공학석사학위논문

가솔린-디젤 예혼합 융합연소 엔진에서 유효압축비 변경을 통한 부하 확장에 대한 연구

Load Expansion by Changing Effective Compression Ratio on Gasoline-Diesel Dual-fuel Premixed Compression Ignition

2020 년 8 월

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이 논문을 공학석사 학위논문으로 제출함

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Abstract

Load Expansion by Changing Effective Compression Ratio on Gasoline-Diesel Dual-fuel Premixed Compression Ignition

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Internal combustion engines have attracted attention as one of the sources of carbon dioxide (CO_2) so that the emission regulations are becoming stringent. Diesel engines, which are advantageous for CO_2 emissions based on high efficiency, have begun to come into the spotlight. However, Volkswagen scandal with the human hazards of nitrogen oxides (NOx) and particulate matter (PM) which are representative exhaust emissions of diesel engines became remarkable, emission regulations based on WLTP and RDE are expected to continue to be severe.

The advantage of dual-fuel combustion as a new advanced combustion technology is that low exhaust emission level can be achieved while maintaining high thermal efficiency based on pre-mixed compression ignition and also the combustion can be controlled by using different reactivity of two types of fuel. The control of the combustion phase can be possible with various

strategies such as fuel substitution rates, fuel injection timing, EGR rate and so on. However, there is a disadvantage that pre-mixed compression ignition causes knocking and high pressure rise rate. So, the operating load range is limited compared to diesel engine. Thus, research for load expansion of dual-fuel combustion is necessary.

In this study, the high load expansion strategy was investigated by changing the effective compression ratio on gasoline-diesel dual-fuel premixed compression ignition engine. The effective compression ratio was adjusted by delaying the intake valve close timing via the intake valve actuator system. First of all, the experiments were conducted to find geometric compression ratio that could show higher thermal efficiency than conventional diesel engines under low load operating condition. The gross thermal efficiency of the EURO 6 engine was 43.8% under low load driving condition. At this time, the comparison of optimization results at the same operating point by applying geometric compression ratio of 14 and 15 respectively. In case of geometric compression ratio 14 showed a lower thermal efficiency of 41.4% while geometric compression ratio 15 showed a higher thermal efficiency as 44.3% than conventional diesel engine. Thus, base geometric compression ratio was applied as 15.

With geometric compression ratio 15, the high load expansion experiments were carried out with the various effective compression ratio at 1,500 and 1,750 rpm. The effective compression ratio through LIVC was continuously controllable from 15.0 to 9.3. At 1,500 rpm high load operating condition, the effective compression ratio 14.1, 12.7, 11.1 and 10.6 were applied respectively. As a result, the full load limit increased by 10.5% while maintaining the equivalent thermal efficiency up to the effective compression ratio of 12.7. And the full load limit can be increased further with the lower

effective compression ratio, but the thermal efficiency also tended to decrease due to reduced combustion reactivity. At 1,750 rpm, high load conditions with certain effective compression ratio based on 1,500 rpm experiment results, the load tended to expand with the equivalent thermal efficiency, and if the effective compression ratio was further reduced, the thermal efficiency tended to decrease at the same time as the load expansion.

Keywords: Dual-fuel, Dual-fuel combustion, Premixed compression ignition, Maximum pressure rise rate, Geometric compression ratio, Effective compression ratio

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Nomenclature

Greek Letters

η	efficiency
$\emptyset$	phi, overall equivalence ratio
λ	lambda, relative air/fuel ratio, inverse of Phi

Acronyms

AFR	air-fuel ratio
ATDC	after top dead center
BD	burn duration
BTDC	before top dead center
CA	crank angle
CDC	conventional diesel combustion
CI	compression ignition
CO	carbon monoxide
CO_2	carbon dioxides
CoV	coefficient of variation
CR	compression ratio
CVVD	continuous variable valve duration
CVVL	continuous variable valve lift
CVVT	continuous variable valve timing
DI	direct injection
DIT	diesel injection timing
DPF	diesel particulate filter
EGR	exhaust gas recirculation
EVC	exhaust valve close
EVO	exhaust valve open
FSN	filtered smoke number

gIMEP	gross indicated mean effective pressure
gITE	gross indicated thermal efficiency
HCCI	homogeneous charge compression ignition
HFR	hydraulic flow rate
HRR	heat release rate
IC	internal combustion
IVC	intake valve close
IVO	intake valve open
LNT	lean NOx trap
LTC	low temperature combustion
LTHR	low temperature heat release
MFB50	mass fraction burned 50 %
mPRR	maximum pressure rise rate
NEDC	new European driving cycle
NOx	nitrogen oxides
NVO	negative valve overlap
PCCI	premixed charge compression ignition
PCI	premixed compression ignition
PFI	port fuel injection
PM	particulate matter
ppm	particles per million
RCCI	reactivity controlled compression ignition
RDE	real driving emission
rpm	revolution per a minute
SCR	selective catalyst reduction
SI	spark ignition
SOC	start of combustion
SOI	start of injection
TDC	top dead center

THC	total hydrocarbon
WLTP	world harmonized light vehicle test procedure

Chapter 1. Introduction

1.1 Research Background

Since the early 2000s, the global Carbon Dioxide (CO_2) regulations are becoming stringent around the globe due to global warming so that CO_2 regulation in the transport sector, which is pointed as one of the emission sources of global warming, are also becoming strict such as Figure 1.1.1.

The diesel compression ignition (CI) engine has advantages in terms of CO_2 compared to gasoline spark ignition (SI) engine. However, due to diesel combustion nature of the diffusion flame characteristics, soot is formed at fuel rich region and nitrogen oxides (NOx) is produced at high temperature of combustion region [3].

After the facts that nitrogen oxides (NOx) and particulate matter (PM) are harmful to the human health were remarkable, emission regulations are being severed around the world. As a result, the test driving cycle has changed from NEDC (new European driving cycle) to WLTP (world harmonized light vehicle test procedure) since 2017 shown in Figure 1.1.2. Also, RDE (real driving emissions) which is the new test cycle on real road has been announced since 2017. On RDE test, the exhaust emission level from real road test should meet the regulations with conformity factor (CF) 2.1. From 2020, this CF decreases to 1.5 for PN and 1.43 for NOx emissions.

To meet emission regulations, after-treatment systems such as Lean NOx trap (LNT), diesel particulate filter (DPF) or selective catalytic reduction (SCR) have been introduced. In addition, advanced combustion technology has been studied to reduce CO_2, NOx and soot emissions in the engine itself. The most common technology is the low temperature combustion (LTC). The main strategy of LTC is premixing a significant of fuel through early fuel injection. Premixed air-fuel mixture makes low flame temperature based on decreased local equivalent ratio (∅) which inhibits NOx. And also, early fuel injection which enables long ignition delay (I.D.) contributes soot reduction. With mentioned advantages regarding emission reduction, LTC is possible to maintain high thermal efficiency similar to conventional diesel combustion (CDC) based on the compression ignition of premixed air-fuel mixture due to low heat losses [4].

The most typical LTC strategies are the diesel premixed charge compression ignition (PCCI) and the gasoline homogeneous charge compression ignition (HCCI). In the PCCI strategy, partial amount of diesel is premixed to reduce soot through early injection and the rest of fuel is injected around TDC to control the combustion phase such as CDC. The flame temperature in the PCCI is lower than CDC, however, it is still high enough to form large amount of NOx. Thus, EGR is supplied to control NOx emission by lowering the flame temperature and O2 concentration. In the view of soot emission, the premixed fuel ratio should be increased. So, PCCI has a problem that the combustion phase control by small amount of main injection around TDC is limited [5].

HCCI makes a well-premixed mixture condition through early injection of all amount of fuel. The combustion is occurred by simultaneous multiple auto-ignition at the air-fuel mixture in cylinder during the compression stroke. The well-premixed condition guarantees very low level of soot and NOx emissions and very short combustion duration offers high thermal efficiency. But knocking is caused by the high volatility of gasoline, which is the main fuel used in HCCI, and combustion phase control is also an issue [6].

Reactivity controlled compression ignition (RCCI) is appropriate method in terms of the combustion control, low level emissions and high thermal efficiency. The low reactivity fuel (e.g. gasoline, natural gas, propane and etc.) is premixed by early injection during intake stroke and the high reactivity fuel such as diesel is injected as an ignition source during the compression stroke. Then, the combustion can be controlled by the reactivity stratification of two fuels and the equivalence ratio [7,8].

However, there is a disadvantage that pre-mixed compression ignition causes negative effect on durability of engine such as knocking and high pressure rise rate. Therefore, the operating load range is limited compared to diesel compression ignition engine. Thus, research for load expansion of dual-fuel combustion is necessary.

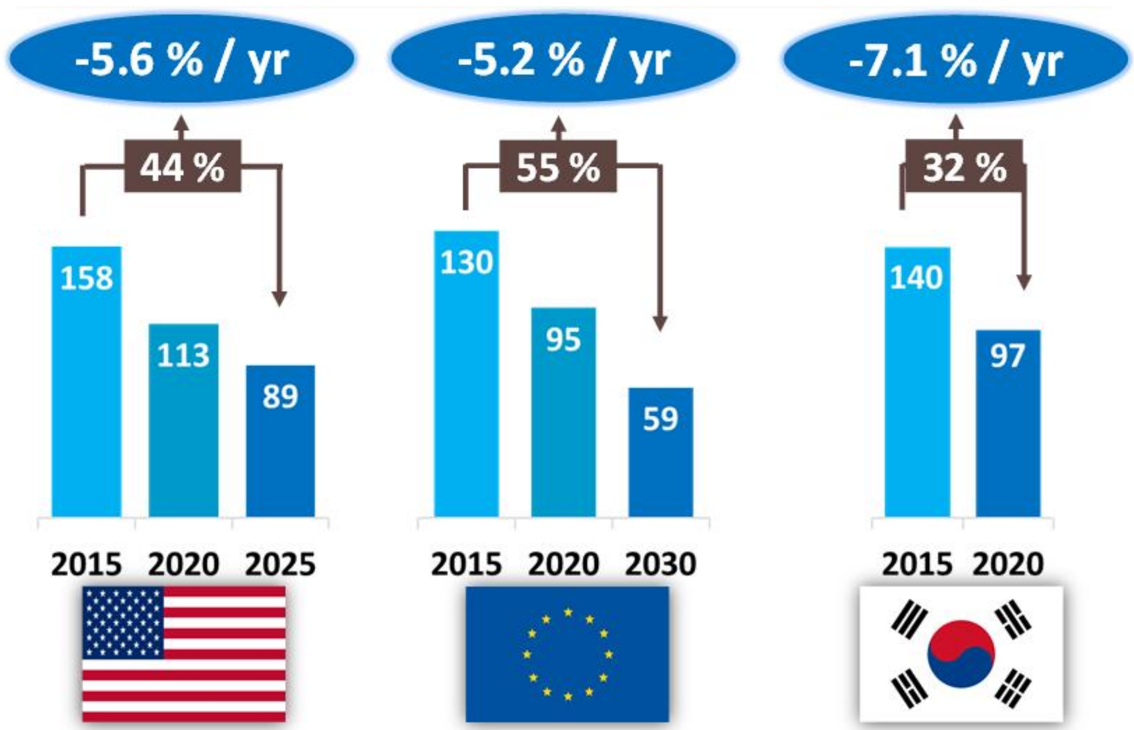

Figure 1.1.1 CO_2 regulation in the transport sector [1]

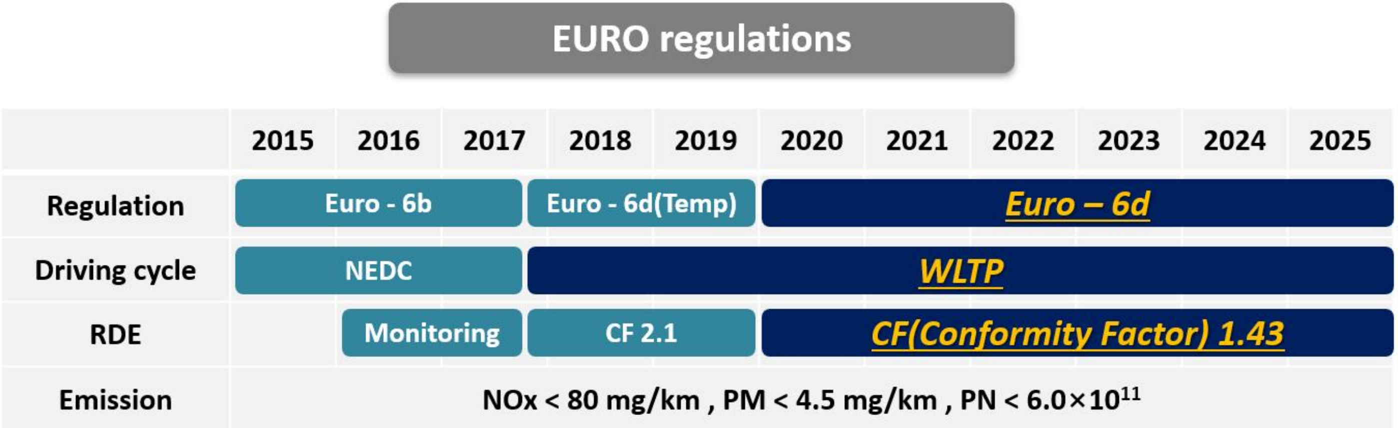

Figure 1.1.2 EURO emission regulation with driving cycle [2]

1.2 Previous Research

RCCI has a wide variety of control variables in that it uses two fuels and can combine the reactivity of the two fuels. Nieman et al. compared the results of various loads with CNG-diesel dual-fuel combustion in heavy-duty engines. This study found that the dual-fuel combustion using CNG has a lower PRR than gasoline and was beneficial for load expansion [9]. Kang et al. worked on the combustion stability, exhaust emissions level under various operating strategies for propane-diesel dual-fuel combustion [10]. In the similar way, Reiter and Kong investigated the combustion and exhaust emissions characteristics through dual-fuel combustion using ammonia, which is carbon-free and more advantageous in terms of production and handling [11]. In addition, Ashok et al. analyzed the effects and characteristics of LPG-diesel dual-fuel combustion [12]. These studies show that difference in the chemical properties between the fuels cause different combustion, emission characteristics and thermal efficiency.

The ratio of the two fuels in the dual-fuel combustion is a factor used as a representative operating strategy. Kokjohn et al. investigated optimal fuel proportion experiment that could control combustion phase with reasonable PRR for part and high loads [13]. Hanson et al. studied fuel fraction effect under low load [14]. They found combustion phase was easily controlled by varying the fuel fraction. Also, high reactivity fuel such as gasoline+2-EHN shows shorter combustion duration that gasoline-diesel combustion. Since combustion can be controlled by the ratio of two fuels with different reactivity, it is reasonable that the combustion can be controlled through the injection type (i.e. PFI or DI) and diesel injection timing. Garcia et al proposed that a fully premixed charging, a partial premixed charging by multiple and late injection, diffusive single injection around TDC are the best ways to obtain optimized combustion at in the order of low, mid and high loads [15].

Introducing EGR is also an effective way in reducing reactivity of combustion reaction in RCCI along with the emission reduction. Splitter et al. conducted experiment to investigate the effect of EGR both gasoline-diesel and E85-diesel dual-fuel combustion from mid to high loads. They demonstrated that EGR rate were increased to maintain low peak pressure and pressure rise rate toward high loads [16]. Also, Chu et al. showed that supplying EGR is effective to obtain reasonable PRR and low emission level [17].

At most researches have commonly demonstrated, dual-fuel combustion has low level of exhaust emissions. And many researchers seek various strategies for combustion control and load expansion while maintaining a high level of efficiency with stable combustion. The fuel reactivity control, fuel proportion, injection strategy and EGR mentioned in all the literature surveys enable combustion control and operation at various loads. However, the high PRR based on premixed mixture combustion still limits expansion toward high loads than CDC operating range.

1.3 Introduction of Dual-fuel PCI strategy

Based on the previous researches, RCCI has many advantages compared to other LTC strategies. Especially, the most advantageous characteristic is the combustion controllability with various operating strategies. However, the combustion control concept of RCCI has limitation on real operating condition. On real operating condition, for example, such as rapidly changed from high load to low load operation in transient mode, the overall combustion reactivity tends to be reduced even if the diesel ratio is increased because the large amount of residual EGR will be remained in cylinder.

Dual-fuel PCI (Premixed Compression Ignition), which was introduced by Inagaki et al [18], has better the combustion controllability compared to RCCI. Refer to Figure 1.3.1, low reactivity fuel is injected with fresh air and EGR during the intake stroke. After that, diesel direct injection is happened during the compression stroke. The major differences between RCCI and Dual-fuel PCI are diesel injection strategies such as diesel injection timing (DIT), single or multiple injection methods and equivalence ratio. As one of them, DIT is happened around 20 to 40 °CA BTDC, which is relatively retarded compared to RCCI. This strategy enables not only control the combustion phase but also exhausts low level of emissions by premixed air-fuel mixture. The advantages of Dual-fuel PCI are listed on Figure 1.3.2. At the ignition process, auto-ignition occurs at diesel rich area, and the power is generated during the expansion stroke with low reactivity fuel's premixed combustion. For the effects of control factors on combustion are described in chapter 3.1.

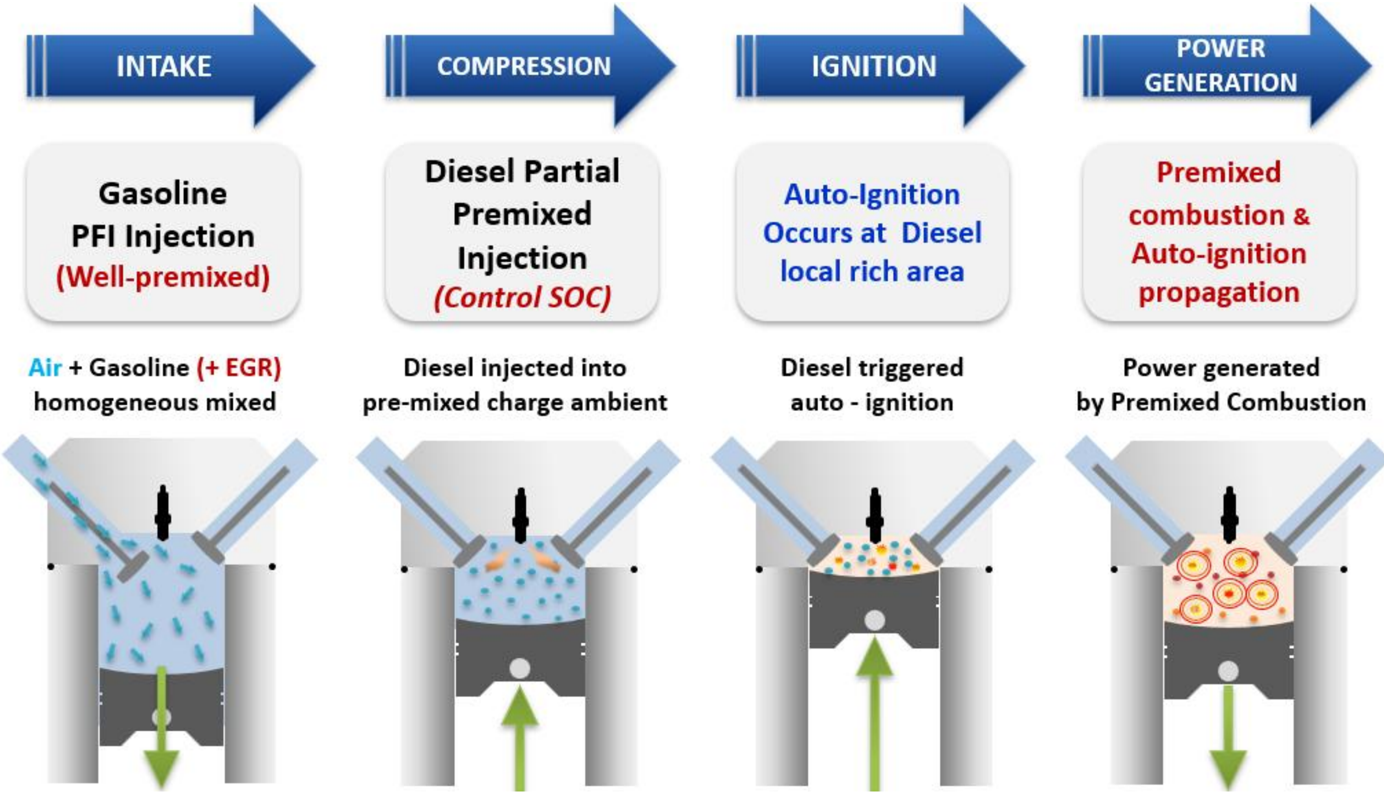

Figure 1.3.1 Process and principle of Dual-fuel PCI

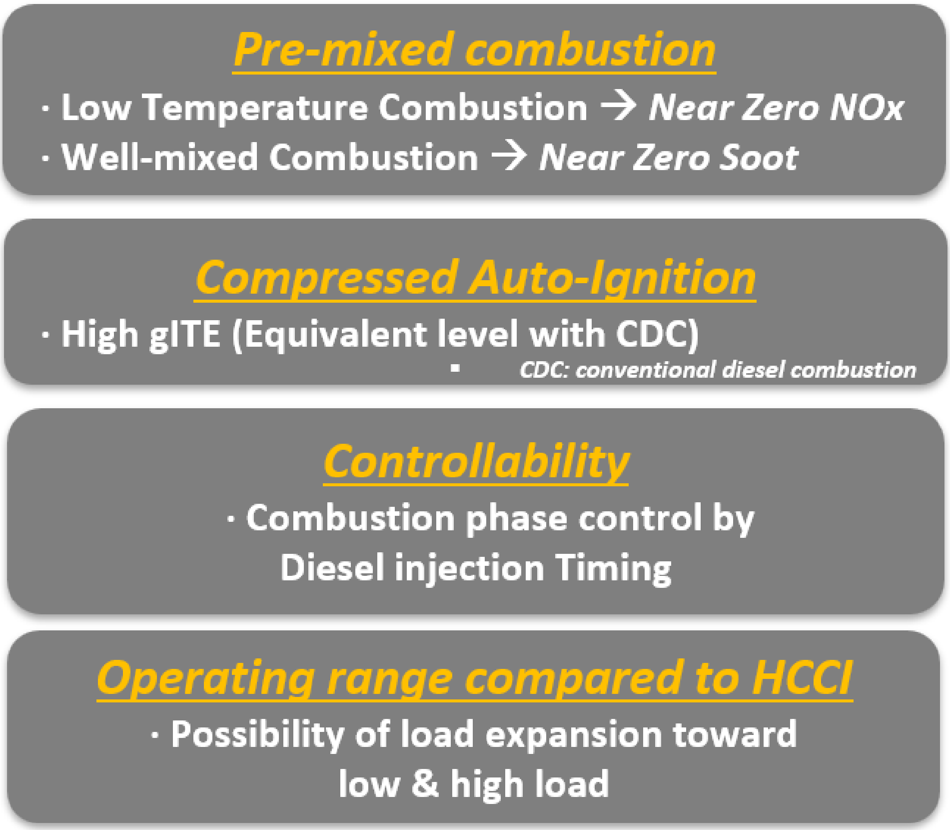

Figure 1.3.2 Merits of Dual-fuel PCI

1.4 Research Motivation and Objective

Dual-fuel PCI has the better combustion controllability compared to RCCI, however, the limitations still exist for high load expansion due to high PRR.

Seoul National University-Automotive laboratory has investigated the optimal hardware which affects the combustion characteristics for dual-fuel combustion. As one of them, which geometric compression ratio (CR) is more suitable for dual-fuel PCI was studied. The geometric CR is kinetic control variables on combustion. Such as diesel engines, high geometric CR around 16 is normally adopted. Dual-fuel PCI is a combustion strategy that is operated based on compression ignition of high reactivity fuel. So, the geometric CR around 16 and 17 are selected similar to diesel engine in most dual-fuel combustion research. Based on SNU-Automotive laboratory research results, with geometric CR around 16, it has high thermal efficiency while maintaining low level of emissions. However, it is impossible to expand high load due to high PRR. With low geometric CR around 14, it has potential for load expansion compared to high CR. In this case, however, thermal efficiency is lower than diesel engine under low load operation. To sum up, it demonstrated that low CR is suitable for the expansion of the high load and high CR is advantageous in terms of high thermal efficiency. The problem is that geometric CR is one of the hardware design factors so that it cannot be adjustable at engine. Thus, there is limit to choose between efficiency and load expansion.

Recently, Miller cycle, which has the longer expansion stroke than the compression stroke, has become common technology in order to improve the fuel consumption. Miller cycle of the past was implemented through a complex mechanical structure design, but is now simply controllable through valve timing control technology via an early intake valve close timing (EIVC) or late intake valve close timing (LIVC). EIVC or LIVC strategies reduce the intake charging air-fuel mixture mass and the pressure at start of combustion (SOC). Finally, it results in adjusting the effective CR.

Therefore, the objective of this research is to achieve high thermal efficiency compared to diesel engine under low load operation and high load expansion with LIVC strategy which can reduce the combustion reactivity through the low effective CR toward high load operation.

Research process were as following;

1) Dual-fuel PCI experiments with geometric CR 14 and 15 were conducted to find which CR has higher thermal efficiency than diesel engine at 1,500 rpm, gIMEP 5.2 bar.

2) Dual-fuel PCI high load operation with various effective CR were done to investigate proper effective CR which can expand gIMEP while maintaining the thermal efficiency.

Chapter 2. Experimental Apparatus

2.1 Experimental Setup

A single cylinder diesel engine based on EURO 6 standard was used for this work. Detailed engine specifications are shown in Figure 2.1.1 and Table 2.1.1. The diesel fuel can be directly injected as 450 bar with a BOSCH solenoid injector which can pressurize up to 1,800 bar with a common rail system. Two solenoid gasoline injectors were installed at intake port. The pressurized gasoline fuel as 6 bar was injected into the cylinder with fresh air and cooled EGR. The air and EGR were boosted by a supercharger and then cooled by intercooler. So, the temperature of fresh air-EGR mixture was kept around 27-30℃. The boosted intake pressure was determined based on EURO 6 diesel engine operating condition. The fuel flowrates were measured by OVAL Altimass CA001, Coriolis type flowmeter, for each diesel and gasoline. The temperature of engine oil and coolant was maintained at 85℃. The geometric CR can be changed by installing shim plate between cylinder block and cylinder head. For this experiment, geometric CR 14 and 15 were adopted. Piston bowl geometry was bathtub shape. Most studies have shown that the bathtub shape piston geometry contributes to the improvement of thermal efficiency due to low heat loss by reducing the inside surface area of the combustion chamber. The concentration of carbon monoxide (CO), carbon dioxide (CO_2), oxygen (O_2), nitrogen oxide (NOx) and total-hydrocarbon (THC) were measured at a HORIBA, MEXA 7100DEGR, an exhaust gas analyzer. The soot was measured by AVL, 415S, a smoke-meter. The schematic diagram of experimental apparatus is shown in Figure 2.1.2.

