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디젤 프로판 융합 연소 엔진에서 연소 최적화에 관한 연구

Combustion Optimization of Diesel and Propane Dual Fueled Engine

2018 년 2 월

서울대학교 대학원

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이 논문을 공학석사 학위논문으로 제출함

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Abstract

Combustion Optimization of Diesel and Propane Dual Fueled Engine

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The regulation for light duty diesel engine is more stricter as time goes on. In Europe, especially regulation for PM emission was reduced about 80% and the NOx was reduced about 70% from EURO-4 to EURO-6 regulation. And also, PN regulation has been started since 2011. To satisfy these regulations, dual fuel combustion has been researched which is based on low combustion temperature and well premixed condition by using two different reactivity fuels.

In this research, improving the diesel and propane dual fuel combustion by optimizing operating strategy and comparison of piston bowl shape in 4 main operation points was conducted. The experiments were comprised of three parts. In first part, investigating effects of three operating parameters (diesel injection timing, propane ratio, and EGR rates) in a diesel-propane dual fuel combustion were

investigated. The characteristics of dual-fuel combustion were analyzed by engine parameters, such as NOx, PM level, thermal efficiency and gross IMEP CoV.

Based on the results, optimization of main 4 operating points was conducted for dual-fuel PCI combustion with different piston bowl shape, CDC piston and bathtub piston. Emission restrictions were implemented to satisfy the EURO-6 regulation without post exhaust treatment, and PRR_{max} restriction was also selected based on the previous research.

Investigation of optimized results with CDC piston was conducted in part 2. Dual-fuel PCI combustion can be available with low NOx, PM emission and the maximum pressure rise rate in relatively low load condition in CDC piston, however, PM and PRR_{max} were not satisfied at high load conditions. By modifying diesel injection strategy can be satisfied due to reducing piston impingement and consecutive reactivity gradient.

In part 3, the same optimization experiment was carried out by using bathtub piston. All the restrictions were satisfied with bathtub piston, and it was shown that lower PM and PRR_{max} levels were appeared than CDC piston results. To check the effect of piston bowl shape, additional experiment was conducted with all operating parameters were fixed except for the difference of piston shape. Same as the previous results, the lower PM and PRR_{max} results appeared in bathtub piston. This can be explained that bathtub piston has potential to reduce PM and PRRmax due to available of formation of consecutive phi gradient and avoiding locally richer region in the combustion chamber

Keywords: Diesel engine, Dual-fuel combustion, PCCI(premixed charge compression ignition), Piston bowl shape, RCCI (reactivity controlled compression ignition)

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Nomenclature

ATC	after top center
BMEP	brake mean effective pressure
BTC	before top center
CA	crank angle
CDC	conventional diesel combustion
CO	carbon monoxide
CO2	carbon dioxides
CoV	coefficient of variation
DI	direct injection
DPF	diesel particulate filter
EGR	exhaust gas recirculation
FSN	filtered smoke number
gIMEP	gross indicated mean effective pressure
HCCI	homogeneous charge compression ignition
HFR	hydraulic flow rate

HSDI	high-speed direct injection
HP-EGR	high pressure EGR
HRR	heat release rate
HTHR	high-temperature heat release
ISNOx	indicated specific nitrogen oxide
IVC	intake valve close
LHV	low heating value
LNT	lean NOx trap
LP-EGR	low pressure EGR
LTC	low-temperature combustion
LTHR	low-temperature heat release
MFB 50	mass fraction burned 50
mPRR	maximum pressure rise rate
NOx	nitrogen oxides
NVO	negative valve overlap
PCI	premixed compression ignition

PCCI	premixed charge compression igniton
PFI	port fuel injection
PM	particulate matter
ppm	part per million
RCCI	reactivity controlled compression ignition
RoHR	rate of heat release
rpm	revolution per a minute
SCR	selective catalytic reduction
SOC	start of combustion
SOI	start of injection
THC	Total hydrocarbon
uHC	unburned hydrocarbon
WLTC	world harmonized light vehicle test cycle

Chapter 1. Introduction

1.1. Research Background

Diesel engines have been widely used in light duty vehicle due to its considerable advantages such as higher fuel efficiency and lower CO2 emission compared to gasoline engines. However, conventional diesel engine combustion process includes formation of large quantity of noxious substances, such as Nitrogen Oxide (NOx) and Particulate Matter (PM). These noxious substances are restricted by emission regulations, and the diesel engine regulations for exhaust gas are becoming more stringent [1].

The EURO-6 regulation, which was implemented in 2014, supports the reduction of NOx emission by approximately 70 % and PM by approximately 80 % compared to the requirements of the previous EURO-4 regulation, which was established 10 years ago. In addition, the driving cycle mode was changed from the New European Driving Cycle (NEDC) to the Worldwide Harmonize Light vehicles test Cycle (WLTC) in 2017. Vehicles will be required to satisfy the same emissions regulation levels under a wider operating range. Furthermore, a Real Driving Emissions (RDE) regulation was also implemented for diesel engines in 2017 with a 2.1point conformity factor. [1]

Many automobile companies have tried to adopt various post exhaust treatment strategies, including Diesel Particulate Filters (DPFs), Selective Catalytic Reduction (SCR) and Lean NOx Traps (LNTs), to satisfy these regulations. However, these[methods require additional cost and reduce thermal efficiency. Therefore, recent studies have focused on reducing NOx and PM emissions within the engine itself [1].

1.2. Previous Research

Many researchers have investigated novel diesel combustion options such as Premixed Charge Compression Ignition (PCCI) [2], Homogeneous Charge Compression Ignition (HCCI) and Reactivity Control Compression Ignition (RCCI) [3, 4]. All of these combustion strategies can be characterized as Low Temperature Combustion (LTC). LTC has the following two characteristics: a lower combustion temperature and a long ignition delay [3]. The first characteristic of LTC can inhibit NOx formation due to the higher activation energy of the NO formation reaction. The second characteristic allows the combustion to have sufficient time for mixing preceding the start of combustion, which avoids the formation of a fuel rich condition that causes PM formation [2, 4]. Thus, HCCI and PCCI were developed to incorporate the benefits of the LTC mode [1].

For these reasons, many current studies focus on Reactivity Controlled Compression Ignition (RCCI). RCCI is a dual fuel combustion strategy that blends two different reactivity fuels in one cylinder, and multiple injections to control the in-cylinder fuel reactivity to optimize combustion phasing, duration and other characteristics [4]. In RCCI combustion, a low-reactivity fuel such as gasoline or a gaseous fuel is introduced using Port Fuel Injection (PFI) or early intake process and a high-reactivity fuel is directly injected into the cylinder. The low reactivity fuel is well-premixed in the cylinder, and the high reactivity fuel takes the role of the ignition source. Thus, the combustion process is controlled by reactivity stratification [1].

RCCI combustion can achieve a high thermal efficiency due to the reduction of heat loss by shortened combustion duration and emits lower NOx and PM emissions

caused by a lower combustion temperature and premixed combustion [5, 6]. In RCCI, high reactivity fuel was early injected by single or split injection [7]. The first injection was conducted near BTC and second ignition was introduced near 30-45 BTC [8]. As the low reactivity fuel was homogeneously distributed in the cylinder, ignition source of high reactivity fuel created reactivity gradient [1, 4].

In current RCCI researches, dual-fuel combustion by using various low reactivity fuels such as Compressed Natural Gas (CNG), E85(85% ethanol and 15% gasoline), methanol and gasoline, were conducted [9-12]. Nieman et al. showed that using CNG for lower reactivity fuel was more controllable the maximum pressure rise rate rather than gasoline fuel due to its lower reactivity in heavy-duty dual-fuel engine [13]. Li et al. reported that methanol/diesel combustion was beneficial ways to improve thermal efficiency and reducing emission [10]. However, most of the study for using alternative low reactivity fuel for gasoline were focused on heavy-duty engine or not adequate for extensive usage fuel [1].

Propane gas is one of the fuel in LPG which is widely used and well-established infrastructure in Korea. Introducing propane gas fuel for dual-fuel combustion has a disadvantage in power output rather than using gasoline fuel since the energy density of propane is lower than that of gasoline fuel for the same volume. However, since propane has higher octane number than that of gasoline, it can be valuable property when the high-load operating condition or high compression ratio engine under dual-fuel combustion, which have a knocking problem. And also, implementing propane fuel does not need in-cylinder chamber and intake port modification [14].

On the other hand, introducing gasoline port fuel injection can make wall-wetting problem which causes higher THC emission unless the optimal injection port is implemented. Thus, it is necessary to investigate the potential of propane fuel for dual-fuel combustion engine [1].

Various operating strategies have been investigated under using propane or gaseous fuel. Jie et al. showed that effects of pilot fuel on NOx and PM characteristics under diesel-CNG dual-fuel combustion [15]. Krishnan et al. analyzed the effects of injection parameter, such as injection timing and common rail pressure, and boost pressure in diesel-propane combustion [16]. Lee et al. suggested that early diesel injection to enhance the premixed phase under dual fuel combustion was effective way to reduce NOx and PM simultaneously retaining with high thermal efficiency at 4 BMEP level [8]. However, considering that RCCI combustion phase is controlled by reactivity stratification, dominant operating factors should be related with the formation of stratified air-fuel mixture in the cylinder. Thus, reactivity stratification is not the only function of diesel injection timing in RCCI combustion. Combustion can be considerably affected by other operating parameters, such as the low reactivity fuel ratio and EGR rates, which affect the degree of in-cylinder reactivity gradient These parameters could be affected by the not only combustion phasing, consequently the NOx and PM emissions but also thermal efficiency and combustion stability [1].

