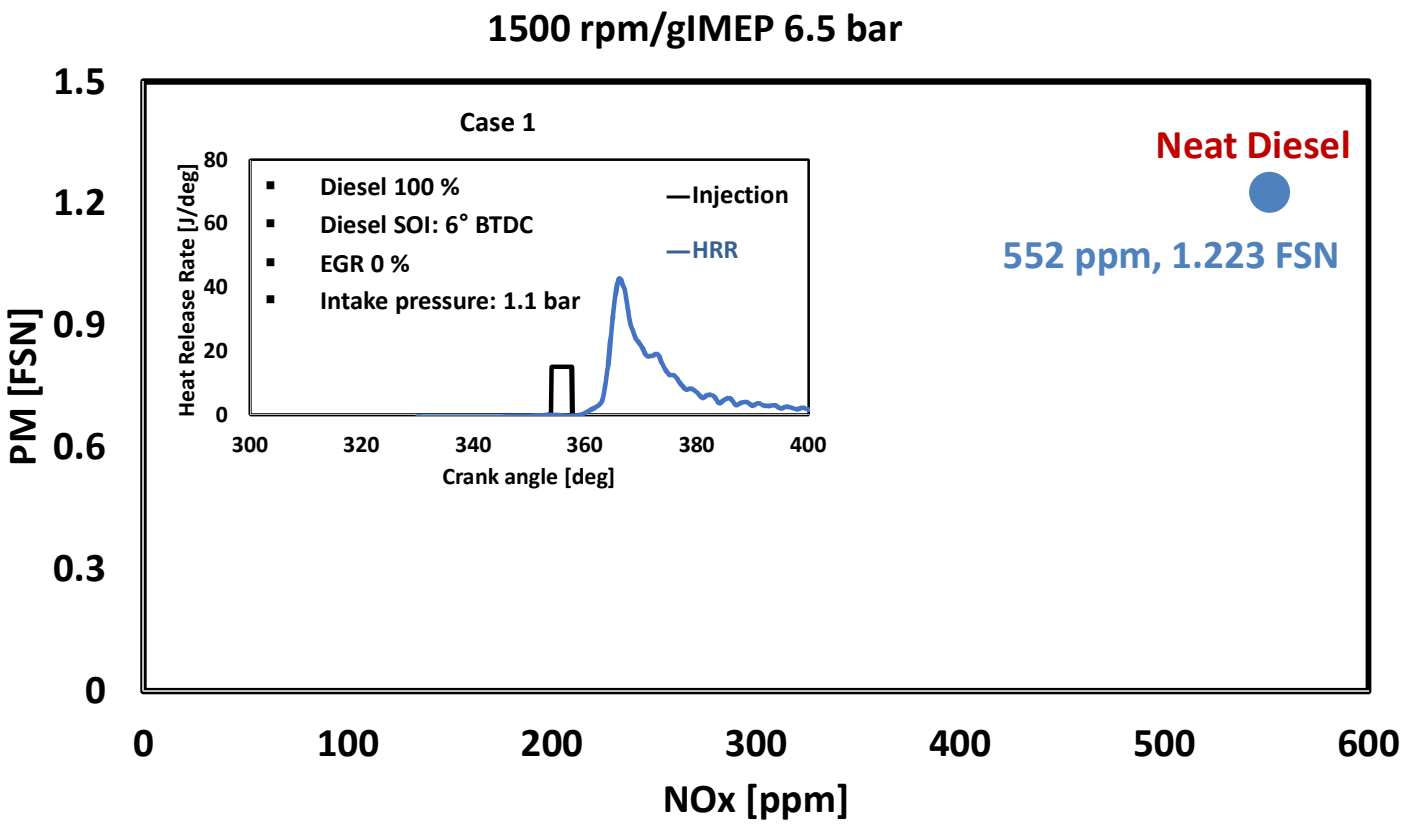


Figure 4.14 Case 1: NOx versus PM emissions and HRR graph of Neat diesel 100 % condition