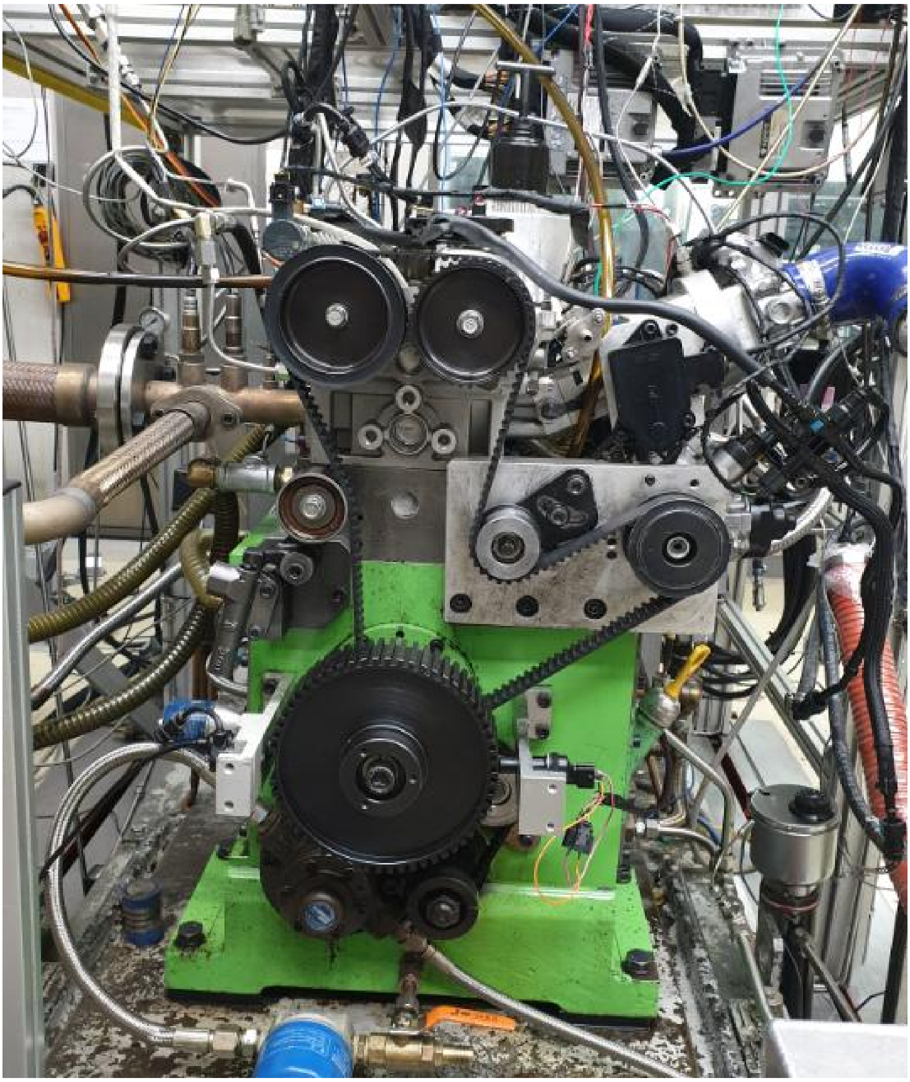

Figure 2.1.1 Dual-fuel Single Cylinder Engine

Table 2.1.1 Engine specifications

Engine type	Single cylinder (four-stroke)
Displacement (cc)	395
Bore ×Stroke (mm)	77.2×84.5
Connecting rod (mm)	140
Compression ratio (-)	14/15
IVO/IVC (Without LIVC)	BTDC 8°/ ABDC 36°
EVO/EVC	BBDC 46° /ATDC 4°
Diesel injection system	Solenoid injector
Injection pressure (bar)	450
No. of injector nozzle holes	8
Cone angle (deg)	149

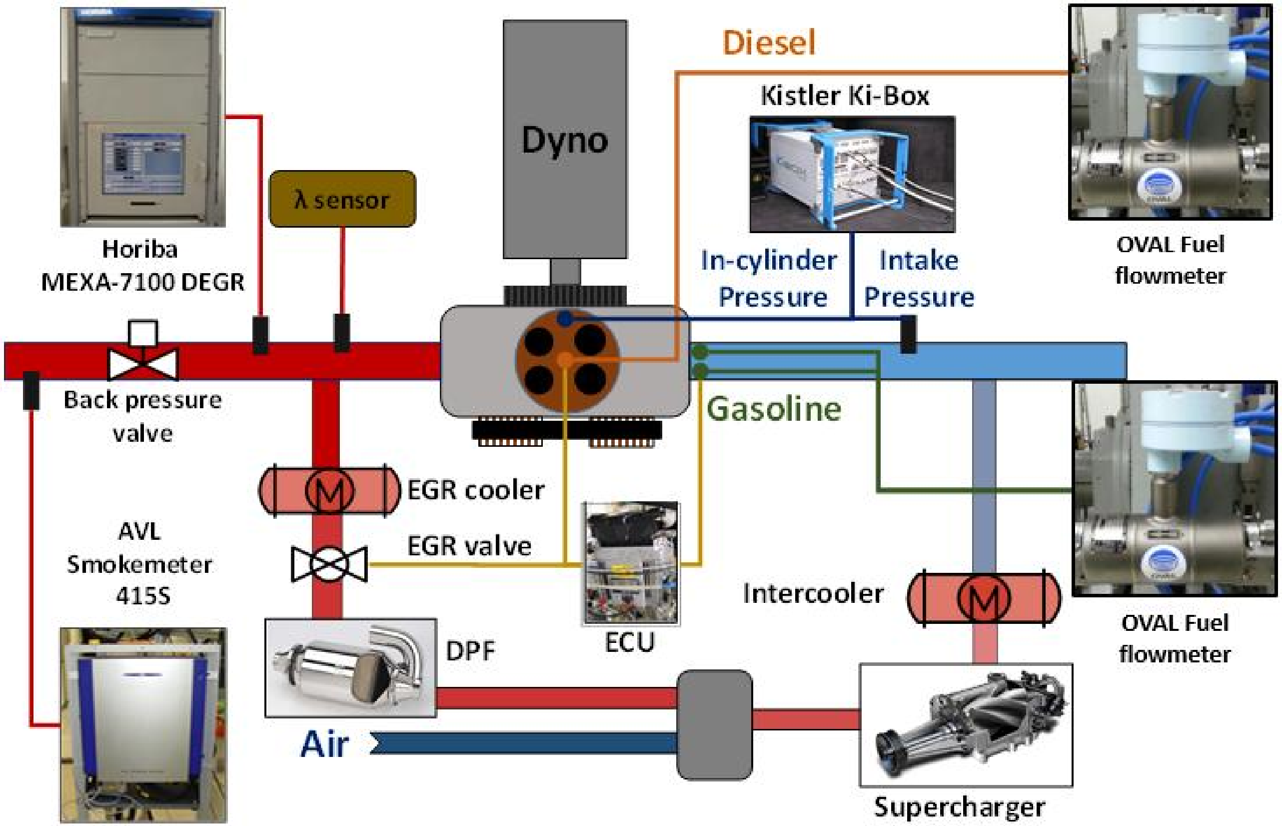

Figure 2.1.2 Schematic diagram of experimental apparatus

Table 2.1.2 Specifications of dynamometer

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

Table 2.1.3 Specifications of fuel flowmeter

Item	Specification
Manufacturer	OVAL
Measurement principle	Coriolis Type
Model	CA001
Range of flow amount (kg/h / max)	0~9
Range of flow amount (kg/h / min)	0~0.45
Maximum allowed pressure (MPa at 20℃)	15
Measurement uncertainty (%)	<0.2
Operating temperature (°C)	-200 ~ +200

Table 2.1.4 Specifications 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

2.2 Fuel properties and substitution rate of dual-fuel

Diesel and gasoline fuel were used in this experiment and properties of fuel are presented in Table 2.2.1. The substitution rate of fuel in dual-fuel combustion is calculated by considering low heating value (LHV) as shown in equation (1).

$$x_{gasoline}(\%) = \frac{\dot{m}_{gasoline} * LHV_{gasoline}}{\dot{m}_{diesel} * LHV_{diesel} - \dot{m}_{gasoline} * LHV_{gasoline}} \quad (1)$$

Table 2.2.1 Properties of diesel and gasoline

Properties	Diesel	Gasoline
Chemical formula	$C_xH_{2.0x}$	$C_xH_{2.0x}$
Density (g/cm^3)	0.831	0.724
Low heating value (MJ/kg)	42.5	42.8
Cetane/Octane number	54 (CN)	91 (RON)
Auto-ignition temperature (K)	483	520-553
Stoichiometric ratio of AF (wt.%)	14.6	14.5

2.3 Experimental Condition

In this study, the dual-fuel combustion was optimized with the various combustion control parameters, such as fuel proportion rate between diesel and gasoline, diesel injection timing and cooled LP-EGR rate. Firstly, the effects of these parameters were studied.

To investigate fuel substitution effect, the experiment was conducted from 0% gasoline fuel (=100% diesel ratio) to the maximum amount of gasoline that can be ignited with stable combustion. As gasoline increases, diesel proportion is reduced followed by Eq (1). And the experiment of diesel injection timing (DIT) was proceeded under dual-fuel combustion condition to study the effect of diesel injection timing as a trigger of auto-ignition on combustion control. In this case, the high level of emissions and high pressure rise rate (PRR) could be occurred due to absence of EGR supplied. So, supplying EGR was considered to be necessary to obtain maximum efficiency through optimization of the dual-fuel combustion such as shifting combustion phase, reducing PRR and control of emission level.

To find the high thermal efficiency than diesel engine under low load operation, dual-fuel PCI experiments with geometric CR 14 and 15 were conducted at 1,500 rpm, gIMEP 5.2 bar. Then, the high load expansion experiment was carried out by applying the LIVC strategy on the geometric CR determined in the low load operation experiment. The engine-out emission of dual-fuel PCI should meet the constraints listed in Table 2.2.2.

Table 2.2.2 Emission constraints

Operating Condition	mPRR [bar/deg]	CoV of IMEP [%]	NOx [ppm]	Soot [FSN]
Low load (1,500 rpm/ gIMEP 5.2 bar)	5	3	40	0.2
High load (1,500 & 1,750 rpm/ full load)	10			

2.4 Variable Valve Actuation System and effective CR

Engine technology has made remarkable progress and a variety of technologies have been focused for efficiency improvement, fuel injection method, engine downsizing with turbocharger and the next generation engine such as HCCI, PCCI, RCCI. Variable valve actuation system is one of the important roles as the key technology that benefits both the performance and emission reduction.

The state-of-the-art VVA system is continuous variable valve duration (CVVD) system. It can control the valve opening duration so that it performs the role of continuous variable valve timing (CVVT) and continuous variable valve lift (CVVL) [21].

The key principle of CVVD system is to use eccentricity. The dis-assembly drawing of CVVD is on Figure 2.4.1. In case cam shaft is rotated, cam shaft slot on cam shaft is rotated and in order of roller, cam slot and cam are rotated. In the view of cam knob, it is rotated by cam slot. The fixed point is cam knob center and cam slot can be shift upward and downward during rotation. If cam slot is close from cam shaft center, the distance for rotation is shorter so that duration will be decreased. On the other hand, cam slot is far from the shaft center, the distance for rotation is long situation which makes duration increased [22].

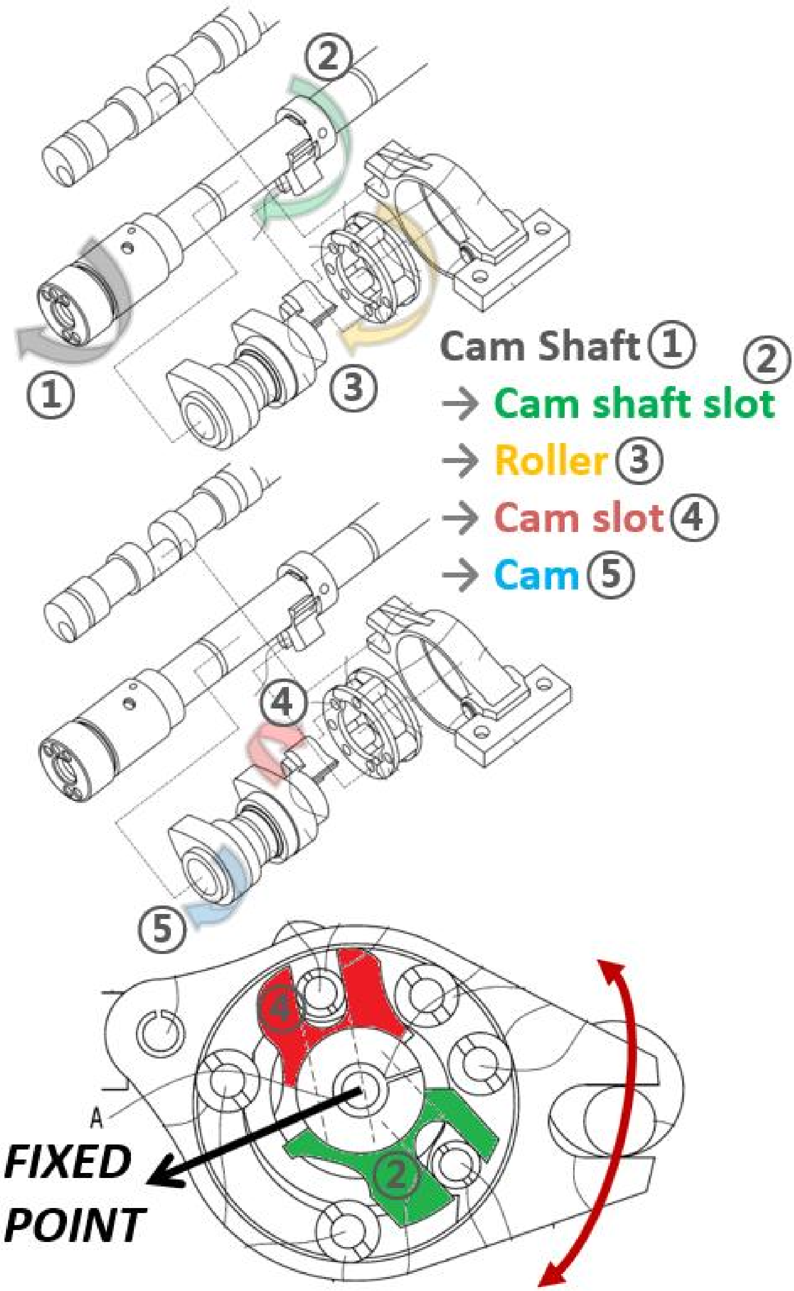

Figure 2.4.1 Disassembly and sectional drawing of CVVD [22]

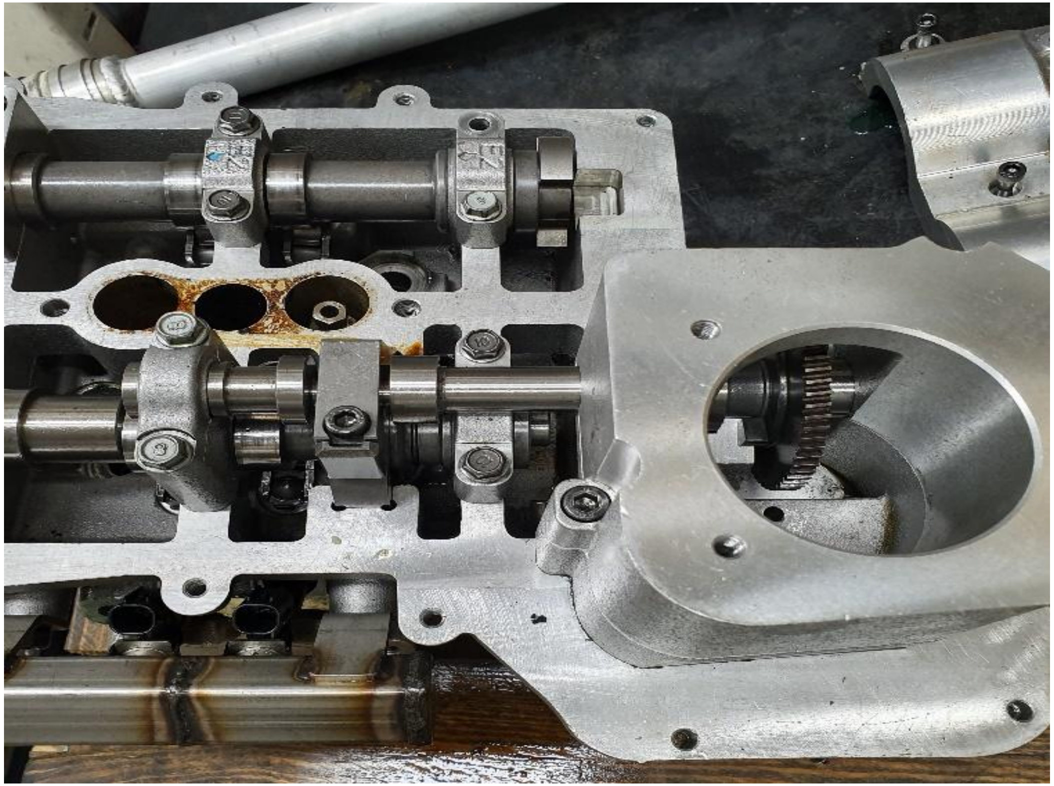

Figure 2.4.2 CVVD installation on dual-fuel single cylinder engine

Figure 2.4.2 shows installed CVVD system which was supported by Hyundai Motor Company. CVVD system on dual-fuel single cylinder engine can retard the intake valve close timing by 60° with fixed intake valve open timing such as Figure 2.4.3. If the intake valve close timing is delayed, the duration of opening in the compression stoke increases. It causes reduced effective CR. Figure 2.4.4 shows the cylinder peak pressure at firing TDC along the LIVC cases. The calculated effective CR according to LIVC cases is shown in Table 2.4.1.

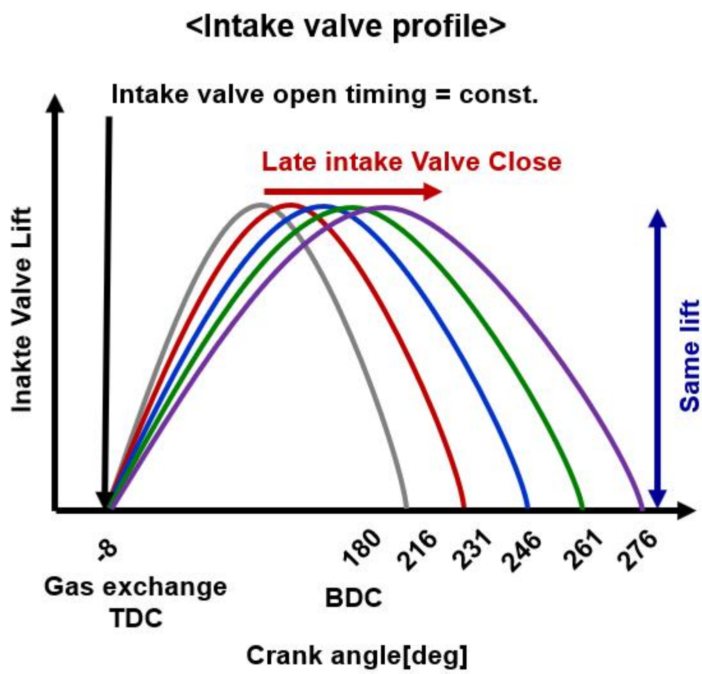

Figure 2.4.3 Intake valve profile with CVVD

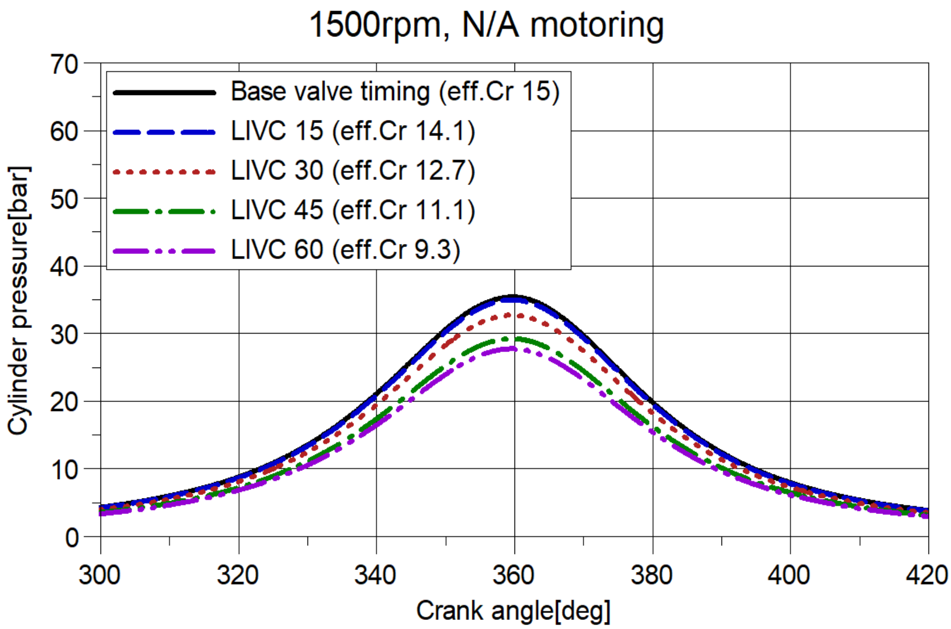

Figure 2.4.4 Cylinder peak pressure at firing TDC along the LIVC cases

Table 2.4.1 Calculated effective CR according to LIVC cases

IVC timing →	Base	LIVC 15	LIVC 30	LIVC 45	LIVC 50	LIVC 60
Effective Cr :	15.0	14.1	12.7	11.1	10.6	9.3

Chapter 3. Experimental Results and Discussion

3.1 Optimization of Dual-fuel PCI combustion

This section introduced the optimization strategy of dual-fuel PCI and studied about the effects of control factors, gasoline substitution rate, DIT and EGR rate.

To optimize the dual-fuel PCI combustion, gasoline substitution rate was increased as first step while maintain target gIMEP. At that time, soot emission is over 0.2FSN. Thus, DIT should be advanced to decrease soot emission and achieve the maximum thermal efficiency with same amount of fuel injections. Once gIMEP approached target value with gasoline injection and advanced DIT, MFB 50 should be controlled for the best thermal efficiency point such as 6~15° CA ATDC and NOx emission is needed to satisfy the constraint, 40ppm. As a result, for the final step, EGR should be supplied until NOx meets 40ppm. During the final step, EGR makes the combustion reactivity weaker so that MFB 50 may be retarded again. So, the combustion optimization should proceed by minor tuning the gasoline substitution rate and DIT adjustment. Based on this process, it can be possible to find the dual-fuel PCI combustion point which shows the best thermal efficiency while satisfying the all constraints listed in Table 2.2.2. The schematic diagram of dual-fuel PCI combustion process is described in Figure 3.1.1.

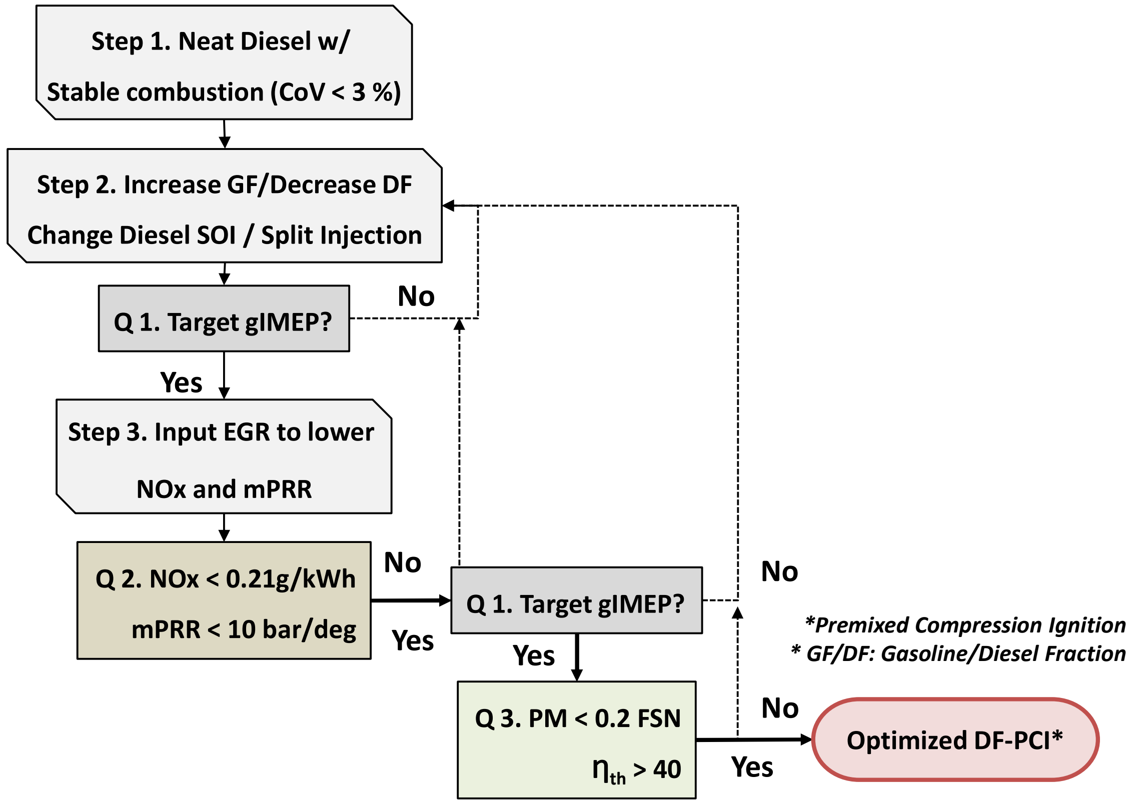

Figure 3.1.1 Schematic diagram of optimization process for Dual-fuel PCI combustion [19]

Gasoline substitution rate is one of the representative control factors on dual-fuel combustion. Figure 3.1.2 shows the cylinder pressure and heat release rate according to gasoline substitution rate at 1,500 rpm, gIMEP 5.2 bar. The cylinder pressure graph was retarded by increasing gasoline proportion. At that time, gIMEP increased since the combustion phase was retarded after TDC. This retard of combustion phase is due to the low reactivity fuel, gasoline, reduces the auto-ignition tendency of air-fuel mixture. With high ratio of diesel injection rate, the shape of cylinder pressure and heat release rate is sharp due to spontaneous auto-ignition triggered by multiple local rich diesel fuel around TDC. However, the gradient of heat release rate becomes lower along the diesel ratio decreased. In this case, the ignition sources from diesel local rich areas in premixed mixture are gradually reduced. Since then, auto-ignition at somewhere of highly premixed mixture by the increased pressure and temperature made by the compression stroke becomes fundamental ignition sources and it causes combustion of rest of fuel as flame propagation [20]. That is, the fundamental dual-fuel combustion mode depends on the fuel ratios between diesel and gasoline. Thus, substitution rate of dual-fuel was considered as a major control factor in this study.

