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CNG/Diesel Dual Fuel Engine Combustion and Emissions Characteristics – An Experimental Analysis According to Common-rail Pressure and Fuel Mixing Ratio

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압축천연가스/디젤 이중 연료 엔진의 연소 및 배기 가스 특성 – 커먼레일 압력 및 연료 혼합비에 따른 실험적 분석

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by

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NOMENCLATURE

ATDC	After top dead centre
BDC	Bottom dead centre
BTDC	Before top dead centre
CA	Crank angle
CI	Compression ignition
CNG	Compressed natural gas
CO	Carbon monoxide
CO_2	Carbon dioxide
COV	Coefficient of variability
ECU	Electronic control unit
HC	Hydrocarbon
IC	Internal combustion
IMEP	Integrated mean effective pressure
NOx	Nitrogen oxides
NO_2	Nitrogen dioxide
PM	Particulate matter
RPM	Revolution per minute
SI	Spark ignition
SOI	Start of injection
TDC	Top dead centre

CNG/Diesel Dual Fuel Engine Combustion and Emissions Characteristics – An Experimental Analysis According to Common-rail Pressure and Fuel Mixing Ratio

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Abstract

디젤 엔진은 일반적으로 발전기 및 이동식 차량 등 수많은 산업 분야에서 사용되고 있다. 그러나 디젤 엔진은 질소 산화물(NOx), 입자상 물질(PM), 이산화탄소(CO_2) 등과 같은 환경오염 가스가 발생한다는 문제점이 있다. 이렇게 발생한 대기 오염 물질은 최근 환경 보호 위원회 등에 의해 엄격하게 규제가 되고 있다. 또한 한정된 화석연료 자원에 대하여 인류의 과도한 사용이 화석연료 자원 고갈을 가속화하고 있다. 따라서 디젤 엔진의 대체 연료 사용 및 개발이 연구자들 및 분야의 종사자들에게 있어 주요한 사항이다. 이러한 분야의 연구 및 발전을 위해, 본 논문은 주로 천연가스가 있는 이중 연료 디젤 엔진의 연소 특성에 대한 커먼레일 압력과 연료 혼합 비율의 변화에 대한 실험을 다룬다. 이 연구를 위해 커먼레일 4 행정 디젤 엔진을 이중 연료 모드로 운전하였으며, 디젤 연료 사용 시 엔진 특성과 비교하였다. 엔진 테스트는 적절한 분사 타이밍으로 11.7kW 및 23kW의 서로 다른 엔진 부하에서 3개의 커먼레일 압력(50MPa, 60MPa, 70MPa)에서 이루어졌으며, 천연가스 비율은 20%에서 90%까지를 사용하였다. 실험 결과는 열 방출 속도와 실린더 압력이 천연가스 비율의 증가와 비례 관계를 보여주었다. 또한, 모든 시험 조건에서 디젤 연료와 비교할 때 이중 연료 모드에서의

NOx와 CO 발생률이 감소하였다. 그러나 이중 연료 모드 결과로만 미루어 볼 때, 커먼레일 압력이 증가함에 따라 CO는 감소하고 NOx는 증가했다. 요약하자면, 이중 연료 모드는 디젤 모드에 비해 NOx와 CO_2 가스 배출이 적지만 HC와 CO의 발생률이 더 높게 나타났다. 또한, 커먼레일 압력이 증가하게 되면 HC와 CO 배출량은 감소하였다. 그러나 NOx와 CO_2 배출량은 약간 증가하였다.

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1 Introduction

In 1876 Sir Nikolaus Otto first developed the spark-ignition engine and in 1892 Sir Rudolf Diesel invented the compression-ignition engine. These SI engines and CI engines are called internal combustion engines. Internal combustion engines produce mechanical power from the chemical energy contained in the fuel. This energy is released by burning or oxidizing the fuel inside the engine. The main theme in this paper is compression-ignition engines (diesel engines). Diesel engines are used as a power source for passenger cars, railroad locomotives, marine vehicles, construction equipment, farming equipment, etc., Because diesel engines are extremely efficient and cost-effective [1,2]. They are designed to operate at higher compression ratios, basically between the ratio of 15 to 20. So, the higher the compression ratio of an engine, the higher the thermal efficiency and diesel engines run on fuel-lean condition. However, diesel engines contribute to environmental pollution gases [3,4]. The main emission from diesel engines is NOx and PM. NOx emission can cause photochemical smog and it can also cause acid rain. Generally, PM from diesel engines consists of numerous types of chemical components such as elemental carbon, organic carbon, inorganic ions, trace elements etc. [5-7]. These particles have a seriously harmful impact on human health and the environment [8-11]. Therefore, emission regulations are becoming tighter to reduce this kind of environmentally harmful emissions. Apart from that, energy demands are continuously increasing, but the oil resources are decreasing. To solve the need for energy demands and the decreasing oil resources, while at the same time reducing the environmental pollution gases, the utilization of alternative fuel has been found to be an effective solution.