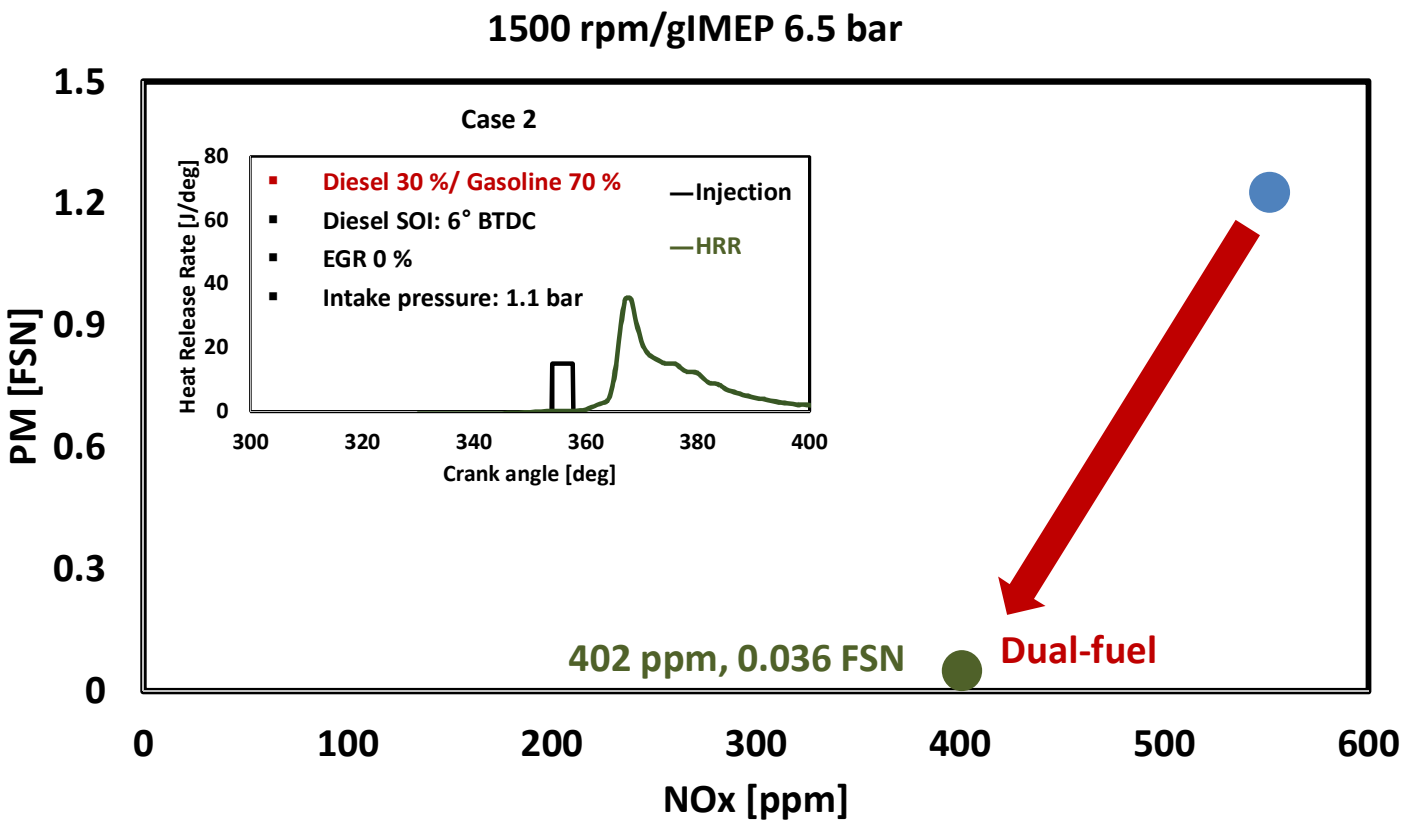


Figure 4.15 Case 2: NOx versus PM emissions and HRR graph

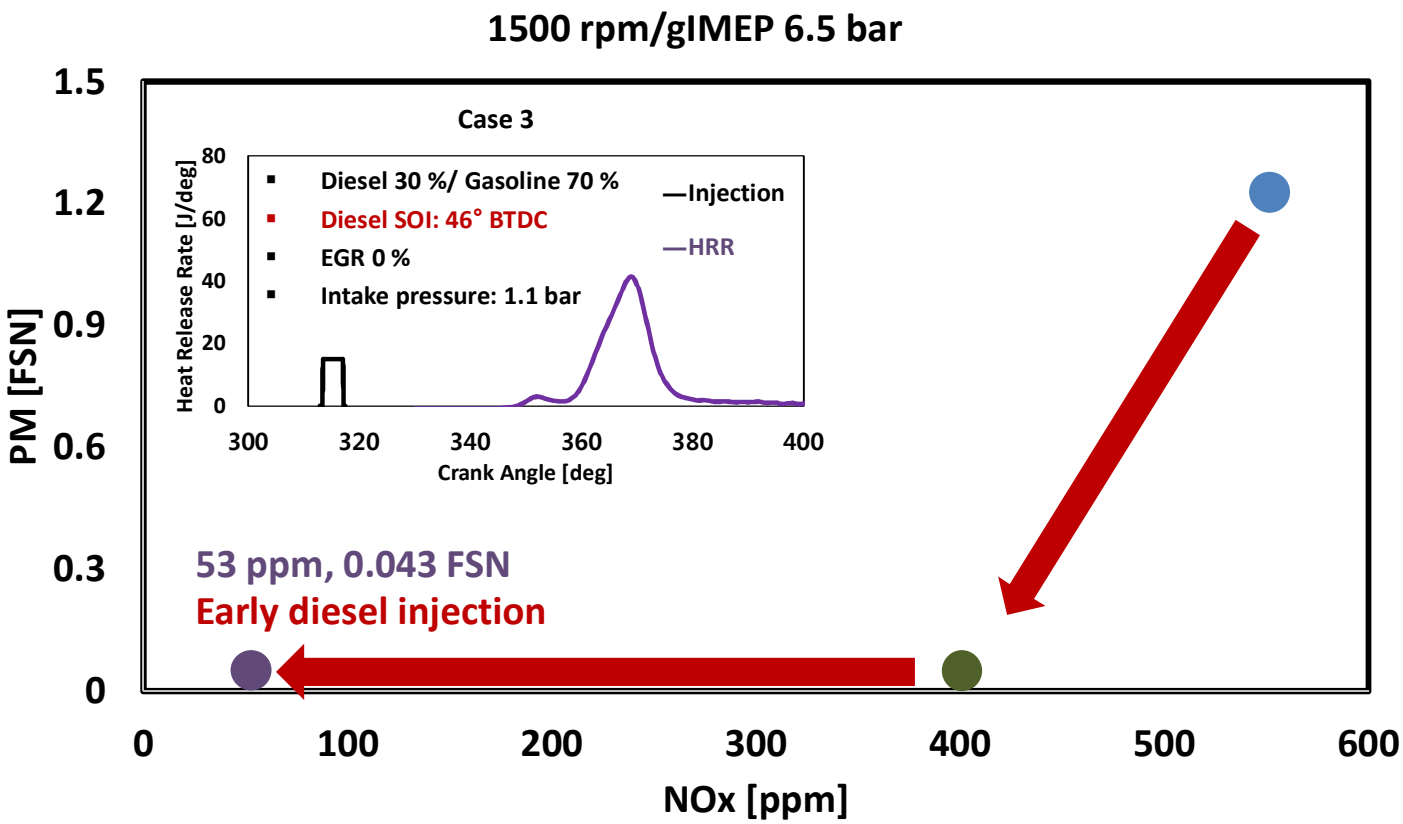


Figure 4.16 Case 3: NOx versus PM emissions and HRR graph

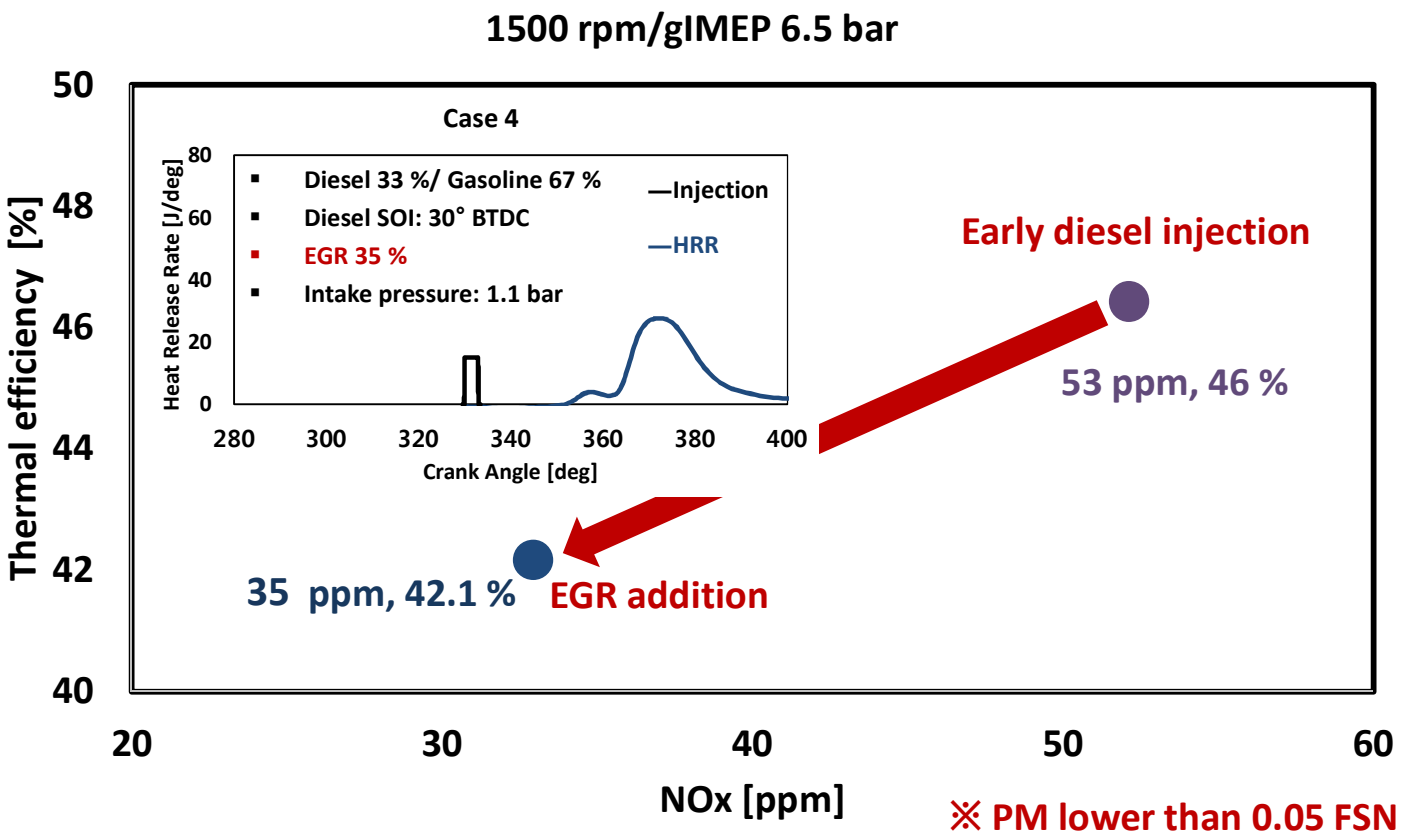


Figure 4.17 Case 4: NOx versus gross thermal efficiency and HRR graph

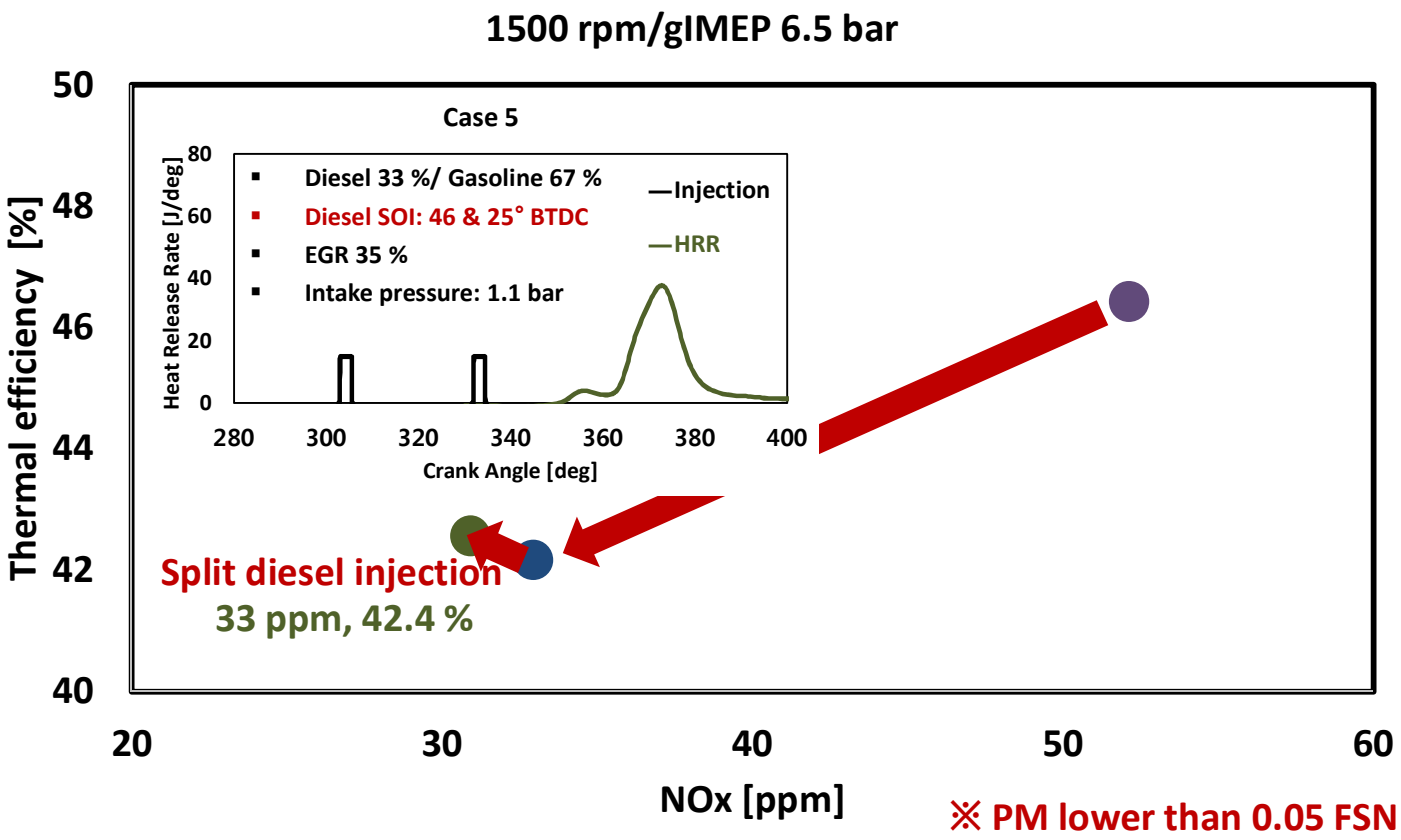


Figure 4.18 Case 5: NOx versus gross thermal efficiency and HRR graph

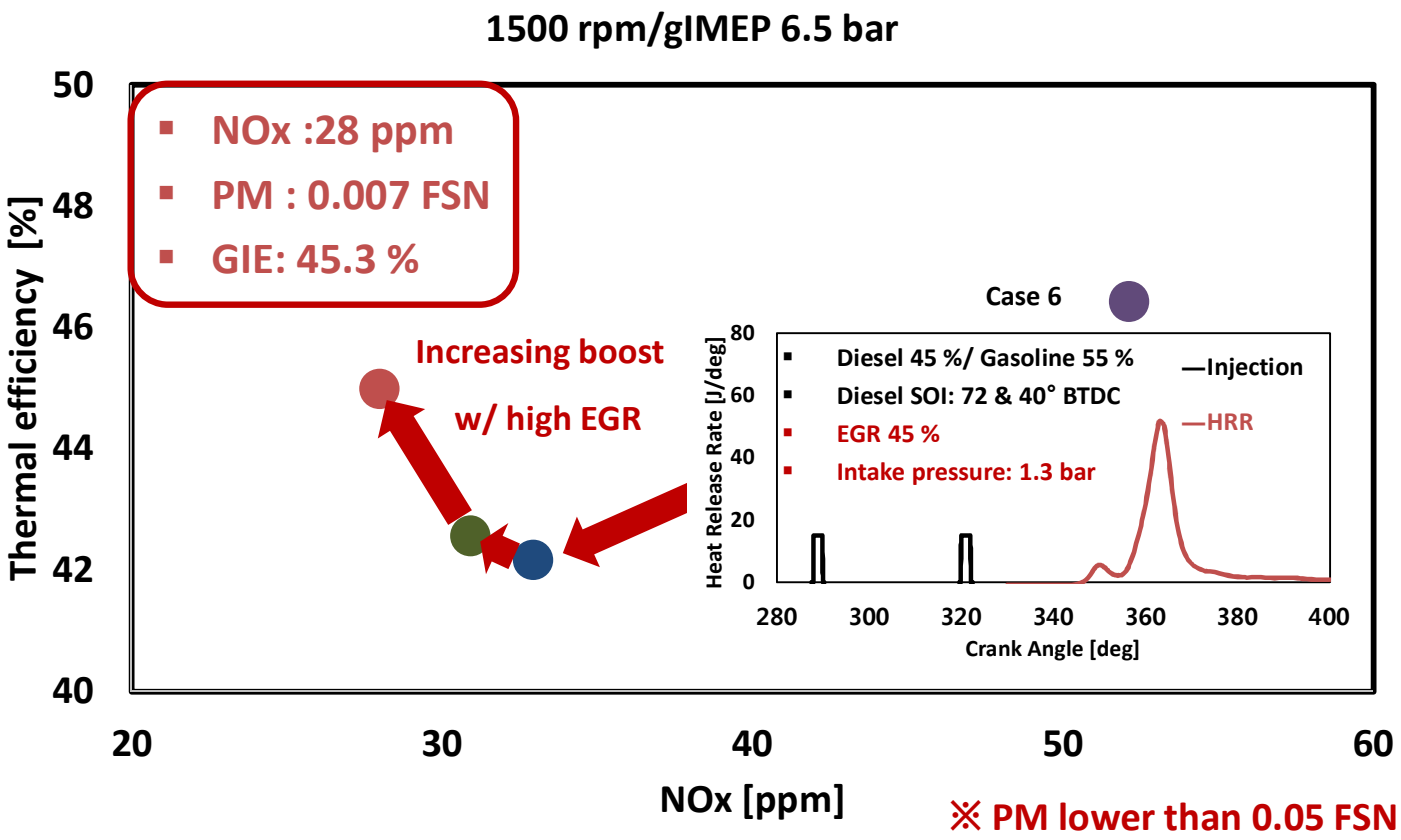


Figure 4.19 Case 6: NOx versus gross thermal efficiency and HRR graph

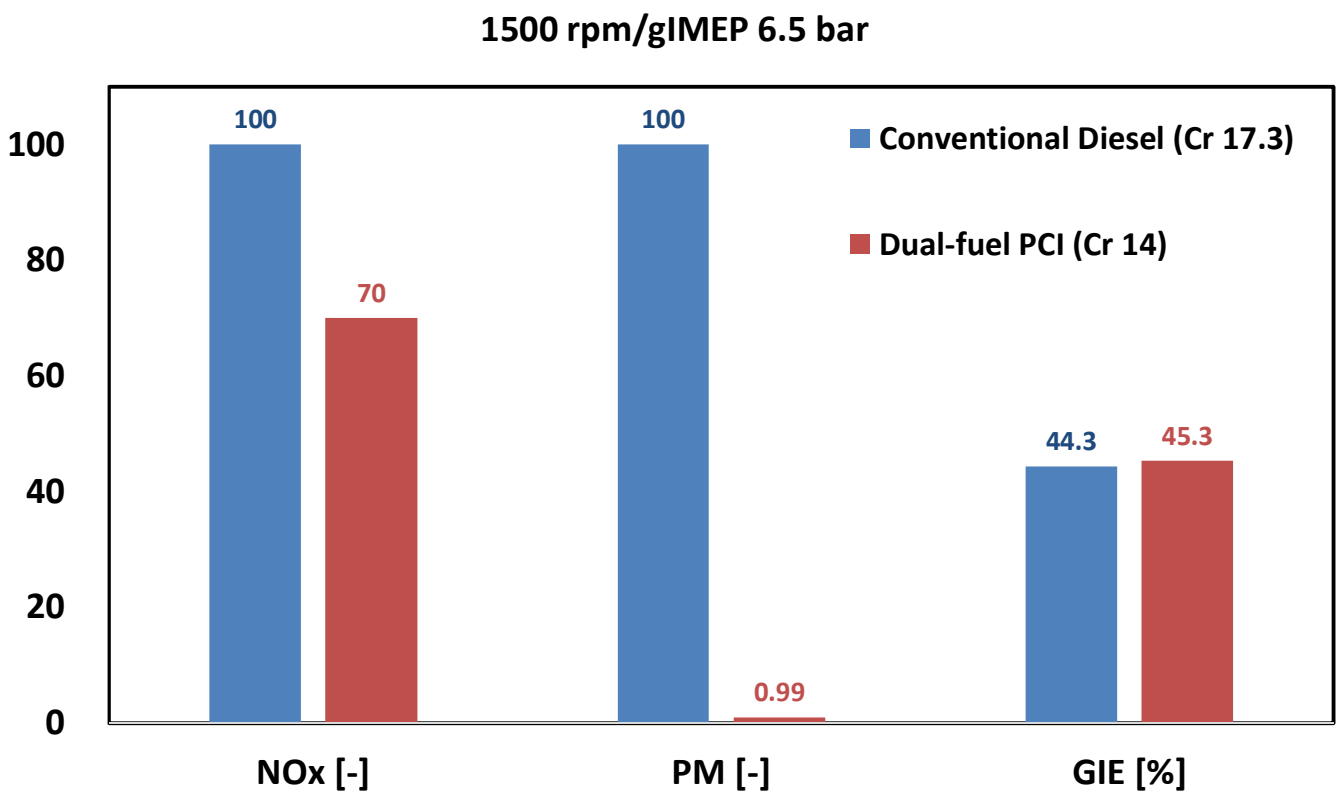


Figure 4.20 Normalized NOx, PM emissions, and gross thermal efficiency of dual-fuel PCI (Cr 14) and conventional diesel (Cr 17.3) (※ Conventional diesel combustion result is from commercial diesel 1.6 L engine)

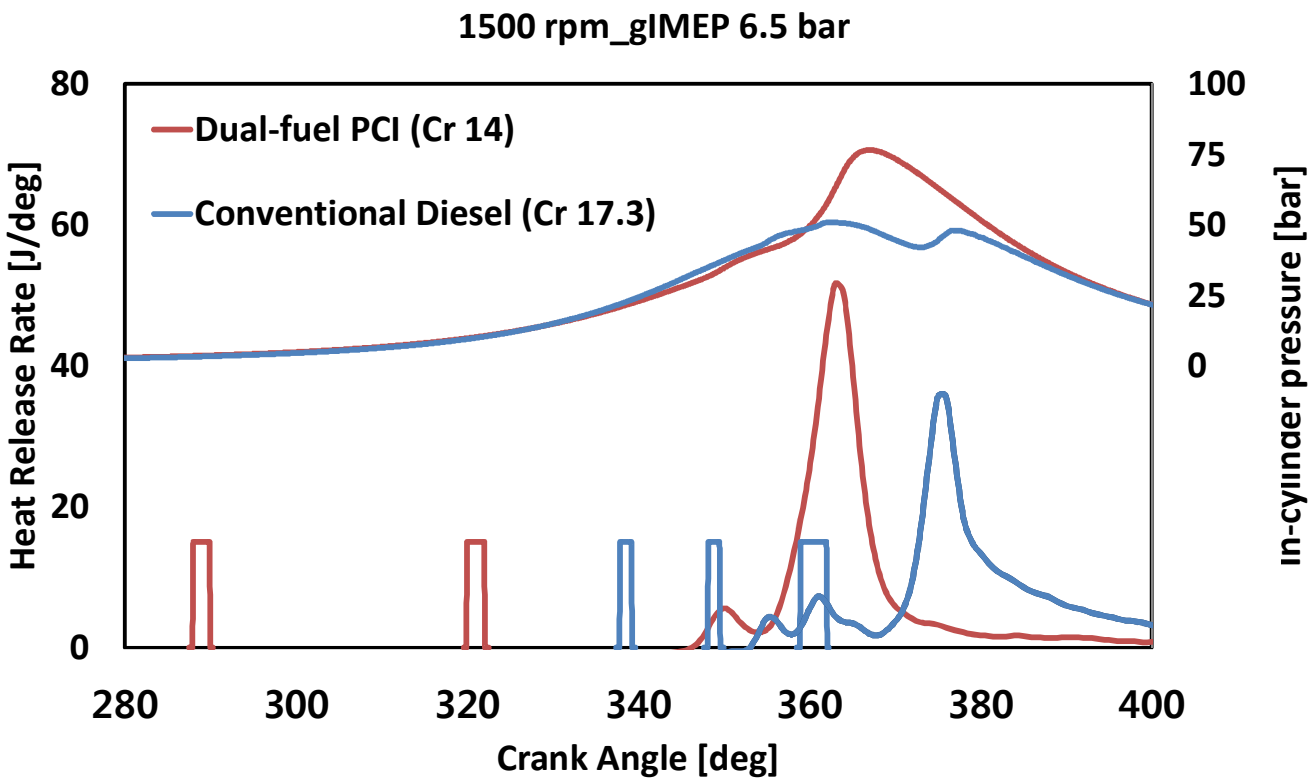


Figure 4.21 Heat release rate and in-cylinder pressures of dual-fuel PCI (Cr 14) and conventional diesel combustion (Cr 17.3) (※Conventional diesel combustion result is from commercial diesel 1.6 L engine)

4.4 Comparison of dual-fuel PCI with diesel PCI

In this chapter, dual-fuel PCI was compared to the classical neat diesel PCI combustion concept to evaluate the competitiveness of a dual-fuel PCI. Especially, two different dual-fuel PCI strategies were introduced. The first one is using a heavy amount of EGR up to 45 % based on the mode 2 and the other one is using low EGR as 15 %, but earlier diesel injection timings which are based on the mode 3. The first one is a kind of low-temperature combustion (LTC) with the low oxygen concentration in intake mixture and the second one is a leaner and more premixed condition by the increasing ignition delay. The details in experimental condition were presented in Table 4.4. Diesel injection strategies for each case are depicted in Figure 4.22.

Table 4.4 Engine operating conditions for the comparison of dual-fuel PCI and diesel PCI combustion concepts

Description	Value
Engine speed [rpm]	1,500
Total LHV of fuels [J/cycle]	580
gIMEP [bar]	6.2 for diesel PCI 6.6 for dual-fuel PCI
Coolant & oil temperature [K]	358
Diesel injection pressure [bar]	450 for dual-fuel PCI 750 bar for diesel PCI
Intake pressure [bar]	1.30
EGR rate [%]	15 for dual-fuel PCI mode 3 45 for dual-fuel PCI mode 2 and diesel PCI
Overall-equivalent ratio	0.44 for dual-fuel PCI mode 3 0.66 for dual-fuel PCI mode 2 and diesel PCI
Gasoline ratio [%]	66 for dual-fuel PCI mode 3 55 for dual-fuel PCI mode 2 0 for diesel PCI
MFB 50 [°ATDC]	3~6

In Figure 4.22, HRR and in-cylinder pressure traces are depicted. Then, it can be shown that all the cases have the LTHR regions. It means these combustions can be classified as a kind of low-temperature combustion. Especially, in diesel PCI combustion, the distance between LTHR and high-temperature heat release (HTHR) was longer than other cases. And it showed more sharp HTHR shape because this combustion was constructed by only high reactivity fuel, i.e. diesel.

For HRR of dual-fuel PCI based on mode 3, the duration of front combustion (MFB 10-50) was longer than that of diesel PCI condition, while late (MFB 50-90) and main (MFB10-90) combustion duration were shorter in Figure 4.22. It was related to the behavior of gasoline. Since low reactivity fuels were entrained into spray under the dual-fuel combustion, initial ignition became slower than that of neat diesel case. On the other hand, as combustion proceeding, residual gasoline and some of the diesel fuels in the unburned area became auto-ignited due to increasing the in-cylinder pressure and temperature. Therefore, dual-fuel PCI was advantageous to faster late combustion (oxidation). LTHR region was clear and the distance between LTHR and HTHR was shorter than that of diesel PCI. On the other hand, HRR of dual-fuel PCI based on mode 2 was more prolonged and retarded comparing to other combustions because of the high EGR rate, which means richer condition due to low oxygen concentration.

Thus, in Figure 4.23, it seems that dual-fuel PCI based on mode 2 with higher EGR showed lower PRRmax which means smoother combustion. On the other hand, PRRmax from diesel PCI was the highest among three combustions, because all the fuels were high reactivity which causes the rapid increase of the in-cylinder pressure. As a result, dual-fuel PCI is superior to the diesel PCI aspect to engine combustion noise from PRRmax, basically. In addition, using a heavy amount of EGR is favorable for the more reduction of PRRmax.

In Figure 4.24, CO emission was very high under diesel PCI condition. Although the overall equivalent ratio of diesel PCI was the same with dual-fuel PCI based on mode 2, the amount of CO emission was three times more than that of dual-fuel PCI. It means the low reactivity fuel coming from intake mixture is advantageous to making more homogeneous mixture condition. Thus, there was a limitation to make a homogeneous mixture by using the only direct injection method. The levels of THC emission from three combustion concepts were similar to each other as 3,000~4,000 ppm. It can be guessed the wall-wetting effect under diesel PCI was the same level of the merging of the crevice and wall-wetting effects of dual-fuel PCI.

Actually, gIMEP from diesel PCI was lower than other dual-fuel PCI cases, even though the same amount of LHV was supplied. Thus, the gross thermal efficiency of diesel PCI was quite lower than other dual-fuel PCI concepts as 3~3.5 % in Figure 4.25. Especially, the heat loss from diesel PCI was higher than dual-fuel PCI. Also, higher PRRmax could be the effect on the acceleration of heat transfer to the wall and the main combustion duration of diesel PCI was longer than that of other cases. From the result, dual-fuel PCI has the competitiveness to get the higher thermal efficiency, comparing to diesel PCI by the reduction of heat loss.