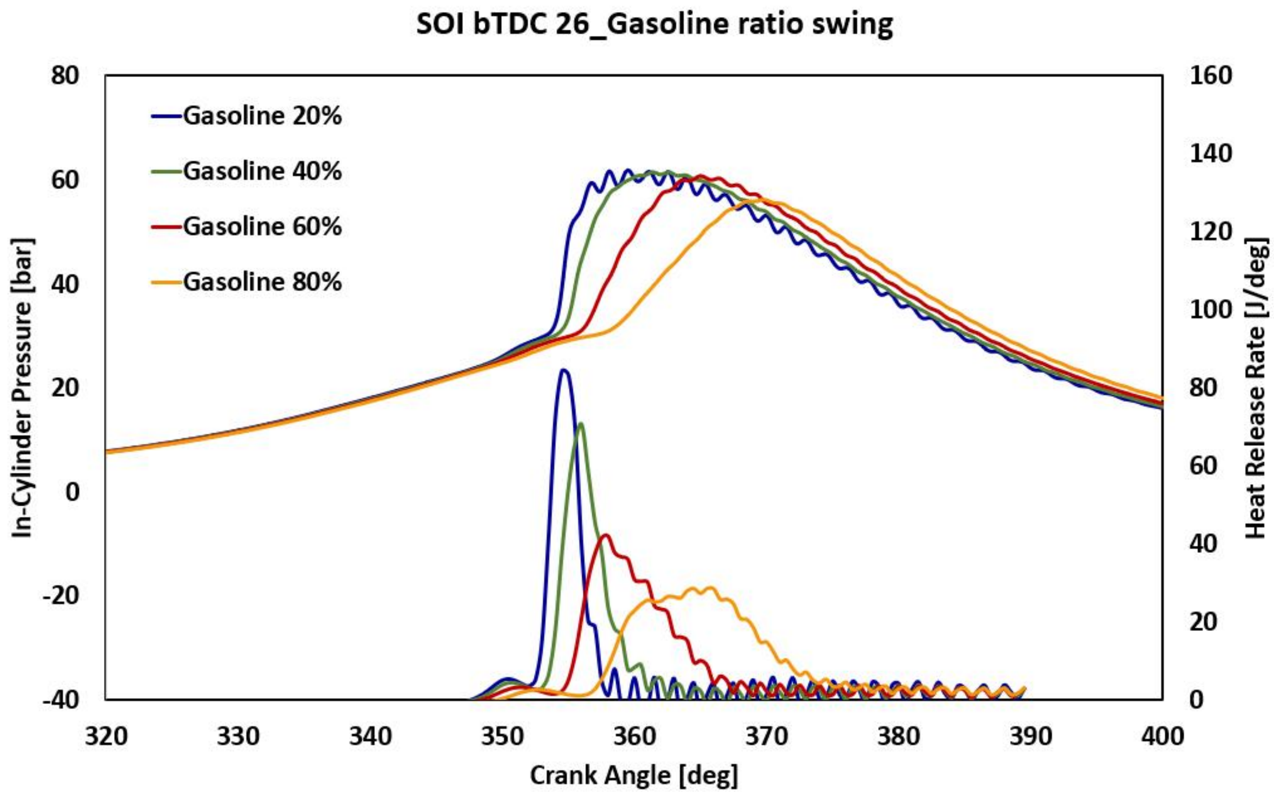

Figure 3.1.2 Cylinder pressure & Heat release rate in accordance with gasoline substitution rate at 1,500 rpm, gIMEP 5.2 bar

Also, DIT is another major control factor. Figure 3.1.3 shows the cylinder pressure and heat release rate in accordance with DIT. When DIT was advanced from 11°CA BTDC to 26°CA BTDC, the combustion phase was advanced until combustion mode transition point from diesel diffusion flame to spontaneous auto-ignition in premixed air-fuel mixture by high temperature and pressure in cylinder due to the compression stroke. When the combustion mode changed with sufficient advanced DIT such as 31° CA BTDC, the combustion phase begun to be retarded. Thus, it was used as an important combustion control factor because the reactivity of combustion can be controlled through the DIT. Also, sufficient advanced DIT around 20~25°CA BTDC, highly premixed air-fuel mixture for overall lean equivalence ratio condition is effective to reduce NOx and soot emissions.

Normally, supplying EGR makes the initial combustion reactivity decreased. It leads to longer I.D. so that maximum cylinder pressure and heat release rate decreased such as Figure 3.1.4. This cause and effect relationship are also applied to dual-fuel PCI combustion. As a result, the proper supply of EGR is a control factor that contributes to securing mPRR through reduction of combustion reactivity toward high load operation as well as reduction of NOx and soot emissions.

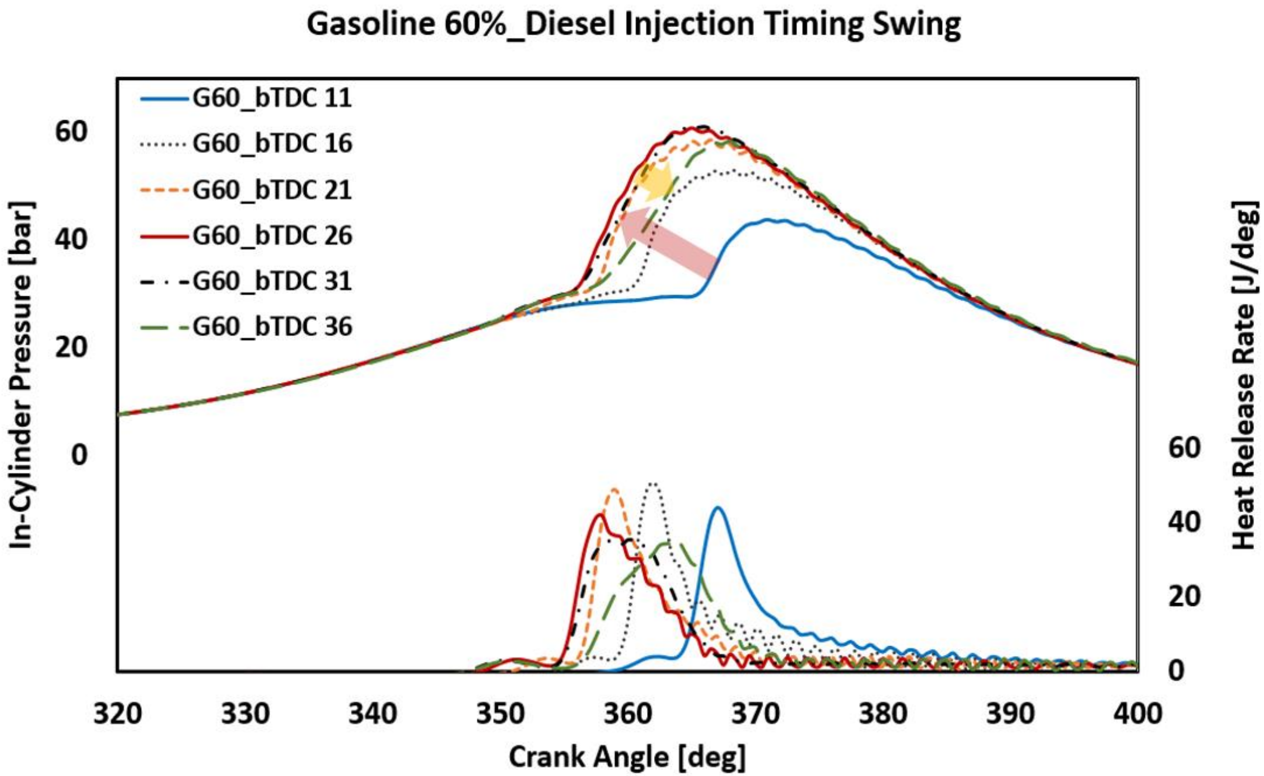

Figure 3.1.3 Cylinder pressure & Heat release rate according to various DIT under 60% gasoline substitution at 1,500 rpm, gIMEP 5.2 bar

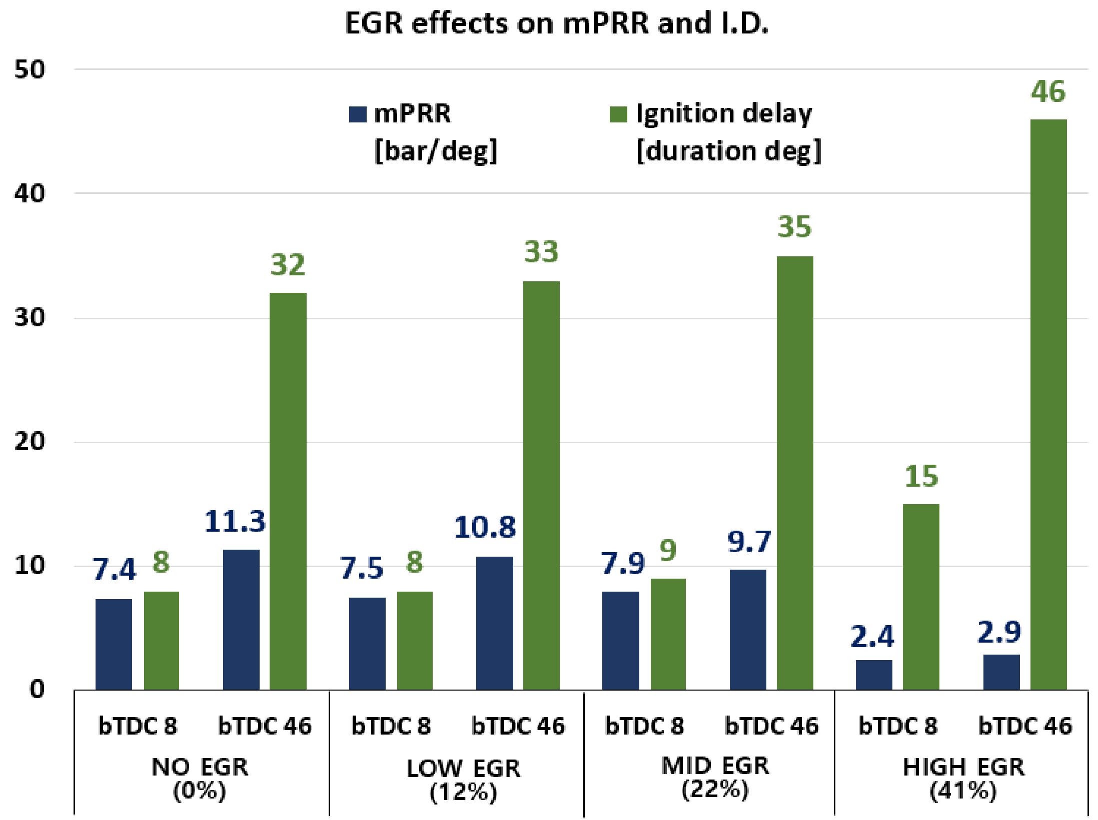

Figure 3.1.4 EGR effect on mPRR and I.D.

3.2 Determination of optimal geometric CR for high thermal efficiency under low load condition

In this section, Dual-fuel PCI experiments with geometric CR 14 and 15 were conducted to find proper CR based on thermal efficiency under low load operation, 1,500 rpm, gIMEP 5.2 bar. As mentioned in “1.4 Research Motivation and Objective”, SNU-Automotive laboratory has found that low geometric CR has potential to expand high load limit, however, it shows lower thermal efficiency than high geometric CR, especially under low load operation. Thus, it was necessary to find the lowest geometric CR among various CR that shows higher thermal efficiency than diesel engine under low load operation. The detailed operating conditions were listed in Table 3.2.1.

Figure 3.2.1 demonstrates gross indicated thermal efficiency (gITE) comparison between geometric CR 14 and 15 at 1,500 rpm, gIMEP 5.2 bar. The combustion phases on figure were optimized by controlling gasoline substitution rate, DIT and EGR supply. Both combustion phases from geometric CR 14 and 15 show very low level of NOx as each 28 ppm and 21 ppm, and soot emissions as each 0.016 FSN and 0.018 FSN. Based on gITE and NOx, soot emissions, the fact that dual-fuel combustion enable high thermal efficiency with near zero NOx and soot emissions can be assured.

As geometric CR increased, the combustion reactivity is better since high in-cylinder pressure and temperature were obtained during the compression stroke. It causes the high self-ignition characteristic from diesel area in cylinder. Therefore, the higher geometric CR, the more DIT was advanced and EGR rate was increased to control MFB 50 to where the point showing the highest gITE. Because advanced DIT decreased diesel local equivalent ratio (Ø) and increased EGR rate reduced the combustion reactivity. So, in geometric

CR 15, DIT and EGR rate were 31° CA BTDC, 50% respectively, and 23° CA BTDC, 46.4% in CR 14.

In geometric CR 15, MFB 50 was controlled at 5.3° CA ATDC. At that time, the combustion shows the best gITE as 44.3%. In case of CR 14, MFB 50 was 11.3° CA ATDC. Relatively retarded MFB 50 of CR 14 was impossible to be advanced due to the low combustion reactivity from lower CR. At that time, the highest gITE was calculated as 41.4%. Base diesel engine of this experiment shows 43.8% of gITE at 1,500 rpm, gIMEP 5.2 bar condition. That is, gITE of CR 15 has higher thermal efficiency than diesel engine, however, gITE of CR 14 was lower than CDC. As a result, geometric CR should be determined as 15 based on high thermal efficiency compared to CDC under low load operating condition.

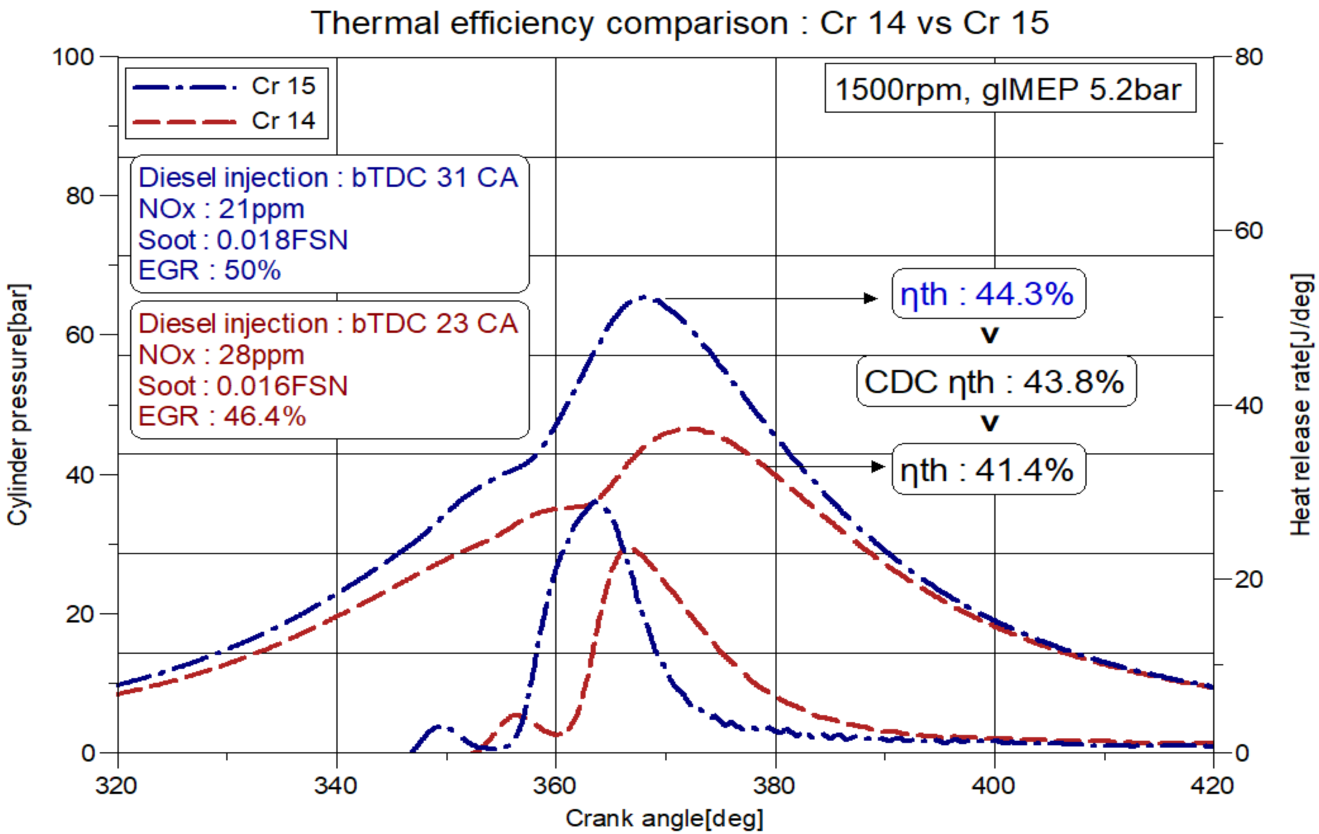

Figure 3.2.1 gITE compassion between geometric CR 14 and 15 at 1,500 rpm, gIMEP 5.2 bar

Table 3.2.1 Experimental conditions of determination of optimal geometric CR

Description	Value
Engine speed [rpm]	1,500
gIMEP [bar]	5.2
Diesel injection pressure [bar]	450
Gasoline Injection pressure [bar]	6
Compression ratio	14 / 15
Intake pressure [bar]	1.06
Engine oil & Coolant [°C]	85
NOx (target) [ppm]	< 40
Soot (target) [FSN]	< 0.2
mPRR (target) [bar/deg]	< 5
CoV of IMEP [%]	< 3

3.3 High load expansion with reduced effective CR via LIVC

The objective of this experiment is to expand the high load limit with reduced effective CR via LIVC strategy. Based on the result from "3.2 Determination of optimal geometric CR for high thermal efficiency under low load condition", base geometric CR was selected as 15. For this engine, normal intake valve opening/close timing were 8° CA BTDC and 36° CA ABDC. With normal IVC/IVO timing, the effective CR shows 15 same as geometric CR. The effective CR was able to be continuously lowered through LIVC. In this experiment, LIVC 15, 30, 35, 45, 50 and 60 were applied, respectively, by retarded unit of 15 degrees. At that time, volume based calculated effective CR of each cases were listed in Table 2.4.1.

Table 2.4.1 Calculated effective CR according to LIVC cases

IVC timing →	Base	LIVC 15	LIVC 30	LIVC 45	LIVC 50	LIVC 60
Effective Cr :	15.0	14.1	12.7	11.1	10.6	9.3

Intake boost pressure was set as the same as the conventional diesel engine. Thus, 1.9 bar was maintained at 1,500 rpm high load condition and 2.0 bar was set at 1,750 rpm high load condition.

High load optimization means the highest gIMEP that can be seen satisfying mPRR and CoV at the applicable rpm so that combustion takes place around lambda (λ) = 1. At high load operating condition, engine-out emissions should also satisfy the constraints listed in Table 2.2.2 and the experiment results were compared to the optimized points with the best gITE.

The cylinder pressure and heat release rate graphs for dual-fuel PCI high load experiments at 1,500 rpm with effective CR 15, 14.1 (LIVC 15 case) and 12.7 (LIVC 30 case) were demonstrated in Figure 3.3.1 and experimental conditions were listed in Table 3.3.1.

The major constraint for dual-fuel PCI combustion toward high load expansion is mPRR. To satisfy it, increasing EGR supply is unavoidable because of too strong combustion stability that exceed the criteria of mPRR and also control of MFB 50. In case of effective CR 15, normal IVC timing, gIMEP showed 13.3 bar with 52% EGR rate. And EGR rate were able to decrease 50.8%, 50% along to reduced effective CR 14.1, 12.7. As EGR rate decreased, a strategy to expand the gIMEP by injecting additional fuel via margin of lambda (λ) was possible. There were three ways to inject additional fuel as below.

1) Additional diesel injection with fixed or decreased gasoline injection
2) Additional injection of both fuels, diesel and gasoline
3) Additional gasoline injection with fixed or decreased diesel injection

Above three methods increased PRR, but showed a difference in the operating strategy to meet under 10 bar/deg again. In the first method, as diesel fuel increases, EGR rate must be increased due to increased combustion reactivity. Although DIT advances to decrease diesel local equivalent ratio, it was not effective due to combustion of premixed air-fuel mixture by spontaneously diesel ignition. It resulted in loss of either gIMEP or gITE compared to effective CR 15 because increased EGR decreases gIMEP or advanced MFB 50 reduces gITE. With the second method, the countermeasure and result were similar to the first method. However, the amount of pre-mixed air-fuel mixture and source of ignition, diesel, increases simultaneously, resulting in a high EGR rate as the mPRR increases more significantly than the first strategy. Therefore, both gIMEP and gITE tend to decrease. In case of third method, the combustion reactivity was reduced due to decreased diesel ratio. So, DIT should be retarded to increase diesel local equivalent ratio. Although soot emission increased by retarded DIT, it was able to meet 0.2 FSN until around 20° CA BTDC. Thus, additional gasoline injection was possible without additional EGR supply. It resulted in increase of gIMEP while maintaining gITE.

The combustion phases of effective CR 14.1 and 12.7 were accomplished with above mentioned third strategy, decreased diesel fuel with retarded DIT. The diesel ratios decreased from 22% to 16% with additional gasoline injection and EGR tended to decrease. gIMEP also expanded from 13.3bar to in the order of 14 bar and 14.7 bar. In the process of increasing gIMEP, gITE was equal to or decreased by only 1% compared to the highest effective CR, 45.9% even if MFB 50 was retarded from TDC. This is because the sum of the increase amount of the expansion work and the decrease amount of the compression work is greater than the work loss due to the cylinder peak pressure drop as effective CR decrease. The amount of work in the area of P-V diagram shown in Figure 3.3.2 demonstrated the total amount of work increases. Thus, gIMEP can be increased while maintaining gITE.