Among many kinds of alternative fuels, natural gas is very potent and highly attractive in the transportation sector. Firstly, natural gas can be extracted in many areas worldwide at affordable prices. Secondly, natural gas is composed of about 87 to 96 percentages of methane, which is well known as a greenhouse gas. So natural gas is an eco-friendly fuel, and it can also generate the reduction of CO_2 emission because it has the lowest carbon to hydrogen ratio than any other fossil fuels. Thirdly, natural gas can also considerably reduce the NOx emission and simultaneously produce almost zero smoke and PM [12-14]; which is hardly difficult to achieve with diesel fuel. Furthermore, it can be used in engines with high compression ratio and get higher thermal efficiency compared with that of the normal gasoline engines.

Natural gas is being used as a supplementary fuel widely in diesel engines for its economic benefits and environmental friendliness [15-18]. The objective of this research is to investigate the combustion and emission characteristics of a CNG/diesel dual fuel engine with different common-rail pressure and natural gas mass ratio.

2 Literature Review

2.1 Operating of a CNG-Diesel Dual Fuel Engine

When a diesel engine is changed into a CNG/diesel dual fuel engine, it operates mostly the same as the original diesel engine. Figure 2.1 shows the idealized cycle of the diesel engine.

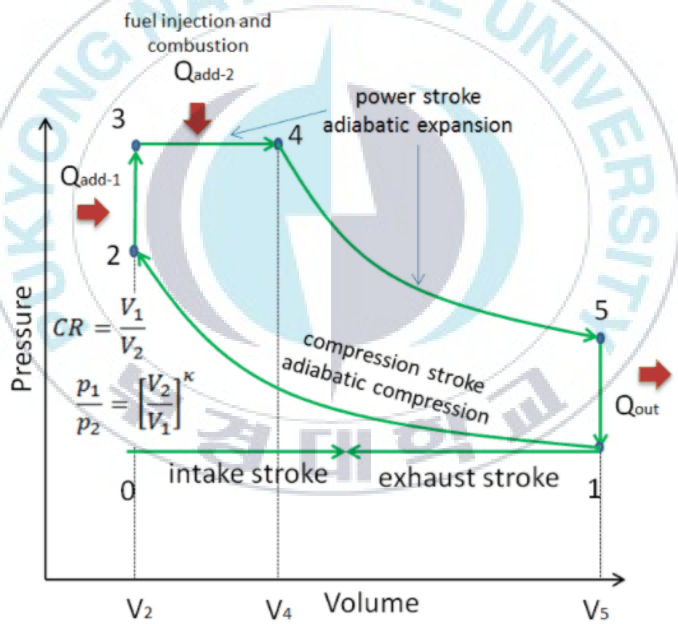

Figure 2. 1 Diesel engine operating cycle [22]

The dual fuel engine has 4 strokes of operation-

1. Intake Stroke: As the piston move BTDC 7°, Inlet valve open and a controlled amount of natural gas is injected into the inlet air manifold and entered into the cylinder as an NG/air mixture.

2. Compression stroke: During this stroke, the piston moving upward to the TDC and compressing the natural gas-air mixture causes the increase in-cylinder temperature. Before the TDC13°, ECU control quantity of pilot diesel is injected by the nozzle into the combustion chamber.

3. Power Stroke: In this stroke, both valves are closed and the temperature is sufficiently high due to the high compression ratio and pilot diesel act as a source of ignition. This causes the combustion of natural gas and air mixture that produce power to piston moves back to the cylinder.

4. Exhaust Stroke: In this stroke the exhaust valve opens and the piston moving upward again. it expels the burning exhaust gases through the exhaust valve, thus completing the 4-stroke cycles. Figure2.2 shows the complete cycle of CNG/Diesel dual fuel operation.