From a point of investigating hardware for dual-fuel combustion, Hanson et al. implemented bathtub shape piston with using diesel and gasoline. As mixing from bowl shape does not need in dual-fuel combustion due to diesel early injection, wide and shallow bathtub was adopted for lower surface area rather than CDC piston that would can be reduced heat transfer [17].

1.3. Research Objective

The objective of this research is to improve diesel and propane dual-fuel combustion by optimizing operating strategy and comparison of piston bowl shape in main 4 operating points. The experiments were comprised of three parts. In first part, investigating effects of three operating parameters (diesel injection timing, propane ratio, and EGR rates) in a diesel-propane dual fuel combustion were investigated. The characteristics of dual-fuel combustion were analyzed by engine parameters, such as NOx, PM level, thermal efficiency and gross IMEP CoV.

Based on the results, optimization of main 4 operating points was conducted for dual-fuel PCI combustion with CDC piston in part 2, and the same optimization experiment was carried out by using bathtub piston in part3 under NOx, PM and the maximum pressure rise rate PRR_{max} restriction. These two emission restrictions are minimum levels that can satisfy the EURO-6 emission regulation without post exhaust treatment. In addition, effects of diesel split injection and lower injection pressure were also tested with CDC piston.

Chapter 2. Experimental Configuration and Conditions

In chapter 2, configuration of experimental setup and experimental condition are introduced. This chapter is based on the Ref. [1].

2.1 Experimental setup

A single-cylinder diesel engine based on the EURO-6 standard was used for these experiments. A solenoid injector with an injection pressure of up to 1,800 bar was equipped with a common rail. In contrast with PFI liquid fuel for dual fuel combustion, injection pressure of propane does not need to be higher because of its gaseous fuel. Therefore, propane was introduced at a pressure of 2 bar upon consideration of intake pressure. The quantity of propane fuel was measured by flowmeter (MK precision Co., MFM (TSM-230)), and the flow rate was controlled by sonic orifices. The detailed specifications of the engine and injector are described in tables 1 and 2, respectively. A 37 KW DC dynamometer was adopted to control the engine. The temperature of the oil and coolant were set at 75 and 85 °C, respectively. The diesel fueling temperature was also maintained at 40 °C during the experiments.

The concentrations of NOx, THC (total hydrocarbon), CO, CO2 and O2 were measured using an exhaust gas analyzer (Horiba, MEXA 7100DEGR), and PM (particulate matter) emission was measured with a smoke meter (AVL, 415S).

An absolute pressure sensor (Kistler 4045A5) was installed in the intake manifold to measure the intake pressure, and a relative pressure sensor (Kistler 6055B) was adapted to measure the in-cylinder pressure. Signals from the pressure transducers were recorded at 0.1 crank angle intervals using a data acquisition (Kistler, KiBox To Go 2893) system.

To measure the diesel flow rate, a mass burette type flow meter (ONO SOKKI, FX-203P) was used. Propane was released into the intake port as a gas. The quantity of propane was measured using a propane flow meter (MK Precision Co., MFM (TSM-230)). Additionally, a 2 L surge chamber was equipped to prevent propane fluctuation and to maintain the gas flow rate throughout the experiments.

An LP - EGR (low pressure EGR) system was adapted in this experiment. A 15 L aluminum chamber was installed after the EGR cooler to maintain the EGR rate. Because of the insufficiency of the exhaust flow to operate a turbo charger system in the single cylinder engine, a super charger system was utilized to simulate a turbo charger system by boosting the intake pressure. The EGR rate was calculated from the CO2 concentration in the exhaust gas and the intake gas.

The schematic diagram of experimental setup is presented in figure 1, and the properties of the diesel fuel and propane gas are given in Table 3.

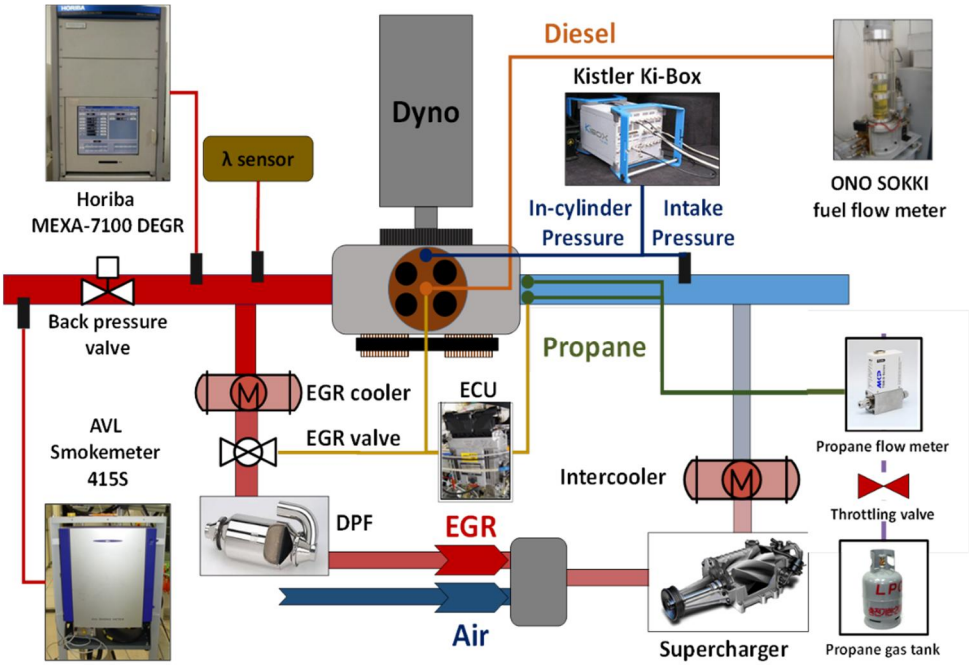

Figure 1. Schematic diagram of the experimental setup [1]

Table 1. Engine Specifications [1]

Displacement [cc]	397.7
Bore × Stroke [mm]	77.2 x 84.5
Compression ratio [-]	16
Con. rod length [mm]	140.0
HFR (Hydraulic Flow Rate) [cc/100 bar/30 s]	380

Table 2. Injector Specifications [1]

Fuel injection system	Bosch Solenoid (Pmax = 1800 bar)
Number of nozzle holes	8
Spray angle [°]	149
Nozzle diameter [mm]	0.119

Table 3. Properties of diesel and propane [1]

Properties	**Diesel**	**Propane**
Chemical formation	C_xH_{2x}	C_3H_8
Molecular weight [g]	190-220	44.1
Density [g/cm3]	0.831	0.51
Low heating value [MJ/kg]	42.5	46.4
Auto-ignition temperature [K]	523	763
Stoichiometric ratio of air to fuel composition [wt. %]	14.6	15.6

2.2 Combustion diagnostics

The ratio of propane fuel, χ %, was defined as the ratio of the propane fuel energy flow to total fuel energy flow based on the lower heating value of each fuel.

$$\chi = \frac{\dot{m}_{propane} * LHV_{propane}}{\dot{m}_{diesel} * LHV_{diesel} + \dot{m}_{propane} * LHV_{propane}}$$

where

$\dot{m}_{propane}$= propane fuel mass per cycle

$\dot{m}_{diesel}$= diesel fuel mass per cycle

Each fuel mass flow rate was measured using individual flow meters. The mass of propane was calculated using the measured volume and the density at room temperature (20 °C). The RoHR (rate of heat release) and gIMEP were calculated using the in-cylinder pressure data. The calculation model of ROHR adopted the single zone model to calculate the cylinder wall heat transfer using the Woschni correlation [18].

2.3 Experimental conditions

In this research, experimental configuration was organized three parts. In first part, the effects of the three operating parameters were evaluated. As dual-fuel combustion uses a mixture of two different reactivity fuels, degree of in-cylinder reactivity gradient also has to be considered along with the local equivalent ratio. Thus, three operating parameters, (diesel injection timing, ratio of propane and EGR rates) which can mainly affect those combustion characteristics, were selected. The parameter study was conducted using the 1500 / 4 [rpm / BMEP] condition on dual fuel combustion.

The investigation of the diesel direct injection timing sweep was gradually advanced from 10 to 45 ° BTC conditions under 58% of propane ratio. The second parameter, the ratio of propane, was examined at 20, 40, 58 and 65 %. The rates of EGR were changed from 12 to 47 %. The diesel injection timings of the propane ratio and EGR rate swing cases were carefully controlled to maintain the MFB50 value of 6 ° ATC.

Second part includes optimized main 4 operating points with CDC piston. By analyzing of effects of these three parameters on dual-fuel combustion, the improving operating strategies to satisfy the emission and the maximum pressure rise rate restrictions of the four main operating conditions were conducted. The four points are highly frequent operating condition in WLTC of commercial diesel engine, so examination of those points can evenly estimate the overall WLTC mode. To meet the

target restriction, injection strategies were modified at 2000 / 10 [rpm / BMEP] case.

In third part, optimization for main 4 operating points was also conducted by using bathtub piston. Except diesel injection timing, propane ratio and EGR rates, all the operating parameters were same with former optimized cases with bathtub piston. More details of operating conditions are described in figure 2, tables 4, 5 and 6.