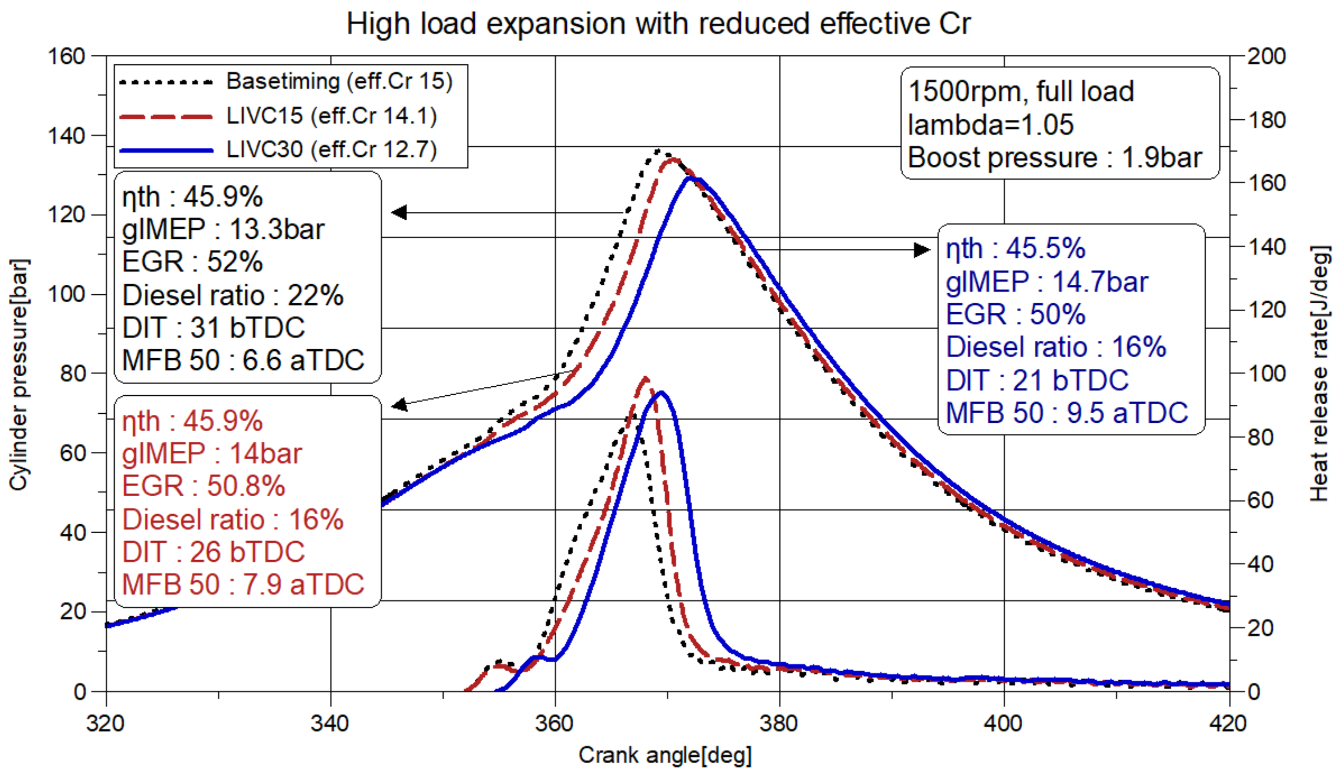

Figure 3.3.1 High load expansion with reduced effective CR 15 / 14.1 / 12.7 at 1,500 rpm

Table 3.3.1 Experimental conditions of high load expansion at 1,500 rpm – (1)

Description	Value
Engine speed [rpm]	1,500
gIMEP [bar]	Full load
Diesel injection pressure [bar]	450
Gasoline Injection pressure [bar]	6
IVO / IVC	8° CA BTDC / 36, 51, 66 CA ABDC
Geometric compression ratio	15
Effective compression ratio	15 / 14.1 / 12.7
Intake pressure [bar]	1.9
Engine oil & Coolant [°C]	85
NOx (target) [ppm]	< 40
Soot (target) [FSN]	< 0.2
mPRR (target) [bar/deg]	< 5
CoV of IMEP [%]	< 3

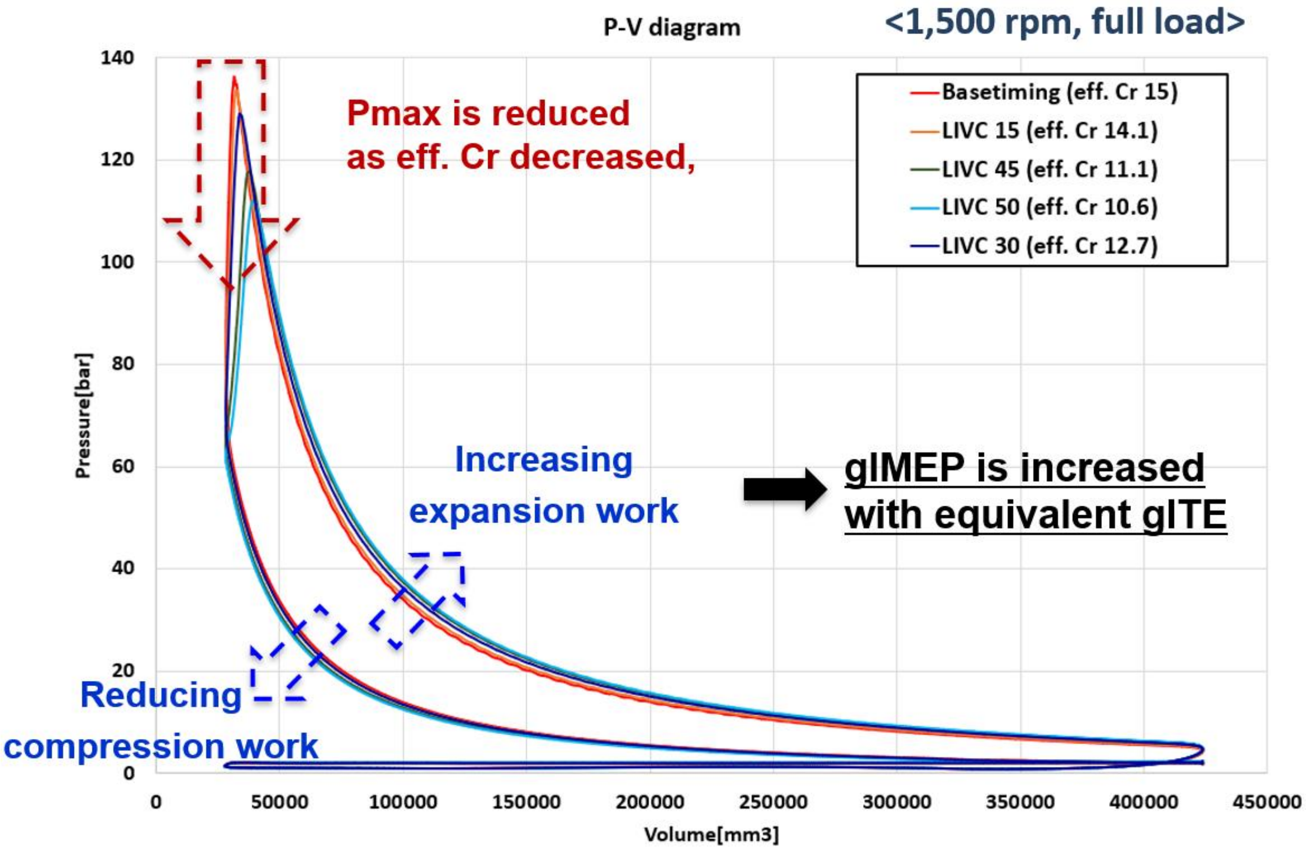

Figure 3.3.2 P-V diagram at 1,500 rpm, full load condition

Figure 3.3.3 shows the cylinder pressure and heat release rate with lower than effective CR 12.7 such as 11.1, 10.6 by applying LIVC 45 and 50. The detailed experimental conditions were listed in Table 3.3.2. If the effective CR is lower than 12.7 by further delaying the IVC timing, gIMEP can be increased, however, gITE tended to decreased due to too low combustion reactivity. At that time, even additional diesel injection was happened with retarded DIT to enhance the combustion stability, it was not able to compensate gITE. Too low combustion reactivity leaded to MFB 50 delayed such as 12.9° CA ATDC, 14.1° CA ATDC. In case of effective CR 10.6, soot emission was measured about 0.132 FSN due to maximum DIT retard limit. Therefore, the effective CR for increasing gIMEP while maintaining gITE at full load condition of the dual-fuel PCI shall be maintained around 12.7.

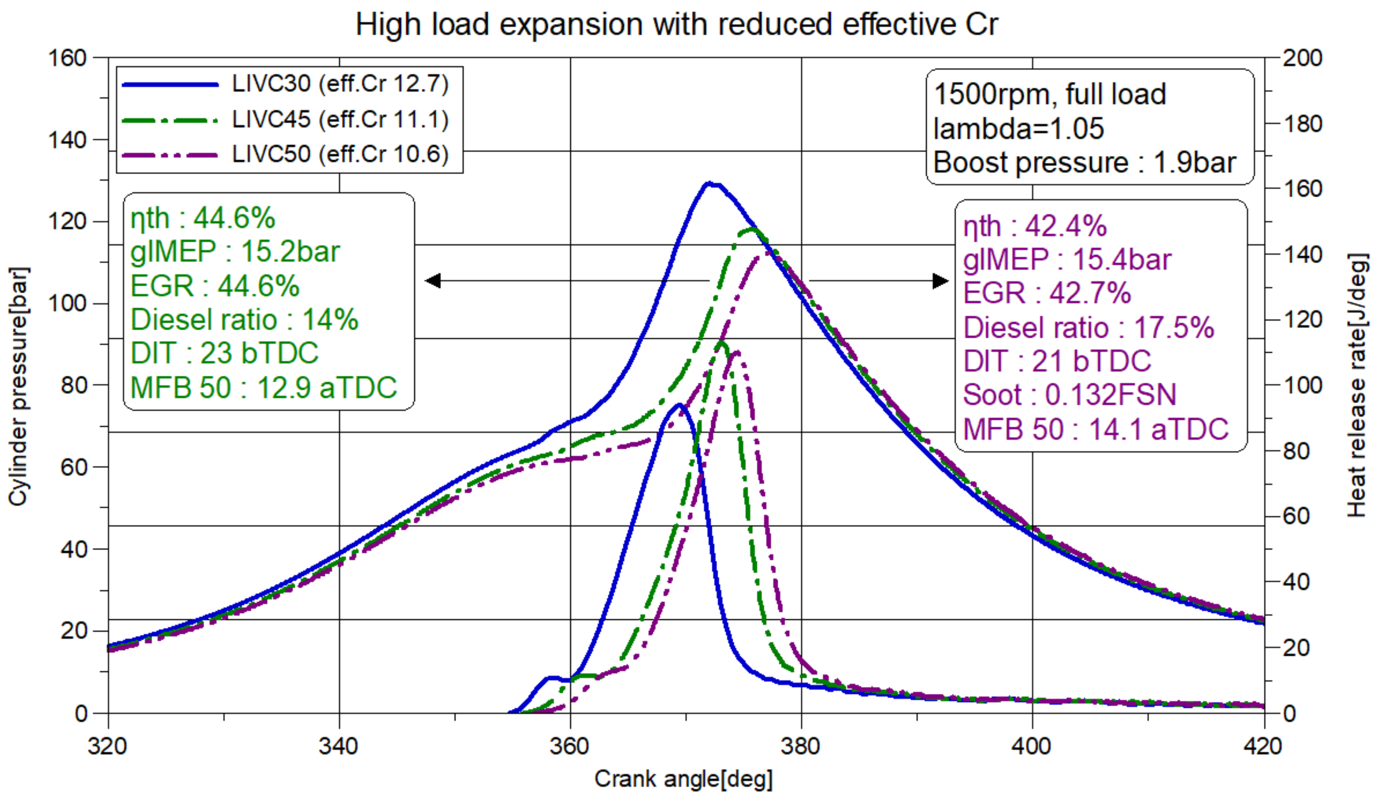

Figure 3.3.3 High load expansion with reduced effective CR 12.7 / 11.1 / 10.6 at 1,500 rpm

Table 3.3.2 Experimental conditions of high load expansion at 1,500 rpm – (2)

Description	Value
Engine speed [rpm]	1,500
gIMEP [bar]	Full load
Diesel injection pressure [bar]	450
Gasoline Injection pressure [bar]	6
IVO / IVC	8° CA BTDC / 66, 81, 86 CA ABDC
Geometric compression ratio	15
Effective compression ratio	12.7 / 11.1 / 10.6
Intake pressure [bar]	1.9
Engine oil & Coolant [°C]	85
NOx (target) [ppm]	< 40
Soot (target) [FSN]	< 0.2
mPRR (target) [bar/deg]	< 5
CoV of IMEP [%]	< 3

Based on the experiment results from high load expansion at 1,500 rpm, the effects of certain effective CR, 15, 12.2, 9.3 were investigated at 1,750 rpm. The cylinder pressure and heat release rate were demonstrated in Figure 3.3.4 at 1,750 rpm, full load condition and Table 3.3.3 listed the detailed experimental conditions.

The trends at 1,500 rpm, full load condition showed the same trends at 1,750 rpm. The most effective combustion appearance was when the effective CR was around 12.7 at 1,500 rpm, high load condition, and as a result of applying the effective compression ratio 12.2 at 1,750 rpm, gIMEP effectively increased from 14.5 bar to 15.4 bar while maintaining equal gITE, 46.5%. However, increase in gIMEP was not higher than 1,500 rpm experiment. It indicates that an additional experiment is necessary to find optimal effective CR for this positive effect. If the effective CR was further reduced, gIMEP can be increased about 0.6 bar but gITE tended to decreased from 46.5% to 44.6% even diesel ratio was increased with retarded DIT until maximum DIT retard limit based on soot constraint.

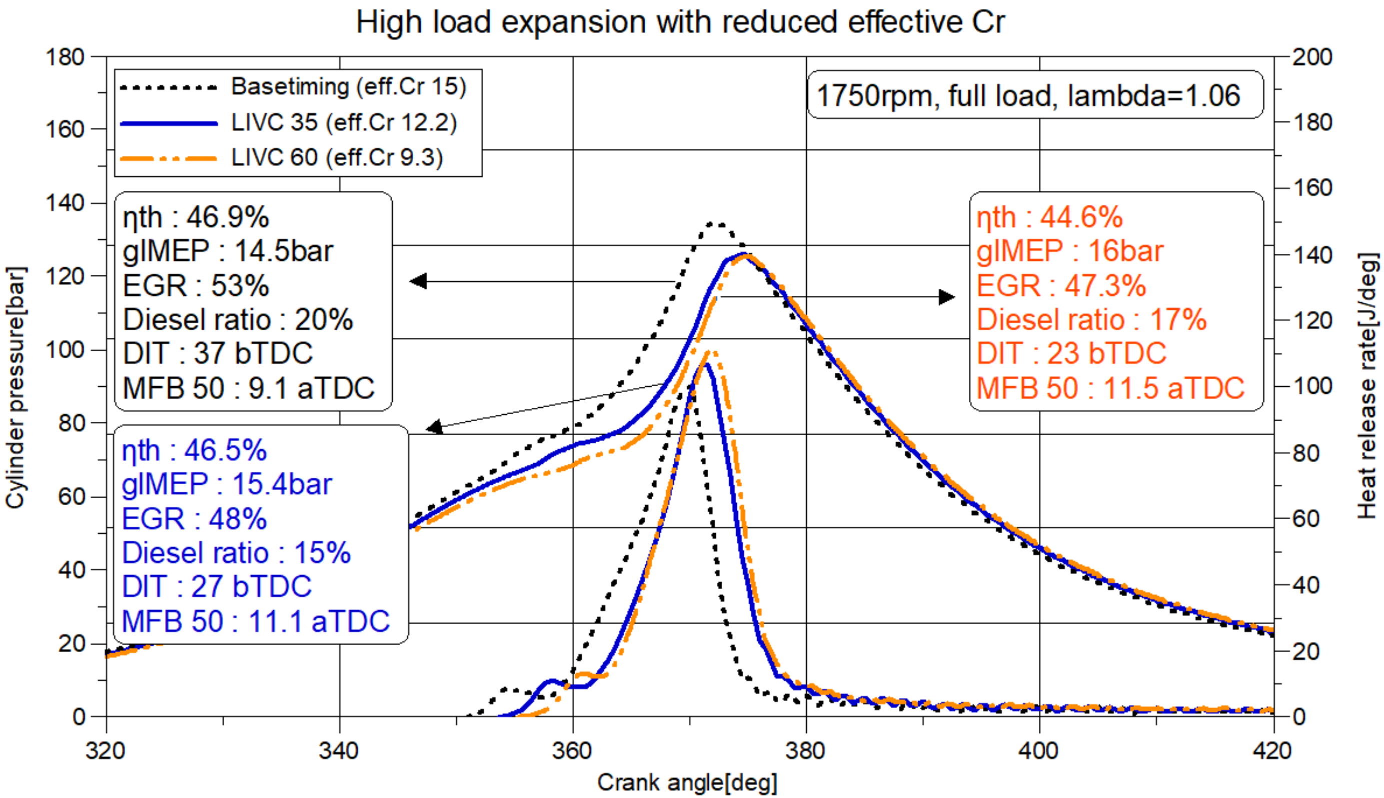

Figure 3.3.4 High load expansion with reduced effective CR 15 / 12.2 / 9.3 at 1,750 rpm

Table 3.3.3 Experimental conditions of high load expansion at 1,750 rpm

Description	Value
Engine speed [rpm]	1,750
gIMEP [bar]	Full load
Diesel injection pressure [bar]	450
Gasoline Injection pressure [bar]	6
IVO / IVC	8° CA BTDC / 36, 71, 96 CA ABDC
Geometric compression ratio	15
Effective compression ratio	15 / 12.2 / 9.3
Intake pressure [bar]	2.0
Engine oil & Coolant [°C]	85
NOx (target) [ppm]	< 40
Soot (target) [FSN]	< 0.2
mPRR (target) [bar/deg]	< 5
CoV of IMEP [%]	< 3

Chapter 4. Conclusions

In this research, high load expansion strategy by changing effective compression ratio was introduced on gasoline-diesel dual-fuel premixed compression ignition combustion. The effective CR was changed via late intake valve close timing (LIVC) strategy by using intake variable valve actuation (VVA) system.

For optimization of dual-fuel PCI, 1) gasoline-diesel substitution rate, 2) diesel injection timing (DIT) and 3) supplying EGR were applied as combustion control factors. The optimized combustion phase should satisfy the constraints listed in Table 2.2.2 regarding the engine-out exhaust emissions and mPRR. Among them, the data with the best thermal efficiency were used to compare the results of the experiments.

Firstly, dual-fuel PCI under low load operation was conducted with geometric CR 14 and 15 to find the lowest CR that shows higher thermal efficiency than diesel engine. The results show that dual-fuel PCI enable high thermal efficiency with near zero NOx and soot emissions. As geometric CR increased, EGR rate had to be increased to reduce the combustion reactivity and advanced DIT was effective. Advanced DIT decreased diesel local equivalent ratio which can reduce self-ignite ability of diesel. Both gITE from geometric CR 14 and 15 show high value as each 41.4% and 44.3%, respectively at 1,500 rpm, gIMEP 5.2 bar condition. At this time, gITE of corresponding EURO 6 diesel engine was 43.8%. Based on gITE under low load operating condition, geometric CR was selected as 15.

As the second part of the research, high load expansion experiments with various reduced effective CR via LIVC strategy were proceeded. For the experiment engine, the effective CR can be adjustable from 15 to 9.3 continuously. At 1,500 rpm, high load expansion experiment was conducted with effective CR 15, 14.1, 12.7, 11.1 and 10.6 cases. As the effective CR decreases, EGR rate can be reduced as mPRR decreases. It allows to inject additional gasoline fuel via margin of lambda (λ). And DIT was retarded to cover the combustion reactivity degradation by reduced effective CR. Also, diesel substitution was able to decreased because of a high diesel local equivalent ratio. Until effective CR 12.7 from 15, gIMEP was increase from 13.3 bar to 14 bar with the same gITE as 46%. This positive phenomenon was possible because the sum of the increase amount of the expansion work and the decrease amount of the compression work is greater than the work loss due to the cylinder peak pressure drop as the effective CR decrease. At too low the effective CR such as 11.1 and 10.6 cases, however, it was impossible to cover the low combustion reactivity by retarded DIT and increased diesel substitution rate. Even gIMEP can be increased due to allowable mPRR, gITE tends to decrease for these cases. Based on the results from 1,500 rpm, high load expansion experiments, certain effective CR were applied to 1,750 rpm. At 1,750 rpm, gIMEP also has been increased with equivalent gITE until a certain effective CR around 12.2. However, if effective CR was further reduced such as 9.3, gIMEP can be increased but gITE was decreased as same as the results of 1,500 rpm experiment. At 1,750 rpm, high load expansion experiment, gIMEP and gITE were increased from 14.5 bar, 46.9% to 15.4 bar, 46.5%.

Figure 4.1.1 demonstrated the overall gITE map according to rpm and gIMEP. By changing the effective CR, gIMEP can be increased about 10.5% at 1,500 rpm and 6.2% at 1,750 rpm while only 1% drop of gITE.

In this study, the results of applying geometric CR which shows higher gITE than diesel engine under low load operating condition of dual-fuel PCI were derived, and also expanding the high load limit by reducing effective CR via LIVC strategy was achieved. This research can contribute to the load expansion which is the main disadvantage of dual-fuel PCI as the next generation advanced combustion system.

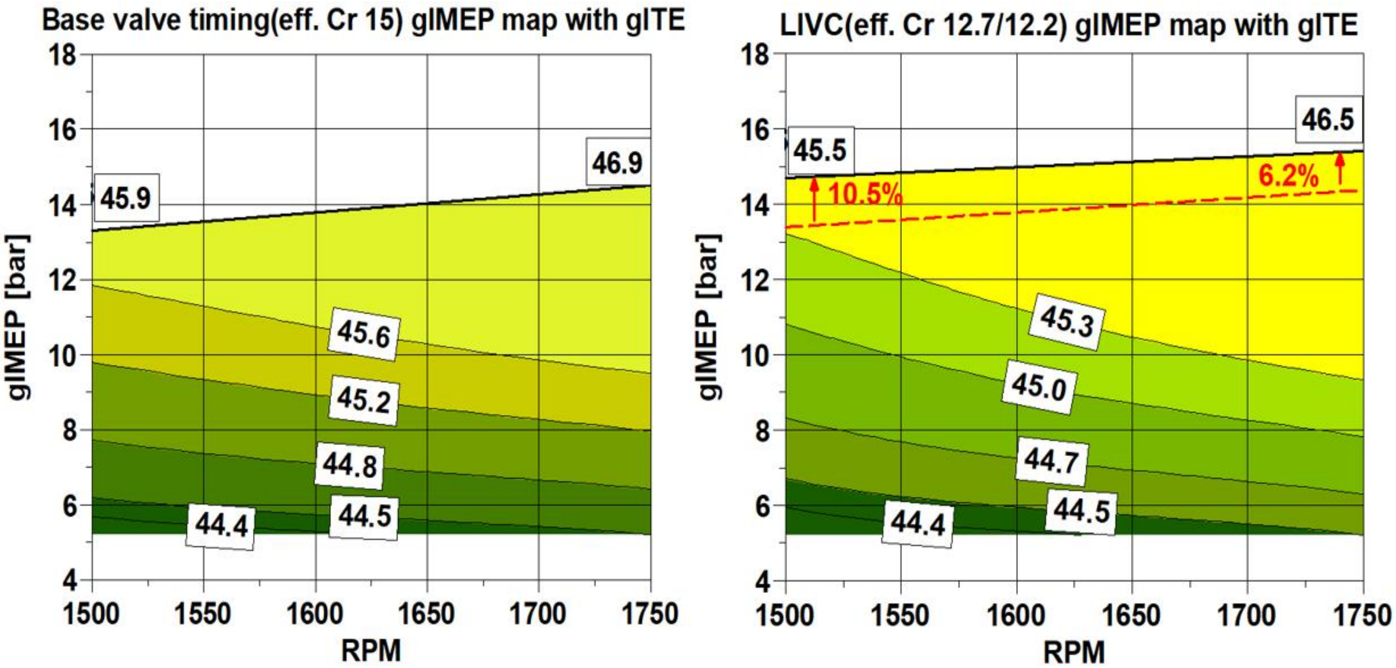

Figure 4.1.1 Comparison of gITE according to RPM and gIMEP between effective CR 15 and reduced effective CR

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국 문 초 록

가솔린-디젤 예혼합 융합연소 엔진에서 유효압축비 변경을 통한 부하 확장에 대한 연구

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2000 년대에 들어 전 세계적으로 지구온난화에 대한 관심이 높아지면서 이산화탄소 규제가 점차 강화되어 왔다. 특히, 내연기관이 이산화탄소 배출원 중 하나로 주목받으며 수송 분야에서 배출가스 규제가 더욱 까다로워졌다. 이에 가솔린 SI 엔진에 비해 높은 열효율 바탕으로 연료 소모가 적어 이산화탄소 배출에 유리한 디젤 압축 착화 엔진이 각광받기 시작하였다. 그러나 디젤 배출가스 조작 파문과 더불어 디젤 엔진의 대표적 배기 배출물인 질소산화물 (NOx) 과 입자상 물질 (PM)의 인체 유해성이 두드러지며, 운전 시험 모드인 WLTP 와 실도로 주행 배기 배출물 시험인 RDE 가 도입되었다. 이러한 규제는 배기가스 재순환 (EGR) 전략과 Lean NOx Trap (LNT), Selective Catalyst Reduction (SCR)을 통해 대응이 가능하지만, 향후 더욱 강화될 예정인 배기가스 규제에 선제적 대응이 필요하다.

따라서 디젤 압축 착화 엔진의 높은 열효율을 유지하면서, 낮은 수준의 배기 배출물을 선보이는 신연소 기술이 개발되었다. 대표적인 신연소 기술로는 가솔린 HCCI 와 디젤 PCCI 기술이 있으나, 연소의 제어가 불가능하다는 측면에서 한계를 보였다.

이러한 단점을 보완한 신연소 기술로서 융합 연소의 개발이 활발히 진행 중이다. 융합 연소는 2 가지 종류의 연료를 사용한 예혼합 압축 착화 방식으로 저배기 고효율 연소를 실현하고, 동시에 2 가지 연료의 다른 반응성을 통해 연료의 비율 및 분사 시기와 같은 다양한 전략들로 연소 제어가 탁월하다는 장점이 있다. 그러나 예혼합 압축 착화로 인해 노킹 및 가파른 압력 상승률로 소음, 진동 유발 및 내구성에 부정적인 영향을 미친다는 단점이 존재한다. 따라서 디젤 압축 착화 엔진 대비 최고 부하의 범위가 한정적이므로 부하 확장을 위한 연구 개발이 필요하다.

본 연구에서는 가솔린-디젤 예혼합 융합연소 엔진에서 유효압축비 변경을 통해 부하 확장 전략을 조사하였다. 유효압축비의 변경은 가변 흡기 밸브 액츄에이터(VVA) 시스템을 통해 흡기 밸브 닫힘 시기를 지연시킴(LIVC)으로써 제어 가능하였다. 저부하 운전 조건에서는 상용 디젤 엔진 대비 높은 열효율을 보일 수 있는 기계적 압축비 (Geometric CR)를 선정하기 위한 실험을 진행하였다. 유로 6 대상 엔진의 저부하 운전 조건에서 열효율은 43.8%을 보였다. 이때, 각각 기계적 압축비 14 와 15 를 적용하여 동일한 운전점에서 최적화 결과를 비교한 결과, 압축비 14 의 경우, 41.4%로 상용 디젤 엔진 대비 낮은 열효율을 보였으나, 압축비 15 는 44.3%로 상용 디젤 엔진 대비 높은 열효율을 보였다. 따라서 기계적 압축비는 15 로 선정한 후, 1,500 rpm 및 1,750 rpm 에서 최고 부하

확장 실험을 진행하였다. 기계적 압축비 15 를 바탕으로 흡기 밸브 닫힘 시기 지연을 통한 유효압축비 감소는 최대 9.3 까지 연속적으로 제어 가능하였다. 저속 최고부하 조건에서 유효압축비는 각각 14.1, 12.7, 11.1 및 10.6 을 적용하였다. 그 결과, 유효압축비 12.7 수준까지는 동등 열효율을 유지하면서 최고부하가 10.5% 증가하였고, 이보다 낮은 유효압축비에서는 최고부하가 더욱 증가하였으나 연소 반응성이 저하되어 열효율 또한 감소하는 경향을 보였다. 1,750 rpm, 최고부하 조건에서 또한 특정 유효압축비까지는 열효율을 동등하게 유지하며 부하가 확장되는 경향을 보였고, 유효압축비를 더욱 감소시킬 경우 부하 확장과 동시에 열효율이 감소하는 경향을 보였다.