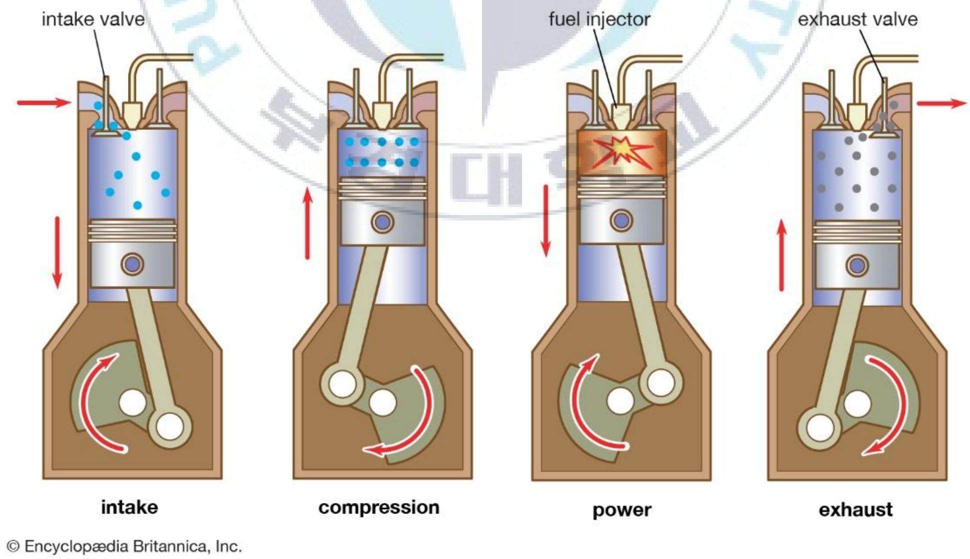

Figure 2. 2 CNG/Diesel dual fuel engine 4-stroke operation [29]

2.2 Natural gas as an alternative fuel

Natural gas can naturally be found in the HC gas mixture including more than 90 percentages of methane, but commonly includes some varying amounts of other minor gases. Using natural gas in internal combustion engines has some advantages. It can be reduced the generating of CO_2 and NOx with a comparison of the other petroleum-derive fuels [19].

2.3 Physicochemical properties of natural gas

Natural gas is a naturally occurring hydrocarbon gas mixture consisting primarily of methane and also contains some kinds of higher alkanes, such as methane, ethane, propane, n-butane and isobutane, and pentanes. It may also contain carbon dioxide, nitrogen and some amounts of water vapor. The composition and content of natural gas vary considerably depending on the source and the production process. The typical composition of natural gas is listed in Table 2.1. Commonly, methane includes 87–96% of natural gas. Therefore, the physicochemical properties of natural gas are pretty similar to methane. Table 2.2 shows the comparison of the properties of natural gas and diesel.

Composition	Formula	Volume Fraction (%)			
		Ref. 1	Ref. 2	Ref. 3	Ref. 4
Methane	CH_4	92.07	94.39	94.00	91.82
Ethane	C_2H_6	4.66	3.29	3.30	2.91
Propane	C_3H_8	1.13	0.57	1.00	-
ISO-Butane	I-C_4H_{10}	0.21	0.11	0.15	-
N-Butane	N-C_4H_{10}	0.29	0.15	0.20	-
ISO-Pentane	I-C_5H_{12}	0.10	0.05	0.02	-
N-Pentane	N-C_5H_{12}	0.08	0.06	0.02	-
Nitrogen	N_2	1.02	0.96	1.00	4.46
Carbon Dioxide	CO_2	0.26	0.28	0.3	0.81
Hexane	C_6+ (C_6H_{14}	0.17	0.13	0.01	-
Oxygen	O_2	0.01	<0.01	-	-
Carbon Monoxide	CO	<0.01	<0.01	-	-
Total	-	100	100	100	100

Table 2. 1 Composition of natural gas [20]

Fuel Properties	CNG	Diesel
Low heating value (Mj/kg)	48.6	42.5
Heating value of stoichiometric mixture (Mj/kg)	2.67	2.79
Stoichiometries air–fuel ratio (kg/kg)	17.2	14.3
Auto-ignition temperature (°C)	650	180-220
Octane number	130	-
Cetane number	-	52.1
Carbon content (%)	75	87

Table 2. 2 Comparison for properties of natural gas and diesel [21]

2.4 Combustion characteristics of dual fuel engine

Dual fuel engine combustion processes are more complicated than the general diesel engines or spark-ignition engines. For the combustion of CNG, it is included more than 90 percentages of methane (CH_4). Under the stoichiometries condition, the general combustion process can be shown by the fuel (hydrocarbon) plus oxidizer (air or oxygen) called the "Reactants", which takes place chemical process and release the "Products" of combustion. In idealized stoichiometric combustion, all of the carbon is formed carbon dioxide (CO_2) and all of the hydrogens are formed water (H_2O) in the products, thus equation 1 shows the chemical reaction for CNG/Diesel dual fuel combustion. When analyzing the combustion of the dual fuel engine, in-cylinder pressure, peak pressure, ignition delay and heat release rate are a significantly important parameter to understand. In the below section, these parameters are explained.