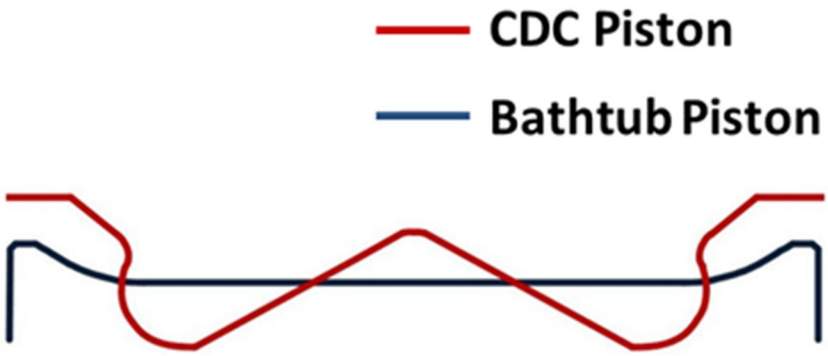

Figure 2. Shape of CDC piston and bathtub piston

Table 4. Experimental condition (1): Effects of operating parameters [1]

Fixed variables	
Engine speed [rpm]	1500
Intake pressure [bar]	1.06
Injection pressure [bar]	450
gIMEP [bar]	Near 5.2
Fuel temperature [k]	315
Oil temperature [K]	348
Coolant temperature [K]	358
Diesel injection timing swing (Ⅰ)	
Diesel injection timing [° BTC]	10/20/30/40/45
The amount of diesel/Propane [mg/str]	4.96/6.28
Diesel/Propane ratio [%]	42/58
EGR rate [%]	0
Various propane ratios (Ⅱ)	
Propane ratio [%]	20/40/58/65
MFB50 [°ATC]	6
Diesel injection timing [° BTC]	10/10/10/7
EGR rates [%]	0
EGR rate swing (Ⅲ)	
EGR ratio [%]	12/35/47

MFB50 [° ATC]	6
Diesel injection timing [° BTC]	9/11/20
Diesel/Propane ratio [LHV base, %]	40/60

Table 5. Experimental conditions (2): Optimization of the four main operating conditions (CDC piston) [1]

Detailed operating conditions	
Operating condition [rpm / BMEP]	1500 / 4, 1750 / 6, 2000 / 8, 2000 / 10
MFB 50 [° ATC]	1.5 (1500 / 4)
	4.0 (1750 / 6)
	7.2 (2000 / 8)
	5.3 (2000 / 10)
Ignition delay [°CA]	30.5 (1500 / 4)
	28.4 (1750 / 6)
	47.4 (2000 / 8)
	41.3 (2000 / 10)
Intake pressure [bar]	1.06 (1500 / 4)
	1.26 (1750 / 6)
	1.53 (2000 / 8)
	1.69 (2000 / 10)
Injection pressure [bar]	450 (1500 / 4)
	590 (1750 / 6)
	790 (2000 / 8)
	940 (2000 / 10)
Diesel/Propane ratio	63/37 (1500 / 4)
	52/48 (1750 / 6)
	46/52 (2000 / 8)
	45/55 (2000 / 10)
EGR rate [%]	52.5 (1500 / 4)
	51.5 (1750 / 6)
	39 (2000 / 8)
	43.2 (2000 / 10)
Diesel injection timing [° BTC]	34 (1500 / 4)
	30 (1750 / 6)
	46 (2000 / 8)
	41.5 (2000 / 10)

Table 6. Experimental conditions (3): Optimization of the four main operating conditions (Bathtub piston)

Detailed operating conditions	
Operating condition [rpm / BMEP]	1500 / 4, 1750 / 6, 2000 / 8, 2000 / 10
MFB 50 [° ATC]	3.7 (1500 / 4)
	1.7 (1750 / 6)
	3.8 (2000 / 8)
	6.9 (2000 / 10)
Ignition delay [°CA]	20.4 (1500 / 4)
	24.7 (1750 / 6)
	44.1 (2000 / 8)
	51.6 (2000 / 10)
Diesel/Propane ratio	61/39 (1500 / 4)
	59/41 (1750 / 6)
	46/54 (2000 / 8)
	42/58 (2000 / 10)
EGR rate [%]	43.4 (1500 / 4)
	44.2 (1750 / 6)
	44.2 (2000 / 8)
	43.9 (2000 / 10)
Diesel injection timing [° BTC]	22.5 (1500 / 4)
	28.5 (1750 / 6)
	46.5 (2000 / 8)
	51.5 (2000 / 10)

Chapter 3. Experimental Results

In chapter 3, experimental results of three parts are discussed. This chapter is based on the Ref. [1].

3.1 The effects of operating parameters

3.1.1 Diesel injection timing

The investigation of operating parameters started with diesel injection timing. Because the BTC 10 ° of diesel injection timing can achieve ATC 6 ° of MFB50 which was the condition of the maximum power in this experimental engine at this gIMEP 5.2 bar condition, diesel injection timing was gradually advanced from 10 to 45° BTC. After advancing over the 45 °BTC cases, CoV of gIMEP exceeded 5%. Thus, investigating the effect of the SOI (start of injection) timing was conducted in that range. Figure 3 shows that the in-cylinder pressure and RoHR for each of the cases. When the SOI came closer to TDC (10, 20° BTC cases), a rapid HRR was indicated in the early stage as a result of premixed combustion, and after that a smooth curve was shown to arise from mixing controlled combustion. These characteristics of the HRR graphs were similar to those of conventional diesel combustion.

On the other hand, when the injection timing was advanced to 30° BTC, a smooth mixing controlled region rarely existed because the premixing rate increased.

Furthermore, an advancing Start Of Combustion (SOC) was also observed when the SOI advanced [19].

However, the combustion phase was rather retarded when the injection timing was fairly advanced over 40 ° BTC. The reason for this result is that ignition delay was prolonged because the peak equivalence ratio wasn't high enough for occurring to auto ignition due to a highly premixed condition formed by early injection [20, 21]. In other words, second ignition delay of premixed diesel fuel which was an ignition source of combustion, occurred as a formation of lower local equivalent ratio before SOC [22]. It seemed that the combustion regime change was occurred from diesel diffusion flame to almost fully premixed combustion at 40 ° BTC. Low temperature heat releases were also shown in figure 3. In 30,40 and 45 ° BTC cases, the beginning of heat release shapes of those three cases which had prolonged ignition delay also were not continuously rising shape rather ascending after inflection section, which is thought to be considerable relation with the formation of premixed combustion. As a result, the combustion phase was retarded.

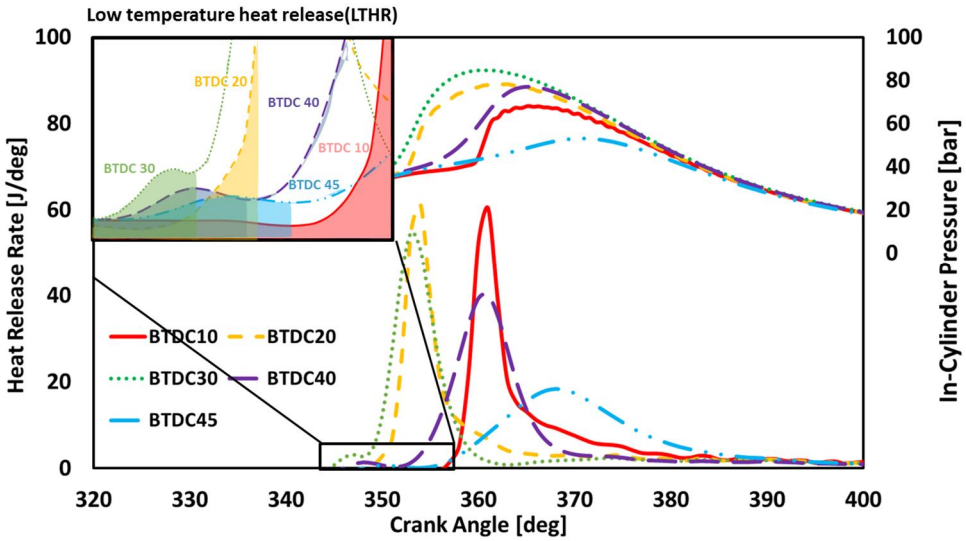

Figure 3. HRR and in-cylinder pressure graphs for various SOI values of diesel [1]

Figure 4 shows the emission levels of NOx and PM in these experimental cases. The NOx emission gradually decreased when the injection timing was advanced from 20 to 45 ° BTC. In the 40 ° BTC case, the NOx emission decreased to approximately 70 ppm. The reason for this result is that the in-cylinder temperature decreased as the local lean area increased due to the increment of the air-fuel premixing rate with prolonged ignition delay. Due to the overall lean premixed mixture formed by prolonged ignition delay, combustion temperature was lower enough to avoid mainly forming NOx and PM temperature. On the other hand, the PM emission increased during the same procedure despite of improving premixing rate. In general PCI diesel combustion, PM formation could be reduced as the decrement of non-homogeneous region of diesel spray area which was the dominant PM formation region. This result can be explained because the piston that was used was the same one from the conventional diesel engine, therefore fuel impingement occurred on the upper side of the piston during the gradually advancing injection timing. However, all the levels of PM emissions remained near the 0.1 FSN.