주요어: 이종 연료, 융합 연소, 예혼합 압축 착화, 최고 압력 상승률, 압축비, 유효압축비

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공학석사학위논문

가솔린-디젤 예혼합 융합연소 엔진에서 유효압축비 변경을 통한 부하 확장에 대한 연구

Load Expansion by Changing Effective Compression Ratio on Gasoline-Diesel Dual-fuel Premixed Compression Ignition

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이 논문을 공학석사 학위논문으로 제출함

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기계공학부

김 기 홍

김기홍의 공학석사 학위논문을 인준함

2020 년 6 월

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부위원장 : ____________________ (인)

위 원 : ____________________ (인)

Abstract

Load Expansion by Changing Effective Compression Ratio on Gasoline-Diesel Dual-fuel Premixed Compression Ignition

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Internal combustion engines have attracted attention as one of the sources of carbon dioxide (CO_2) so that the emission regulations are becoming stringent. Diesel engines, which are advantageous for CO_2 emissions based on high efficiency, have begun to come into the spotlight. However, Volkswagen scandal with the human hazards of nitrogen oxides (NOx) and particulate matter (PM) which are representative exhaust emissions of diesel engines became remarkable, emission regulations based on WLTP and RDE are expected to continue to be severe.

The advantage of dual-fuel combustion as a new advanced combustion technology is that low exhaust emission level can be achieved while maintaining high thermal efficiency based on pre-mixed compression ignition and also the combustion can be controlled by using different reactivity of two types of fuel. The control of the combustion phase can be possible with various

strategies such as fuel substitution rates, fuel injection timing, EGR rate and so on. However, there is a disadvantage that pre-mixed compression ignition causes knocking and high pressure rise rate. So, the operating load range is limited compared to diesel engine. Thus, research for load expansion of dual-fuel combustion is necessary.

In this study, the high load expansion strategy was investigated by changing the effective compression ratio on gasoline-diesel dual-fuel premixed compression ignition engine. The effective compression ratio was adjusted by delaying the intake valve close timing via the intake valve actuator system. First of all, the experiments were conducted to find geometric compression ratio that could show higher thermal efficiency than conventional diesel engines under low load operating condition. The gross thermal efficiency of the EURO 6 engine was 43.8% under low load driving condition. At this time, the comparison of optimization results at the same operating point by applying geometric compression ratio of 14 and 15 respectively. In case of geometric compression ratio 14 showed a lower thermal efficiency of 41.4% while geometric compression ratio 15 showed a higher thermal efficiency as 44.3% than conventional diesel engine. Thus, base geometric compression ratio was applied as 15.

With geometric compression ratio 15, the high load expansion experiments were carried out with the various effective compression ratio at 1,500 and 1,750 rpm. The effective compression ratio through LIVC was continuously controllable from 15.0 to 9.3. At 1,500 rpm high load operating condition, the effective compression ratio 14.1, 12.7, 11.1 and 10.6 were applied respectively. As a result, the full load limit increased by 10.5% while maintaining the equivalent thermal efficiency up to the effective compression ratio of 12.7. And the full load limit can be increased further with the lower

effective compression ratio, but the thermal efficiency also tended to decrease due to reduced combustion reactivity. At 1,750 rpm, high load conditions with certain effective compression ratio based on 1,500 rpm experiment results, the load tended to expand with the equivalent thermal efficiency, and if the effective compression ratio was further reduced, the thermal efficiency tended to decrease at the same time as the load expansion.

Keywords: Dual-fuel, Dual-fuel combustion, Premixed compression ignition, Maximum pressure rise rate, Geometric compression ratio, Effective compression ratio

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Nomenclature

Greek Letters

η	efficiency
$\emptyset$	phi, overall equivalence ratio
λ	lambda, relative air/fuel ratio, inverse of Phi

Acronyms

AFR	air-fuel ratio
ATDC	after top dead center
BD	burn duration
BTDC	before top dead center
CA	crank angle
CDC	conventional diesel combustion
CI	compression ignition
CO	carbon monoxide
CO_2	carbon dioxides
CoV	coefficient of variation
CR	compression ratio
CVVD	continuous variable valve duration
CVVL	continuous variable valve lift
CVVT	continuous variable valve timing
DI	direct injection
DIT	diesel injection timing
DPF	diesel particulate filter
EGR	exhaust gas recirculation
EVC	exhaust valve close
EVO	exhaust valve open
FSN	filtered smoke number

gIMEP	gross indicated mean effective pressure
gITE	gross indicated thermal efficiency
HCCI	homogeneous charge compression ignition
HFR	hydraulic flow rate
HRR	heat release rate
IC	internal combustion
IVC	intake valve close
IVO	intake valve open
LNT	lean NOx trap
LTC	low temperature combustion
LTHR	low temperature heat release
MFB50	mass fraction burned 50 %
mPRR	maximum pressure rise rate
NEDC	new European driving cycle
NOx	nitrogen oxides
NVO	negative valve overlap
PCCI	premixed charge compression ignition
PCI	premixed compression ignition
PFI	port fuel injection
PM	particulate matter
ppm	particles per million
RCCI	reactivity controlled compression ignition
RDE	real driving emission
rpm	revolution per a minute
SCR	selective catalyst reduction
SI	spark ignition
SOC	start of combustion
SOI	start of injection
TDC	top dead center

THC	total hydrocarbon
WLTP	world harmonized light vehicle test procedure

Chapter 1. Introduction

1.1 Research Background

Since the early 2000s, the global Carbon Dioxide (CO_2) regulations are becoming stringent around the globe due to global warming so that CO_2 regulation in the transport sector, which is pointed as one of the emission sources of global warming, are also becoming strict such as Figure 1.1.1.

The diesel compression ignition (CI) engine has advantages in terms of CO_2 compared to gasoline spark ignition (SI) engine. However, due to diesel combustion nature of the diffusion flame characteristics, soot is formed at fuel rich region and nitrogen oxides (NOx) is produced at high temperature of combustion region [3].

After the facts that nitrogen oxides (NOx) and particulate matter (PM) are harmful to the human health were remarkable, emission regulations are being severed around the world. As a result, the test driving cycle has changed from NEDC (new European driving cycle) to WLTP (world harmonized light vehicle test procedure) since 2017 shown in Figure 1.1.2. Also, RDE (real driving emissions) which is the new test cycle on real road has been announced since 2017. On RDE test, the exhaust emission level from real road test should meet the regulations with conformity factor (CF) 2.1. From 2020, this CF decreases to 1.5 for PN and 1.43 for NOx emissions.

To meet emission regulations, after-treatment systems such as Lean NOx trap (LNT), diesel particulate filter (DPF) or selective catalytic reduction (SCR) have been introduced. In addition, advanced combustion technology has been studied to reduce CO_2, NOx and soot emissions in the engine itself. The most common technology is the low temperature combustion (LTC). The main strategy of LTC is premixing a significant of fuel through early fuel injection. Premixed air-fuel mixture makes low flame temperature based on decreased local equivalent ratio (∅) which inhibits NOx. And also, early fuel injection which enables long ignition delay (I.D.) contributes soot reduction. With mentioned advantages regarding emission reduction, LTC is possible to maintain high thermal efficiency similar to conventional diesel combustion (CDC) based on the compression ignition of premixed air-fuel mixture due to low heat losses [4].

The most typical LTC strategies are the diesel premixed charge compression ignition (PCCI) and the gasoline homogeneous charge compression ignition (HCCI). In the PCCI strategy, partial amount of diesel is premixed to reduce soot through early injection and the rest of fuel is injected around TDC to control the combustion phase such as CDC. The flame temperature in the PCCI is lower than CDC, however, it is still high enough to form large amount of NOx. Thus, EGR is supplied to control NOx emission by lowering the flame temperature and O2 concentration. In the view of soot emission, the premixed fuel ratio should be increased. So, PCCI has a problem that the combustion phase control by small amount of main injection around TDC is limited [5].

HCCI makes a well-premixed mixture condition through early injection of all amount of fuel. The combustion is occurred by simultaneous multiple auto-ignition at the air-fuel mixture in cylinder during the compression stroke. The well-premixed condition guarantees very low level of soot and NOx emissions and very short combustion duration offers high thermal efficiency. But knocking is caused by the high volatility of gasoline, which is the main fuel used in HCCI, and combustion phase control is also an issue [6].

Reactivity controlled compression ignition (RCCI) is appropriate method in terms of the combustion control, low level emissions and high thermal efficiency. The low reactivity fuel (e.g. gasoline, natural gas, propane and etc.) is premixed by early injection during intake stroke and the high reactivity fuel such as diesel is injected as an ignition source during the compression stroke. Then, the combustion can be controlled by the reactivity stratification of two fuels and the equivalence ratio [7,8].

However, there is a disadvantage that pre-mixed compression ignition causes negative effect on durability of engine such as knocking and high pressure rise rate. Therefore, the operating load range is limited compared to diesel compression ignition engine. Thus, research for load expansion of dual-fuel combustion is necessary.

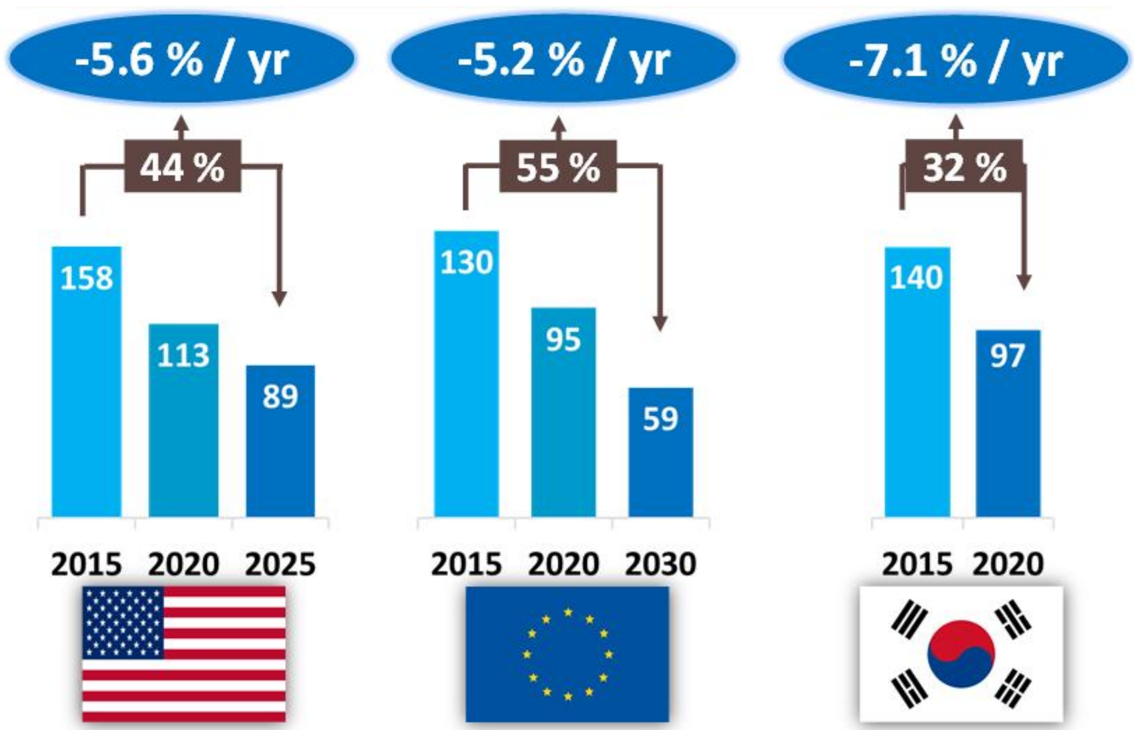

Figure 1.1.1 CO_2 regulation in the transport sector [1]

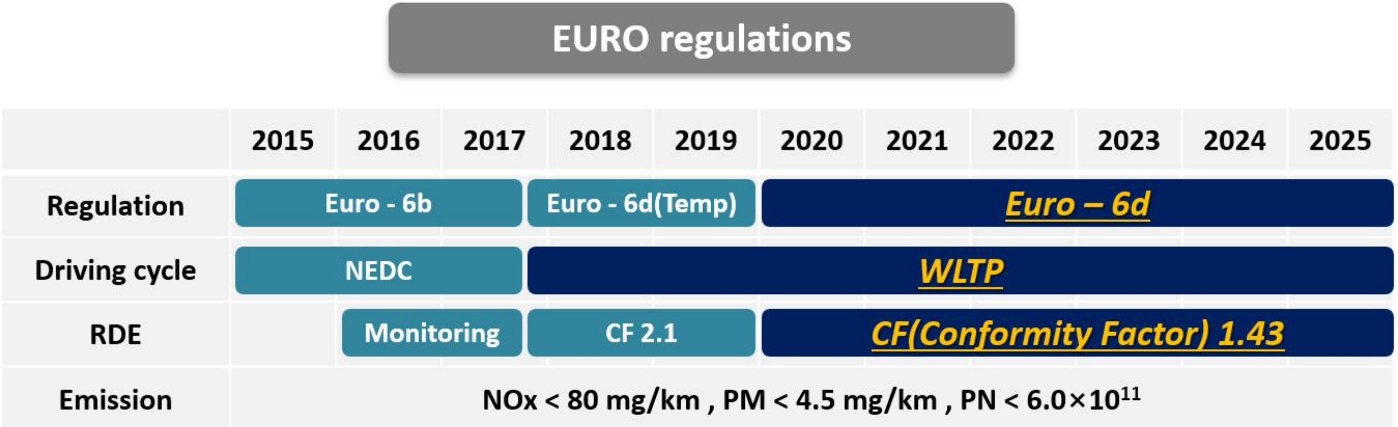

Figure 1.1.2 EURO emission regulation with driving cycle [2]

1.2 Previous Research

RCCI has a wide variety of control variables in that it uses two fuels and can combine the reactivity of the two fuels. Nieman et al. compared the results of various loads with CNG-diesel dual-fuel combustion in heavy-duty engines. This study found that the dual-fuel combustion using CNG has a lower PRR than gasoline and was beneficial for load expansion [9]. Kang et al. worked on the combustion stability, exhaust emissions level under various operating strategies for propane-diesel dual-fuel combustion [10]. In the similar way, Reiter and Kong investigated the combustion and exhaust emissions characteristics through dual-fuel combustion using ammonia, which is carbon-free and more advantageous in terms of production and handling [11]. In addition, Ashok et al. analyzed the effects and characteristics of LPG-diesel dual-fuel combustion [12]. These studies show that difference in the chemical properties between the fuels cause different combustion, emission characteristics and thermal efficiency.

The ratio of the two fuels in the dual-fuel combustion is a factor used as a representative operating strategy. Kokjohn et al. investigated optimal fuel proportion experiment that could control combustion phase with reasonable PRR for part and high loads [13]. Hanson et al. studied fuel fraction effect under low load [14]. They found combustion phase was easily controlled by varying the fuel fraction. Also, high reactivity fuel such as gasoline+2-EHN shows shorter combustion duration that gasoline-diesel combustion. Since combustion can be controlled by the ratio of two fuels with different reactivity, it is reasonable that the combustion can be controlled through the injection type (i.e. PFI or DI) and diesel injection timing. Garcia et al proposed that a fully premixed charging, a partial premixed charging by multiple and late injection, diffusive single injection around TDC are the best ways to obtain optimized combustion at in the order of low, mid and high loads [15].

Introducing EGR is also an effective way in reducing reactivity of combustion reaction in RCCI along with the emission reduction. Splitter et al. conducted experiment to investigate the effect of EGR both gasoline-diesel and E85-diesel dual-fuel combustion from mid to high loads. They demonstrated that EGR rate were increased to maintain low peak pressure and pressure rise rate toward high loads [16]. Also, Chu et al. showed that supplying EGR is effective to obtain reasonable PRR and low emission level [17].

At most researches have commonly demonstrated, dual-fuel combustion has low level of exhaust emissions. And many researchers seek various strategies for combustion control and load expansion while maintaining a high level of efficiency with stable combustion. The fuel reactivity control, fuel proportion, injection strategy and EGR mentioned in all the literature surveys enable combustion control and operation at various loads. However, the high PRR based on premixed mixture combustion still limits expansion toward high loads than CDC operating range.

1.3 Introduction of Dual-fuel PCI strategy

Based on the previous researches, RCCI has many advantages compared to other LTC strategies. Especially, the most advantageous characteristic is the combustion controllability with various operating strategies. However, the combustion control concept of RCCI has limitation on real operating condition. On real operating condition, for example, such as rapidly changed from high load to low load operation in transient mode, the overall combustion reactivity tends to be reduced even if the diesel ratio is increased because the large amount of residual EGR will be remained in cylinder.

Dual-fuel PCI (Premixed Compression Ignition), which was introduced by Inagaki et al [18], has better the combustion controllability compared to RCCI. Refer to Figure 1.3.1, low reactivity fuel is injected with fresh air and EGR during the intake stroke. After that, diesel direct injection is happened during the compression stroke. The major differences between RCCI and Dual-fuel PCI are diesel injection strategies such as diesel injection timing (DIT), single or multiple injection methods and equivalence ratio. As one of them, DIT is happened around 20 to 40 °CA BTDC, which is relatively retarded compared to RCCI. This strategy enables not only control the combustion phase but also exhausts low level of emissions by premixed air-fuel mixture. The advantages of Dual-fuel PCI are listed on Figure 1.3.2. At the ignition process, auto-ignition occurs at diesel rich area, and the power is generated during the expansion stroke with low reactivity fuel's premixed combustion. For the effects of control factors on combustion are described in chapter 3.1.

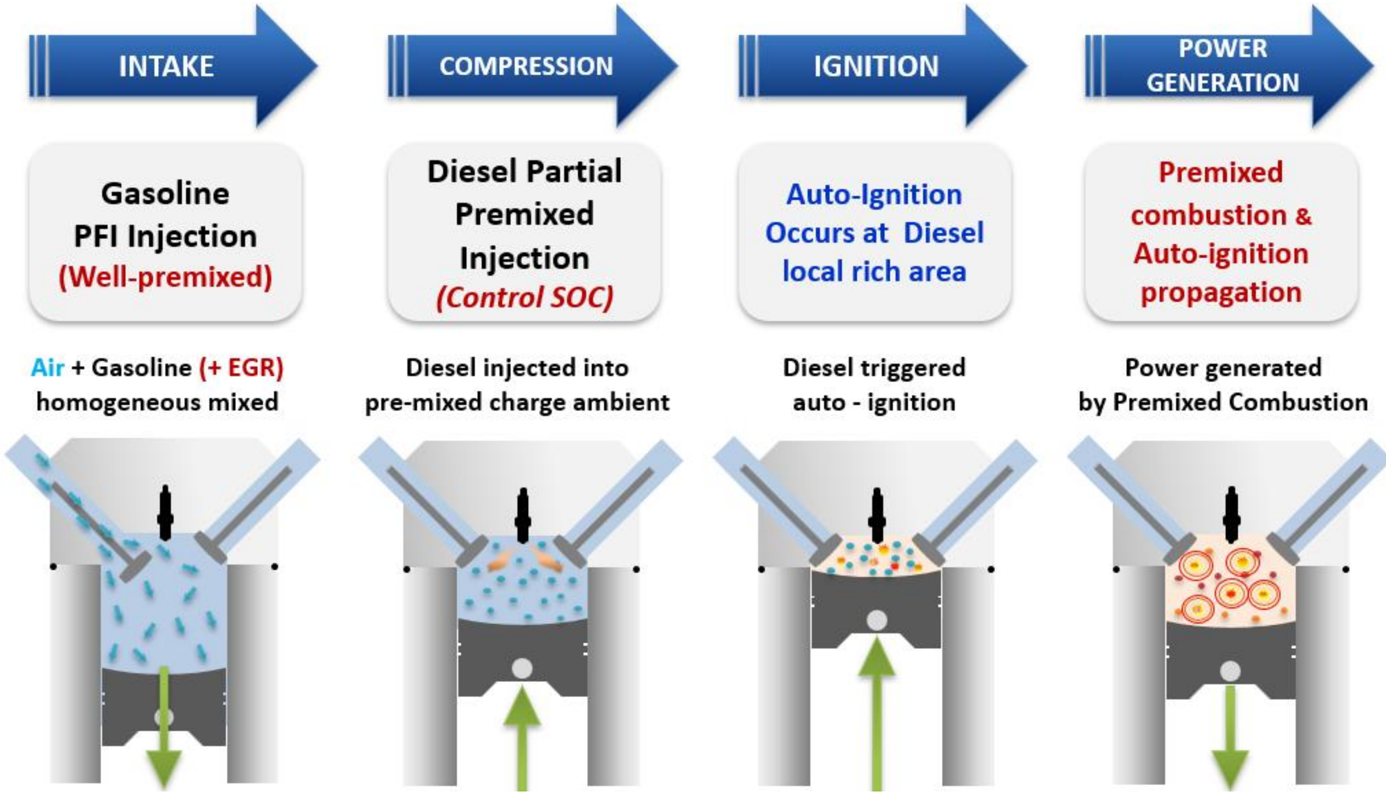

Figure 1.3.1 Process and principle of Dual-fuel PCI

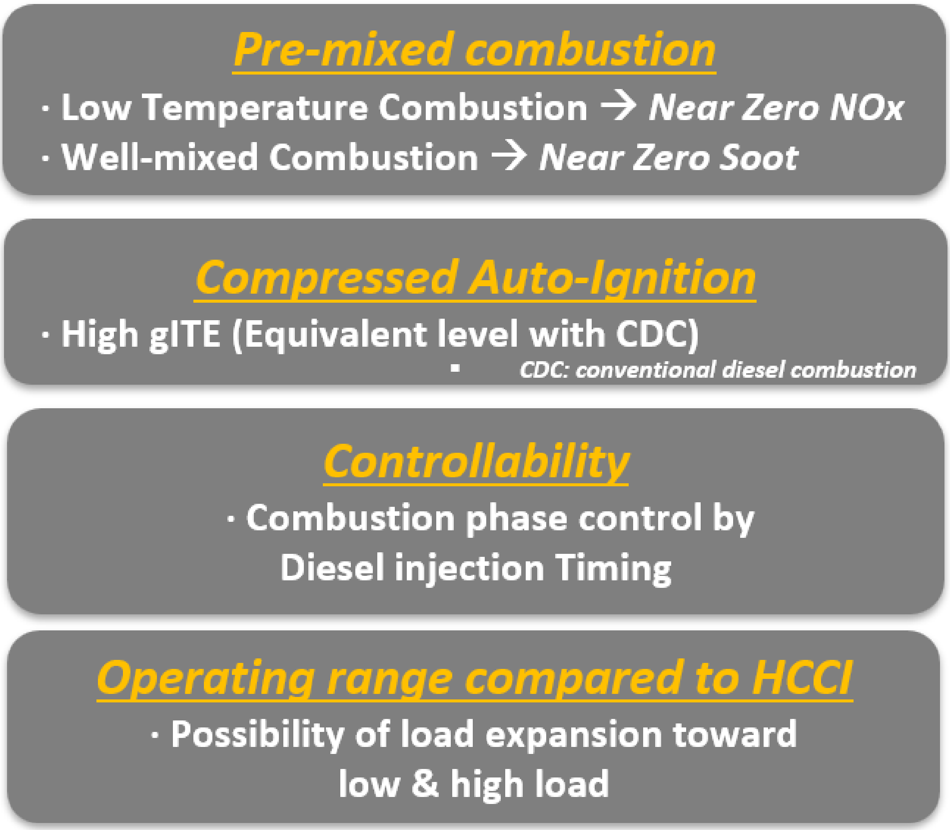

Figure 1.3.2 Merits of Dual-fuel PCI

1.4 Research Motivation and Objective

Dual-fuel PCI has the better combustion controllability compared to RCCI, however, the limitations still exist for high load expansion due to high PRR.

Seoul National University-Automotive laboratory has investigated the optimal hardware which affects the combustion characteristics for dual-fuel combustion. As one of them, which geometric compression ratio (CR) is more suitable for dual-fuel PCI was studied. The geometric CR is kinetic control variables on combustion. Such as diesel engines, high geometric CR around 16 is normally adopted. Dual-fuel PCI is a combustion strategy that is operated based on compression ignition of high reactivity fuel. So, the geometric CR around 16 and 17 are selected similar to diesel engine in most dual-fuel combustion research. Based on SNU-Automotive laboratory research results, with geometric CR around 16, it has high thermal efficiency while maintaining low level of emissions. However, it is impossible to expand high load due to high PRR. With low geometric CR around 14, it has potential for load expansion compared to high CR. In this case, however, thermal efficiency is lower than diesel engine under low load operation. To sum up, it demonstrated that low CR is suitable for the expansion of the high load and high CR is advantageous in terms of high thermal efficiency. The problem is that geometric CR is one of the hardware design factors so that it cannot be adjustable at engine. Thus, there is limit to choose between efficiency and load expansion.

Recently, Miller cycle, which has the longer expansion stroke than the compression stroke, has become common technology in order to improve the fuel consumption. Miller cycle of the past was implemented through a complex mechanical structure design, but is now simply controllable through valve timing control technology via an early intake valve close timing (EIVC) or late intake valve close timing (LIVC). EIVC or LIVC strategies reduce the intake charging air-fuel mixture mass and the pressure at start of combustion (SOC). Finally, it results in adjusting the effective CR.

Therefore, the objective of this research is to achieve high thermal efficiency compared to diesel engine under low load operation and high load expansion with LIVC strategy which can reduce the combustion reactivity through the low effective CR toward high load operation.

Research process were as following;

1) Dual-fuel PCI experiments with geometric CR 14 and 15 were conducted to find which CR has higher thermal efficiency than diesel engine at 1,500 rpm, gIMEP 5.2 bar.

2) Dual-fuel PCI high load operation with various effective CR were done to investigate proper effective CR which can expand gIMEP while maintaining the thermal efficiency.