$$C_xH_y + z(O_2 + 3.76N_2) \longrightarrow aCO_2 + bH_2O + cN_2 + O_2 \quad (1)$$

This is just an ideal reaction and never happens in reality. As follow the stage of lean and rich fuel, the composition of the combustion products is remarkably different because the stoichiometric fuel-air ratio relied on the fuel composition, in the other parameter called the equivalence ratio (∅), as shown in equation 2.

$$\emptyset = \frac{(\frac{F}{A})_{actual}}{(\frac{F}{A})_{stoich}} \quad (2)$$

Figure 2.3 shows the different stages of dual fuel combustion, which are explained by the heat release rate graph.

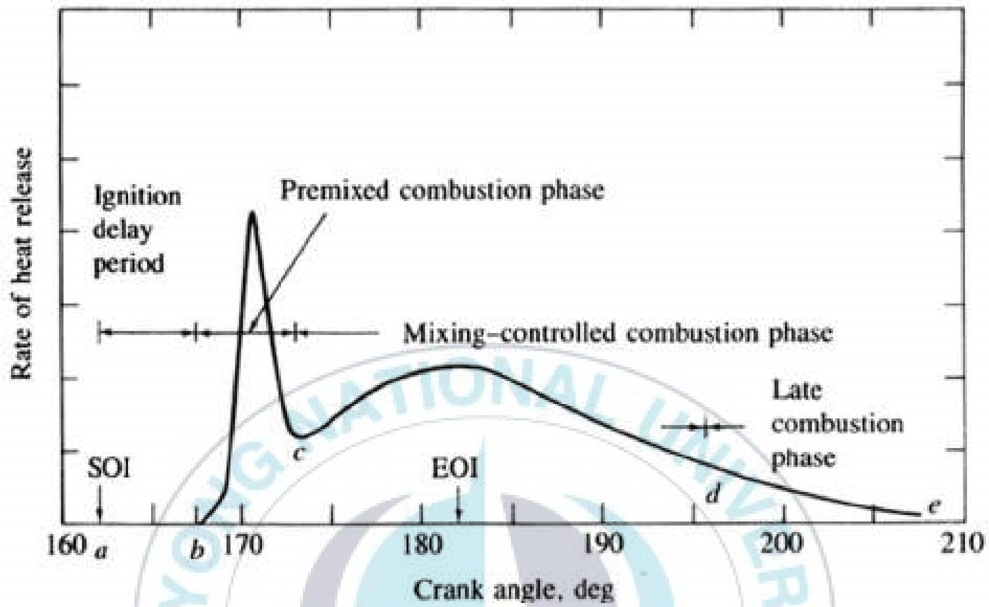

Figure 2. 3 The different stages of dual fuel combustion [22]

1. Ignition delay: The ignition delay of dual fuel engine can be defined as the crank angle within the start of injection and the start of combustion (a to b). A continuous reaction of physical and chemical processes take place within the ignition delay before the combustion. During this stage, the rate of heat release is dropped to below zero due to absorption of the heat by vaporizing fuel. Equation (3) shows the formula of ignition delay.

$$\tau_{id} = a\emptyset^{-k^*}P^{-n^*}\exp\left(\frac{E_a}{R_u T_{cyl}}\right) \qquad (3)$$

τ_{id} = Ignition delay

$\emptyset$ = Equivalence ratio

E_a = Activation energy

T_{cyl} = Cylinder charge temperature

R_u = Gas constant

a, k^* and n^* = =Empirical constants

2. Rapid combustion or premixed combustion phase: This stage happens during very slight range of crank angle. Pilot diesel mixes with air and natural gas mixture and then burns rapidly, resulting in a high heat release rate, as can be seen from the highest peak (b to c).

3. Mixing controlled combustion phase: In this stage, the heat release rate is not high compared to the peak during the rapid combustion phase, but it takes place over a larger period of crank angle. Too many processes that take place in this phase. Pilot diesel atomizes, vaporizes, and mixes with air and natural gas mixture and then burns with a diffusion flame (c to d).

4. Late combustion phase: This is the last stage of heat release. This stage takes place within a few crank angle degrees. It could be because of some remaining fuel, or some energy remains in soot and fuel-rich combustion products (d to e).