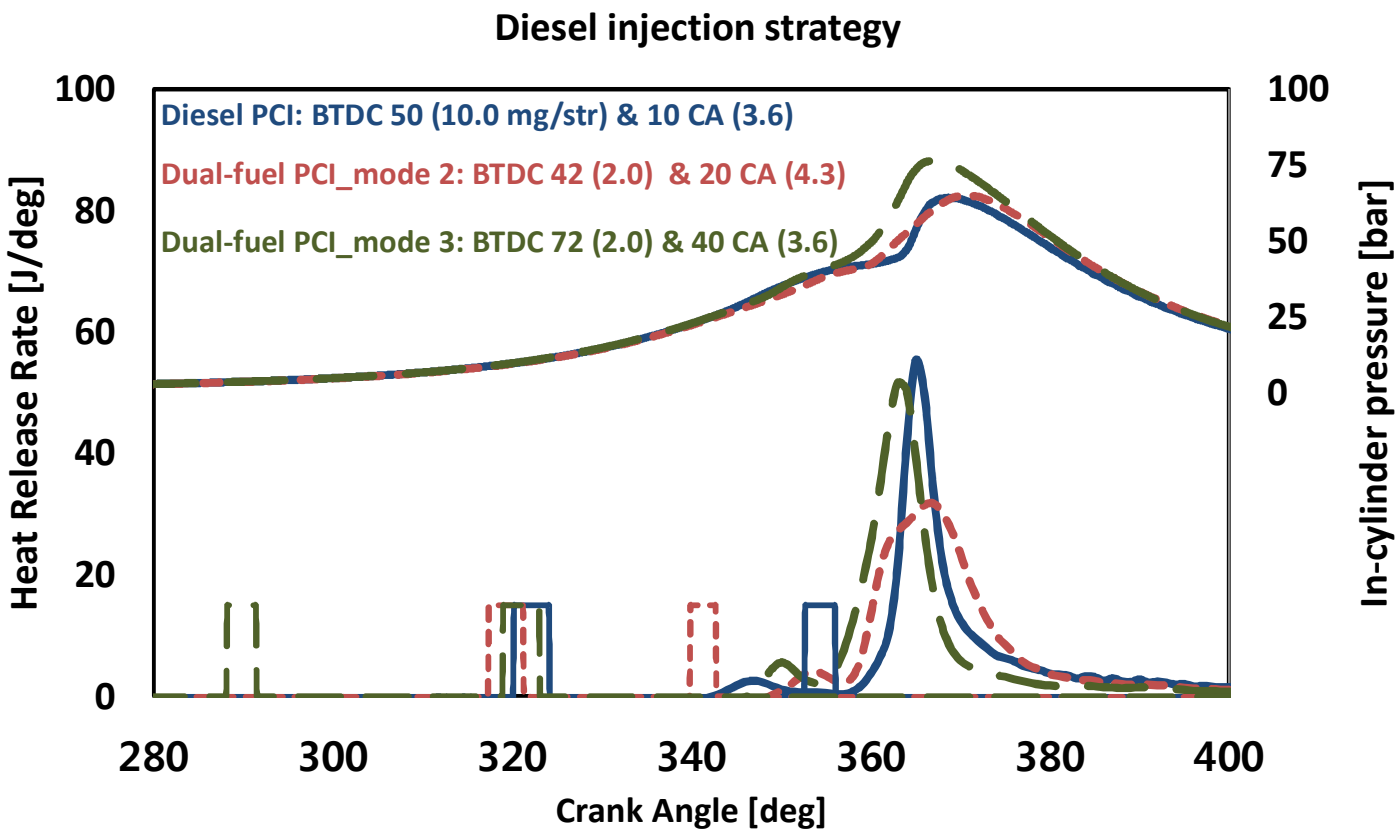


Figure 4.22 Diesel injection signal, heat release rate and in-cylinder pressure of three different combustion conditions

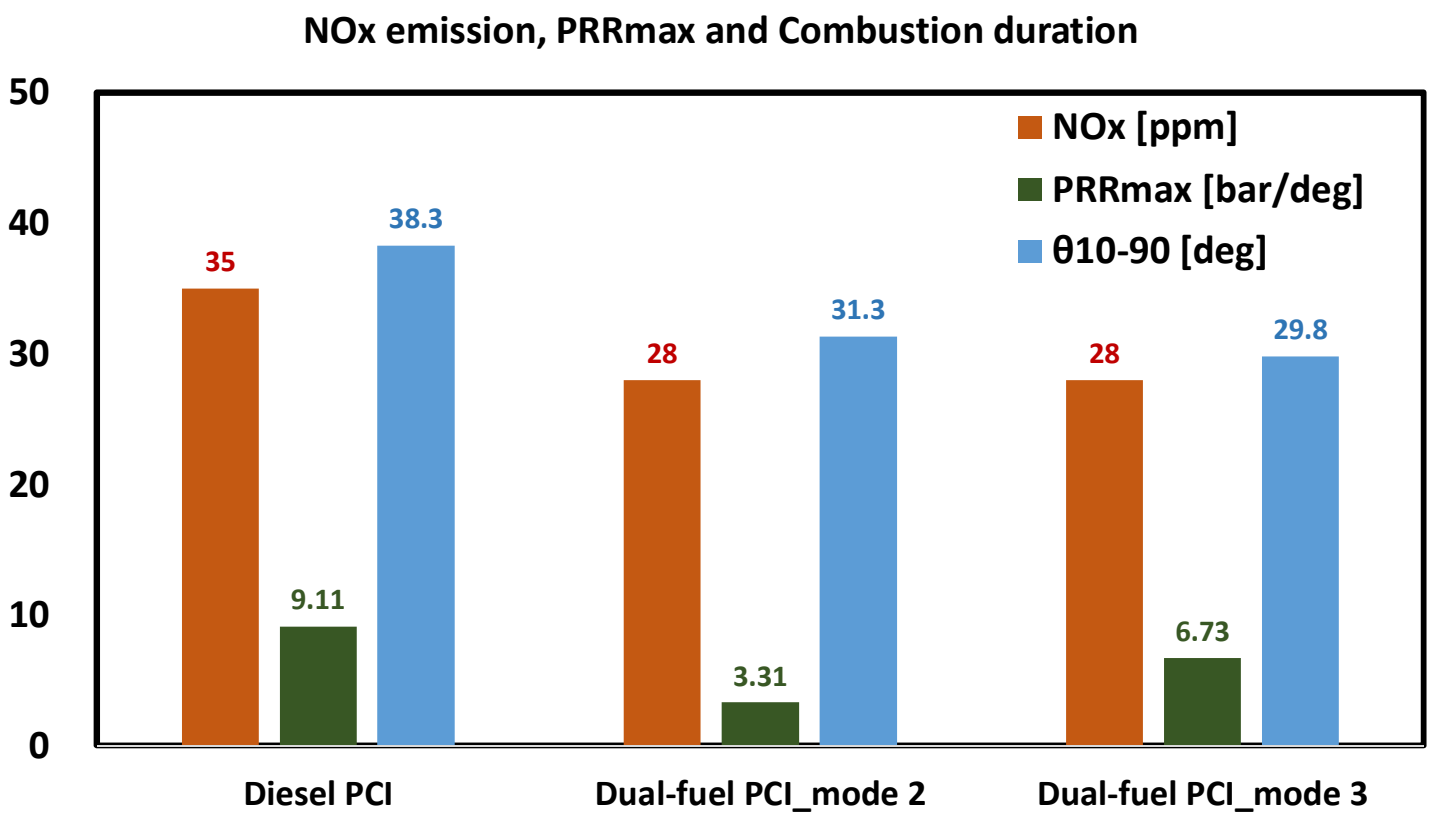


Figure 4.23 NOx emission, PRRmax and main combustion duration (θ10-90) of three different combustion conditions

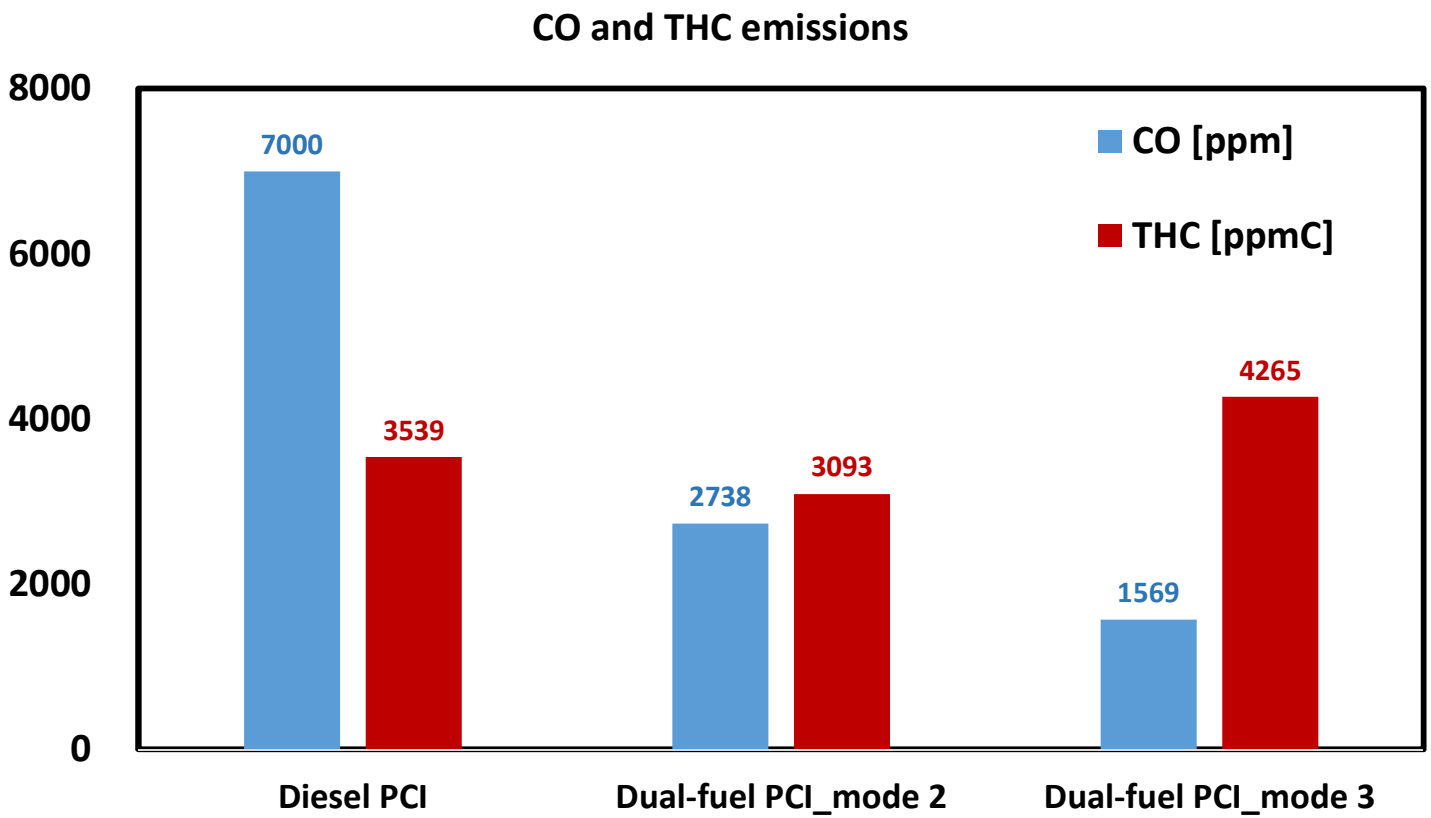


Figure 4.24 CO and THC emissions of three different combustion conditions

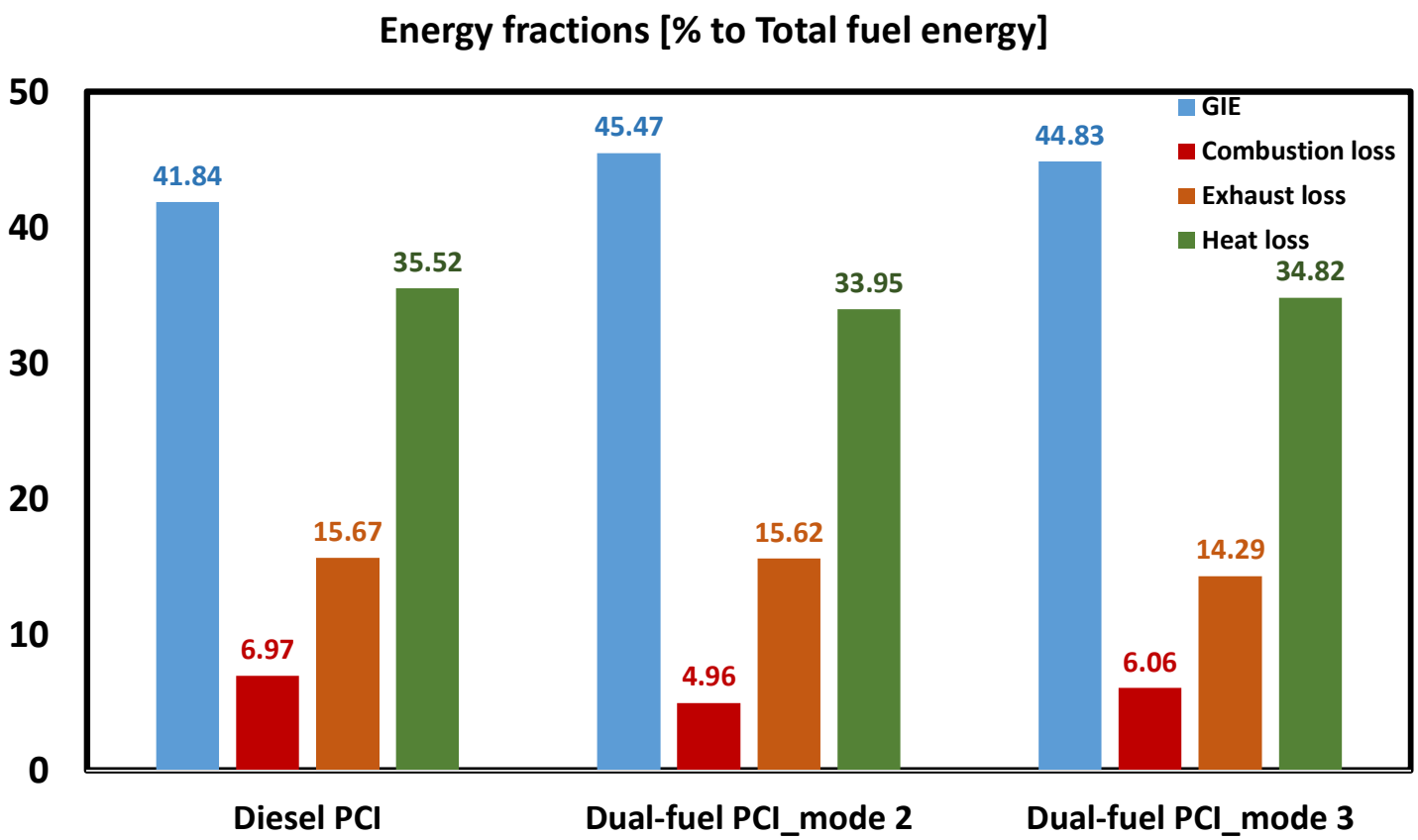


Figure 4.25 Energy budget for diesel PCI and dual-fuel PCI combustions

5. Dual-fuel PCI toward High Thermal Efficiency and Extension of Operating Range

In the last chapter of this research, the investigation for the implementation of dual-fuel PCI under the relatively low load condition was systemically evaluated. Under the low load condition, PRRmax is not a issue, but how to get the higher thermal efficiency is the main concern with low NOx and PM emissions. On the other hand, under the high load condition, since the large portion of gasoline fuel were used, unintended auto-ignition from gasoline-related with PRRmax is mainly investigated [34, 62, 63].

Therefore, in this chapter, the way to achieve higher thermal efficiency under the low load condition and how to reach the highest gIMEP with the dual-fuel PCI are introduced. Before the research, the first of this chapter will introduce the verification of dual-fuel PCI under various operating ranges to evaluate the robustness of this combustion concept. Especially, dual-fuel PCI were adjusted under conventional boost pressure conditions at each operating condition, because the higher boost needs to work from the turbocharger which can influence the fuel consumption in multi-cylinder engines. Therefore, futher practical approach of dual-fuel PCI for the mass production is introduced.

5.1 Verification of robustness of dual-fuel PCI in various operating ranges with low emissions and PRRmax

For the first experiment, the dual-fuel PCI concept was applied to various engine operating conditions. Eight representative points were selected, which were frequently used during the NEDC mode test of a conventional 1.6 L diesel engine. Criteria for emissions, GIE and PRRmax are shown in Table 5.1. Criteria for NOx and PM emissions in this research can satisfy the EURO-6 regulation without any after-treatment systems. In addition, criteria for CO, THC, and PRRmax are extracted from the research of RCCI, Wisconsin ERC [34].

Three main operating parameters were changed to meet the criteria. The EGR rate, the fraction of each fuel and diesel injection timings were adjusted. Especially, two different diesel injection strategies were applied. The first one was a single diesel injection strategy and the second one was a split diesel injection strategy. From the results of different diesel injection strategies, it can be found which diesel injection strategy is favorable under each operating condition.

Table 5.1 Engine operating conditions for the verification of robustness of dual-fuel PCI under various operating conditions

Description	Value
Engine speed [rpm]	1,500/1,750/2,000
gIMEP [bar]	5.3/7.5/9.5/11.5
Coolant & oil temperature [K]	358
Diesel injection pressure [bar]	450
Diesel injection strategy	Single or Split
Limited criteria for the dual-fuel PCI	
gISNOx [g/kWh]	Below 0.21
PM [FSN]	Below 0.1
CO & THC [ppm & ppmC]	Below 5,000
GIE [%]	Near 42 (conventional diesel level)
PRRmax [bar/deg]	Below 10
※ Boost pressures for each case do not exceed the conventional conditions.	

For the four representative cases, all emissions were below the criteria. Combustion phases and in-cylinder pressures of each case are introduced in Figure 5.1~5.4. Usually, MFB 50 was settled near 10 °ATDC for all cases, because the low NOx emission and PRRmax could be achieved by the retardation of combustion phase.

Also, since boost pressures were the same with those of conventional diesel operating conditions, overall-equivalent ratios were relatively richer than the optimized dual-fuel PCI conditions. However, gross thermal efficiency was still acceptable, comparing to those of conventional diesel combustion conditions (Figure 5.5). In other words, the margin of gross thermal efficiency by the dual-fuel PCI concept was changed to low NOx, PM emissions, and PRRmax by using more EGR rate.

For most of cases, the gasoline fraction was increased as a load condition increased for each engine speed in Figure 5.6. Since under lower speed or load conditions there was a problem with the ignitability, the diesel fraction could not be decreased. Also, there was no tendency of the EGR rate, but the higher EGR rate over 40 % was needed to all cases in Figure 5.7. Actually, since there is a limitation of NOx reduction just by an early diesel injection strategy, using heavy EGR was effective as the result of Chapter 4.3 indicated

In Figure 5.8 and 5.9, except for two cases of 2,000 rpm conditions, CO, and THC emissions can be reduced by applying the split diesel injection strategy. Thus, gross thermal efficiency could be increased by the split diesel injection strategy as the amount of recovering combustion losses in Figure 5.5. One of the advantages of the split diesel injection is the improvement of diesel fuel stratification, which means well-distribution, for preventing local rich regions [60, 61]. Also, shortened

diesel spray penetration length is one of beneficial for the reduction of combustion losses.