Chapter 2. Experimental Apparatus

2.1 Experimental Setup

A single cylinder diesel engine based on EURO 6 standard was used for this work. Detailed engine specifications are shown in Figure 2.1.1 and Table 2.1.1. The diesel fuel can be directly injected as 450 bar with a BOSCH solenoid injector which can pressurize up to 1,800 bar with a common rail system. Two solenoid gasoline injectors were installed at intake port. The pressurized gasoline fuel as 6 bar was injected into the cylinder with fresh air and cooled EGR. The air and EGR were boosted by a supercharger and then cooled by intercooler. So, the temperature of fresh air-EGR mixture was kept around 27-30℃. The boosted intake pressure was determined based on EURO 6 diesel engine operating condition. The fuel flowrates were measured by OVAL Altimass CA001, Coriolis type flowmeter, for each diesel and gasoline. The temperature of engine oil and coolant was maintained at 85℃. The geometric CR can be changed by installing shim plate between cylinder block and cylinder head. For this experiment, geometric CR 14 and 15 were adopted. Piston bowl geometry was bathtub shape. Most studies have shown that the bathtub shape piston geometry contributes to the improvement of thermal efficiency due to low heat loss by reducing the inside surface area of the combustion chamber. The concentration of carbon monoxide (CO), carbon dioxide (CO_2), oxygen (O_2), nitrogen oxide (NOx) and total-hydrocarbon (THC) were measured at a HORIBA, MEXA 7100DEGR, an exhaust gas analyzer. The soot was measured by AVL, 415S, a smoke-meter. The schematic diagram of experimental apparatus is shown in Figure 2.1.2.

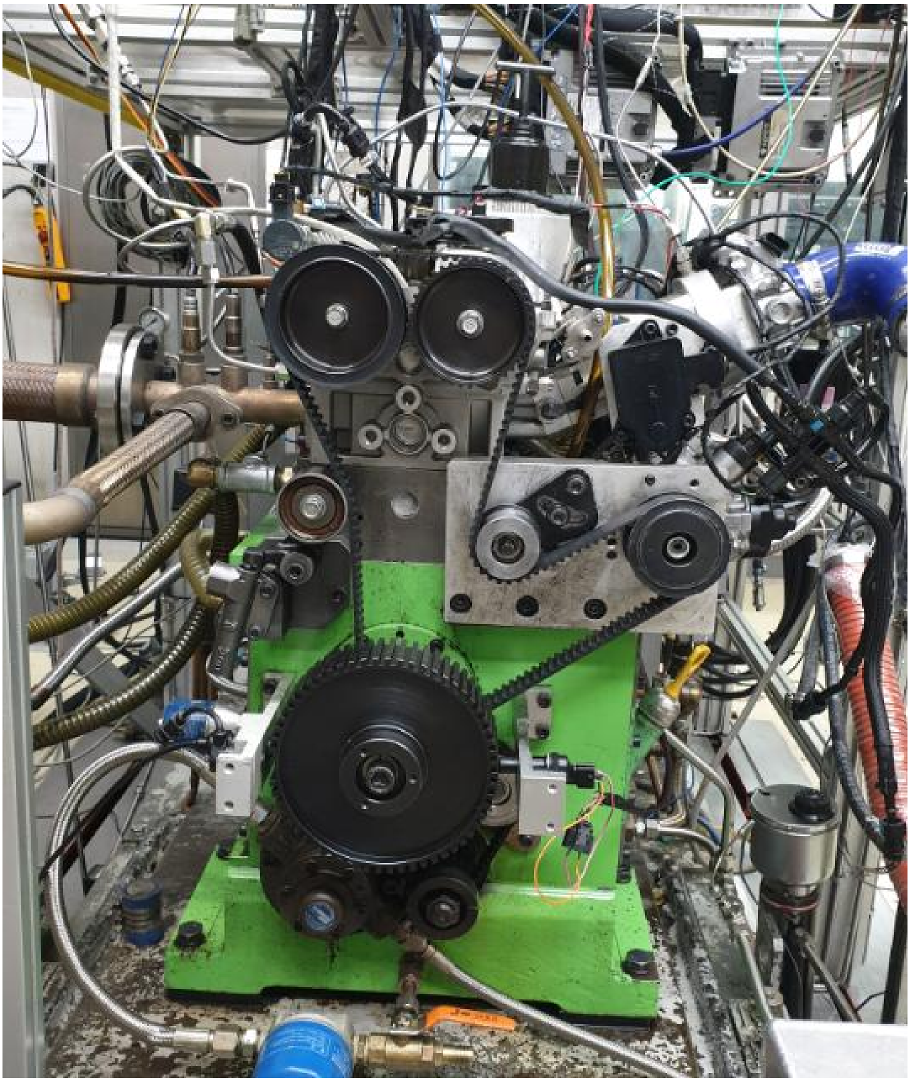

Figure 2.1.1 Dual-fuel Single Cylinder Engine

Table 2.1.1 Engine specifications

Engine type	Single cylinder (four-stroke)
Displacement (cc)	395
Bore ×Stroke (mm)	77.2×84.5
Connecting rod (mm)	140
Compression ratio (-)	14/15
IVO/IVC (Without LIVC)	BTDC 8°/ ABDC 36°
EVO/EVC	BBDC 46° /ATDC 4°
Diesel injection system	Solenoid injector
Injection pressure (bar)	450
No. of injector nozzle holes	8
Cone angle (deg)	149

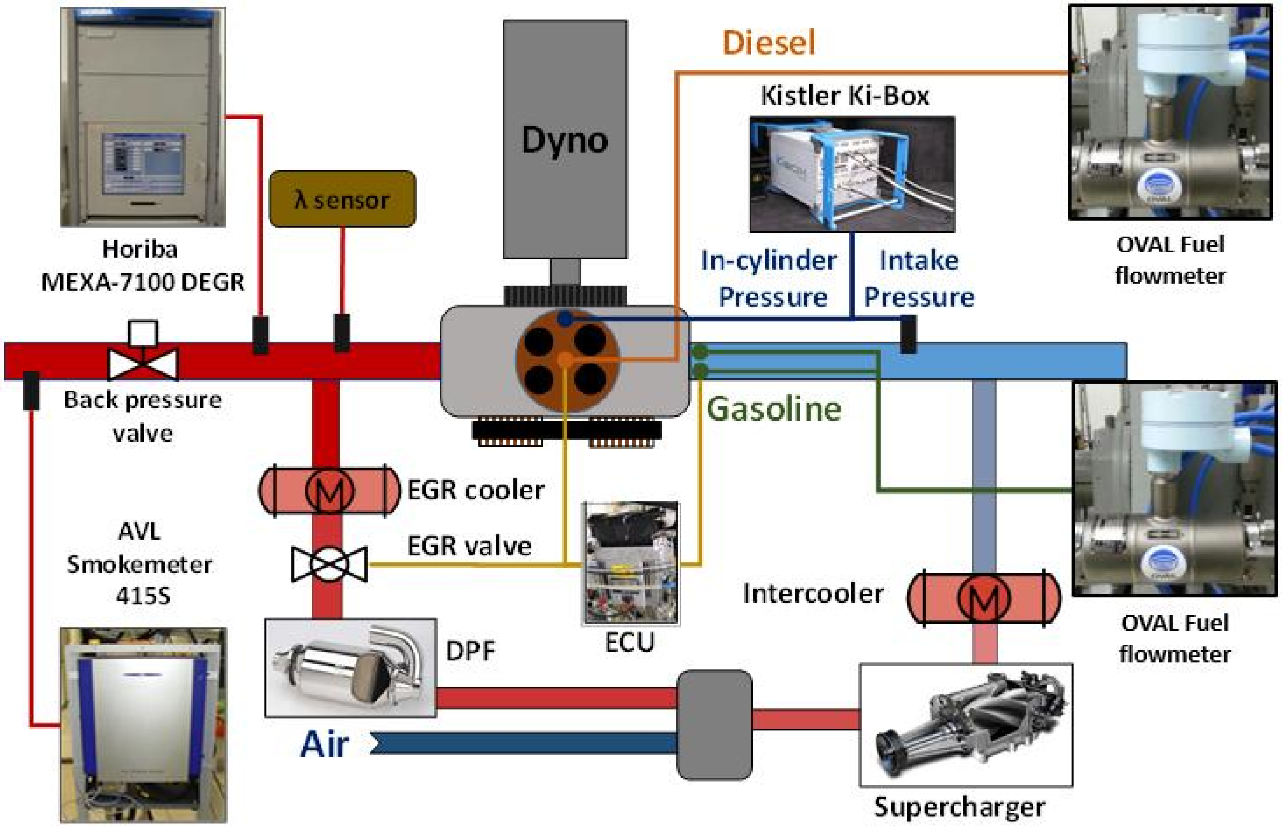

Figure 2.1.2 Schematic diagram of experimental apparatus

Table 2.1.2 Specifications of dynamometer

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

Table 2.1.3 Specifications of fuel flowmeter

Item	Specification
Manufacturer	OVAL
Measurement principle	Coriolis Type
Model	CA001
Range of flow amount (kg/h / max)	0~9
Range of flow amount (kg/h / min)	0~0.45
Maximum allowed pressure (MPa at 20℃)	15
Measurement uncertainty (%)	<0.2
Operating temperature (°C)	-200 ~ +200

Table 2.1.4 Specifications 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

2.2 Fuel properties and substitution rate of dual-fuel

Diesel and gasoline fuel were used in this experiment and properties of fuel are presented in Table 2.2.1. The substitution rate of fuel in dual-fuel combustion is calculated by considering low heating value (LHV) as shown in equation (1).

$$x_{gasoline}(\%) = \frac{\dot{m}_{gasoline} * LHV_{gasoline}}{\dot{m}_{diesel} * LHV_{diesel} - \dot{m}_{gasoline} * LHV_{gasoline}} \quad (1)$$

Table 2.2.1 Properties of diesel and gasoline

Properties	Diesel	Gasoline
Chemical formula	$C_xH_{2.0x}$	$C_xH_{2.0x}$
Density (g/cm^3)	0.831	0.724
Low heating value (MJ/kg)	42.5	42.8
Cetane/Octane number	54 (CN)	91 (RON)
Auto-ignition temperature (K)	483	520-553
Stoichiometric ratio of AF (wt.%)	14.6	14.5

2.3 Experimental Condition

In this study, the dual-fuel combustion was optimized with the various combustion control parameters, such as fuel proportion rate between diesel and gasoline, diesel injection timing and cooled LP-EGR rate. Firstly, the effects of these parameters were studied.

To investigate fuel substitution effect, the experiment was conducted from 0% gasoline fuel (=100% diesel ratio) to the maximum amount of gasoline that can be ignited with stable combustion. As gasoline increases, diesel proportion is reduced followed by Eq (1). And the experiment of diesel injection timing (DIT) was proceeded under dual-fuel combustion condition to study the effect of diesel injection timing as a trigger of auto-ignition on combustion control. In this case, the high level of emissions and high pressure rise rate (PRR) could be occurred due to absence of EGR supplied. So, supplying EGR was considered to be necessary to obtain maximum efficiency through optimization of the dual-fuel combustion such as shifting combustion phase, reducing PRR and control of emission level.

To find the high thermal efficiency than diesel engine under low load operation, dual-fuel PCI experiments with geometric CR 14 and 15 were conducted at 1,500 rpm, gIMEP 5.2 bar. Then, the high load expansion experiment was carried out by applying the LIVC strategy on the geometric CR determined in the low load operation experiment. The engine-out emission of dual-fuel PCI should meet the constraints listed in Table 2.2.2.

Table 2.2.2 Emission constraints

Operating Condition	mPRR [bar/deg]	CoV of IMEP [%]	NOx [ppm]	Soot [FSN]
Low load (1,500 rpm/ gIMEP 5.2 bar)	5	3	40	0.2
High load (1,500 & 1,750 rpm/ full load)	10			

2.4 Variable Valve Actuation System and effective CR

Engine technology has made remarkable progress and a variety of technologies have been focused for efficiency improvement, fuel injection method, engine downsizing with turbocharger and the next generation engine such as HCCI, PCCI, RCCI. Variable valve actuation system is one of the important roles as the key technology that benefits both the performance and emission reduction.

The state-of-the-art VVA system is continuous variable valve duration (CVVD) system. It can control the valve opening duration so that it performs the role of continuous variable valve timing (CVVT) and continuous variable valve lift (CVVL) [21].

The key principle of CVVD system is to use eccentricity. The dis-assembly drawing of CVVD is on Figure 2.4.1. In case cam shaft is rotated, cam shaft slot on cam shaft is rotated and in order of roller, cam slot and cam are rotated. In the view of cam knob, it is rotated by cam slot. The fixed point is cam knob center and cam slot can be shift upward and downward during rotation. If cam slot is close from cam shaft center, the distance for rotation is shorter so that duration will be decreased. On the other hand, cam slot is far from the shaft center, the distance for rotation is long situation which makes duration increased [22].

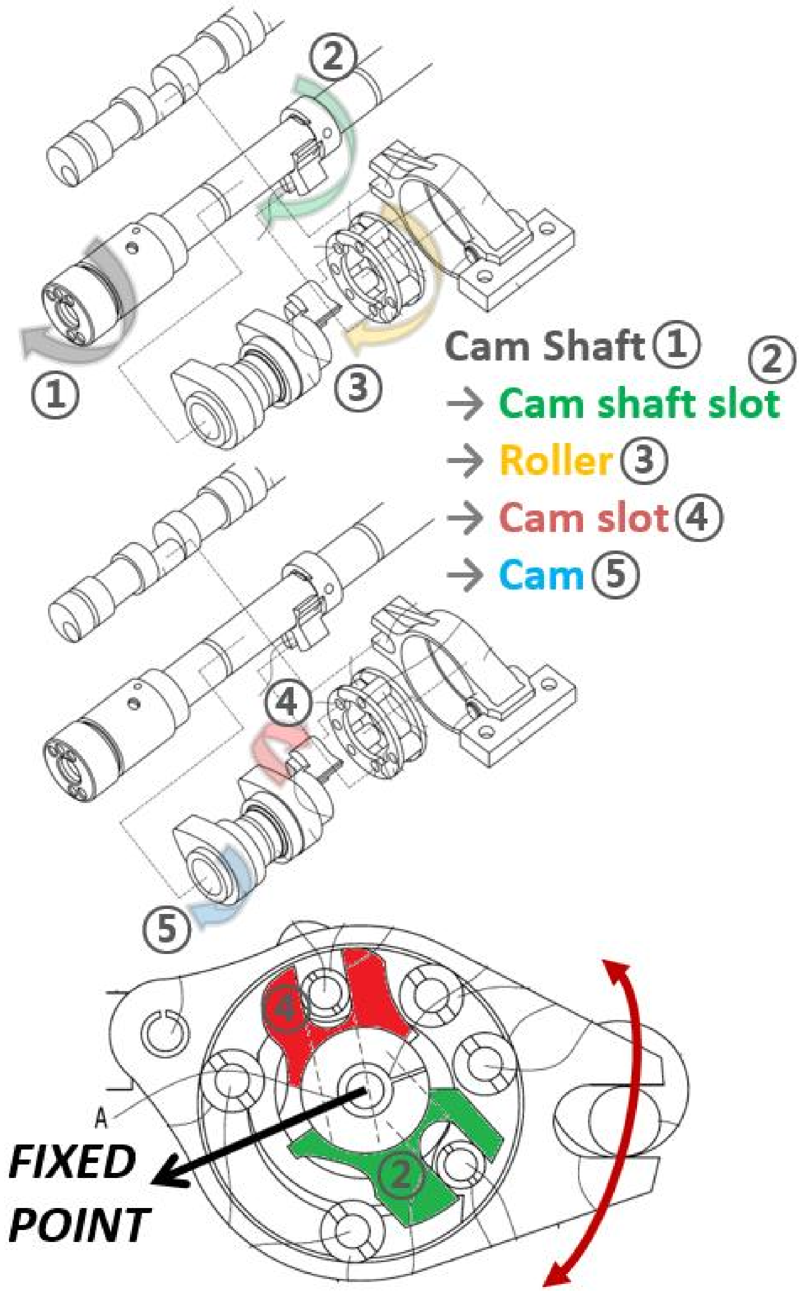

Figure 2.4.1 Disassembly and sectional drawing of CVVD [22]

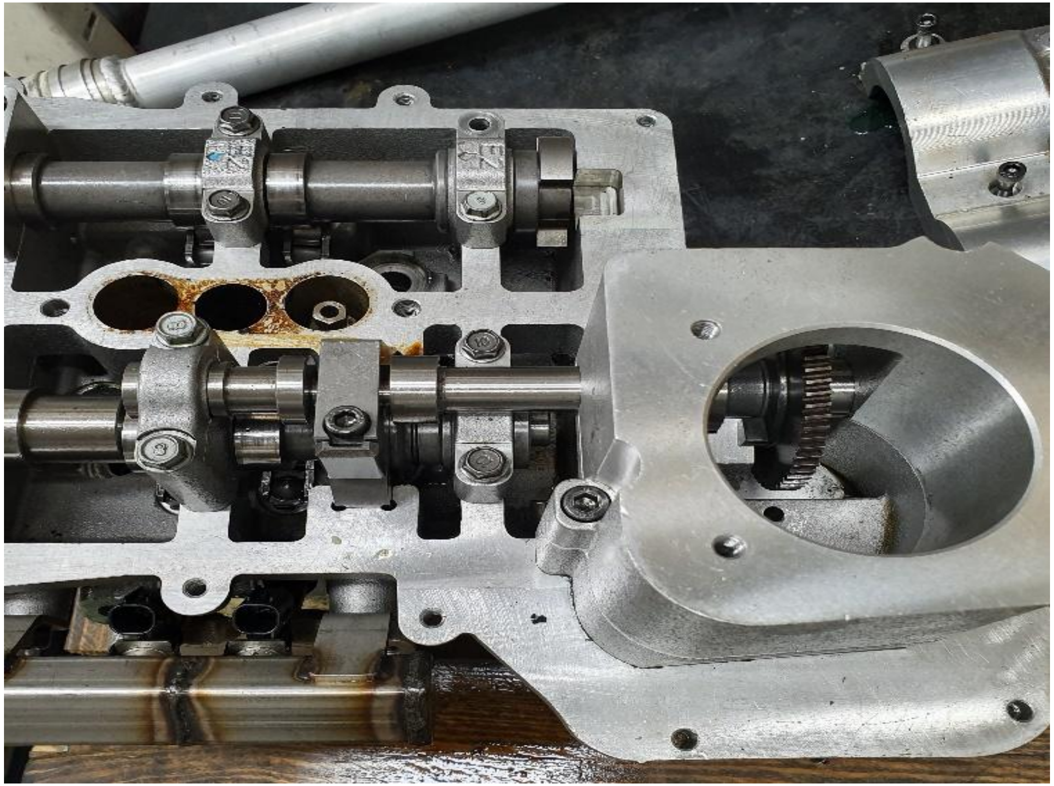

Figure 2.4.2 CVVD installation on dual-fuel single cylinder engine

Figure 2.4.2 shows installed CVVD system which was supported by Hyundai Motor Company. CVVD system on dual-fuel single cylinder engine can retard the intake valve close timing by 60° with fixed intake valve open timing such as Figure 2.4.3. If the intake valve close timing is delayed, the duration of opening in the compression stoke increases. It causes reduced effective CR. Figure 2.4.4 shows the cylinder peak pressure at firing TDC along the LIVC cases. The calculated effective CR according to LIVC cases is shown in Table 2.4.1.

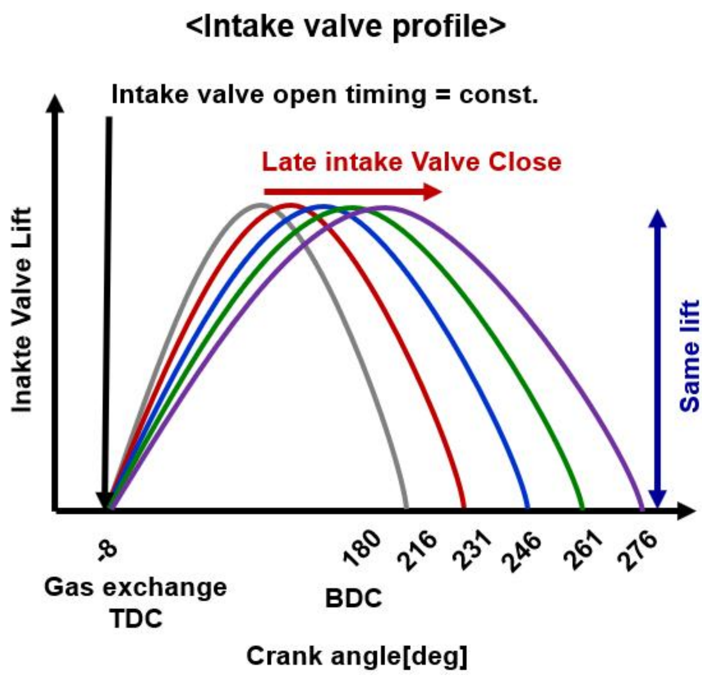

Figure 2.4.3 Intake valve profile with CVVD

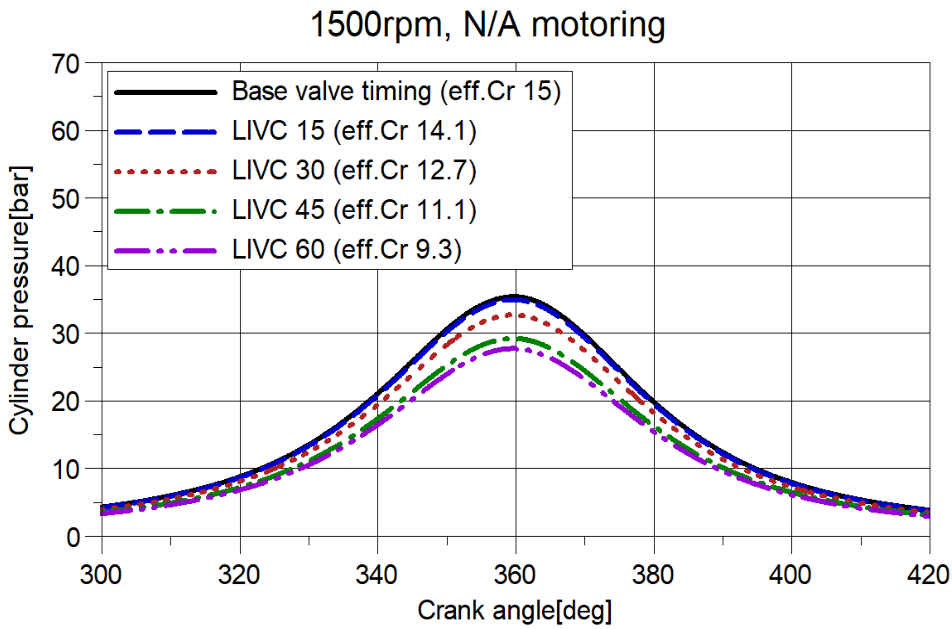

Figure 2.4.4 Cylinder peak pressure at firing TDC along the LIVC cases

Table 2.4.1 Calculated effective CR according to LIVC cases

IVC timing →	Base	LIVC 15	LIVC 30	LIVC 45	LIVC 50	LIVC 60
Effective Cr :	15.0	14.1	12.7	11.1	10.6	9.3

Chapter 3. Experimental Results and Discussion

3.1 Optimization of Dual-fuel PCI combustion

This section introduced the optimization strategy of dual-fuel PCI and studied about the effects of control factors, gasoline substitution rate, DIT and EGR rate.

To optimize the dual-fuel PCI combustion, gasoline substitution rate was increased as first step while maintain target gIMEP. At that time, soot emission is over 0.2FSN. Thus, DIT should be advanced to decrease soot emission and achieve the maximum thermal efficiency with same amount of fuel injections. Once gIMEP approached target value with gasoline injection and advanced DIT, MFB 50 should be controlled for the best thermal efficiency point such as 6~15° CA ATDC and NOx emission is needed to satisfy the constraint, 40ppm. As a result, for the final step, EGR should be supplied until NOx meets 40ppm. During the final step, EGR makes the combustion reactivity weaker so that MFB 50 may be retarded again. So, the combustion optimization should proceed by minor tuning the gasoline substitution rate and DIT adjustment. Based on this process, it can be possible to find the dual-fuel PCI combustion point which shows the best thermal efficiency while satisfying the all constraints listed in Table 2.2.2. The schematic diagram of dual-fuel PCI combustion process is described in Figure 3.1.1.

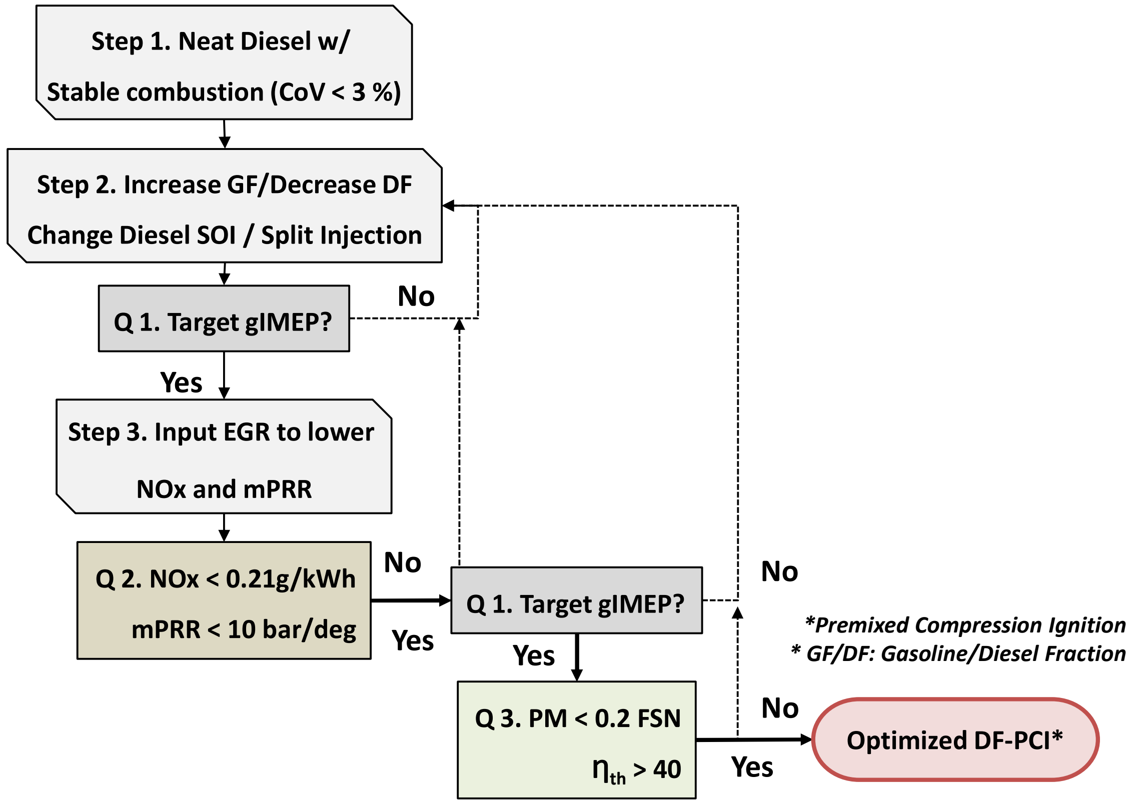

Figure 3.1.1 Schematic diagram of optimization process for Dual-fuel PCI combustion [19]

Gasoline substitution rate is one of the representative control factors on dual-fuel combustion. Figure 3.1.2 shows the cylinder pressure and heat release rate according to gasoline substitution rate at 1,500 rpm, gIMEP 5.2 bar. The cylinder pressure graph was retarded by increasing gasoline proportion. At that time, gIMEP increased since the combustion phase was retarded after TDC. This retard of combustion phase is due to the low reactivity fuel, gasoline, reduces the auto-ignition tendency of air-fuel mixture. With high ratio of diesel injection rate, the shape of cylinder pressure and heat release rate is sharp due to spontaneous auto-ignition triggered by multiple local rich diesel fuel around TDC. However, the gradient of heat release rate becomes lower along the diesel ratio decreased. In this case, the ignition sources from diesel local rich areas in premixed mixture are gradually reduced. Since then, auto-ignition at somewhere of highly premixed mixture by the increased pressure and temperature made by the compression stroke becomes fundamental ignition sources and it causes combustion of rest of fuel as flame propagation [20]. That is, the fundamental dual-fuel combustion mode depends on the fuel ratios between diesel and gasoline. Thus, substitution rate of dual-fuel was considered as a major control factor in this study.