Although the NOx emission decreased when the injection timing was advanced from 20 to ,45 ° BTC, the CO and THC emissions increased when the injection timing was excessively advanced. Figure 5 shows the CO and THC levels, and figure 5 presents the Gross Indicated Thermal Efficiency (GIE) values for various SOI values. In the 45 ° BTC case, the CO and THC levels increased, and the GIE was lower than 40%. This result can be explained because when the diesel is injected with extremely

advanced timing, the ignition stability can be lost due to a locally over-lean condition. Thus, the CO and THC emissions, which are products of incomplete oxidation, dominantly increased. On the other hand, the GIE remained near 43 % at 40 ° BTC, and the CO and THC emissions remained at low levels.

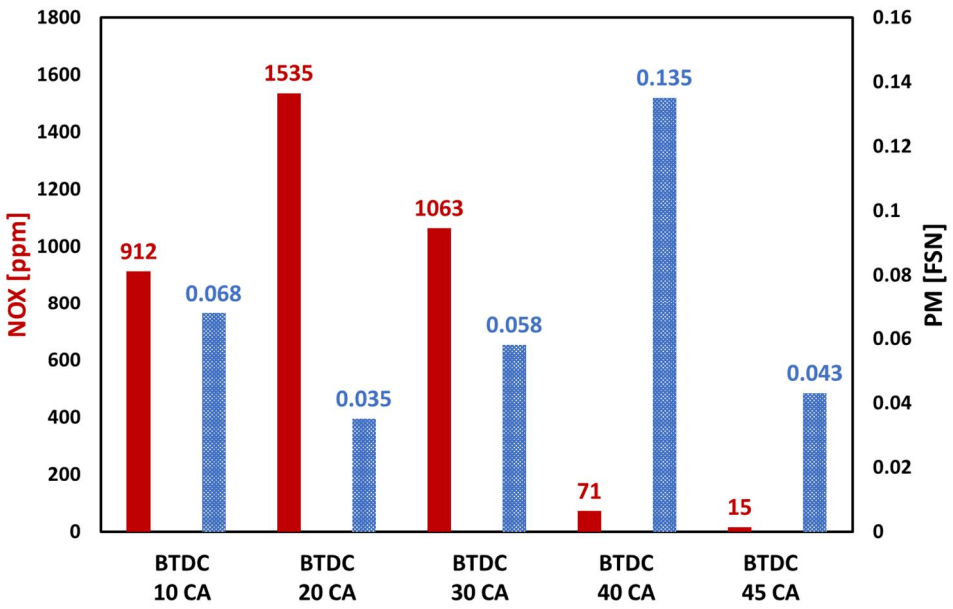

Figure 4. NOx and PM emissions for various SOI values of diesel [1]

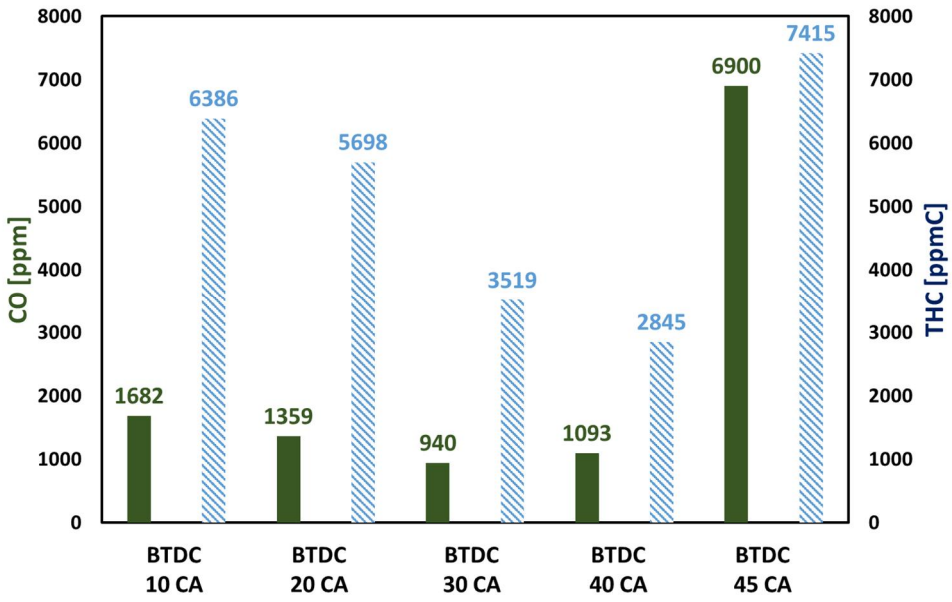

Figure 5. CO and THC emissions for various SOI values of diesel [1]

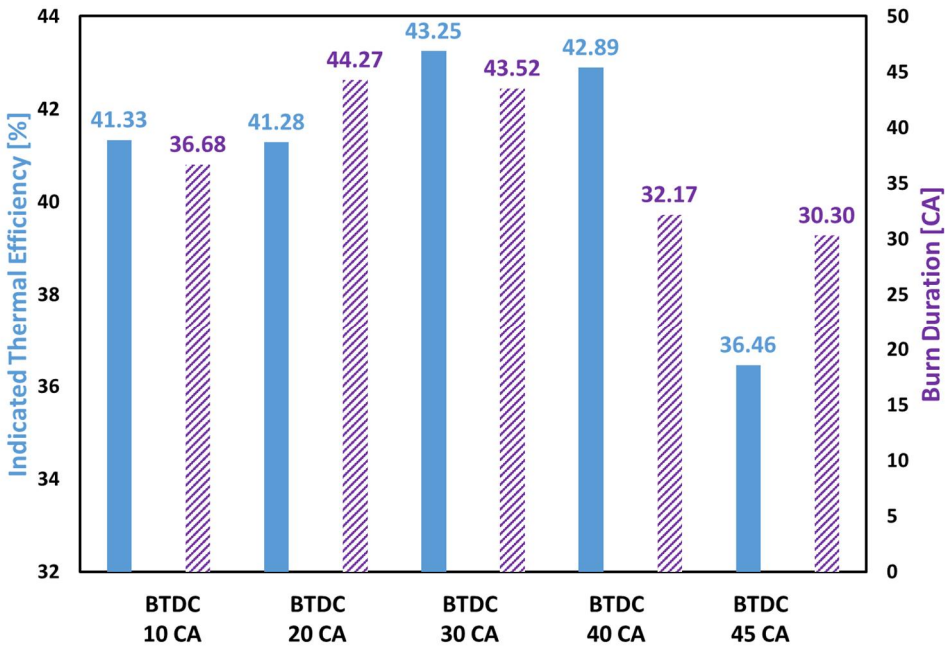

Figure 6. Indicated thermal efficiency for various SOI values of diesel [1]

3.1.2 Propane ratio

Examinations of the effects of the propane ratio were conducted in cases of 20/40/58/65 % of propane. The diesel injection timing was adjusted to fix the Mass Fraction Burned (MFB50) of all the cases at 6 ° ATC.

Figure 7 indicates the NOx and PM emissions for various propane ratios. The PM emission decreased with increasing propane ratio as a result of the decrement of the diffusion flame and the improvement of the premixing rate. The NOx emission remained at similar levels during the changing of this operating parameter [8, 13]. The reason is that the diesel injection timing was advanced when the propane ratio increased to maintain the MFB50 at 6 ° ATC. Thus, the diesel fuel which is main source of NOx formation decreased during the procedure, while combustion temperature was also increased as local region which had unity of equivalent ratio increased. These two effects cancelled out each other, the NOx emission remain similar level during variation of propane ratio.

On the other hand, with the reduction of the high reactivity diesel fuel fraction, the combustion stability worsened. As a result, the products of incomplete combustion, CO and THC emissions, increased with increasing propane fraction, as shown in figure 8. In addition, figure 9 shows that the CoV of the Indicated Mean Effective Pressure (IMEP), which is related with the combustion stability, also gradually increased with increasing propane fraction. The reason of this phenomena can be

explained that decreasing of diesel fuel fraction, which is the main ignition source of dual-fuel combustion, caused lower combustion instability. However, as shown in figure 9, the GIE values of all the dual-fuel combustion cases were higher than those of the diesel single combustion

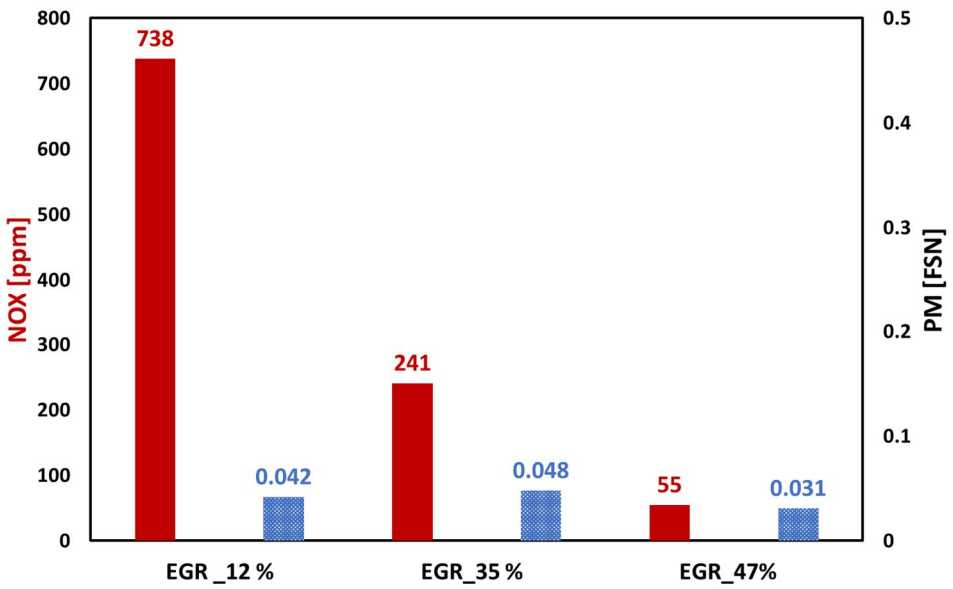

Figure 7. NOx and THC emissions for various propane ratios

(D: diesel, P: propane, #: fraction of each fuel) [1]