However, since the diesel fraction was small under higher speed and load conditions, the diesel split injection strategy was more suitable for relatively lower load and speed conditions. Also, due to improved combustion efficiency from the split diesel injection strategy, PRRmax became higher under most of the cases, especially for the high load (Figure 5.10). Therefore, under the relatively low speed and load condition, the split diesel injection strategy was recommendable, while single diesel injection was suitable for higher engine speed and load conditions.

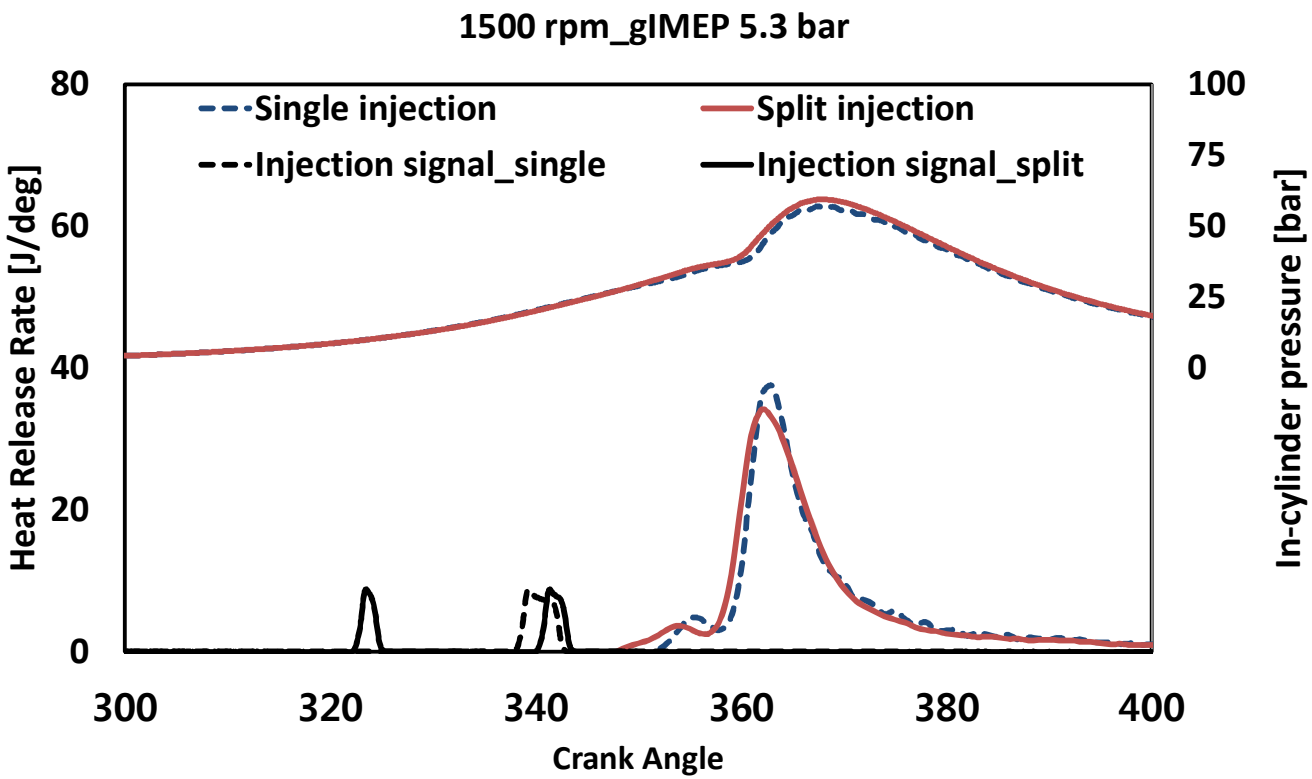


Figure 5.1 In-cylinder pressure and heat release rate graphs as different diesel injection strategy under 1500 rpm/gIMEP 5.3 bar

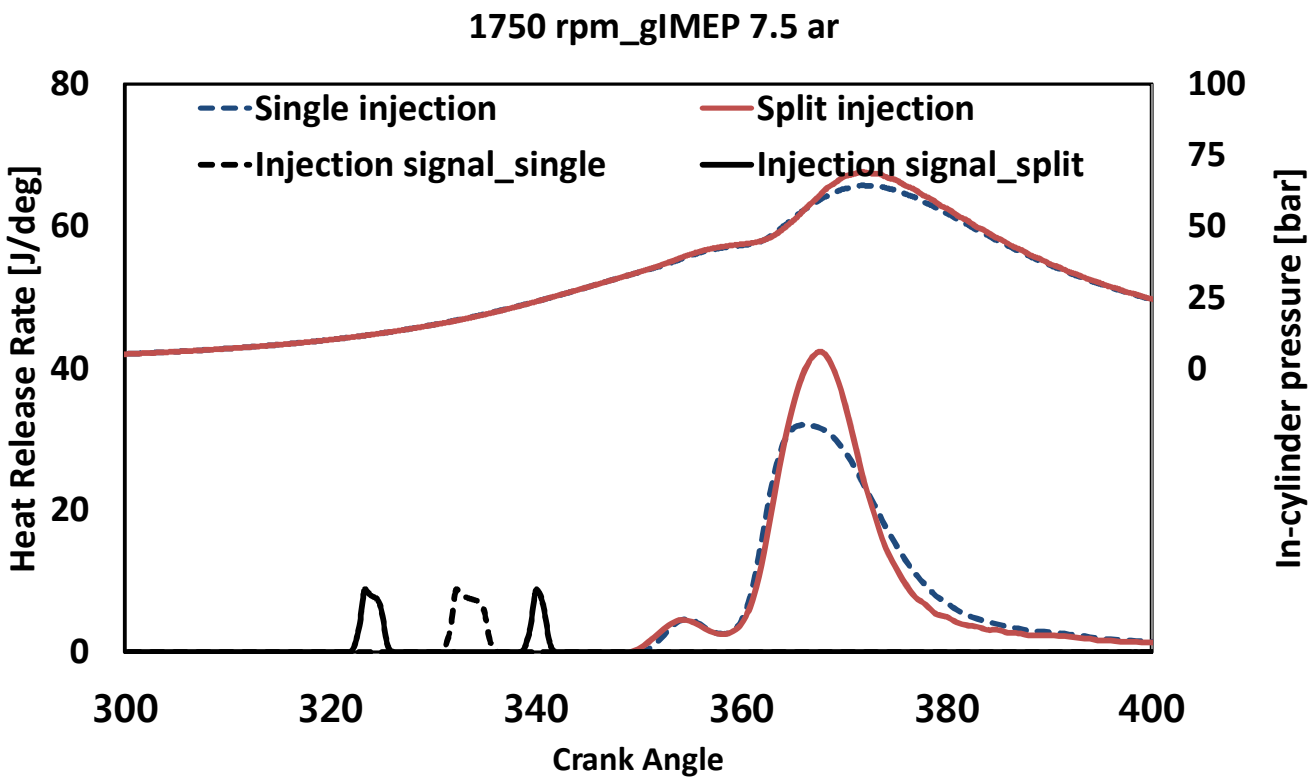


Figure 5.2 In-cylinder pressure and heat release rate graphs as different diesel injection strategy under 1750 rpm/gIMEP 7.5 bar

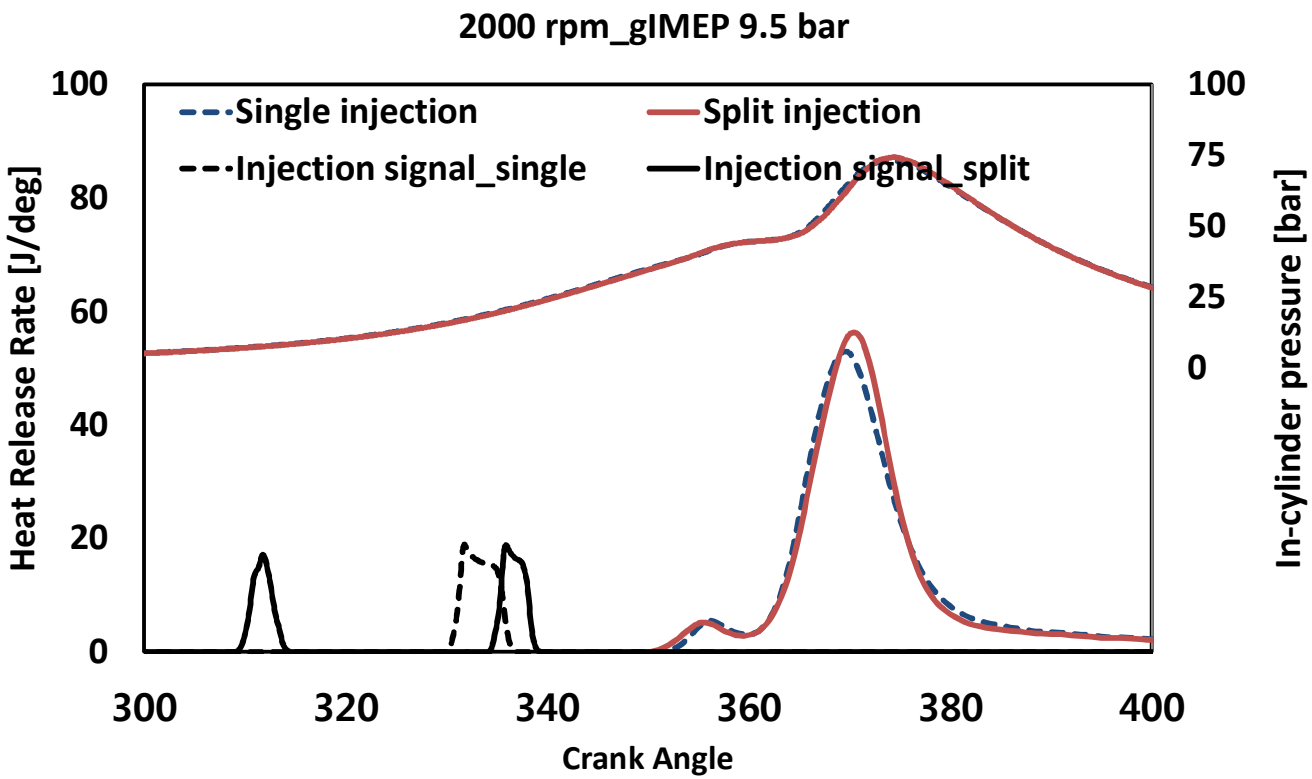


Figure 5.3 In-cylinder pressure and heat release rate graphs as different diesel injection strategy under 2000 rpm/gIMEP 9.5 bar

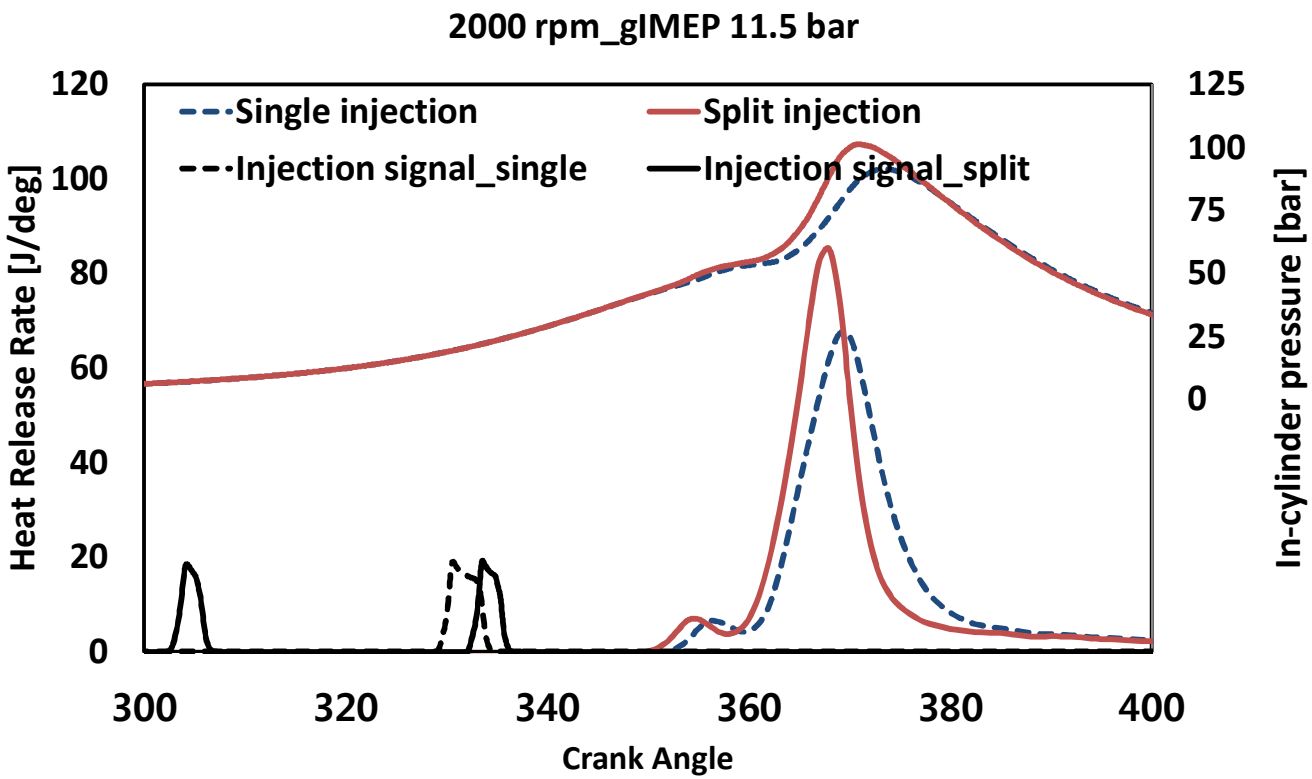


Figure 5.4 In-cylinder pressure and heat release rate graphs as different diesel injection strategy under 2000 rpm/gIMEP 11.5 bar

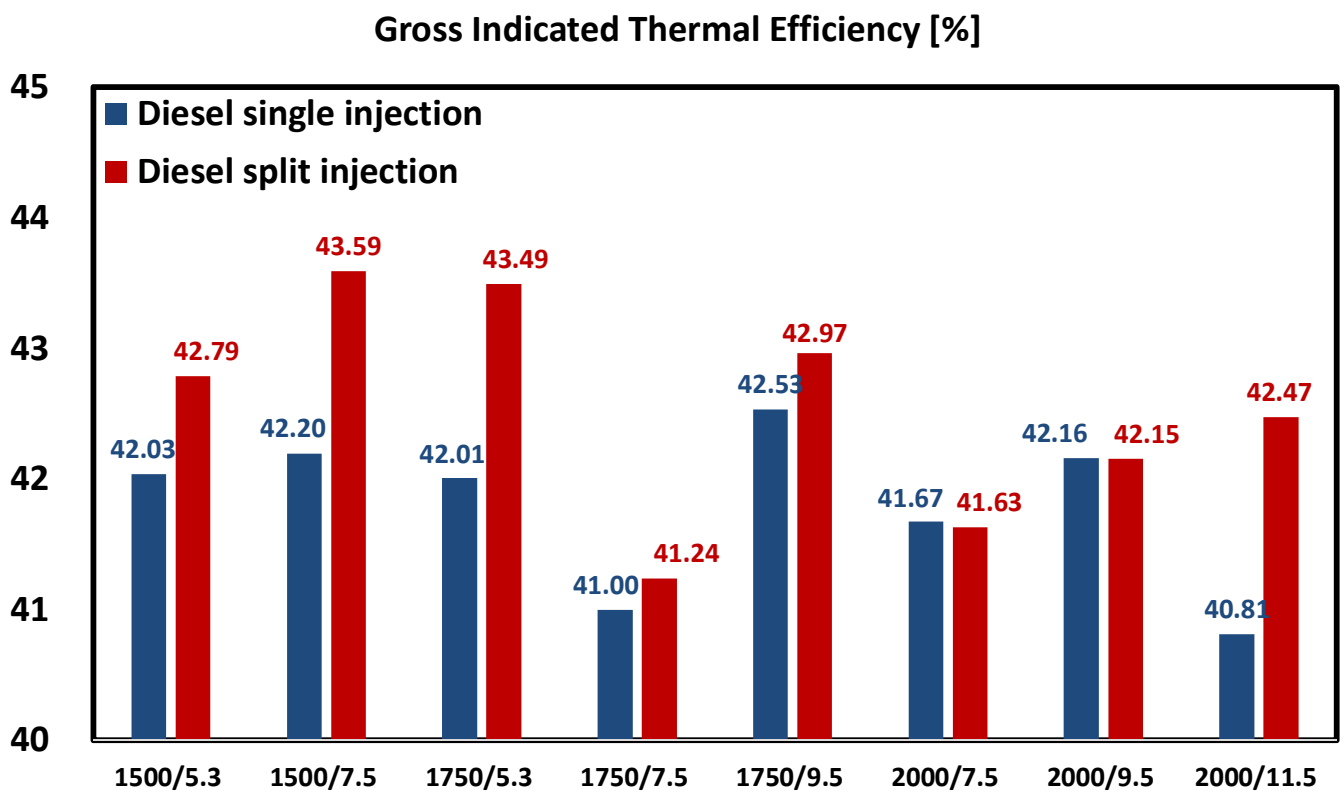


Figure 5.5 Gross indicated thermal efficiency of representative eight operating points as different diesel injection strategy