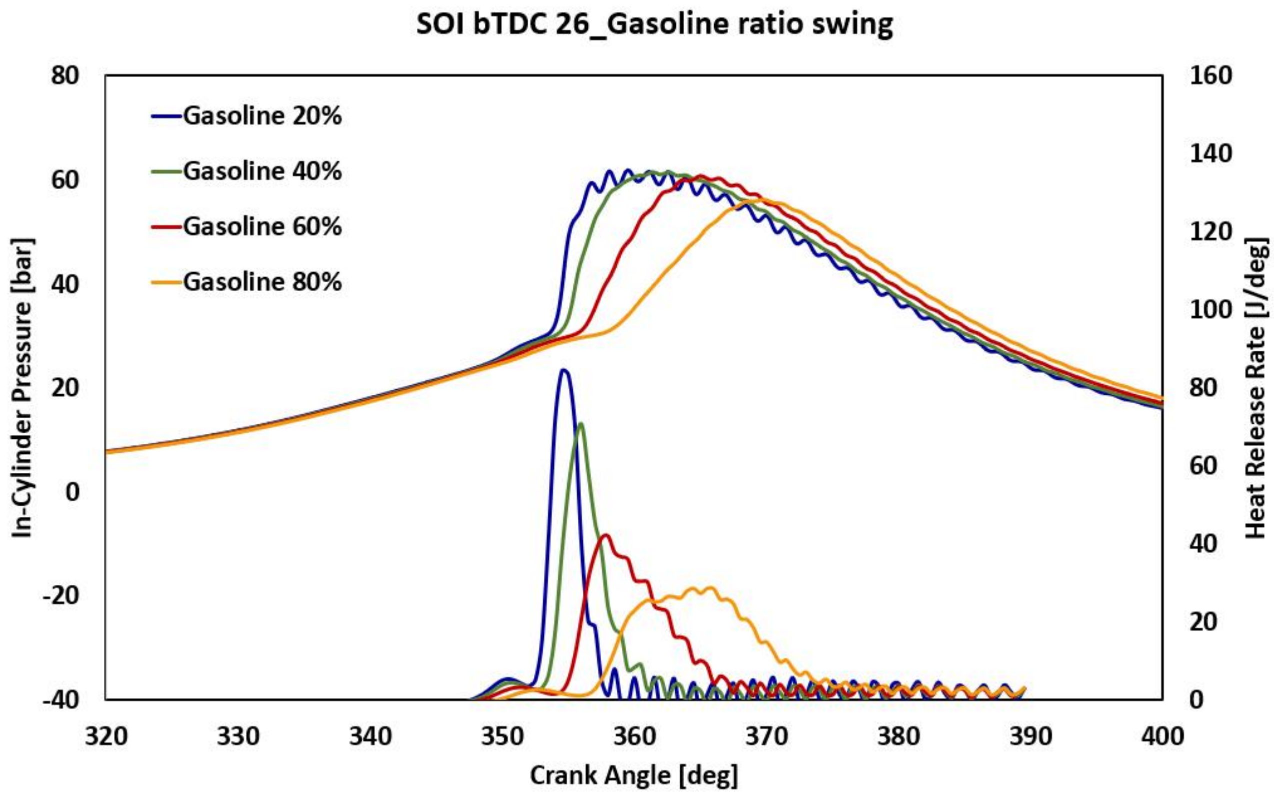

Figure 3.1.2 Cylinder pressure & Heat release rate in accordance with gasoline substitution rate at 1,500 rpm, gIMEP 5.2 bar

Also, DIT is another major control factor. Figure 3.1.3 shows the cylinder pressure and heat release rate in accordance with DIT. When DIT was advanced from 11°CA BTDC to 26°CA BTDC, the combustion phase was advanced until combustion mode transition point from diesel diffusion flame to spontaneous auto-ignition in premixed air-fuel mixture by high temperature and pressure in cylinder due to the compression stroke. When the combustion mode changed with sufficient advanced DIT such as 31° CA BTDC, the combustion phase begun to be retarded. Thus, it was used as an important combustion control factor because the reactivity of combustion can be controlled through the DIT. Also, sufficient advanced DIT around 20~25°CA BTDC, highly premixed air-fuel mixture for overall lean equivalence ratio condition is effective to reduce NOx and soot emissions.

Normally, supplying EGR makes the initial combustion reactivity decreased. It leads to longer I.D. so that maximum cylinder pressure and heat release rate decreased such as Figure 3.1.4. This cause and effect relationship are also applied to dual-fuel PCI combustion. As a result, the proper supply of EGR is a control factor that contributes to securing mPRR through reduction of combustion reactivity toward high load operation as well as reduction of NOx and soot emissions.

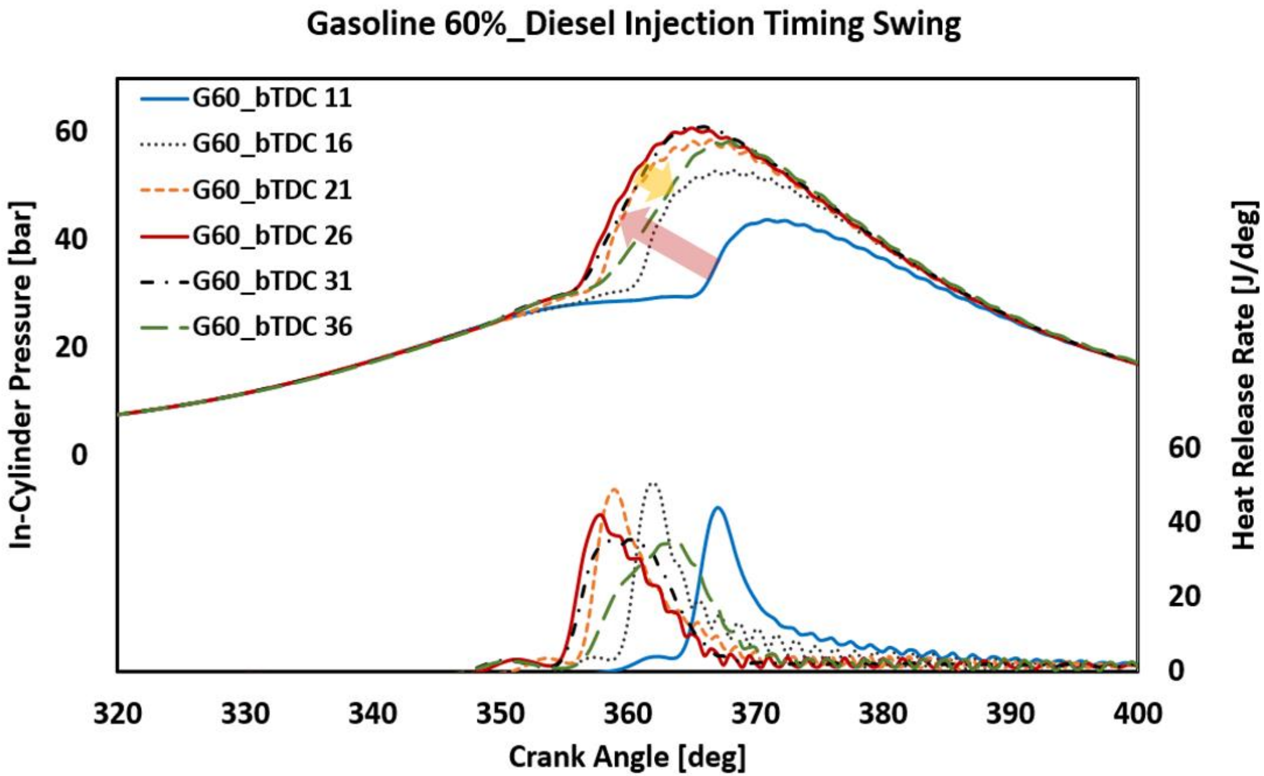

Figure 3.1.3 Cylinder pressure & Heat release rate according to various DIT under 60% gasoline substitution at 1,500 rpm, gIMEP 5.2 bar

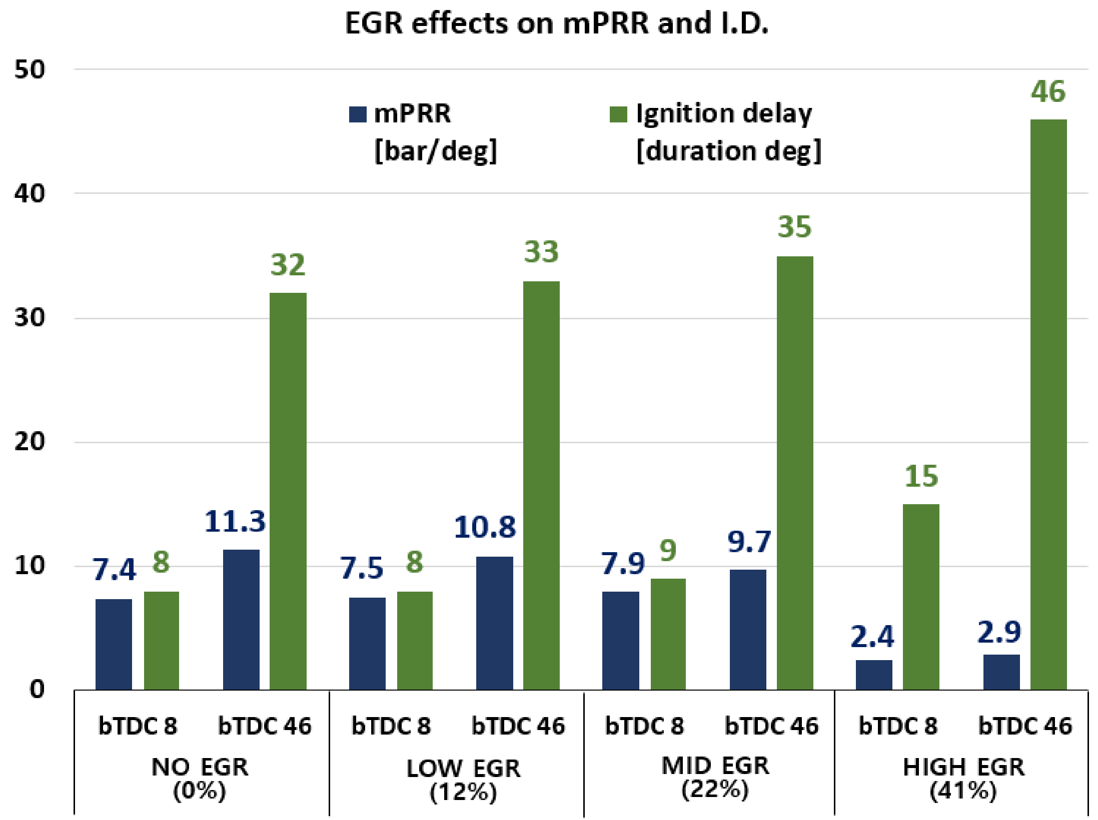

Figure 3.1.4 EGR effect on mPRR and I.D.

3.2 Determination of optimal geometric CR for high thermal efficiency under low load condition

In this section, Dual-fuel PCI experiments with geometric CR 14 and 15 were conducted to find proper CR based on thermal efficiency under low load operation, 1,500 rpm, gIMEP 5.2 bar. As mentioned in "1.4 Research Motivation and Objective", SNU-Automotive laboratory has found that low geometric CR has potential to expand high load limit, however, it shows lower thermal efficiency than high geometric CR, especially under low load operation. Thus, it was necessary to find the lowest geometric CR among various CR that shows higher thermal efficiency than diesel engine under low load operation. The detailed operating conditions were listed in Table 3.2.1.

Figure 3.2.1 demonstrates gross indicated thermal efficiency (gITE) comparison between geometric CR 14 and 15 at 1,500 rpm, gIMEP 5.2 bar. The combustion phases on figure were optimized by controlling gasoline substitution rate, DIT and EGR supply. Both combustion phases from geometric CR 14 and 15 show very low level of NOx as each 28 ppm and 21 ppm, and soot emissions as each 0.016 FSN and 0.018 FSN. Based on gITE and NOx, soot emissions, the fact that dual-fuel combustion enable high thermal efficiency with near zero NOx and soot emissions can be assured.

As geometric CR increased, the combustion reactivity is better since high in-cylinder pressure and temperature were obtained during the compression stroke. It causes the high self-ignition characteristic from diesel area in cylinder. Therefore, the higher geometric CR, the more DIT was advanced and EGR rate was increased to control MFB 50 to where the point showing the highest gITE. Because advanced DIT decreased diesel local equivalent ratio (Ø) and increased EGR rate reduced the combustion reactivity. So, in geometric

CR 15, DIT and EGR rate were 31° CA BTDC, 50% respectively, and 23° CA BTDC, 46.4% in CR 14.

In geometric CR 15, MFB 50 was controlled at 5.3° CA ATDC. At that time, the combustion shows the best gITE as 44.3%. In case of CR 14, MFB 50 was 11.3° CA ATDC. Relatively retarded MFB 50 of CR 14 was impossible to be advanced due to the low combustion reactivity from lower CR. At that time, the highest gITE was calculated as 41.4%. Base diesel engine of this experiment shows 43.8% of gITE at 1,500 rpm, gIMEP 5.2 bar condition. That is, gITE of CR 15 has higher thermal efficiency than diesel engine, however, gITE of CR 14 was lower than CDC. As a result, geometric CR should be determined as 15 based on high thermal efficiency compared to CDC under low load operating condition.

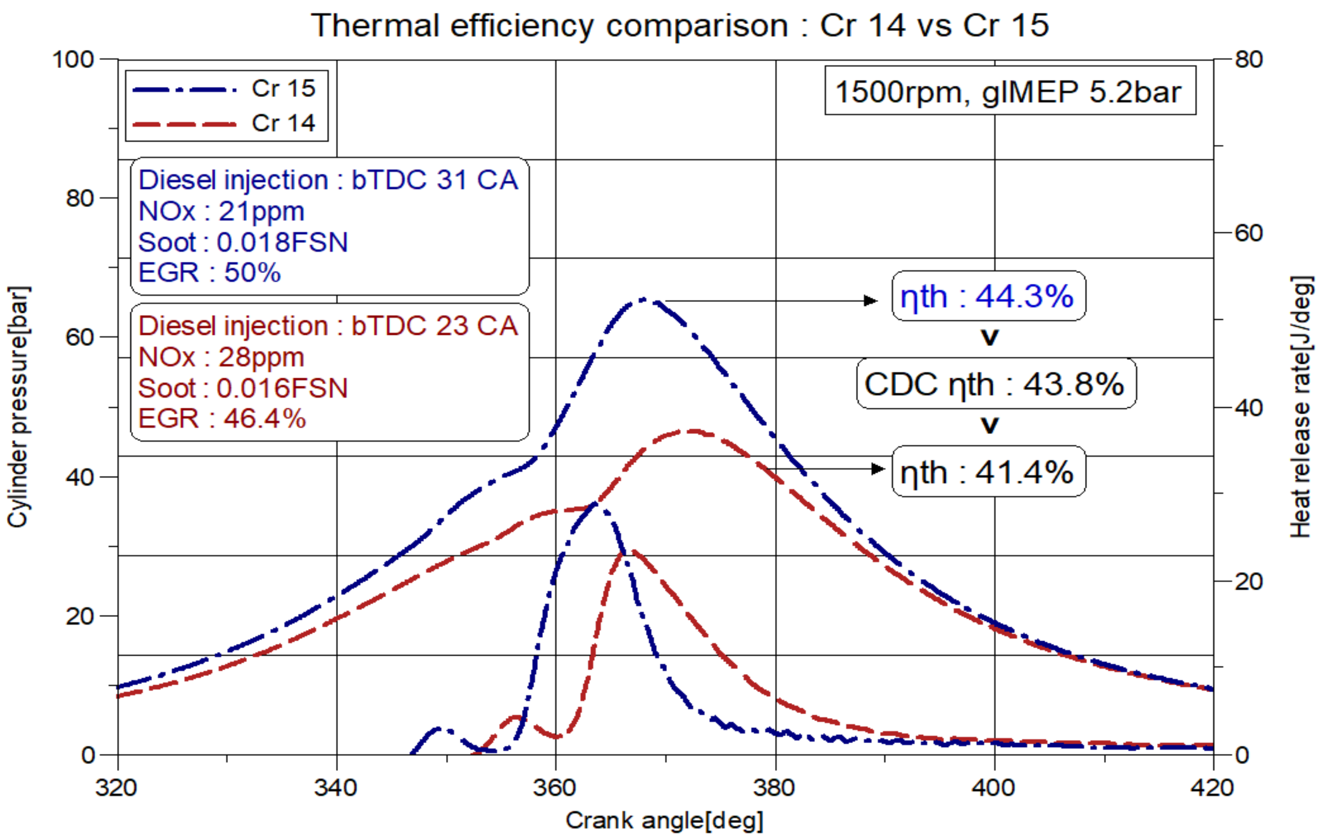

Figure 3.2.1 gITE compassion between geometric CR 14 and 15 at 1,500 rpm, gIMEP 5.2 bar

Table 3.2.1 Experimental conditions of determination of optimal geometric CR

Description	Value
Engine speed [rpm]	1,500
gIMEP [bar]	5.2
Diesel injection pressure [bar]	450
Gasoline Injection pressure [bar]	6
Compression ratio	14 / 15
Intake pressure [bar]	1.06
Engine oil & Coolant [°C]	85
NOx (target) [ppm]	< 40
Soot (target) [FSN]	< 0.2
mPRR (target) [bar/deg]	< 5
CoV of IMEP [%]	< 3

3.3 High load expansion with reduced effective CR via LIVC

The objective of this experiment is to expand the high load limit with reduced effective CR via LIVC strategy. Based on the result from "3.2 Determination of optimal geometric CR for high thermal efficiency under low load condition", base geometric CR was selected as 15. For this engine, normal intake valve opening/close timing were 8° CA BTDC and 36° CA ABDC. With normal IVC/IVO timing, the effective CR shows 15 same as geometric CR. The effective CR was able to be continuously lowered through LIVC. In this experiment, LIVC 15, 30, 35, 45, 50 and 60 were applied, respectively, by retarded unit of 15 degrees. At that time, volume based calculated effective CR of each cases were listed in Table 2.4.1.

Table 2.4.1 Calculated effective CR according to LIVC cases

IVC timing →	Base	LIVC 15	LIVC 30	LIVC 45	LIVC 50	LIVC 60
Effective Cr :	15.0	14.1	12.7	11.1	10.6	9.3

Intake boost pressure was set as the same as the conventional diesel engine. Thus, 1.9 bar was maintained at 1,500 rpm high load condition and 2.0 bar was set at 1,750 rpm high load condition.

High load optimization means the highest gIMEP that can be seen satisfying mPRR and CoV at the applicable rpm so that combustion takes place around lambda (λ) = 1. At high load operating condition, engine-out emissions should also satisfy the constraints listed in Table 2.2.2 and the experiment results were compared to the optimized points with the best gITE.

The cylinder pressure and heat release rate graphs for dual-fuel PCI high load experiments at 1,500 rpm with effective CR 15, 14.1 (LIVC 15 case) and 12.7 (LIVC 30 case) were demonstrated in Figure 3.3.1 and experimental conditions were listed in Table 3.3.1.

The major constraint for dual-fuel PCI combustion toward high load expansion is mPRR. To satisfy it, increasing EGR supply is unavoidable because of too strong combustion stability that exceed the criteria of mPRR and also control of MFB 50. In case of effective CR 15, normal IVC timing, gIMEP showed 13.3 bar with 52% EGR rate. And EGR rate were able to decrease 50.8%, 50% along to reduced effective CR 14.1, 12.7. As EGR rate decreased, a strategy to expand the gIMEP by injecting additional fuel via margin of lambda (λ) was possible. There were three ways to inject additional fuel as below.

1) Additional diesel injection with fixed or decreased gasoline injection
2) Additional injection of both fuels, diesel and gasoline
3) Additional gasoline injection with fixed or decreased diesel injection

Above three methods increased PRR, but showed a difference in the operating strategy to meet under 10 bar/deg again. In the first method, as diesel fuel increases, EGR rate must be increased due to increased combustion reactivity. Although DIT advances to decrease diesel local equivalent ratio, it was not effective due to combustion of premixed air-fuel mixture by spontaneously diesel ignition. It resulted in loss of either gIMEP or gITE compared to effective CR 15 because increased EGR decreases gIMEP or advanced MFB 50 reduces gITE. With the second method, the countermeasure and result were similar to the first method. However, the amount of pre-mixed air-fuel mixture and source of ignition, diesel, increases simultaneously, resulting in a high EGR rate as the mPRR increases more significantly than the first strategy. Therefore, both gIMEP and gITE tend to decrease. In case of third method, the combustion reactivity was reduced due to decreased diesel ratio. So, DIT should be retarded to increase diesel local equivalent ratio. Although soot emission increased by retarded DIT, it was able to meet 0.2 FSN until around 20° CA BTDC. Thus, additional gasoline injection was possible without additional EGR supply. It resulted in increase of gIMEP while maintaining gITE.

The combustion phases of effective CR 14.1 and 12.7 were accomplished with above mentioned third strategy, decreased diesel fuel with retarded DIT. The diesel ratios decreased from 22% to 16% with additional gasoline injection and EGR tended to decrease. gIMEP also expanded from 13.3bar to in the order of 14 bar and 14.7 bar. In the process of increasing gIMEP, gITE was equal to or decreased by only 1% compared to the highest effective CR, 45.9% even if MFB 50 was retarded from TDC. This is because the sum of the increase amount of the expansion work and the decrease amount of the compression work is greater than the work loss due to the cylinder peak pressure drop as effective CR decrease. The amount of work in the area of P-V diagram shown in Figure 3.3.2 demonstrated the total amount of work increases. Thus, gIMEP can be increased while maintaining gITE.