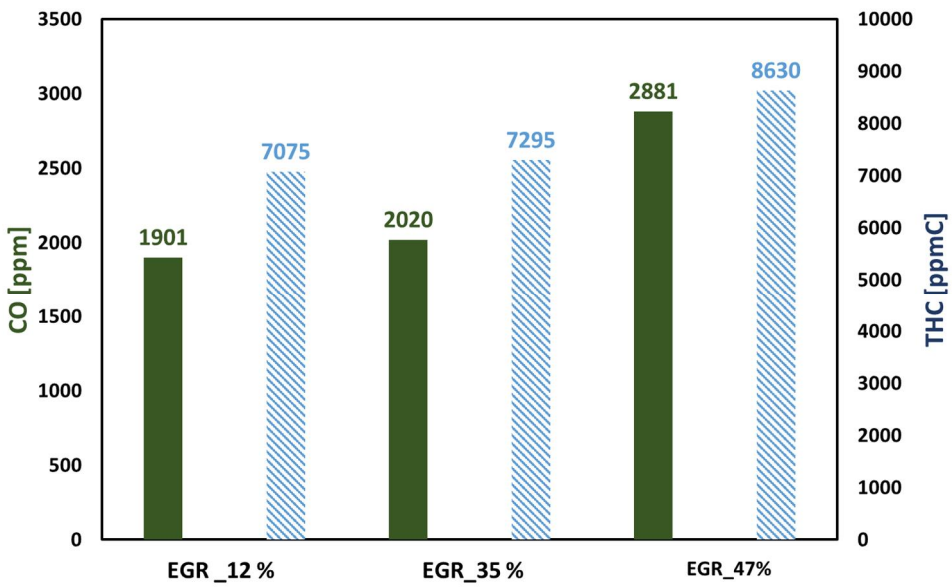

Figure 8. CO and THC emissions for various propane ratios

(D: diesel, P: propane, #: fraction of each fuel) [1]

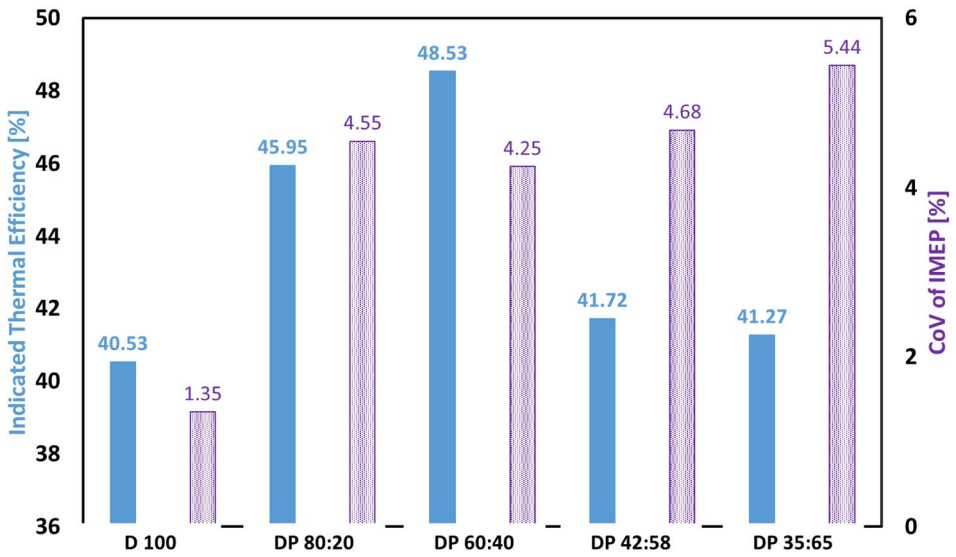

Figure 9. Indicated thermal efficiency and CoV of IMEP for various propane ratios

(D: diesel, P: propane, #: fraction of each fuel) [1]

3.1.3 EGR rates

To investigate the effect of the EGR rates on dual fuel combustion, the EGR rates were changed from 12 to 47 %. The diesel injection timings under various EGR rate swing cases were controlled to maintain the MFB50 value of 6 ° ATC.

Figure 10 presents the NOx and PM emissions for various EGR rates. In a manner similar to Conventional Diesel Combustion (CDC), the NOx emissions decreased as the EGR rates increased due to the heat absorption and reduction of oxygen ratio by the recirculation of exhaust gas. The PM emissions, in contrast to those of CDC which has a NOx and PM emission trade off relationship, were also maintained at a considerably low level.

The reason for this finding is that well premixed gaseous fuel comprises the highest amount of power in dual-fuel combustion. In other words, the amount of directly injected diesel fuel, which is the major cause of the formation of PM emissions, can be reduced [23, 24]. In addition, the LP-EGR system, which is cooled by two inter-coolers, results in a lower EGR temperature and causes a relatively lean condition compared with the HP-EGR system. The incremental levels of PM emissions due to the EGR rate increases were approximately 0.005 FSN, which is the same as the error range of the measuring equipment.

However, CO and THC emissions increase whereas Indicated Thermal Efficiency (ITE) decrease as supplying higher EGR rates, as shown in figures 11 and 12. The reason of this tendency is that combustion temperature became lower as the fraction

of inert gas s increased due to the higher EGR rate. As a result, combustion efficiency deteriorated and higher levels of CO and THC emissions occurred as increasing EGR rate, and ITE decreased [24].

On the other hands, CoV of IMEP was not shown obvious tendency, the results indicated under deviation level.

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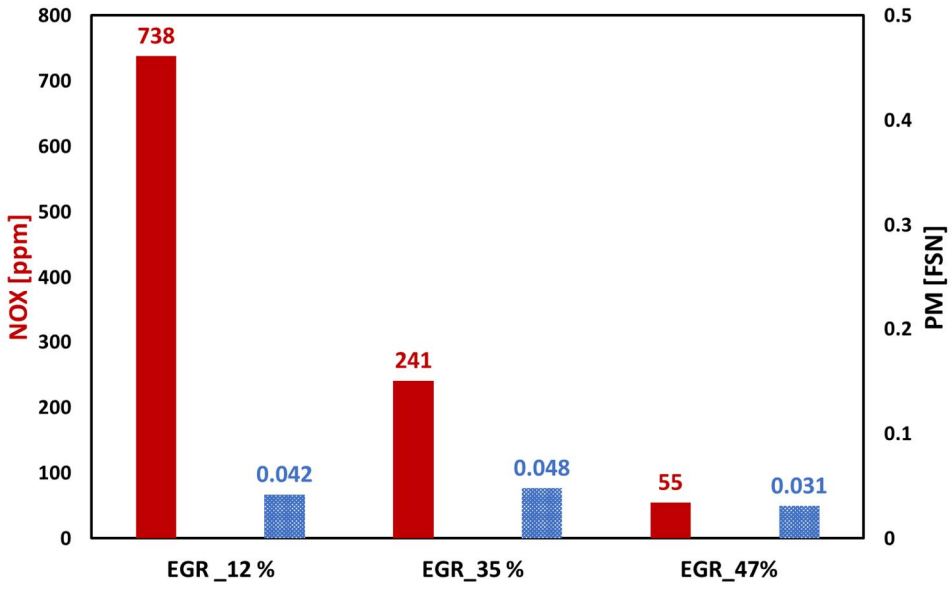

Figure 10. NOx and PM emissions for EGR rates [1]

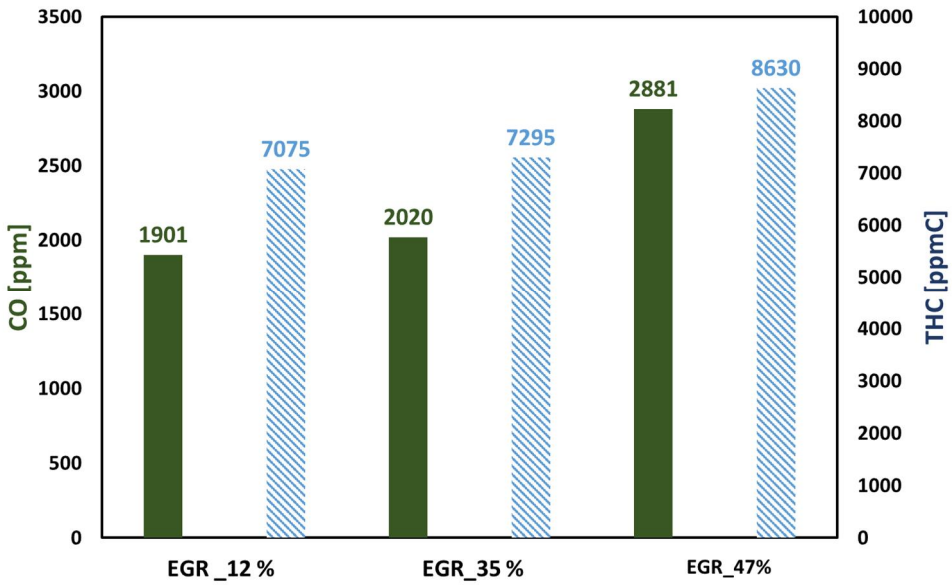

Figure 11. CO and THC emissions for EGR rates [1]