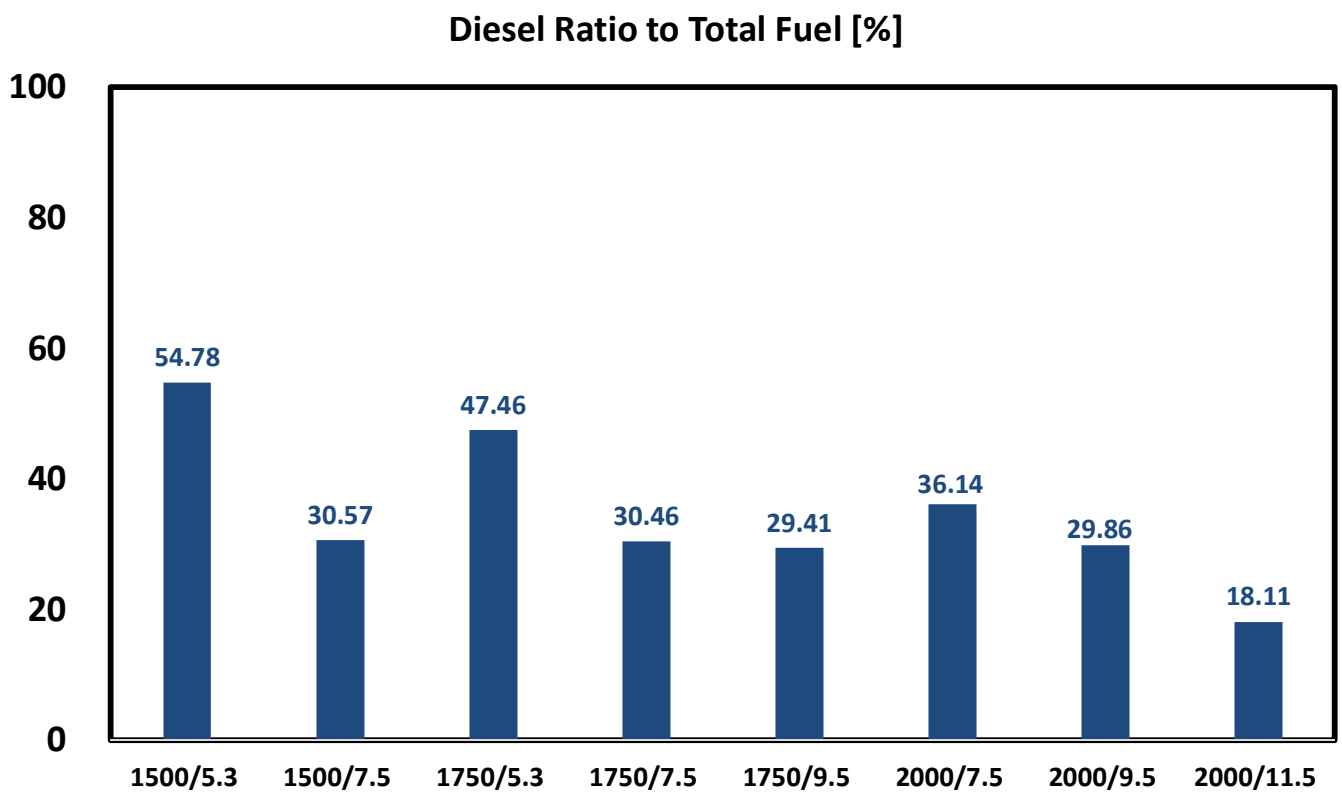


Figure 5.6 Diesel ratio to total fuel amount of representative eight operating points

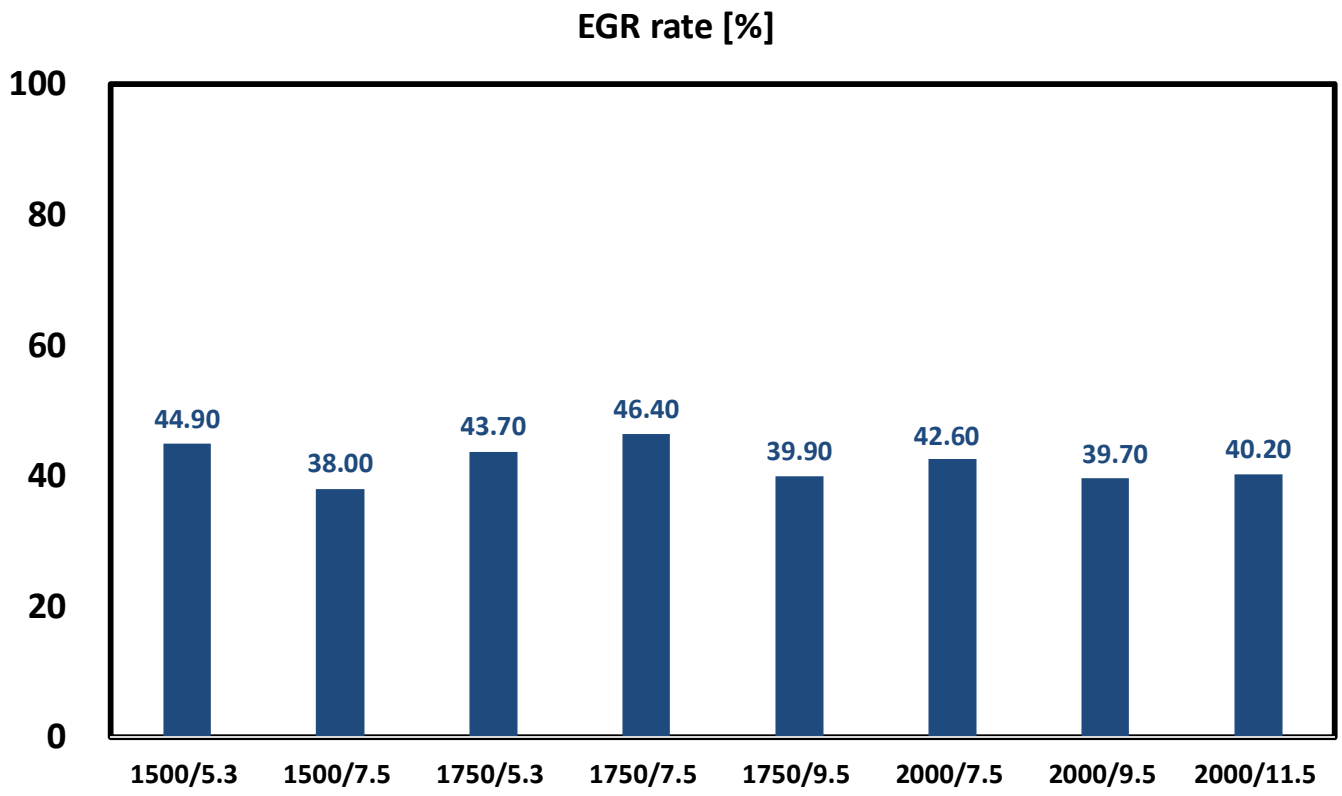


Figure 5.7 EGR rates of representative eight operating points

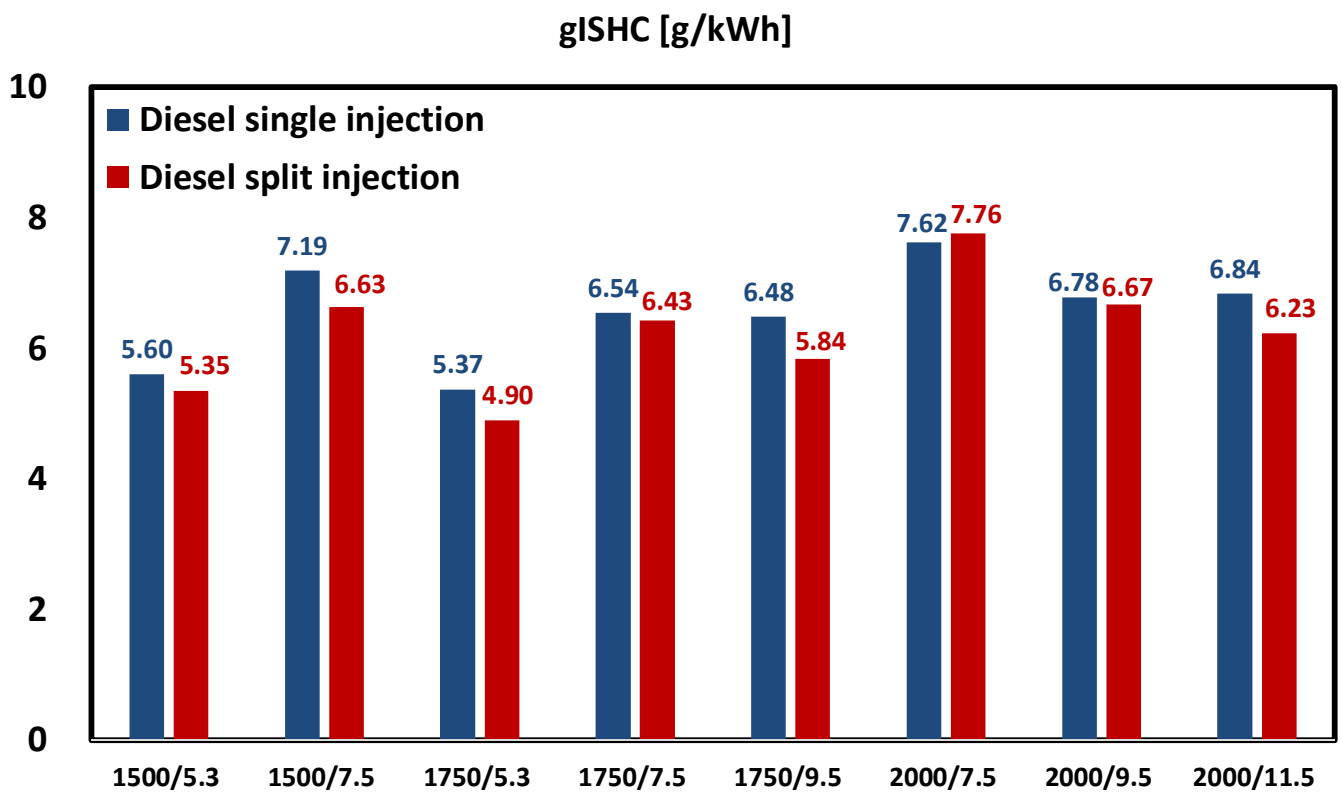


Figure 5.8 Gross indicated specific total hydrocarbon of representative eight operating points as different diesel injection strategy

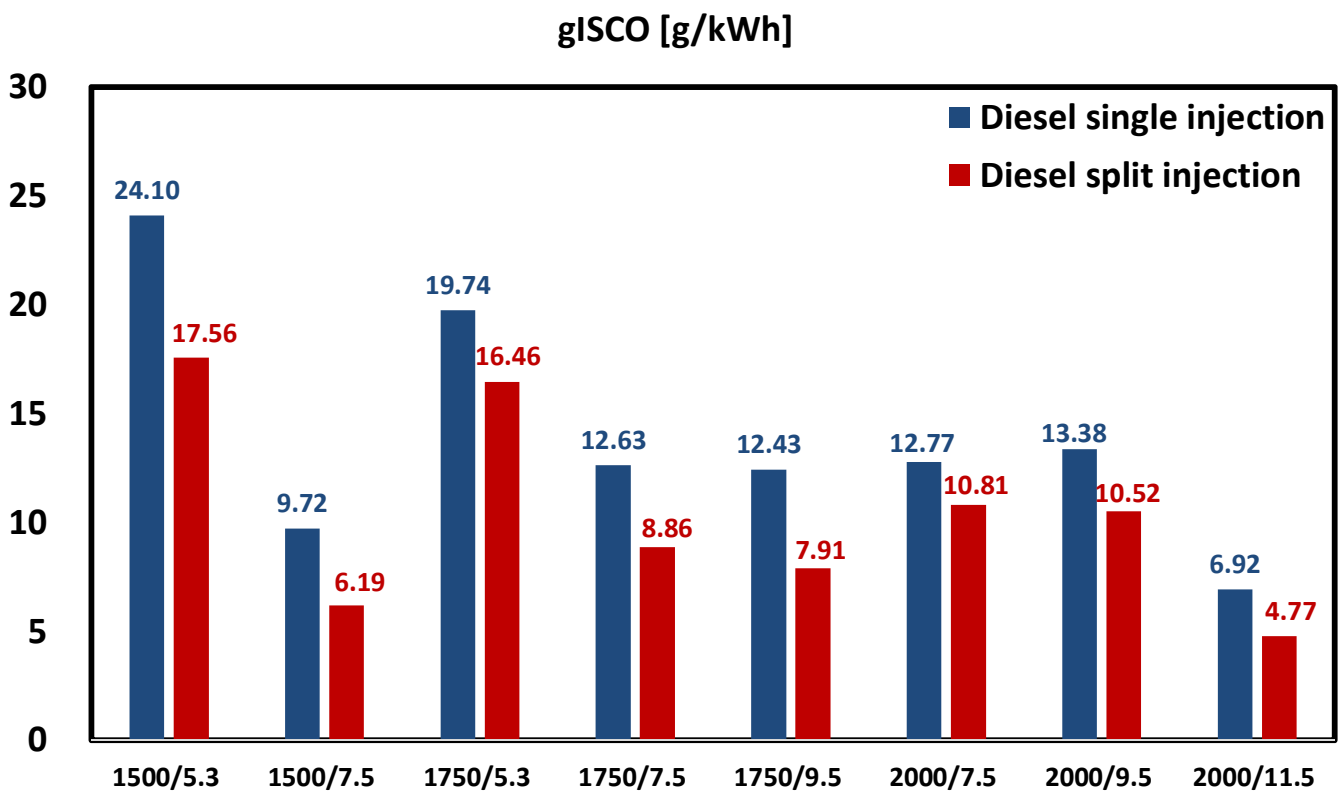


Figure 5.9 Gross indicated specific carbon monoxide of representative eight operating points as different diesel injection strategy

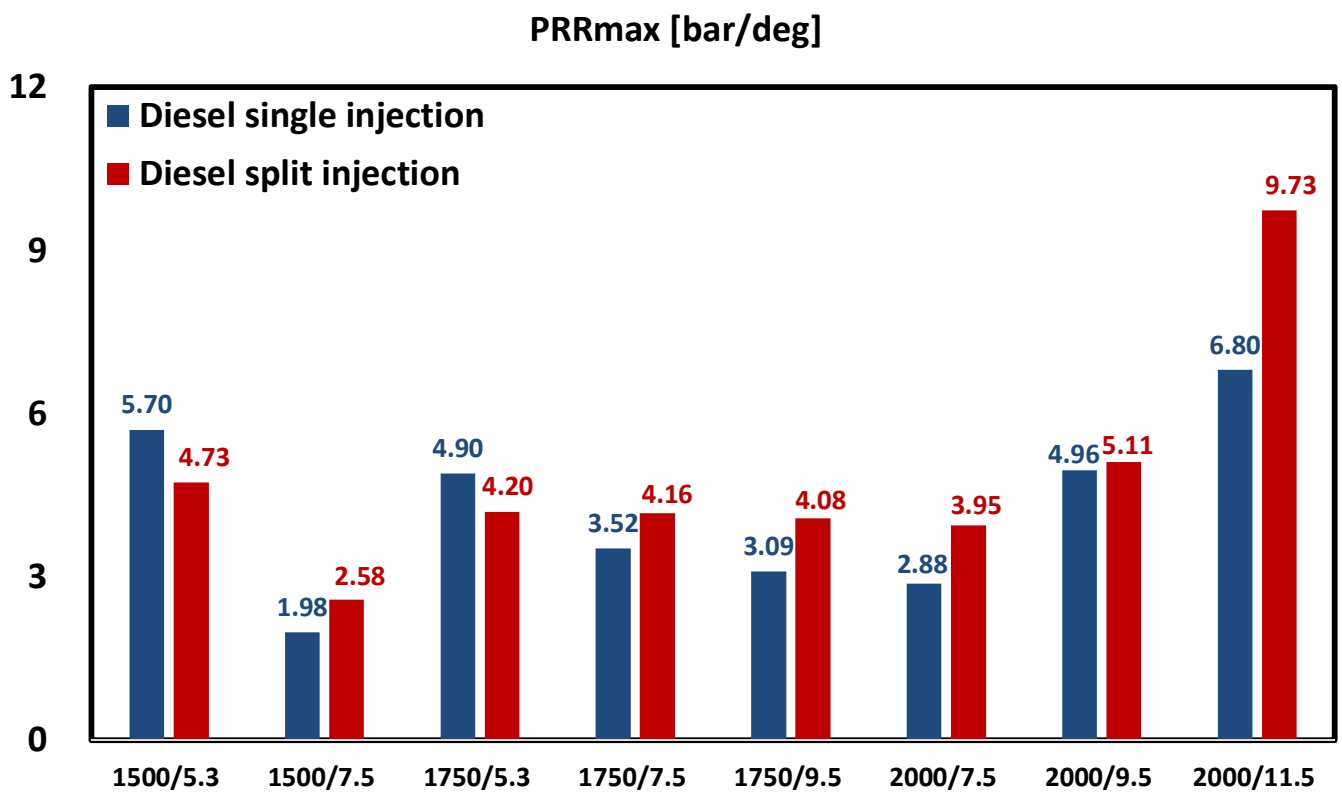


Figure 5.10 The maximum in-cylinder pressure rise rate of representative eight operating points as different diesel injection strategy

5.2 Improvement of dual-fuel PCI under low load condition toward high thermal efficiency

Following the findings from Chapter 5.1., under the relatively low engine speed and low load condition, the diesel split injection strategy was suitable for the improvement of combustion and thermal efficiencies. Thus, in this research, the boost pressure was varied under the lower load dual-fuel PCI condition, based on the diesel split injection strategy. While diesel injection timings were fixed, the fraction of gasoline and EGR rate were adjusted to various boost pressures. The detailed experimental conditions are in Table 5.2.

Table 5.2 Engine operating conditions for the higher thermal efficiency of dual-fuel PCI as increasing boost pressure

Description	Value
Engine speed [rpm]	1,500
gIMEP [bar]	6.5
Total LHV [J/cycle]	580
Coolant & oil temperature [K]	358
Diesel injection pressure [bar]	450
EGR rate [%]	38 for boost 1.1 15 for boost 1.3 0 for boost 1.5
Gasoline fraction [%]	35 for boost 1.1 55 for boost 1.3 67 for boost 1.5
Diesel injection strategy	Split diesel injection
Diesel injection timing	The 1st injection: 60 °BTDC The 2nd injection: 30 °BTDC
※ The amount of the 1st diesel injection was fixed as 2 mg/cycle.	

In Figure 5.11, the peak of HRR became higher and the combustion duration was shortened as increasing a boost pressure. Not only has the effect of increasing the boost pressure, but also the reduction of EGR had the effect on the faster combustion. It could be shown in Figure 5.12 that as the combustion duration became shorter, log scaled P-V curve became similar with ideal Otto cycle which means a higher efficiency cycle.