2.5 In-cylinder pressure

By using a piezoelectric transducer, in-cylinder pressure can be measured and any other combustion parameters such as maximum-cylinder pressure, indicated mean effective pressure, coefficient of variation of mean effective pressure, heat release rate and so on can be analyzed based on in-cylinder pressure. Basically, in-cylinder pressure under dual fuel mode was slightly lower than diesel mode within the period of the compression stroke and the initial stage of combustion. Figure 2.4 shows the cylinder pressure and heat release rate under diesel mode and dual fuel mode.

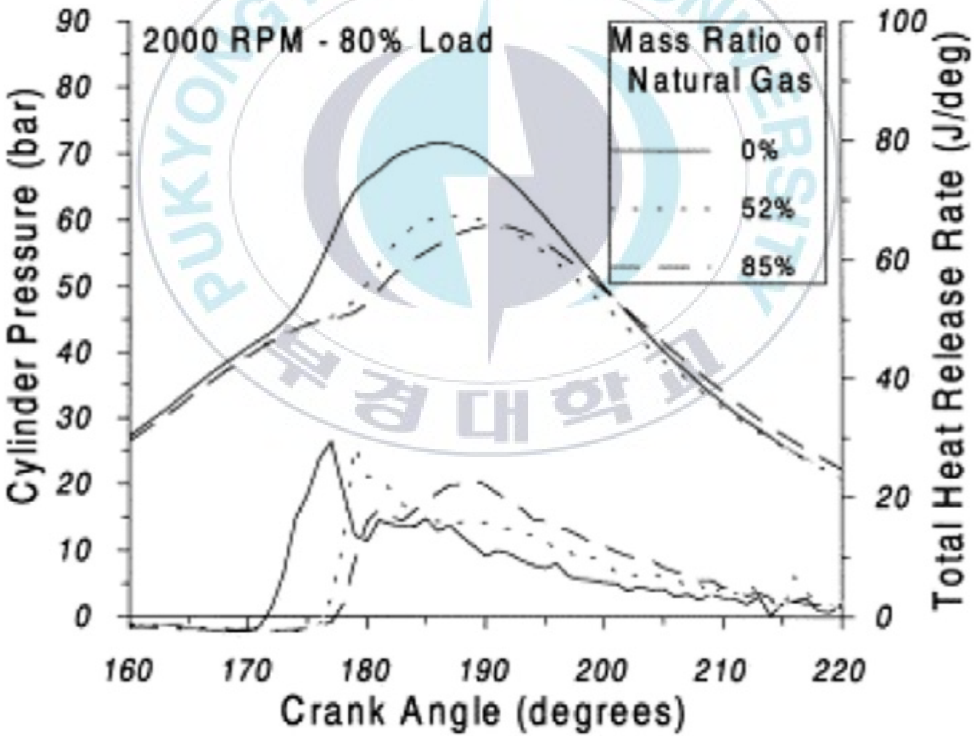

Figure 2. 4 The cylinder pressure and heat release rate under diesel mode and dual fuel mode [21]

2.6 NOx emission

Nitrogen oxides generally produced from the reaction between oxygen and nitrogen within the combustion of hydrocarbons, especially at high temperatures. The mixture of nitric oxide (NO) and nitrogen dioxide (NO_2) is represented as NOx. In the sector of diesel engines, produced huge amounts of NOx emission because of combustion temperature is high above 1800k and lack of oxygen and nitrogen in the combustion chamber also known as the Zeldovich mechanism [23]. NOx can be formed in flames by the three mechanisms: the thermal mechanism, the prompt mechanism and the nitrous oxide (N_2O) mechanism [23] as described in equations (4), (5), and (6).

$$O+N_2 \leftrightarrow NO+N \tag{4}$$

$$N+O_2 \leftrightarrow NO+O \tag{5}$$

$$N+OH \leftrightarrow NO+H \tag{6}$$

equations (3) and (4) were described by Zeldovich (1946) and equation (5) was noted by Lavoie el al. [24] as they summarized those chemical equations contributed under equilibrium conditions. There are too many strategies have been discovered to reduce NO_x emissions in an IC engine. Cenk Sayin et al. [25] showed that a way to reduce NO_x emissions were by retarding the injection timing because it resulted to decrease in the peak cylinder pressure and cause the lower peak temperatures. Jie Liu et al. [26] showed the using CNG and Diesel dual fuel can be reduced NOx emissions, as it would be most of the fuel is burned under lean premixed conditions which results in lower cylinder temperature.