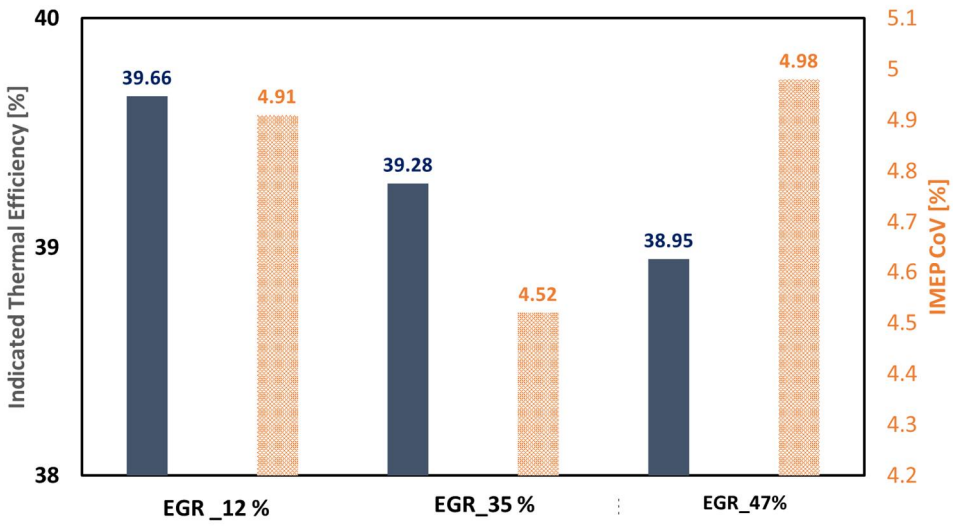

Figure 12. Indicated thermal efficiency and CoV of IMEP for EGR [1]

3.2 Optimization of dual-fuel combustion under the four main operating condition with CDC piston

3.2.1 Optimization result

Based on the parameter study results, optimization of the four main operating conditions was conducted (1500 / 4, 1750 / 6, 2000 / 8, 2000 / 10 [rpm / BMEP]). In all the cases, the intake pressure and injection pressure used the same values of the conventional diesel operating condition. In addition, in all the cases, the operating ranges were constrained by 0.21 g/kWh-in terms of the Indicated Specific NOx (ISNOx), 0.2 FSN for PM emissions and 10 bar/deg for the Maximum Pressure Rise Rate (MPRR) [25]. The two restrictions of the emissions were the minimum levels that can satisfy the EURO-6 emission regulation without any post exhaust treatment, and the limitation of the PRR_{max} was selected from the criteria of the Wisconsin ERC group [7]. The diesel injection timing, propane ratio and EGR rates were adjusted to satisfy all of these restrictions.

Figure 13 shows RoHR and in-cylinder pressure graphs under dual-fuel combustion at main 4 operating point. All the cases satisfy NOx emission level under 0.21 g/kWh. Comparing with single diesel combustion results in the same single cylinder engine, NOx levels was decreased about 30% to as much as 70%. It should be noticed that fast combustion can be available in all cases and the premixed combustion phase is dominant during the entire combustion process. Those heat release shape are gradual

bell-shape form in contrast with conventional diesel engine combustion which has long-late combustion process. And also, LTHR region was occurred in all cases, one of the factors strongly related with premixed combustion.

The ignition delays, which are the duration between SOI SOC, were also remarkably prolonged near 30 °CA in all cases which were achieved fully premixed combustion.

Figure 14 indicates that PM emission and the maximum pressure rise rate of 4 cases. Although all the PM level of 4 cases were lower than 1 FSN, 2000 / 10 [rpm / BMEP] cases could not satisfy the PM constraint for EURO-6 regulation. Due to retaining high diesel fraction to maintain combustion stability with early injection, spray impingement was occurred on the upper side of the piston. However, PM emission was reduced by at least 80% compared with the single diesel combustion results in all cases.

Early diesel injection strategy could be effective way to reduce NOx and PM emission, which can achieve higher mixing rate and lower combustion temperature by overall lean premixed mixture combustion. Therefore, design for optimal piston bowl shape to be available of diesel early injection can be considered to overcome the exceeded PM levels. The pressure rise rate was prevented by the inclusion of low reactivity fuel under relatively low load of 2 cases. On the other hand, 2000 / 8 and 2000 / 10 [rpm / BMEP] cases could not achieve the maximum pressure rise rate under 10 bar / deg. The reason of this result is that locally rich region would be formed in CDC piston, that could cause simultaneous auto ignition. To reduce PRR_{max}, formation of consecutive phi gradient was needed to sequential auto ignition in

cylinder.

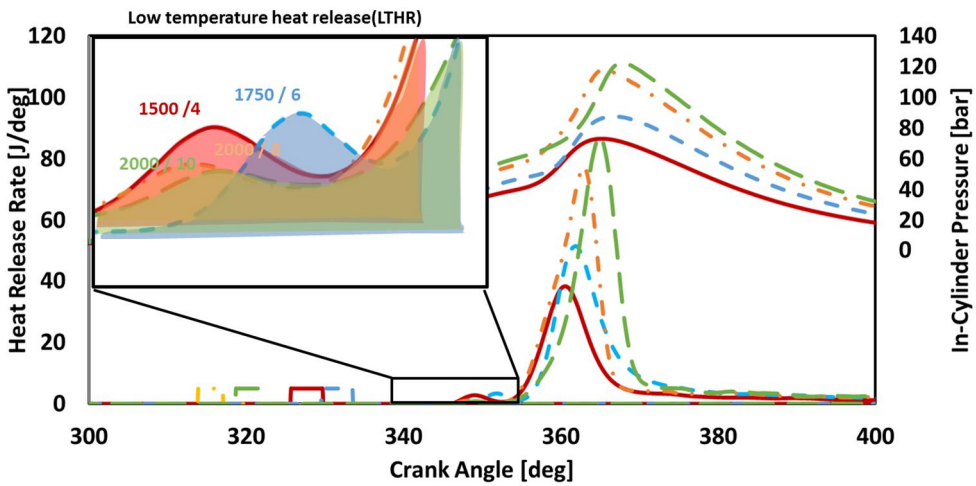

Figure 13. HRR and in-cylinder pressure graphs for dual-fuel combustion at main 4 operating points

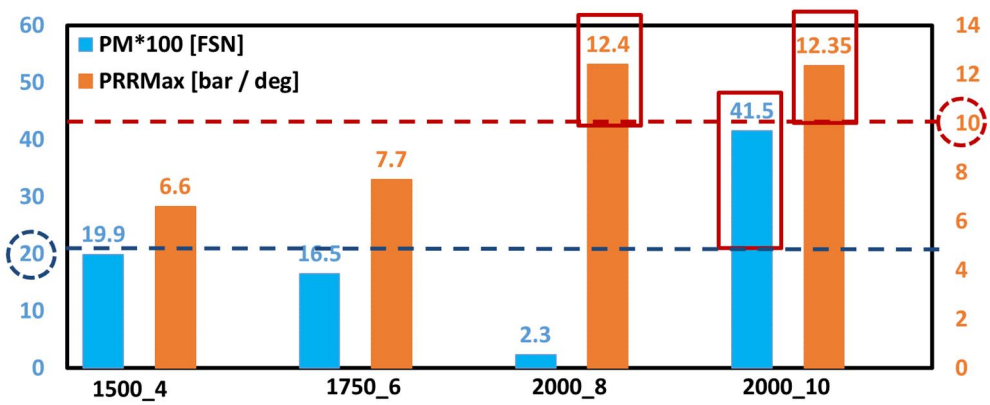

Figure 14. Comparison of PM and PRR_{max} values of dual-fuel combustion at main 4 points (CDC piston)

3.2.2 Modified injection strategy

As the PM and PRR_{max} regulations were not satisfied at 2000 / 10 [rpm / BMEP] case, modified injection strategies were conducted with CDC piston. Due to early diesel injection for PCI combustion, diesel was injection with lower ambient pressure. As a result, diesel spray impingement would be occurred due to enlarged penetration length. Therefore, effects of lower diesel injection pressure were carried out.

Figure 15 shows the PM and PRR results as various diesel injection pressure. Diesel injection pressure of optimized result at 2000 / 10 [rpm / BMEP] was 940 bar, that was the same operating value of commercial diesel engine. During the changing diesel injection pressure, diesel injection duration was also changed to maintain diesel fuel quantity.

From 940 to 800 bar, due to shorten penetration length, PM formation was decreased as reduction of piston impingement. However, when the injection pressure dropped as 450 bar, PM emission increased again. It is thought relatively lower injection pressure caused deterioration of fuel atomization.

As PM level was reduced at 800 bar case, application of diesel split injection was conducted with 800 bar of diesel injection pressure. The experiment was carried out in a way that increased first pilot injection. Figure 16 shows the PM and PRR_{max} values of split injection cases. Due to formation of sequential phi gradient by early

pilot injection, PM and PRR_{max} reduction can be available below each restriction. By these modified injection strategy, there is an optimal point that affected by diesel penetration length and reactivity gradient. Indicated thermal efficiency was retained similar level during the procedure.