For the low boost condition, the higher EGR rate is required to achiever lower NOx emission than conventional multi-cylinder diesel engines which satisfies EURO-6 regulation. Thus, there was a lack of oxygen concentration in the intake mixture. As a result, the fraction of diesel should be increased. From the reasons mentioned above, combustion became retarded and the duration of combustion was prolonged and it brought low gross thermal efficiency in Figure 5.13. Especially, the exhaust loss of this condition was much higher than other cases, because rich combustion occurred [59, 60].

Under the medium boost pressure condition (1.3 bars), although the combustion loss was increased compared to boost 1.1 bars condition, the exhaust loss was reduced by 2.5 %. Also, the heat loss was decreased by 1.1 %. Since leaner combustion was implemented which brought the low exhaust temperature and increasing gasoline fraction for the bulky combustion as mentioned in Chapter 4.4, the gain of recovering the heat and exhaust loss caused higher thermal efficiency. This value is even higher than that of conventional diesel combustion, although the compression ratio was lower from 17.3 to 14 (Figure 5.14).

For the highest boost pressure condition (1.5 bars), there was no need to supply EGR to reduce NOx emission and the gasoline fraction could be increased up to

67 %. Then, gross thermal efficiency could be achieved as approximately 48 % from the reduction of the exhaust loss in Figure 5.14.

Therefore, the higher boost pressure for the lean mixture condition and the gasoline fraction without an EGR usage was recommended for the low load condition toward higher thermal efficiency with near-zero emissions in 5.14 [35, 61].

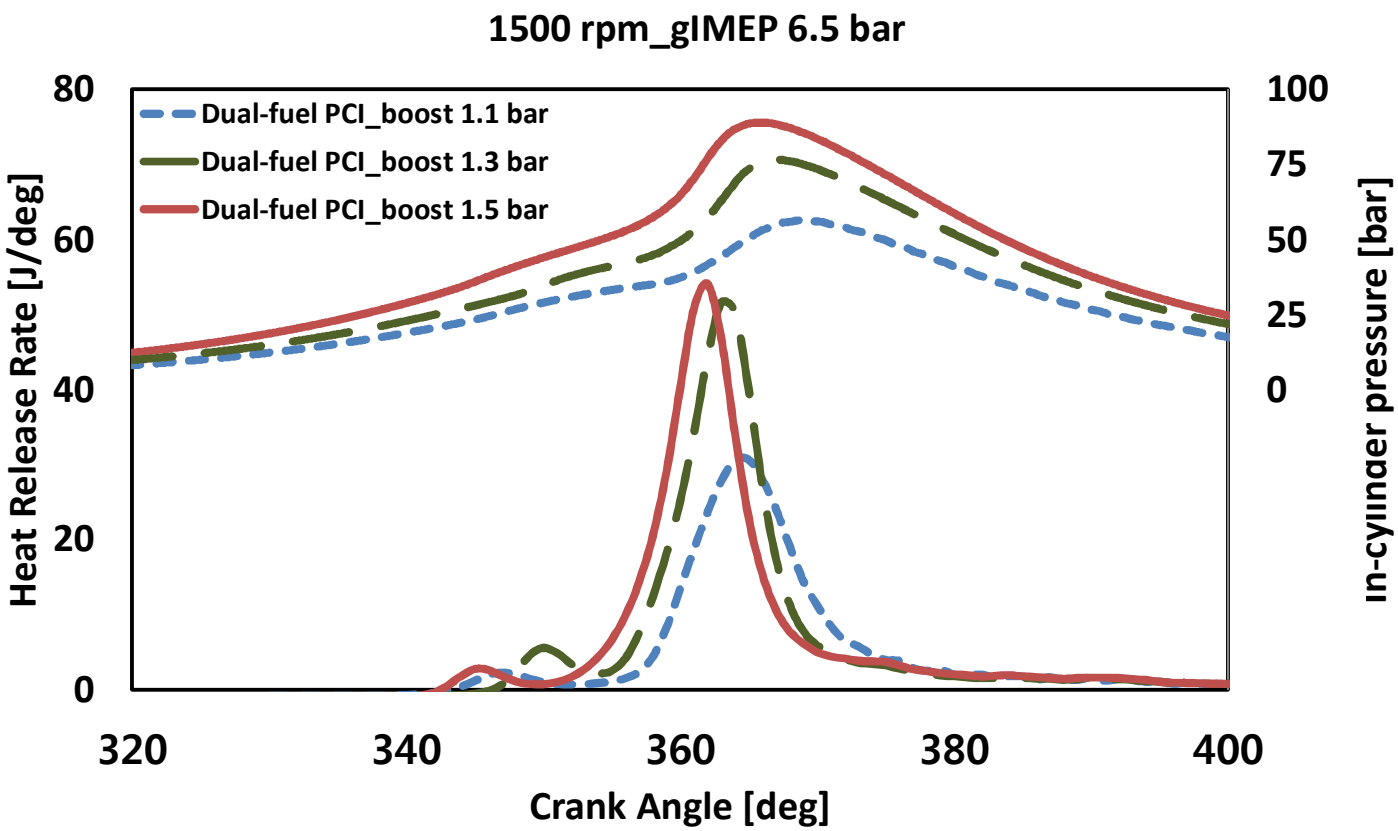


Figure 5.11 Heat release rate and in-cylinder pressure as various boost pressures

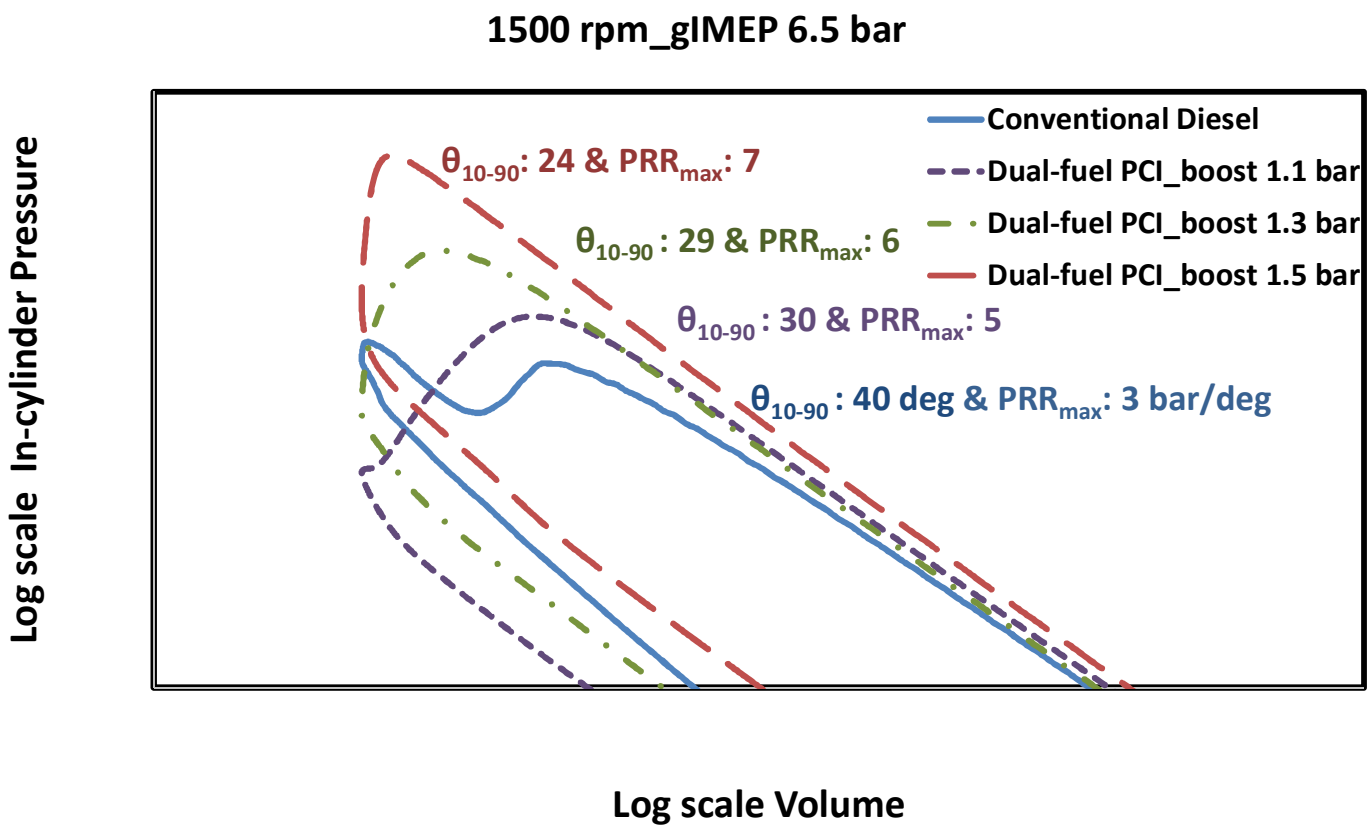


Figure 5.12 Log scaled in-cylinder pressure versus volume graphs as intake pressures

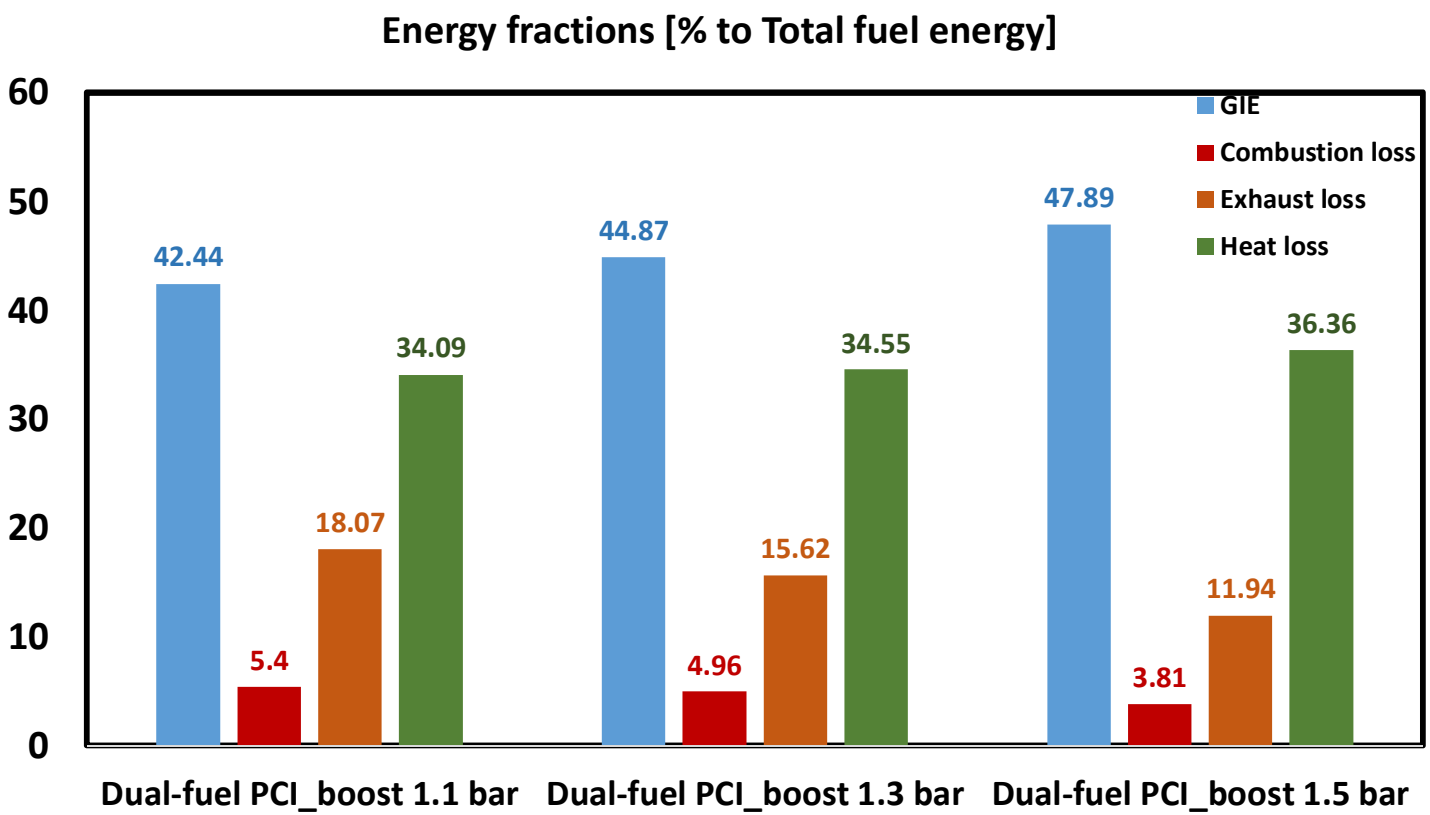


Figure 5.13 Energy budget of dual-fuel PCI as various boost pressures

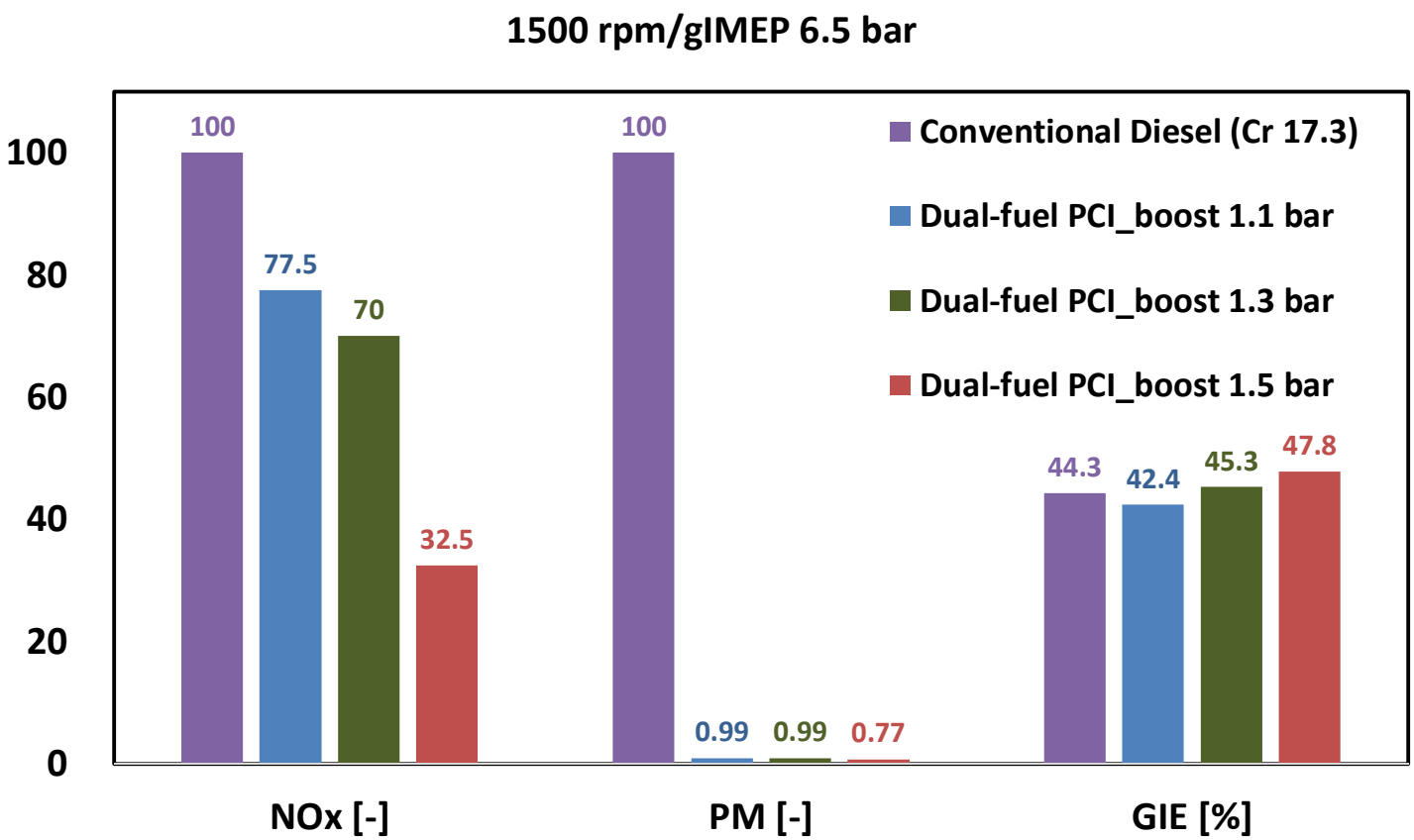


Figure 5.14 Normalized NOx and PM emissions, and gross indicated thermal efficiency as various boost pressures

5.3 Improvement of dual-fuel PCI under high load condition toward extension of operating range

From the results of Chapter 5.1, the investigation of the highest load of dual-fuel PCI with satisfying all the criteria from Chapter 5.1 was evaluated under 1,500 and 2,000 rpm conditions. Experimental conditions and results are introduced in Table 5.3.

For the implementation of the single auto-ignition condition under 2,000 rpm, the valve overlap was controlled as a 'negative valve overlap (NVO)' in Figure 5.15 [55]. Also, the split diesel injection strategy was applied for this condition. The amount of 1st diesel injection was 2 mg/str to avoid wall-wetting from the diesel spray.

In Figure 5.16, unintended combustion occurred from the single auto-ignition condition, which means one of the knocking conditions. PRRmax was increased up to 23 bar/deg in Table 5.3. Since the most of gasoline and diesel fuels were burned as bulky premixed combustion like an HCCI combustion, there was a knocking and a pulsation of in-cylinder pressure.

As a result, the single diesel injection strategy at 22 °BTDC was applied and the valve overlap was controlled as 12 deg. Then, PRRmax was decreased as 6.5 bar/deg and there were two peaks of HRR, which indicates split auto-ignition as mentioned in Chapter 4.1. At the first combustion, 52 % of total heat occurred. It means all of the diesel fuel and some of the gasoline fuel was ignited. Then, at the second combustion phase, residual gasoline fuel which is 48 % of total heat was burned as an auto-ignition. The splitting of the combustion means 'dual auto-

ignition system' and this is important because it suppress the knocking phenomenon under the high load condition.