2.7 CO emission

Carbon monoxide emission is one of the harmful emissions released from the engine. It's can be generated from incomplete combustion and lower cylinder temperatures. Too much CO is generally produced in the fuel-rich region due to lack of oxygen. Nonetheless, a huge amount of CO can also be generated in the fuel-lean region due to the combustion chamber temperature is less than 1450 k. CO formation mechanism can be summarized by the equation (7)

$$RH \rightarrow R \rightarrow RO_2 \longrightarrow RCHO \rightarrow RCO \rightarrow CO \quad (7)$$

Where R indicates for the hydrocarbon radical. Equation (8) shows the principal CO oxidation reaction in hydrocarbon air flames.

$$CO + OH = CO_2 + H \quad (8)$$

3 Experimental Setup and Test Conditions

3.1 Test Engine

To examine the effect of common-rail pressure and fuel mixing ratio on combustion characteristics in dual fuel diesel engine, a turbocharged, intercooled, in-line 4-cylinder diesel engine with 83 mm bore, 130mm stroke, 1991cc displacement and a compression ratio of 17.2 are used in this study. It was built with a high-pressure common-rail injection system. The main specifications of the engine are shown in Table 3.1. Figure 3.1 demonstrates a schematic diagram of the experimental apparatus.

Description	Specification
Bore × Stroke	83 × 130mm
Number of cylinders	4
Piston displacement	1991cc
Maximum power	84/4000 kW/rpm
Compression ratio	17.7
Inlet valve opening timing	7°CA
Inlet valve closing timing	43°CA
Exhaust valve opening timing	52°CA
Exhaust valve closing timing	6°CA