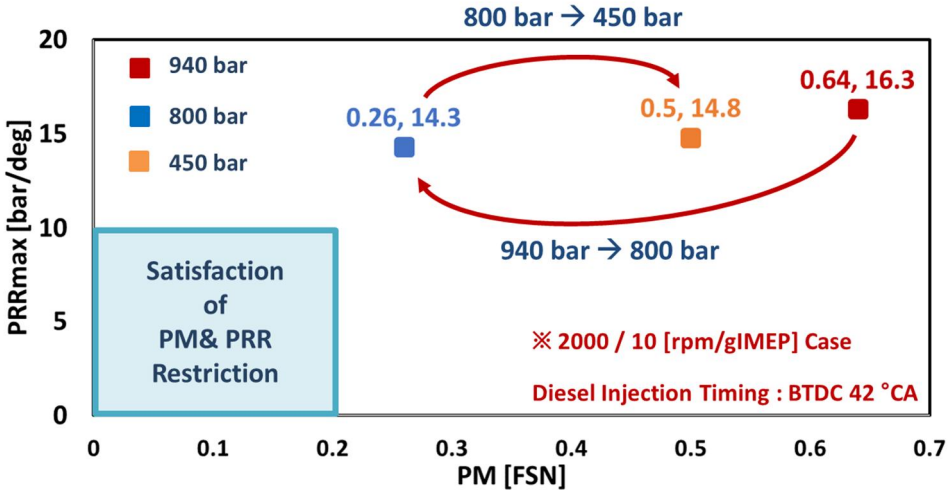

Figure 15. PM and PRRmax values of various diesel injection pressure

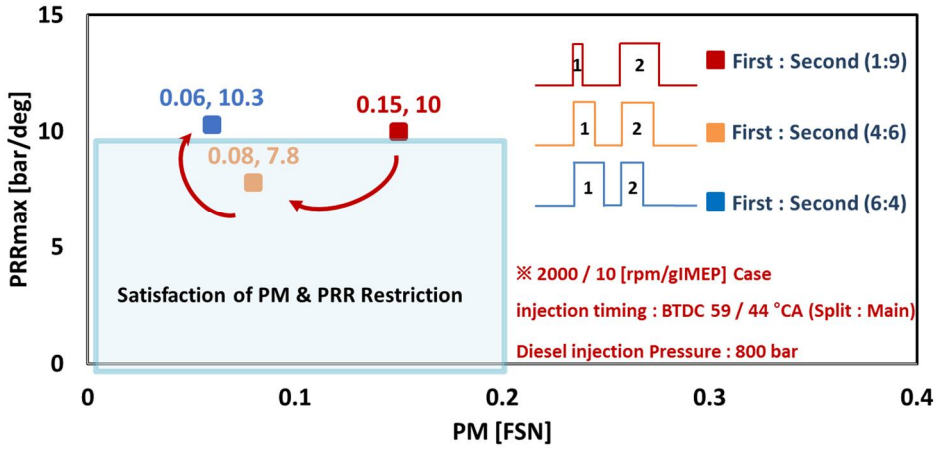

Figure 16. PM and PRRmax values of various diesel split injection

3.3 Optimization of dual-fuel combustion under the four main operating condition with bathtub piston

3.3.1 Optimization result

To compare the effects of piston bowl shape, optimization of 4 main points was also conducted with bathtub piston. As diesel fuel was early injected that can be enough time to air-fuel mixing in dual-fuel PCI combustion, mixing from bowl shape does not need [17]. Therefore, removed center edge was suitable for the premixed combustion.

Figure 17 shows the PM emissions and the PRR_{max} of 4 operating points by using bathtub piston. NOx, PM and PRR_{max} restriction can be satisfied without modifying injection strategies such as diesel injection pressure and pilot injection. On the other hand, comparing to optimized results using CDC piston (figure 14), PM emissions level was generally lower than CDC piston. The reason for this phenomena is that occurrence of diesel spray impingement on the piston bowl is higher in CDC piston comparing with bathtub piston.

Therefore, additional experiment was conducted to examine the combustion characteristics. Figure 18 shows the combustion phase of different pistons at 2000 / 10 [rpm / BMEP] case. All operating parameters, such as diesel injection timing, diesel and propane fuel quantity, intake pressure and EGR rates, are equivalent within each cases. In combustion phase of bathtub piston, smoother combustion phase can be

achieved rather than CDC piston. On the other hand, in CDC piston, it seems that there are two stage combustion is held, the propane fuel burn after auto-ignition of diesel fuel. This phenomena means that fuel mixing was deteriorated in CDC bowl shape compare to bathtub piston. For this reason, there appears to be a higher PM and PRR_{max} levels than bathtub piston.

Considering the piston bowl geometry, there is possibility of reducing PM and PRR_{max} from bathtub piston. Figures 19 and 20 show the spray targeting location as various diesel injection timing with CDC and bathtub piston. From a point of PM formation, possibility of occurrence of diesel spray impingement on the piston bowl shape is higher in CDC piston rather than bathtub piston. These impinged fuel can cause local weak flame near the piston surface which can cause PM emission. And also, locally richer region also can be formed during the early injection in the CDC piston due to existence of untargeted combustion bowl chamber. Bathtub piston have advantages of escaping these phenomena.

From the point of $PRR_{max,}$ smoother combustion can be realized by sequential auto ignition through the reactivity gradient. In that point, CDC bowl shape has an obstacle for the formation of consecutive diesel fuel gradient. Therefore, lower PRR_{max} can be achieved bathtub piston by smoother transition shape from the piston center to squish region.

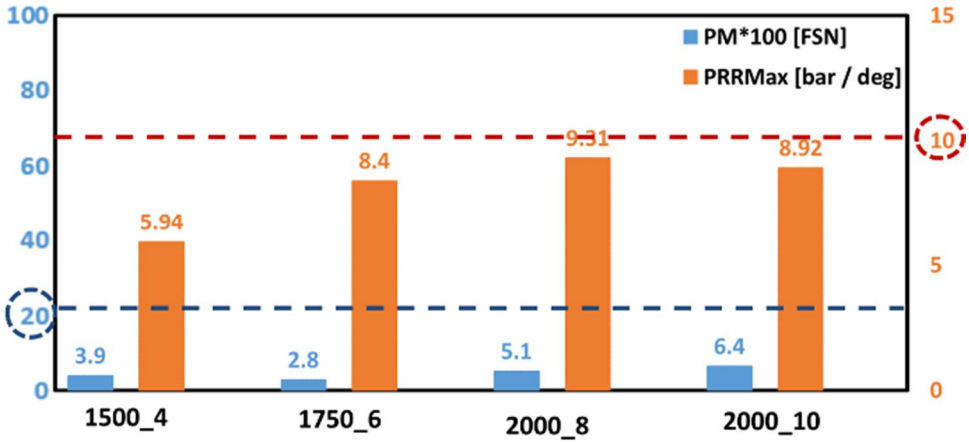

Figure 17. Comparison of PM and maximum PRR values of dual-fuel combustion at main 4 points (Bathtub piston)

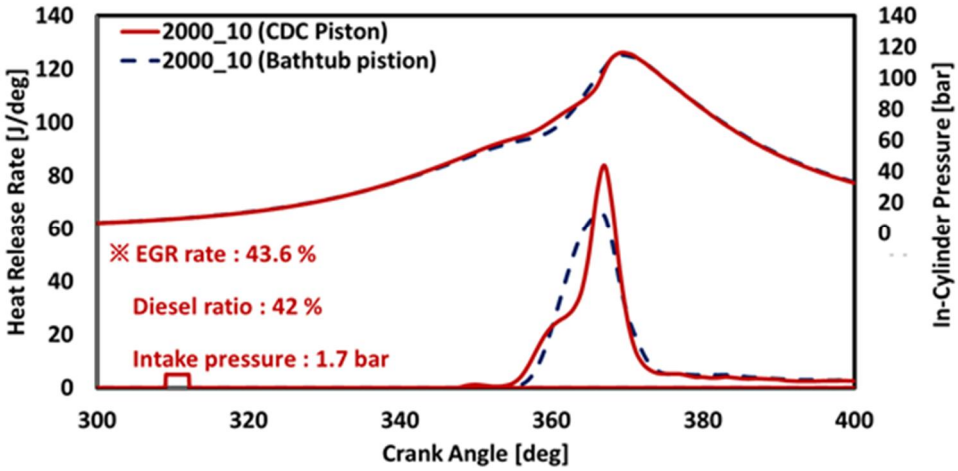

Figure 18. Comparison of combustion phase between CDC piston and bathtub piston

Table 7. Combustion result of same operating parameter with different pistons

Piston	Nox [ppm]	PM [FSN]	PRR_{max} [bar/deg]	ITE [%]	BD [°CA]	EGR [%]
CDC	22	1.212	17.42	41.1	46.52	43.67
Bathtub	19	0.064	8.92	42.7	45.86	43.87

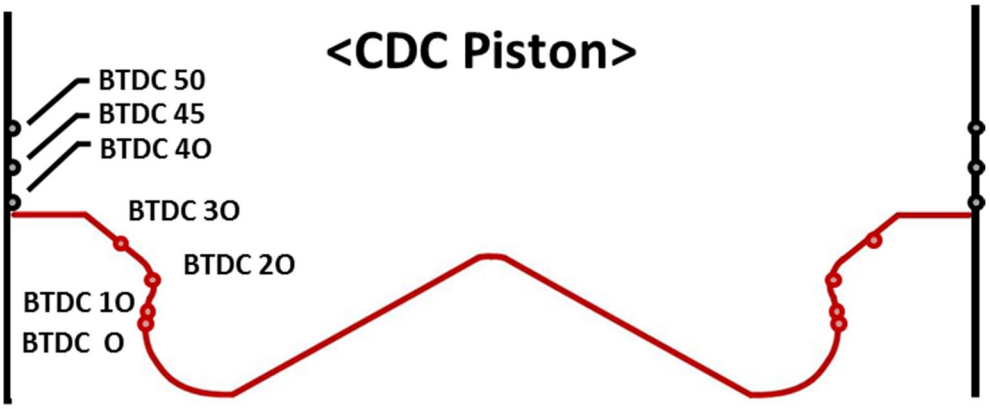

Figure 19. Spray targeting location as diesel early injection (CDC piston)