By using the same way, gIMEP 10.5 bars were achieved under 1,500 rpm in Figure 5.17. In this case, 'dual auto-ignition' was also shown and PRRmax was below 10 bar/deg.

Therefore, dual-fuel PCI concept can be extended up to gIMEP 14 bars which is corresponding to near BMEP 12.5 bars with low NOx, PM emissions and PRRmax maintaining higher gross thermal efficiency in Figure 5.18.

In Figure 5.19, dual-fuel PCI operating points which were verified in this research are depicted and compared to those of conventional diesel engine during the NEDC mode. It can be shown that dual-fuel PCI has the potential to cover the all of the operating range of NEDC mode satisfying criteria.

Table 5.3 Engine operating conditions and results for the higher loads of dual-fuel PCI

Description	Values		
Engine speed [rpm]	1,500	2,000	2,000
gIMEP [bar]	10.48	14.09	13.98
Valve overlap [deg]	12		0 (NVO)
Diesel injection pressure [bar]	420		
Boost pressure [bar]	1.94	2.01	
Overall equivalent ratio	0.76	0.82	
Diesel SOI [°BTDC]	22		24 & 53
EGR rate [%]	51.5	46.9	45.8
Gasoline fraction [%]	70	81	81
NOx [ppm]	33	27	29
PM [FSN]	0.05	0.10	0.14
CO [ppm]	1709	4460	4707
THC [ppmC]	2512	4830	4129
PRRmax [bar/deg]	7.42	6.55	23.39
GIE [%]	45.7	44.0	44.4

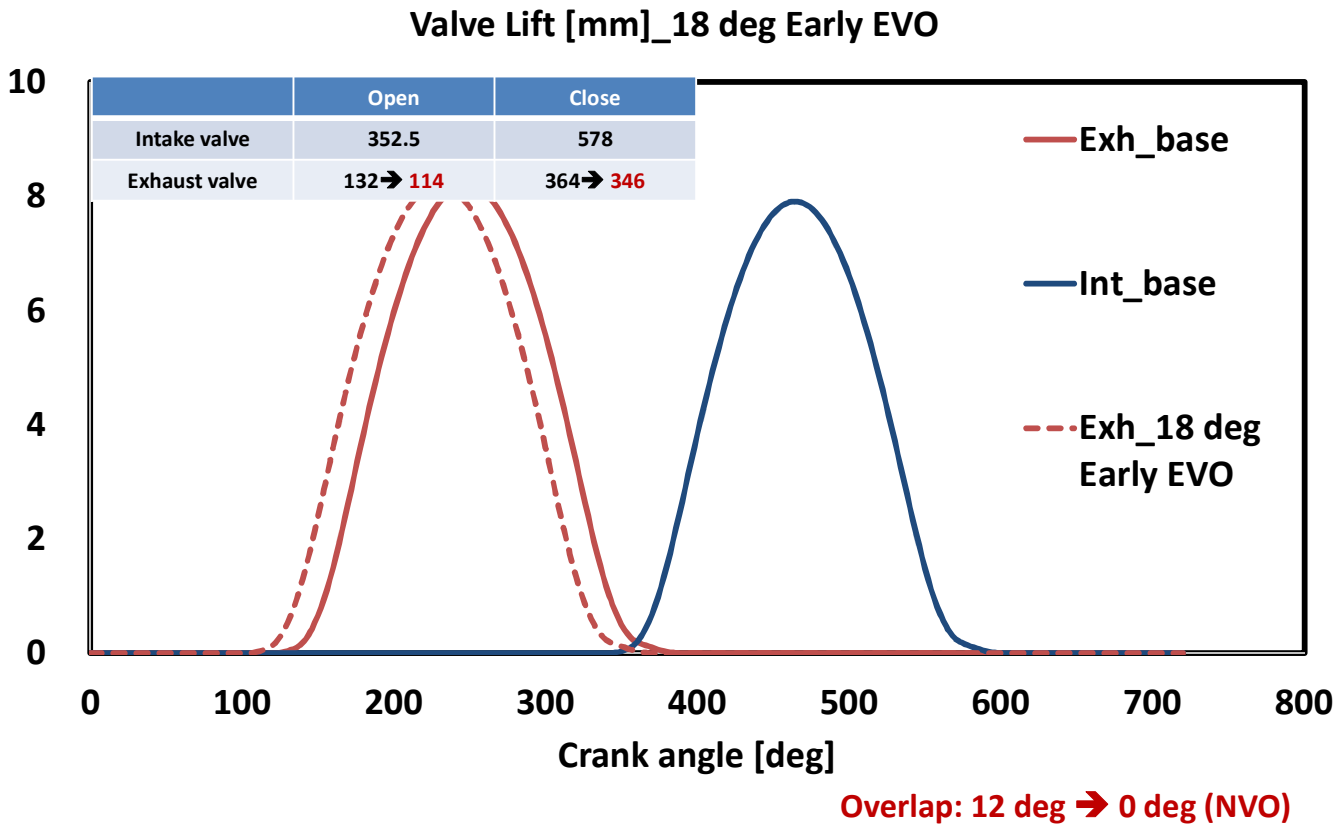


Figure 5.15 Exhaust and intake valve timings under base (dash line) and NVO (dot line) conditions

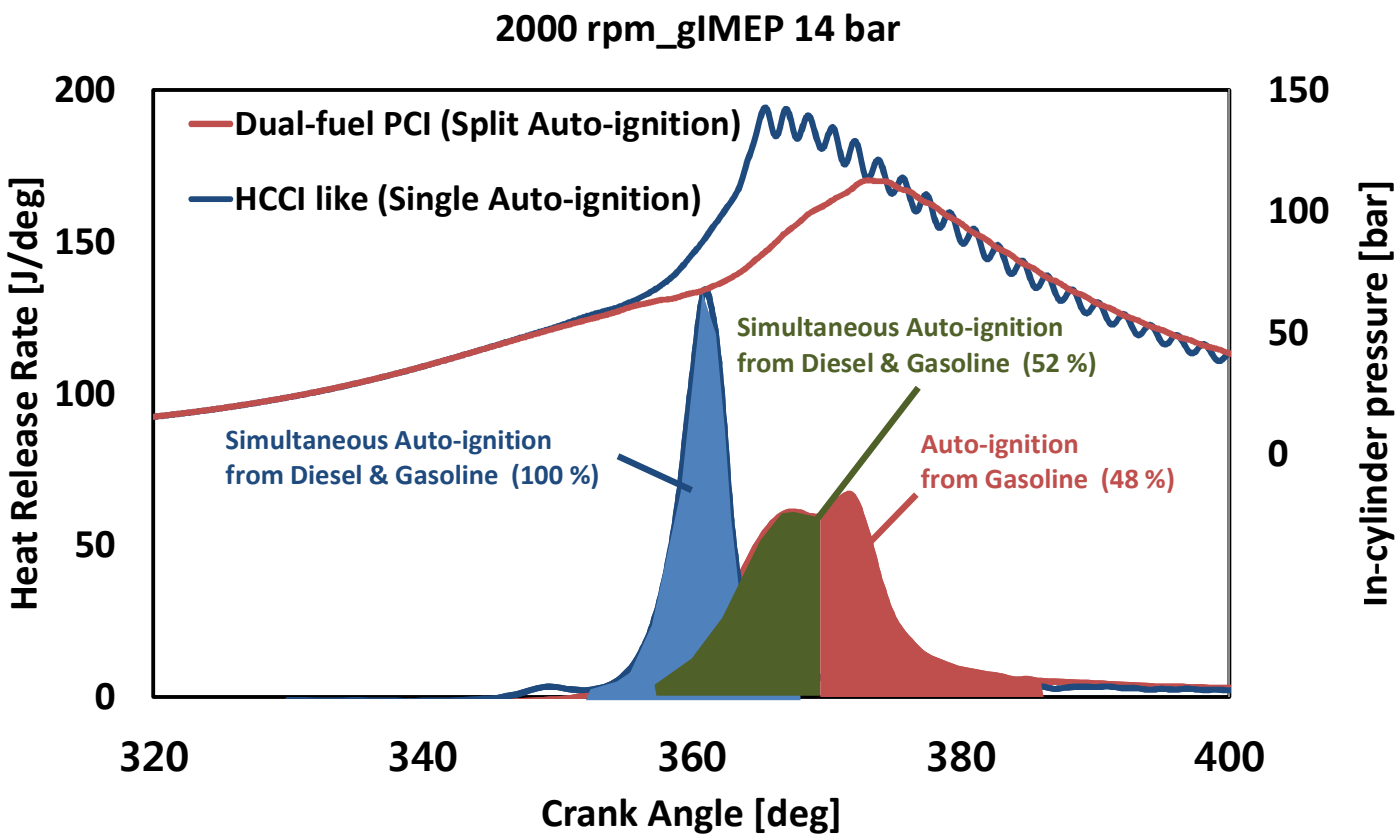


Figure 5.16 Heat release rate and in-cylinder pressures under the 2,000 rpm and gIMEP 14 bar condition

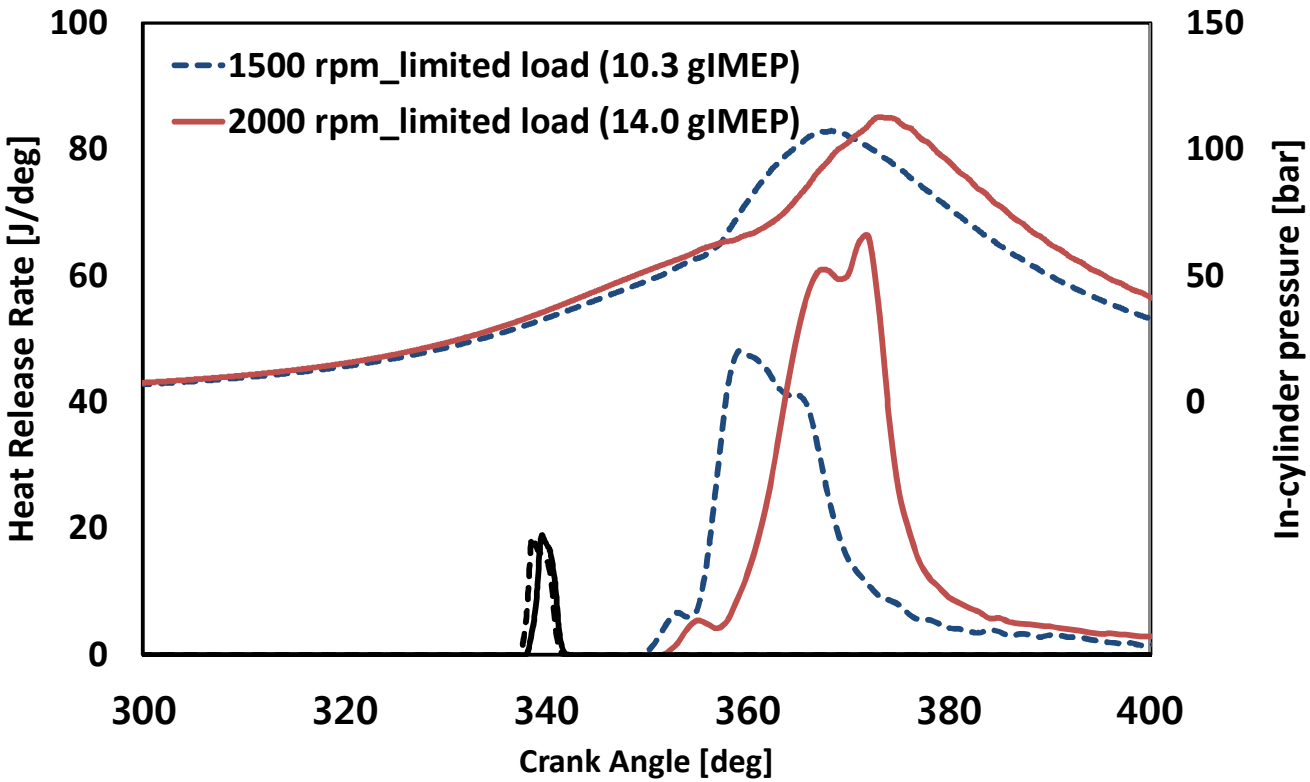


Figure 5.17 Heat release rate and in-cylinder pressures under the maximum load conditions of 1,500 and 2,000 rpm by dual-fuel PCI (Black lines mean the current signals for diesel injections)

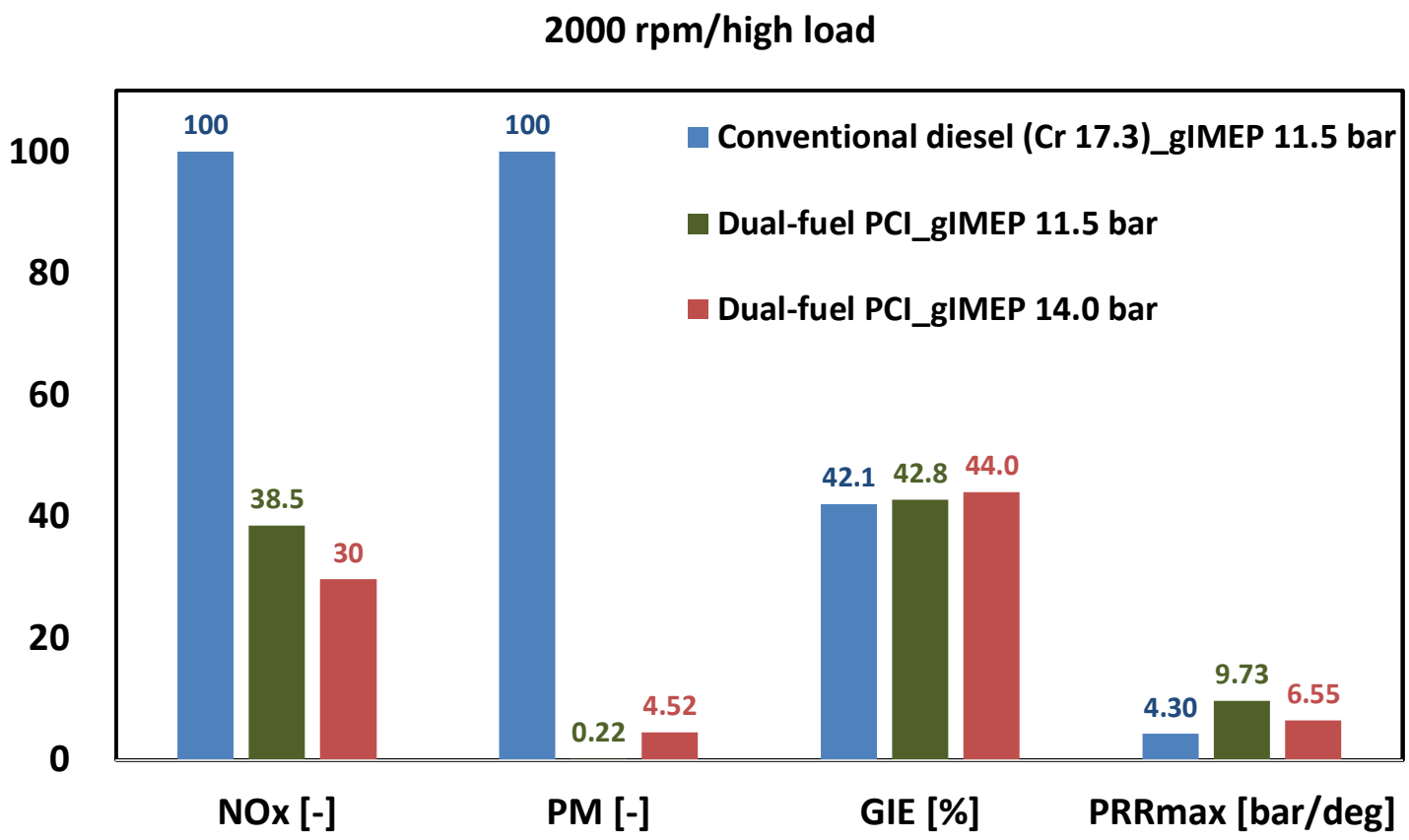


Figure 5.18 Normalized NOx, PM emissions, GIE and PRRmax of each case of 2,000 rpm/high loads

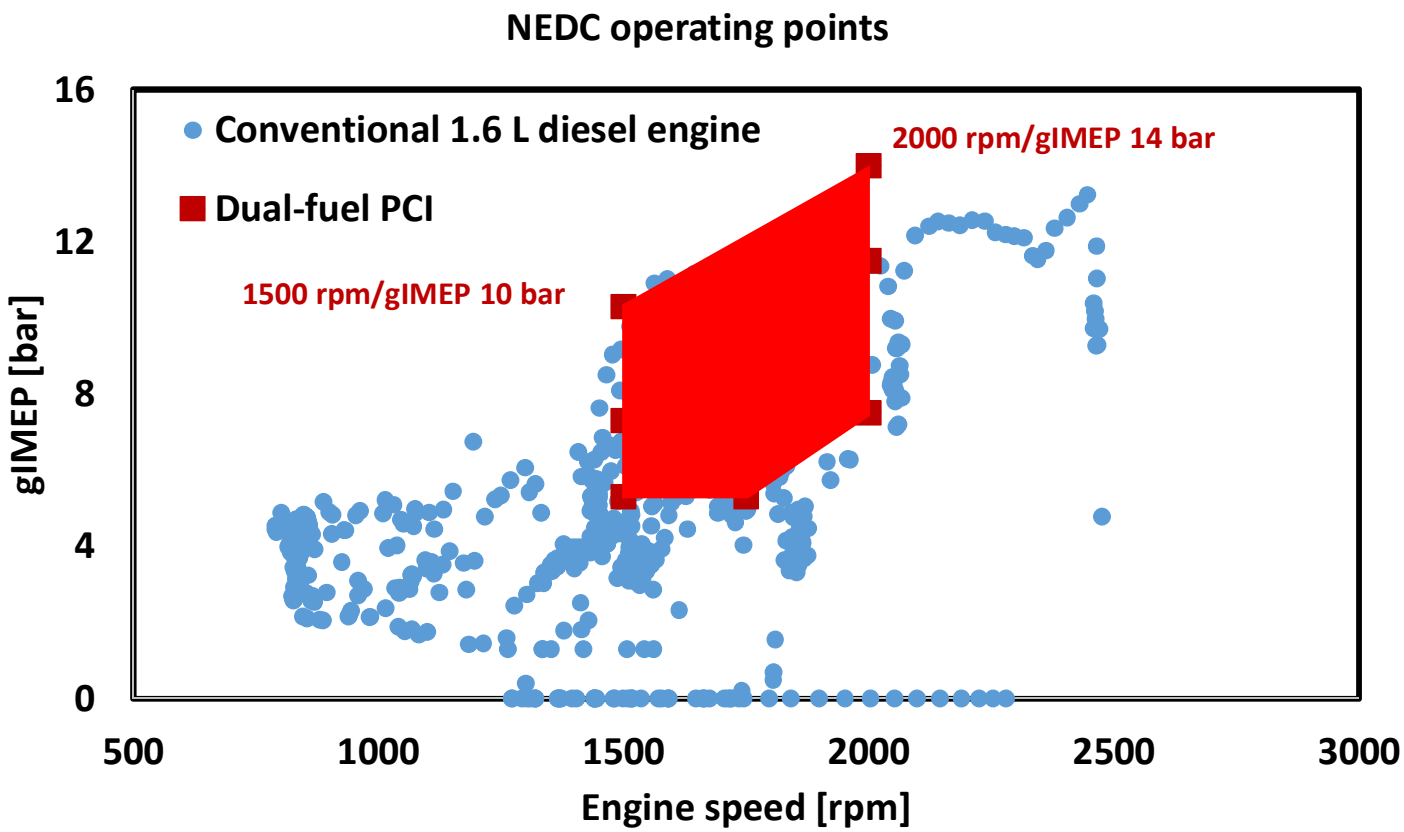


Figure 5.19 NEDC operating points of conventional 1.6 L diesel engine and dual-fuel PCI in this research

6. Conclusions

In this study, the characteristics of dual-fuel combustion under various modes were studied and the investigation into a higher thermal efficiency and the load of dual-fuel PCI was verified. Especially, from this research, the definition of dual-fuel PCI was introduced and the potential to more practical applications of dual-fuel combustion for commercial CI engines can be confirmed. The followings are the summary and conclusion of this research.