Table 3. 1 Test Engine Specification

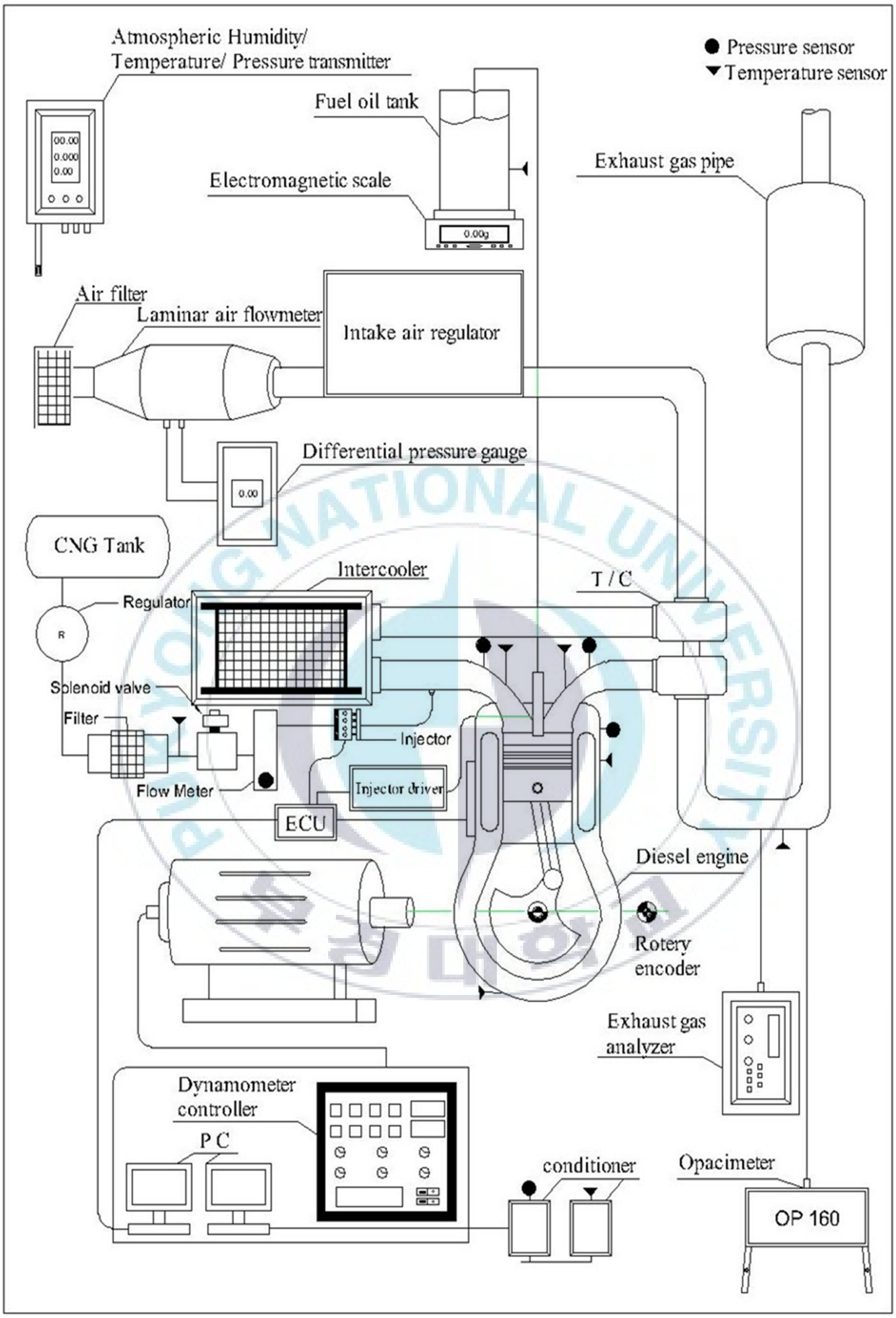

Figure 3. 1 Schematic diagram of the experimental apparatus

3.2 Dynamometer

The load on the engine was controlled by using a DC dynamometer (130 kW) (manufactured by Hwan Woong Mechatronic Co., Ltd.) coupled to the engine. It can measure up to 130 kW/10000rpm.Table 3.2 shows the specifications of this dynamometer. It was water-cooled and the engine and the dynamometer were controlled by adjusting the settings on a Dialog Testmate 25 dyno and throttle controller. Cooling water temperatures were monitored during the test to prevent the overheating of the dynamometer. There are two types of adjustment RPM constant and Torque constant. In this experiment, the RPM was fixed to change the torque and adjust the load. Figure 3.2 shows the cross section of the dynamometer.

Description	Specification
Model	HWANWOONG DYTEK-130
Absorption power	130 kW (180 ps)
Absorption torque	35 kgf.m (343 Nm)
Maximum speed	10000 rpm
Rotor inertia	0.14 kg/m^2
Weight	500 kg

Table 3. 2 Specification of Dynamometer

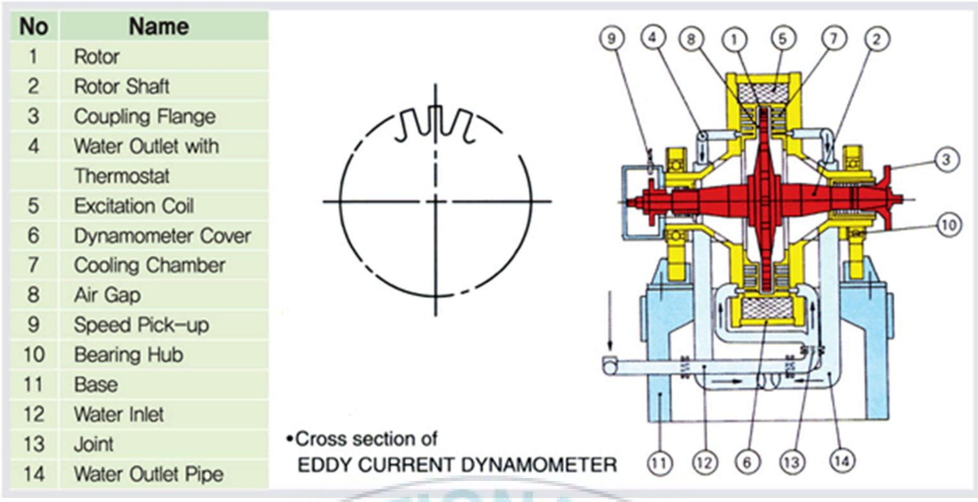

Figure 3. 2 Cross-section of the dynamometer

3.3 Electronic Control Unit (ECU)

As for hardware, ECM565-128 electronic control unit and UID800 a general-purpose injector driver was used. MotoHawk, a Simulink-based development software was used to implement a control mechanism. In this study, we configured six factors to control Fuel Injection Timing (IT), Throttle Position (TP), Pilot Injection (PI), Exhaust Gas Recirculation (EGR), Variable Geometry Turbocharger (VGT), and Common-rail pressure (CR.P).

3.4 Data acquisition

Real-time engine data acquisition was possible with custom programs written in National Instruments Labview. Signals such as the mass of airflow (MAF), emissions, temperatures, etc., were read by a series of Field Point modules. Data were saved in a format that could be easily processed in Microsoft Excel.

For most conditions, a sampling interval of 2 seconds was selected with a total sampling time of 3-4 minutes, once steady-state conditions were achieved.