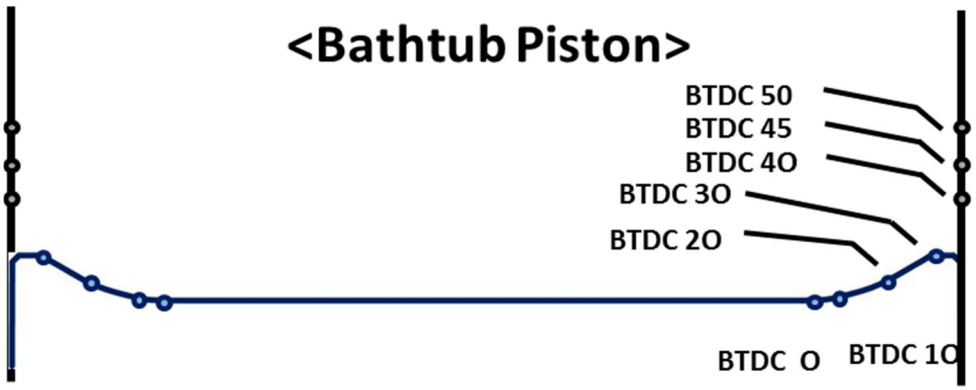

Figure 20. Spray targeting location as diesel early injection (Bathtub piston)

Chapter 4. Conclusion

In this study, the effects of operating parameters which are strongly related with reactivity stratification, diesel injection timing, propane ratio, and EGR rates, were investigated for diesel propane dual fuel combustion [1]. Based on each result, improvement the operating strategies for four main operating points with dual fuel combustion were conducted with different pistons. Lower injection pressure and pilot injection strategy were needed to satisfy the target PM and PRR_{max} restriction in high load operating condition. Using bathtub piston can be potential to reduce PM emission and PRR_{max}, since locally rich region proper to be formed during the early injection in the CDC piston due to existence of untargeted bowl chamber. On the other hand, bathtub piston is free of former phenomena, and also consecutive fuel gradient which would occur sequential auto ignition can be available. The experimental results can be summarized as follows.

1) Advancing the diesel injection timing to realize premixed combustion was an effective way to obtain a higher GIE and a lower NOx emission. However, excessively advancing the SOI can cause ignition stability to deteriorate due to the formation of a locally over-lean region. Consequently, the products of incomplete combustion, CO and THC, increased, and the indicated thermal efficiency decreased [1].

2) Increasing the propane ratio could reduce the PM emissions as a result of a

decrement of the diffusion flame and an improvement of the premixing rate. However, there is no effect on reducing the NOx emissions, and due to the decrement of the high reactivity fuel, the combustion stability declined. As a result, the CO and THC emissions as well as the IMEP CoV values increased, so a relatively high propane ratio was restricted for the preceding reasons [1].

3) Using aggressive EGR rates could be used as one of the ways to reduce NOx emissions in the same manner as is typical of diesel combustion, however, combustion efficiency would be deteriorated for a reason of lower combustion temperature On the other hand, there is no NOx and PM emission trade off relationship. It is considered that a well premixed gaseous fuel takes the dominant possession of the power in dual fuel combustion. Additionally, the LP-EGR system had a positive effect on reducing the PM emissions via the formation of an overall lean mixture condition [1].

4) Optimization results of dual-fuel combustion under the 4 main operating condition with CDC piston can be satisfied NOx emission. NOx levels was effectively decreased about 30% to as much as 70% compared to the conventional diesel combustion.

PM emission was significantly reduced by at least 80% compared with the single diesel combustion results in all cases. However, PM emission exceeded restriction level at 2000 / 10 [rpm / BMEP] case, and PRR_{max} was not satisfied at 2000 / 8 and 2000 / 10 [rpm / BMEP] cases. As retaining high diesel fuel quantity to maintain combustion stability, piston bowl spray targeting problem would be occurred in high

load condition. And also, locally rich region would be formed in CDC piston, that could cause simultaneously auto ignition. As a result, PRR_{max} was relatively high level at high load condition. In the case of Indicated thermal efficiency, it has risen by about 1 to 5 % or maintained at equivalent level compared with the conventional diesel combustion through the 4 cases.

By implementing lower diesel injection pressure as 800bar, PM emission can be satisfied with target restriction as the decreased impingement of piston surface. On the other hand, when the diesel injection pressure was under 450bar, PM emission increased again as the deterioration of fuel atomization.

5) Optimization results of dual-fuel combustion under the 4 main operating condition with bathtub piston was also conducted. All the cases can be satisfied NOx, PM and PRR_{max} restriction. Especially, PM and PRR_{max} levels significantly lower than optimizing results of CDC piston, reduced about 90% compared to CDC piston. As following additional experimental results which is all the operating parameters are same except for piston shape, PM and PRR_{max} levels are remarkably lower at bathtub piston. One possibility for this result is that bathtub piston has an advantage of reducing formation of local richer region during the early diesel injection by smoother transition shape form the piston center to squish region. These shape has also an advantage of sequential reactivity gradient, that can affect whether formation of smoother combustion phase which is held through successive auto ignition through the reactivity gradient.

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국 문 초 록

디젤 프로판 융합 연소 엔진에서 연소 최적화에 관한 연구

서울대학교 기계항공공학부
기계공학과
강재구

요 약

승용 디젤엔진에 대한 배기규제가 갈수록 심화되고 있다. 유럽 배기 규제의 경우, 이전 10 년 대비 PM 은 80% 수준, NOx 는 70%수준으로 낮은 기준을 만족시켜야 한다. 또한, 2011 년부터 입자상 물질에 대해 PN 규제도 시작 되었다. 점차 강화되는 배기규제를 만족하기 위해서 저온 연소 및 예혼합 연소를 바탕으로 하는 융합연소 연구가 활발하게 진행되고 있다.

본 연구에서는 디젤 프로판 융합 연소개선을 위해 운전 변수 최적화 및 피스톤 보울 형상 변화를 바탕으로 주요 운전 4 지점에서 최적화를 진행하였다. 실험은 크게 세 파트로 구성되어있다. 첫 번 째 파트에서는 디젤 분사 시기, 프로판 분율, EGR 율 변화에 따른 효과를 파악했다. 융합 연소 특성 파악을 위해 NOx, PM, PRR_{max}, 열효율, IMEP CoV 인자들을 사용하였다.

이를 바탕으로 예혼합 융합연소를 통한 주요 운전영역 4 지점 최적화를 진행하였고, 최적화는 CDC 보울 피스톤과 bathtub 보울 피스톤을 사용하여 각각의 결과를 비교하였다. 해당 영역 최적화를 위해서 배기 후처리 장치 없이 Euro-6 배기 규제 수준을 만족시킬 수 있는 NOx, PM 기준을 정했고, 선행 연구 논문을 바탕으로 PRR_{max} 기준을 선정했다.

두 번째 파트에서는 CDC 보울 피스톤을 이용한 최적화 결과를 분석하였다. 예혼합 융합 연소를 바탕으로 저부하 영역에서 낮은 NOx, PM, PRR_{max} 수준을 확보할 수 있었지만, PM 과 PRR_{nax} 의 경우 고부하 영역에서 선정 기준을

초과하는 결과를 얻었다. 이에 디젤 분사압을 낮추고, 스플릿 분사전략을 도입하는 등 디젤 분사 전략 변화를 통해 디젤 연료의 피스톤 충돌을 피하고 연소실 전 영역에 반응성 성층화를 형성함으로써 해당 PM, PRR_{max} 기준을 만족시킬 수 있었다.

세 번째 파트에서는 동일한 최적화 실험을 Bathtub 형상 피스톤을 이용한 진행하였다. Bathtub 형상 피스톤을 사용 시 전 영역에서 앞선 배기 및 PRR_{max} 규제를 만족시킬 수 있었고, CDC 형상 피스톤 대비 낮은 PM 및 PRR_{max} 수준이 나타남을 확인할 수 있었다. 피스톤 형상 효과를 확인하기 위해 모든 운전 변수가 동일한 조건에서 피스톤만 변경한 상태로 실험을 진행하였고, bathtub 형상 피스톤에서 PM 및 PRR_{max} 저감이 확연하게 나타났다. 이는 디젤 연료를 조기분사할 경우 연료가 라이너 부근에 도달하게 되는데, 피스톤 중심부를 향하는 각이 진 형태보다 완만한 형태의 보울 형상에서 연료가 더 완만하게 분포되어 국부적으로 농후한 영역 생성이 줄어들고, 연료 자발화가 연료 구배에 따른 연속적인 방향으로 진행되어 낮은 PM 및 PRR_{max} 수준을 확보가 가능했을 것이라 판단된다.

주요어: 디젤 엔진, 이종 연료 연소, 예혼합 압축착화, 피스톤 보울 형상, 반응성 조정 압축착화 연소

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