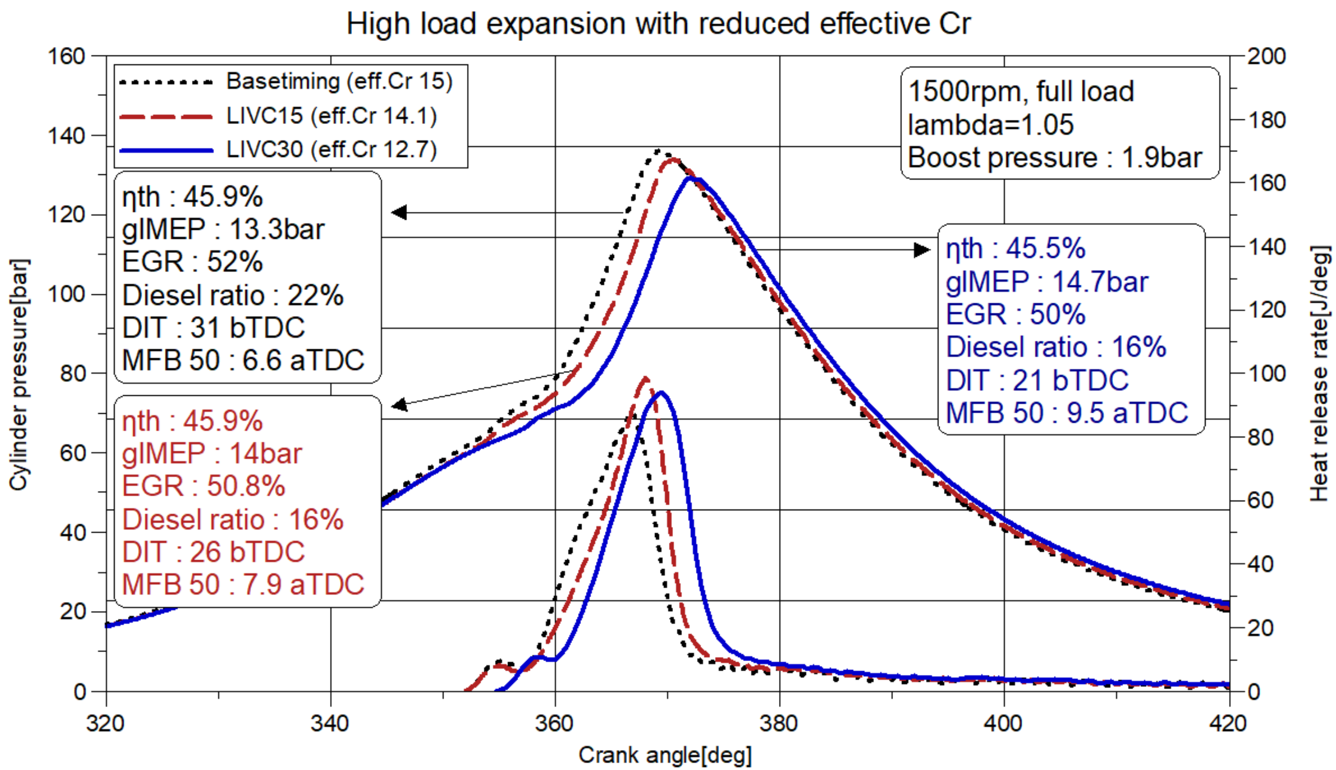

Figure 3.3.1 High load expansion with reduced effective CR 15 / 14.1 / 12.7 at 1,500 rpm

Table 3.3.1 Experimental conditions of high load expansion at 1,500 rpm – (1)

Description	Value
Engine speed [rpm]	1,500
gIMEP [bar]	Full load
Diesel injection pressure [bar]	450
Gasoline Injection pressure [bar]	6
IVO / IVC	8° CA BTDC / 36, 51, 66 CA ABDC
Geometric compression ratio	15
Effective compression ratio	15 / 14.1 / 12.7
Intake pressure [bar]	1.9
Engine oil & Coolant [°C]	85
NOx (target) [ppm]	< 40
Soot (target) [FSN]	< 0.2
mPRR (target) [bar/deg]	< 5
CoV of IMEP [%]	< 3

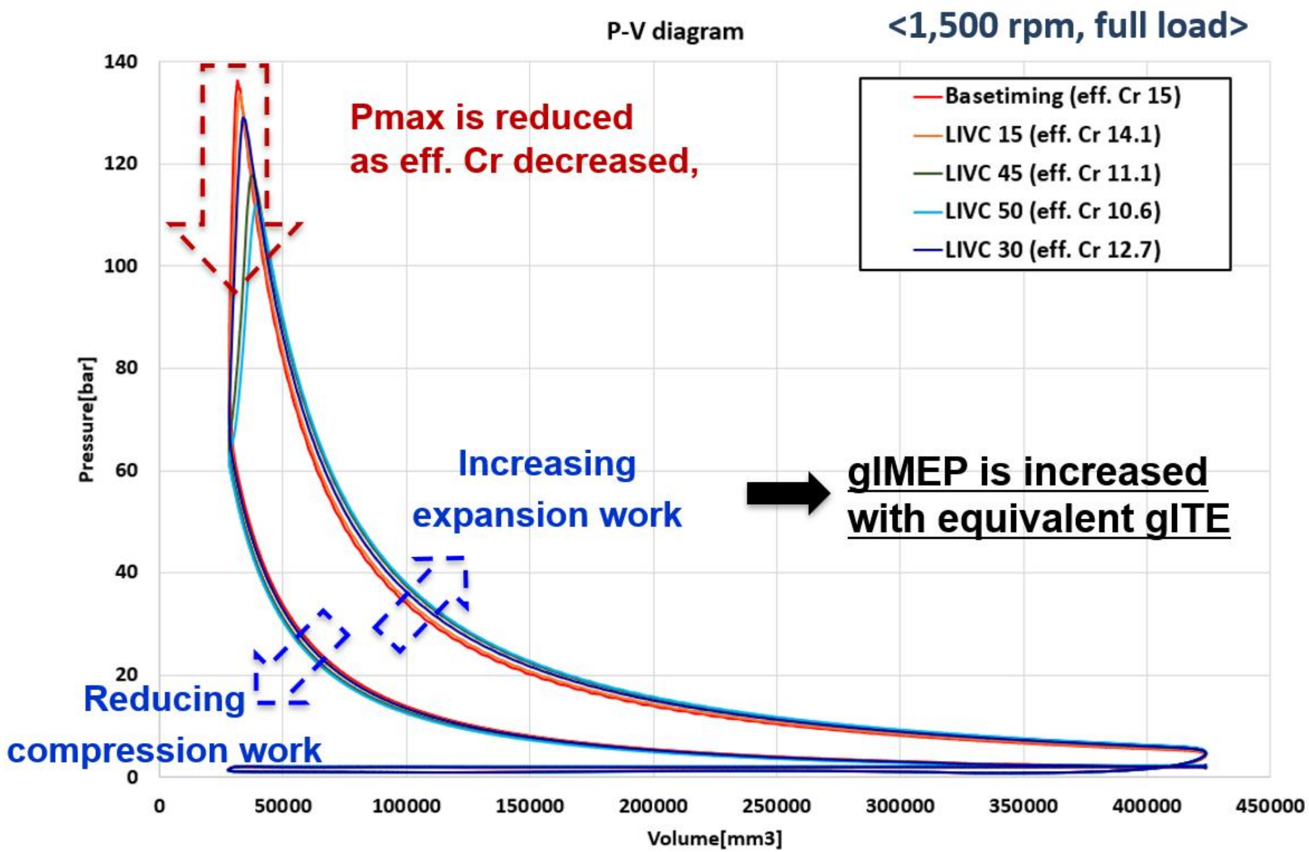

Figure 3.3.2 P-V diagram at 1,500 rpm, full load condition

Figure 3.3.3 shows the cylinder pressure and heat release rate with lower than effective CR 12.7 such as 11.1, 10.6 by applying LIVC 45 and 50. The detailed experimental conditions were listed in Table 3.3.2. If the effective CR is lower than 12.7 by further delaying the IVC timing, gIMEP can be increased, however, gITE tended to decreased due to too low combustion reactivity. At that time, even additional diesel injection was happened with retarded DIT to enhance the combustion stability, it was not able to compensate gITE. Too low combustion reactivity leaded to MFB 50 delayed such as 12.9° CA ATDC, 14.1° CA ATDC. In case of effective CR 10.6, soot emission was measured about 0.132 FSN due to maximum DIT retard limit. Therefore, the effective CR for increasing gIMEP while maintaining gITE at full load condition of the dual-fuel PCI shall be maintained around 12.7.

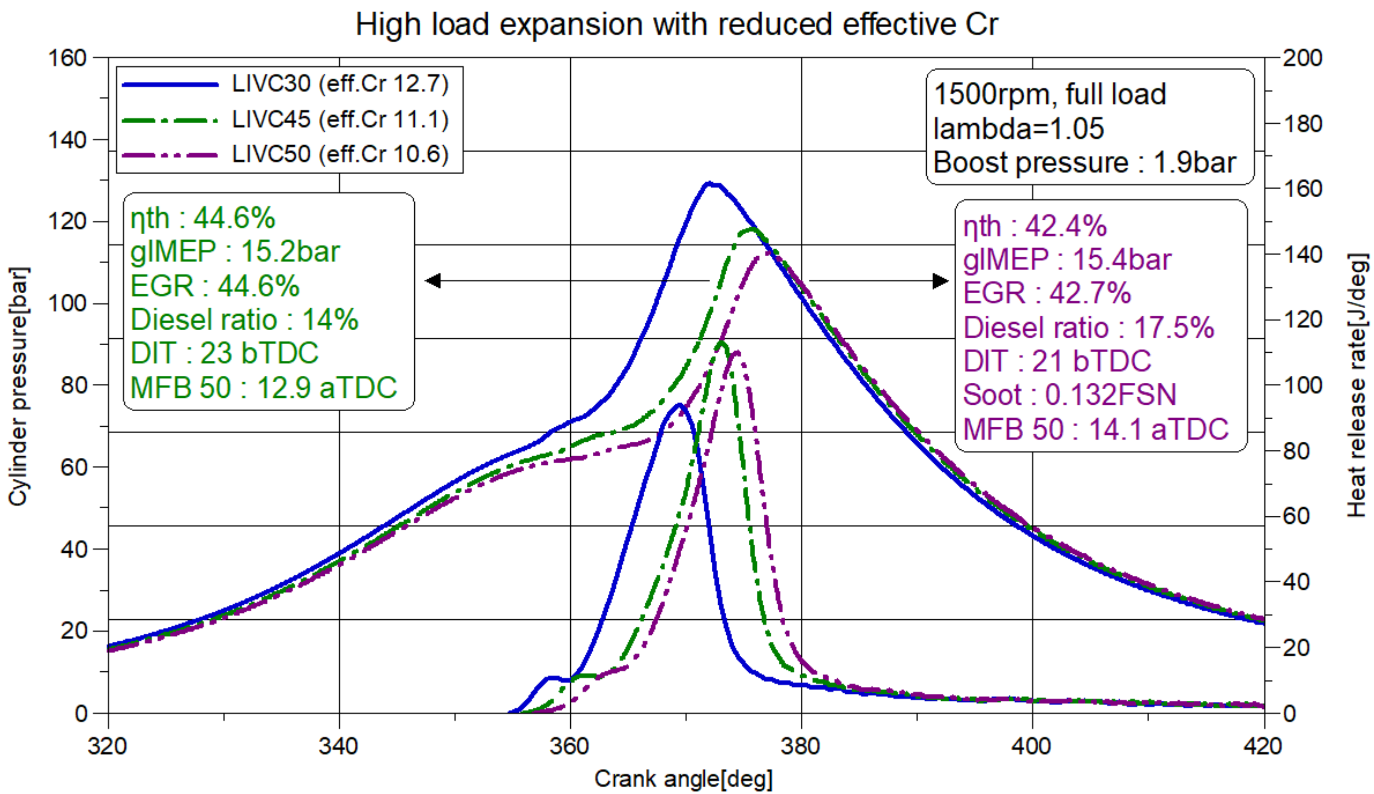

Figure 3.3.3 High load expansion with reduced effective CR 12.7 / 11.1 / 10.6 at 1,500 rpm

Table 3.3.2 Experimental conditions of high load expansion at 1,500 rpm – (2)

Description	Value
Engine speed [rpm]	1,500
gIMEP [bar]	Full load
Diesel injection pressure [bar]	450
Gasoline Injection pressure [bar]	6
IVO / IVC	8° CA BTDC / 66, 81, 86 CA ABDC
Geometric compression ratio	15
Effective compression ratio	12.7 / 11.1 / 10.6
Intake pressure [bar]	1.9
Engine oil & Coolant [°C]	85
NOx (target) [ppm]	< 40
Soot (target) [FSN]	< 0.2
mPRR (target) [bar/deg]	< 5
CoV of IMEP [%]	< 3

Based on the experiment results from high load expansion at 1,500 rpm, the effects of certain effective CR, 15, 12.2, 9.3 were investigated at 1,750 rpm. The cylinder pressure and heat release rate were demonstrated in Figure 3.3.4 at 1,750 rpm, full load condition and Table 3.3.3 listed the detailed experimental conditions.

The trends at 1,500 rpm, full load condition showed the same trends at 1,750 rpm. The most effective combustion appearance was when the effective CR was around 12.7 at 1,500 rpm, high load condition, and as a result of applying the effective compression ratio 12.2 at 1,750 rpm, gIMEP effectively increased from 14.5 bar to 15.4 bar while maintaining equal gITE, 46.5%. However, increase in gIMEP was not higher than 1,500 rpm experiment. It indicates that an additional experiment is necessary to find optimal effective CR for this positive effect. If the effective CR was further reduced, gIMEP can be increased about 0.6 bar but gITE tended to decreased from 46.5% to 44.6% even diesel ratio was increased with retarded DIT until maximum DIT retard limit based on soot constraint.

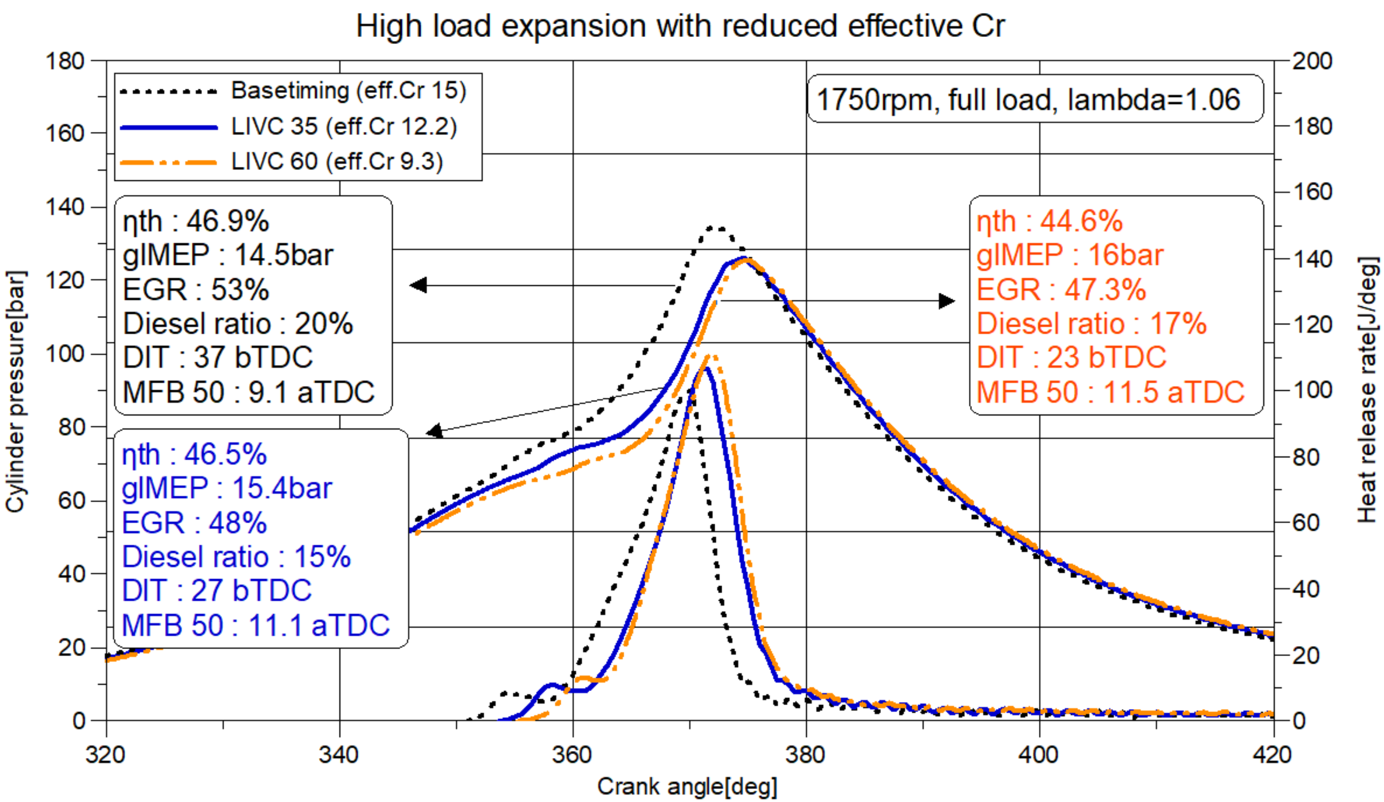

Figure 3.3.4 High load expansion with reduced effective CR 15 / 12.2 / 9.3 at 1,750 rpm

Table 3.3.3 Experimental conditions of high load expansion at 1,750 rpm

Description	Value
Engine speed [rpm]	1,750
gIMEP [bar]	Full load
Diesel injection pressure [bar]	450
Gasoline Injection pressure [bar]	6
IVO / IVC	8° CA BTDC / 36, 71, 96 CA ABDC
Geometric compression ratio	15
Effective compression ratio	15 / 12.2 / 9.3
Intake pressure [bar]	2.0
Engine oil & Coolant [°C]	85
NOx (target) [ppm]	< 40
Soot (target) [FSN]	< 0.2
mPRR (target) [bar/deg]	< 5
CoV of IMEP [%]	< 3

Chapter 4. Conclusions

In this research, high load expansion strategy by changing effective compression ratio was introduced on gasoline-diesel dual-fuel premixed compression ignition combustion. The effective CR was changed via late intake valve close timing (LIVC) strategy by using intake variable valve actuation (VVA) system.

For optimization of dual-fuel PCI, 1) gasoline-diesel substitution rate, 2) diesel injection timing (DIT) and 3) supplying EGR were applied as combustion control factors. The optimized combustion phase should satisfy the constraints listed in Table 2.2.2 regarding the engine-out exhaust emissions and mPRR. Among them, the data with the best thermal efficiency were used to compare the results of the experiments.

Firstly, dual-fuel PCI under low load operation was conducted with geometric CR 14 and 15 to find the lowest CR that shows higher thermal efficiency than diesel engine. The results show that dual-fuel PCI enable high thermal efficiency with near zero NOx and soot emissions. As geometric CR increased, EGR rate had to be increased to reduce the combustion reactivity and advanced DIT was effective. Advanced DIT decreased diesel local equivalent ratio which can reduce self-ignite ability of diesel. Both gITE from geometric CR 14 and 15 show high value as each 41.4% and 44.3%, respectively at 1,500 rpm, gIMEP 5.2 bar condition. At this time, gITE of corresponding EURO 6 diesel engine was 43.8%. Based on gITE under low load operating condition, geometric CR was selected as 15.

As the second part of the research, high load expansion experiments with various reduced effective CR via LIVC strategy were proceeded. For the experiment engine, the effective CR can be adjustable from 15 to 9.3 continuously. At 1,500 rpm, high load expansion experiment was conducted with effective CR 15, 14.1, 12.7, 11.1 and 10.6 cases. As the effective CR decreases, EGR rate can be reduced as mPRR decreases. It allows to inject additional gasoline fuel via margin of lambda (λ). And DIT was retarded to cover the combustion reactivity degradation by reduced effective CR. Also, diesel substitution was able to decreased because of a high diesel local equivalent ratio. Until effective CR 12.7 from 15, gIMEP was increase from 13.3 bar to 14 bar with the same gITE as 46%. This positive phenomenon was possible because the sum of the increase amount of the expansion work and the decrease amount of the compression work is greater than the work loss due to the cylinder peak pressure drop as the effective CR decrease. At too low the effective CR such as 11.1 and 10.6 cases, however, it was impossible to cover the low combustion reactivity by retarded DIT and increased diesel substitution rate. Even gIMEP can be increased due to allowable mPRR, gITE tends to decrease for these cases. Based on the results from 1,500 rpm, high load expansion experiments, certain effective CR were applied to 1,750 rpm. At 1,750 rpm, gIMEP also has been increased with equivalent gITE until a certain effective CR around 12.2. However, if effective CR was further reduced such as 9.3, gIMEP can be increased but gITE was decreased as same as the results of 1,500 rpm experiment. At 1,750 rpm, high load expansion experiment, gIMEP and gITE were increased from 14.5 bar, 46.9% to 15.4 bar, 46.5%.

Figure 4.1.1 demonstrated the overall gITE map according to rpm and gIMEP. By changing the effective CR, gIMEP can be increased about 10.5% at 1,500 rpm and 6.2% at 1,750 rpm while only 1% drop of gITE.

In this study, the results of applying geometric CR which shows higher gITE than diesel engine under low load operating condition of dual-fuel PCI were derived, and also expanding the high load limit by reducing effective CR via LIVC strategy was achieved. This research can contribute to the load expansion which is the main disadvantage of dual-fuel PCI as the next generation advanced combustion system.

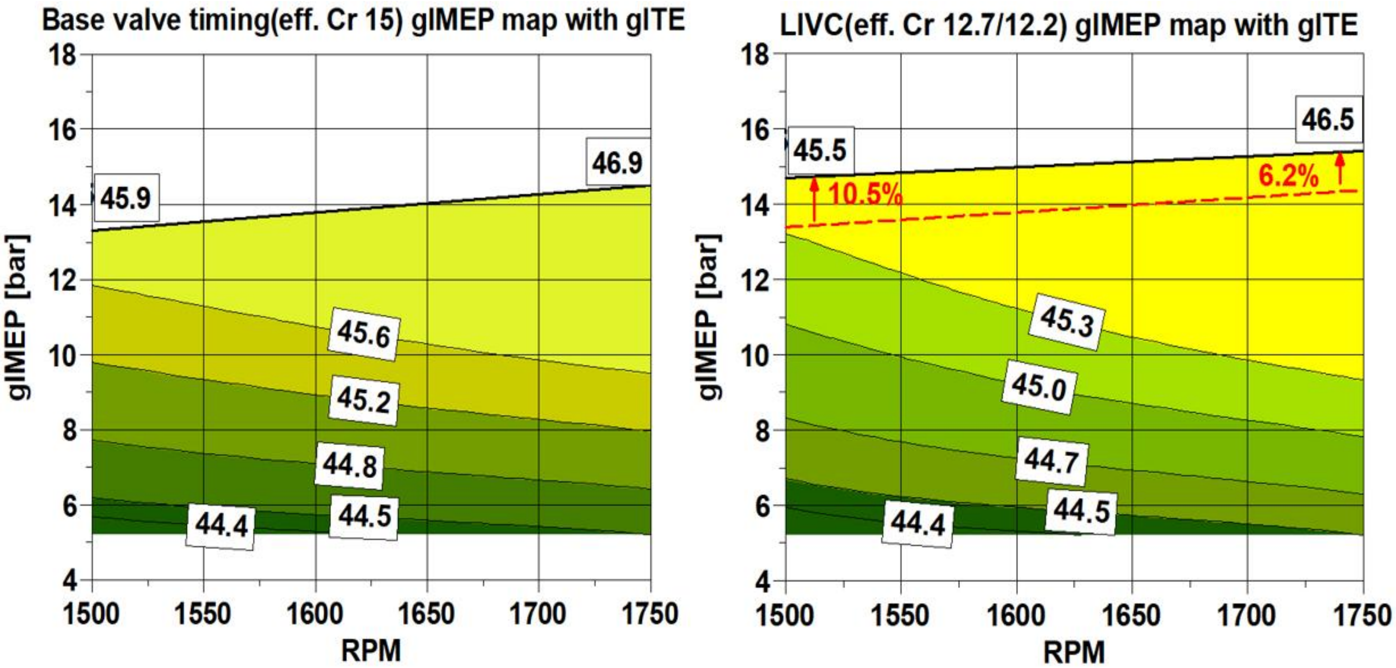

Figure 4.1.1 Comparison of gITE according to RPM and gIMEP between effective CR 15 and reduced effective CR

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국 문 초 록

가솔린-디젤 예혼합 융합연소 엔진에서 유효압축비 변경을 통한 부하 확장에 대한 연구

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2000 년대에 들어 전 세계적으로 지구온난화에 대한 관심이 높아지면서 이산화탄소 규제가 점차 강화되어 왔다. 특히, 내연기관이 이산화탄소 배출원 중 하나로 주목받으며 수송 분야에서 배출가스 규제가 더욱 까다로워졌다. 이에 가솔린 SI 엔진에 비해 높은 열효율 바탕으로 연료 소모가 적어 이산화탄소 배출에 유리한 디젤 압축 착화 엔진이 각광받기 시작하였다. 그러나 디젤 배출가스 조작 파문과 더불어 디젤 엔진의 대표적 배기 배출물인 질소산화물 (NOx) 과 입자상 물질 (PM)의 인체 유해성이 두드러지며, 운전 시험 모드인 WLTP 와 실도로 주행 배기 배출물 시험인 RDE 가 도입되었다. 이러한 규제는 배기가스 재순환 (EGR) 전략과 Lean NOx Trap (LNT), Selective Catalyst Reduction (SCR)을 통해 대응이 가능하지만, 향후 더욱 강화될 예정인 배기가스 규제에 선제적 대응이 필요하다.

따라서 디젤 압축 착화 엔진의 높은 열효율을 유지하면서, 낮은 수준의 배기 배출물을 선보이는 신연소 기술이 개발되었다. 대표적인 신연소 기술로는 가솔린 HCCI 와 디젤 PCCI 기술이 있으나, 연소의 제어가 불가능하다는 측면에서 한계를 보였다.

이러한 단점을 보완한 신연소 기술로서 융합 연소의 개발이 활발히 진행 중이다. 융합 연소는 2 가지 종류의 연료를 사용한 예혼합 압축 착화 방식으로 저배기 고효율 연소를 실현하고, 동시에 2 가지 연료의 다른 반응성을 통해 연료의 비율 및 분사 시기와 같은 다양한 전략들로 연소 제어가 탁월하다는 장점이 있다. 그러나 예혼합 압축 착화로 인해 노킹 및 가파른 압력 상승률로 소음, 진동 유발 및 내구성에 부정적인 영향을 미친다는 단점이 존재한다. 따라서 디젤 압축 착화 엔진 대비 최고 부하의 범위가 한정적이므로 부하 확장을 위한 연구 개발이 필요하다.

본 연구에서는 가솔린-디젤 예혼합 융합연소 엔진에서 유효압축비 변경을 통해 부하 확장 전략을 조사하였다. 유효압축비의 변경은 가변 흡기 밸브 액츄에이터(VVA) 시스템을 통해 흡기 밸브 닫힘 시기를 지연시킴(LIVC)으로써 제어 가능하였다. 저부하 운전 조건에서는 상용 디젤 엔진 대비 높은 열효율을 보일 수 있는 기계적 압축비 (Geometric CR)를 선정하기 위한 실험을 진행하였다. 유로 6 대상 엔진의 저부하 운전 조건에서 열효율은 43.8%을 보였다. 이때, 각각 기계적 압축비 14 와 15 를 적용하여 동일한 운전점에서 최적화 결과를 비교한 결과, 압축비 14 의 경우, 41.4%로 상용 디젤 엔진 대비 낮은 열효율을 보였으나, 압축비 15 는 44.3%로 상용 디젤 엔진 대비 높은 열효율을 보였다. 따라서 기계적 압축비는 15 로 선정한 후, 1,500 rpm 및 1,750 rpm 에서 최고 부하

확장 실험을 진행하였다. 기계적 압축비 15 를 바탕으로 흡기 밸브 닫힘 시기 지연을 통한 유효압축비 감소는 최대 9.3 까지 연속적으로 제어 가능하였다. 저속 최고부하 조건에서 유효압축비는 각각 14.1, 12.7, 11.1 및 10.6 을 적용하였다. 그 결과, 유효압축비 12.7 수준까지는 동등 열효율을 유지하면서 최고부하가 10.5% 증가하였고, 이보다 낮은 유효압축비에서는 최고부하가 더욱 증가하였으나 연소 반응성이 저하되어 열효율 또한 감소하는 경향을 보였다. 1,750 rpm, 최고부하 조건에서 또한 특정 유효압축비까지는 열효율을 동등하게 유지하며 부하가 확장되는 경향을 보였고, 유효압축비를 더욱 감소시킬 경우 부하 확장과 동시에 열효율이 감소하는 경향을 보였다.

주요어: 이종 연료, 융합 연소, 예혼합 압축 착화, 최고 압력 상승률, 압축비, 유효압축비

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