There were mainly two parts of this work. The first one was a study on the dual-fuel combustion by using propane and diesel fuels in a single cylinder general research engine. Since there were no exclusive PFI injectors, propane gas was selected as a low reactivity fuel and this gas was supplied as the fumigation. After the basic work on dual-fuel combustion by diesel and propane, dual-fuel combustion researches with diesel and gasoline were performed with an improved single cylinder research engine. Using this engine which was improved by the dual-fuel combustion system, combustion modes of dual-fuel combustion was investigated and the potential to the highest thermal efficiency and load conditions were verified.

The first finding of this research was that dual-fuel combustion modes can be divided into three cases. The first one was dual-fuel combustion which was comprised of diesel premixed combustion and slow late combustion due to residual low reactivity fuels. This combustion mode was usually observed when the reactivity stratification in the cylinder was high and the combustion phase was retarded. Since the combustion phase was retarded, which means the lower combustion temperature, there was no auto-ignition of gasoline and gasoline was burned with a flame propagation. However, this late combustion is not a flame propagation due to the overall lean mixture condition. It can be guessed that the late

combustion was a simple oxidation of low reactivity fuels, i.e. gasoline slowly, because heat transfer with turbulent flow from diesel combustion caused the combustion of low reactivity fuels.

On the other hand, when the reactivity stratification was adjusted and the higher in-cylinder temperature and pressure were comprised, the split auto-ignition occurred, which was verified by two peaks of HRR. The first peak of HRR came from diesel and some of the gasoline fuel which was entrained during diesel spray penetration, but the second peak of HRR came from the simultaneous auto-ignition of residual end gas of gasoline and diesel fuels. Thus, in this combustion mode, the duration of combustion was shortened and PRRmax was reduced. However, gross thermal efficiency from this split auto-ignition dual-fuel combustion mode was usually lower than the third combustion mode, which meant the entire premixed combustion case.

As it was mentioned, the last dual-fuel combustion mode was entirely premixed combustion from two fuels simultaneously. As reactivity stratification of in-cylinder became smooth and gradual, then stratified auto-ignition occurred from diesel and gasoline fuels. It looks like one of HCCI combustions, but this mode is usually smoother than HCCI combustion because of low reactivity fuel. Although this combustion mode showed low NOx and PM emissions with higher thermal efficiency, higher PRRmax was observed because of simultaneous auto-ignition of two fuels.

In this work, the second and third dual-fuel combustion modes were selected to achieve a higher thermal efficiency with low NOx and PM emissions. Since the ignition delay from these two combustion modes were sufficiently longer than the diesel injection period, these dual-fuel combustion modes which are based on the

earlier diesel injection strategy and a large amount of low reactivity fuel are called as ‘dual-fuel PCI’.

The second work was evaluating the relation between combustion index and dual-fuel combustion modes. As varying the total equivalence ratio, gasoline fraction, and diesel injection timings, LTHR (Low-Temperature Heat Release) region occurred from the third mode (single auto-ignition) and the tendency of MFB 50 point became the opposite behavior of the diesel injection timing. Therefore, although the second and third modes were based on the PCI region, the second mode was still related to diesel injection timing (spray motion) which meant ‘late injection PCI’ and the third mode was seemed as an ‘early injection PCI’ which is based on the chemical reaction rather than combustion from spray.

Additional finding was the source of THC emission under dual-fuel combustion. Similar with the other advanced diesel combustion concepts, wall-wetting from the early diesel injection strategy which is earlier than 40 °BTDC was one of the sources for THC emission, especially under higher load conditions. Thus, THC emission was rather reduced as the gasoline fraction was increased in this case.

However, except for the higher load condition, usually the crevice effect was dominant for THC emission. In reality, since the diesel spray penetration length became shortened as increasing the low reactivity fuel, i.e. gasoline, fraction, the problem with the wall-wetting effect was diminished. Also, the overall equivalent ratio has a strong influence on THC emission. One of the reasons is that there were locally over-lean pockets due to the low overall equivalent ratio.

Another reason is that there was not the sufficient in-cylinder temperature to oxidize THC emission. Therefore, from the results, too much leaner equivalent ratio

condition is not suitable for the dual-fuel combustion. Also, there might be a criteria for the maximum fraction of gasoline and diesel injection timings.

In addition, in this research, the split diesel injection strategy can be suitable for an improvement of combustion efficiency in dual-fuel combustion concepts. Using the split diesel injection strategy, the distribution of diesel, which has a major role of ignition source, improved and wall-wetting can be reduced by shortening the diesel spray length. However, under the high load condition, the split diesel injection strategy was not recommended because of the higher PRRmax.

The last experimental achievement of this research was the optimization of dual-fuel PCI under low and high loads, respectively. Under the low load condition, since PRRmax was low enough to satisfy the criteria (especially, 10 bar/deg), the third dual-fuel combustion mode which means simultaneous premixed combustion of diesel and gasoline fuels was favorable. Thus, a higher gasoline fraction, the early and split diesel injection strategy and increasing boost pressure are effective methods toward higher thermal efficiency. In this research, 48 % of the gross indicated thermal efficiency was achieved under 1,500 rpm/gIMEP 6.5 bar condition with low NOx (13 ppm) and near-zero PM emissions (Below 0.05 FSN).

On the other hand, under the higher load condition, PRRmax which is related with a knocking from gasoline fuel is the main challenge. Thus, the second dual-fuel combustion mode which was the split auto-ignition system was applied to reach the highest loads. Although the first dual-fuel combustion mode which was constructed with diesel premixed combustion and the late combustion of gasoline also prevented the knocking phenomenon due to the slow burn rate, gross thermal efficiency decreased due to the prolonged combustion duration. Using the split auto-ignition combustion strategy, gIMEP 14 bar was successfully achieved without the knocking

phenomenon under the 2,000 rpm condition. This maximum value implied that there is a potential of dual-fuel PCI to cover the entire region of NEDC test mode.

This research includes the study of dual fuel combustion characteristics under different modes and the investigation of improved operating strategies of dual-fuel PCI. From the results of combustion characteristics, appropriate combustion modes were suggested as load conditions to achieve low NOx and PM emissions, higher thermal efficiency and especially the highest operating loads. Thus, this research can contribute to the practical application of dual-fuel combustion in passenger cars with light-duty diesel engines.

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국 문 초 록

압축착화 방식(CI, compression ignition)을 이용하는 디젤 엔진은 전기점화 방식(SI, spark ignition)을 채택하는 가솔린 엔진에 비해 우수한 내구성과 높은 열효율로 인해 승용차량분야에 있어서 적용의 폭을 넓혀가고 있다. 그러나 일반 디젤 연소의 경우 디젤 연료의 분사 시점과 연소 시작 시점 간의 간격이 짧아 연소실 내 국부적으로 농후한 지역을 유발하여 다량의 입자상 물질(PM, particulate matter)의 발생을 초래하며, 연료의 자발화 특성에 기인하므로 가솔린 전기점화 방식에 비해 높은 압력상승률로 인한 소음 문제를 지니고 있다. 또한 이론 공연비 운전으로 삼원촉매장치(TWC, three way catalyst)를 통해 질소산화물(NOx, nitrogen oxides) 및 일산화탄소(CO, carbon monoxide), 미연탄화수소(THC, total hydrocarbon)을 동시에 저감하는 가솔린 엔진에 비해 희박연소를 기반으로 하는 디젤 엔진은 별도의 산화 및 환원 후처리 장치를 필요로 한다.

따라서 연소 자체에서 발생하는 유해 배기물질의 저감을 위해 높은 열효율과 저배기를 만족하는 이상적인 연소인 균일 혼합 압축착화(HCCI, homogeneous charge compression ignition)를 목표로 하는 예혼합 압축착화(PCI, premixed compression ignition) 연소를 위시한 디젤 신연소에 대한 연구가 진행되어 왔다. 대부분의 디젤 신연소 개념들은 다량의 배기재순환(EGR, exhaust gas recirculation)을 이용하거나 디젤 조기분사를 통한 예혼합기 형성에 초점을 두었으므로 연소 자체에서 발생하는 배기저감은 효과적이었으나, 높은 압력상승률의 문제 및 저온 연소(LTC, low temperature combustion)로 인한 연료소모율 악화 문제를 해결하지 못하였다. 특히 예혼합 기간의 확보를 위해 상대적으로

고속/고부하 운전조건에서는 구현하기 어려운 운전영역 상의 문제점도 한계로 지적되어 왔다.

이에 최근 대두되고 있는 반응성 조정 압축착화 연소(RCCI, reactivity controlled compression ignition)의 경우 이종 연료를 이용한 예혼합 압축착화 연소를 구현함으로써 기존의 타 신연소들에 비해 상대적으로 높은 부하의 운전조건에서도 연소를 이룰 수 있으며, 도시 열효율 (GIE, gross indicated thermal efficiency) 역시 50 % 가까이 향상 시킬 수 있음을 제안하였다. 특히 RCCI 연소는 전체적인 예혼합 연소로 인해 짧은 연소 기간을 구현하여 연소실 측의 열손실을 저감하여 열효율을 향상시키는 목적이 있으나, 짧은 연소로 인해 이에 따른 급격한 연소실 내 최대 압력상승률(Max PRR, the maximum pressure rise rate)이 동반되는 문제점을 안고 있다.

따라서 본 연구에서는 압축착화 엔진에서 이종 연료를 적용한 연소 시, 열발생률(HRR, heat release rate)을 바탕으로 연소를 구성하는 방식에 대해 분석 및 분류하고, 운전조건에 따라 적합한 이종 연료 연소 방식에 대해 제안하였다.

첫째로, 이종 연료 연소는 열발생률의 형태에 따라 세 가지 모드로 구분이 가능하다. 첫번째 연소모드는 디젤 예혼합 연소와 느린 후연소로 이루어진 모드이다. 디젤 분사시기가 상사점에 가까운 경우 저반응성 연료에 의해 연소상이 팽창과정 중으로 지연되기 때문에, 점차 낮아지는 연소온도로 인해 예혼합된 디젤 연료와 일부 가솔린 연료의 연소 직후, 남은 가솔린 연료들의 잔류 연소가 이어진다. 이 때 가솔린 연소는 해당 연료의 자발화 연소라기 보다는 디젤 연소에 의한 열전달과 난류유동의 영향을 받은 것으로 보이며, 이 경우 연소 기간이 길어지기 때문에 벽면으로의 열전달이 증가하고, 효과적인 최대 연소 압력을 얻기 어렵기 때문에 효율 및 출력 면에서 문제를 안고 있다.

두번째 연소모드는 열발생률의 형태가 두 봉우리로 분리되어 자발화 연소 (Auto-ignition)를 이루는 모드이다. 첫번째 연소와 같이 시작은 디젤 연료와 일부 가솔린 연료의 예혼합 연소로 진행되나, 이후 선행 연소로 인해 상승한 연소실 내부의 온도와 압력으로 인해 잔류 연료들의 자발화 현상이 발생한다. 이와 같은 분할 자발화 연소를 통해 첫번째 연소 모드에 비해 짧은 연소 기간과 효율적인 연소가 가능하여, 상대적으로 높은 열효율과 출력을 확보할 수 있다.

마지막 세번째 연소모드는 디젤 연료와 가솔린 연료 모두 동시에 예혼합 압축착화를 이루는 모드이다. 이는 일종의 균일 혼합 압축착화와 유사한 조건으로 일반적인 반응성 조정 압축착화 연소의 지향하는 바와 같다. 이를 통해 낮은 질소산화물과 입자상 물질 배출 수준은 물론, 높은 열효율도 얻을 수 있으나 높은 연소실 내 압력상승률이 문제 시 될 수 있다. 이에 본 연구에서는 두번째 및 세번째 연소모드를 이용하여 이종 연료 연소의 최적화 및 개선을 이루었으며, 다량의 가솔린 연료 이용과 디젤 조기 분사에 기반한 이종 연료 연소를 특별히 '이종 연료 예혼합 연소 (dual-fuel PCI)'라 칭하였다.

두번째 연구는 이종 연료 연소 모드와 연소 지표간의 상관관계에 관한 연구이다. 두번째와 세번째 연소모드는 점화지연기간이 디젤 분사기간 보다 길다는 공통점이 있기에 예혼합 연소인 공통점이 존재하지만, 세번째 연소 모드에서만 저온연소반응영역 (low temperature heat release)가 존재하였으며, 디젤 분사시기와 연소상 (MFB50)간의 상반된 거동이 존재하였다. 이에 따라 두번째 연소 모드는 디젤 분문에 의해 영향을 받는 '지각 분사 예혼합 연소(Late injection PCI combustion)'으로 볼 수 있으며, 세번째 연소 모드는 연료들의 혼합과 반응성 성층화 정도에 영향을 받는 '조기 분사 예혼합 연소(Early injection PCI combustion)'으로 판단할 수 있다.

관련된 결과로서 이종 연료 연소 시 미연탄화수소의 발생 원인에 대해 분석하였다. 이종 연료 연소 역시 일반적인 디젤 신연소 기술들과 같이 디젤 조기 분사로 인한 벽면 분무 적심(wall-wetting) 현상이 발생하기도 하지만, 흡기와 함께 공급된 저반응성 연료가 틈새 체적으로부터 연소되지 않고 배출되는 일반적인 가솔린 엔진과 동일한 문제를 겪고 있다. 따라서 본 연구 결과로부터 상대적으로 고부하의 농후한 연소 시 디젤 조기 분사 조건을 제외하고, 대체로 이종 연료 연소 시 다수의 미연탄화수소는 틈새체적 효과 (crevice effect)에 기인함을 알 수 있었다. 이 때, 벽면 분무 적심 현상을 줄이기 위해 디젤 분사량을 줄일 시, 필연적으로 가솔린 연료량이 증가하여 틈새체적 효과가 커지게 되고, 반면 가솔린 연료를 줄일 시 디젤 연료의 증가로 인한 분무 적심 현상이 발생할 가능성이 커지므로, 상호 간에 상반관계(trade-off relation)가 존재함을 확인할 수 있다. 따라서 이종 연료 연소 시, 각 운전조건에 따라 최적의 가솔린과 디젤 연료 비율 및 디젤 분사시기가 존재함을 알 수 있다.

마지막으로 앞서 연구한 이종 연료 예혼합 연소를 이용하여 미연탄화수소를 최소화할 수 있는 최적의 운전조건을 바탕으로, 저부하 시 열효율 향상 및 고부하 운전 영역 확장에 대한 연구를 진행하였다. 이에 따라 저부하 운전조건에서는 압력상승률은 큰 문제가 되지 않으므로, 세번째 연소모드를 이용하여 최적화를 진행하였다. 이 경우, 높은 가솔린 연료 비율 및 흡기압력과 디젤 분할 분사를 이용하여 1,500 rpm/ gIMEP 6.5 bar 조건에서 최대 48 %의 열효율을 확인하였다.

또한 고부하 운전 조건에서는 두번째 연소 모드를 이용하여, 디젤과 가솔린 연료의 동시 다발적인 자발화 연소를 피함으로써, 최대 압력상승률을 낮추어 노킹(knocking) 현상을 피하고 2,000 rpm 에서 최대 gIMEP 14 bar 조건을 확보하였다. 이를 통해 이종 연료 예혼합

연소를 이용하여 기존의 NEDC 모드 운전 영역의 전반을 아우를 수 있도록 운전 영역을 넓힐 수 있다는 가능성을 타진하였다.

따라서 본 연구는 이종 연료의 연소 모드를 구분 짓는 연구와 동시에 각 운전조건에 적합한 연소모드를 제시하여 높은 열효율 및 운전 영역의 확대를 확인하였다. 본 연구의 결과를 통해 이종 연료 연소를 승용 디젤 엔진에 실질적으로 적용하기 위한 초석을 마련하였다.

주요어 : 디젤 엔진, 디젤 분사전략, 이종 연료 연소, 질소산화물, 입자상 물질, 예혼합 압축착화연소, 반응성 조정 압축착화연소

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