3.5 Cylinder Pressure and Temperature measurement

In order to investigate the operating condition of the engine, the pressure and temperature inside and outside of the engine were measured. In this experiment, in-cylinder pressure was saved at every 0.5 CAD by a piezoelectric pressure transducer (Kistler 6125C) for high pressure, (Kistler 4045A5) for low pressure and rotary encoder Omoron E6B2-CWZ6C connected with a charge amplifier (Kistler, 5011B). Table 3.3 shows the specifications of the pressure sensor. The inside cylinder pressures of 100 cycles were used to calculate the heat release rate. K-type thermocouple was used to measure the inlet and outlet cooling water temperature, intake manifold air temperature, exhaust gas temperature, fuel temperature, and lubrication oil temperature. The temperature value signal was obtained one signal per second by the labview program and stored on the PC for 10 minutes. Figure 3.3 shows the pressure and the temperature signal coded by the labview program.

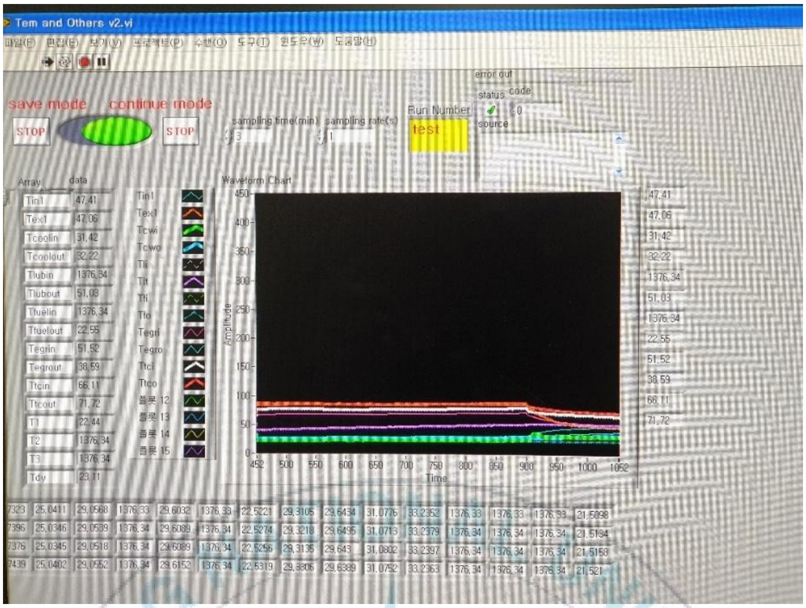

Figure 3.3 (a) Measurement of temperature

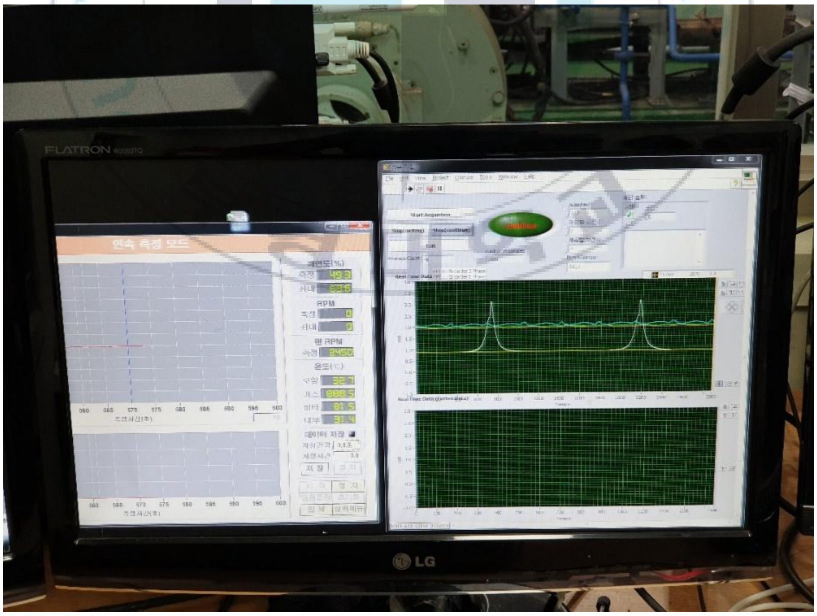

Figure 3.3 (b) Measurement of pressure

Figure 3. 3 Measurement of pressure and temperature

Model		6152C	4045A
Measuring range	bar	0 ... 250	0…5
Calibrated partial ranges	bar	0…50,0…100, 0…150,0…250	
Overload	bar	300	12,5
Sensitivity	PC/bar	≈–20	25
Natural frequency (measuring element)	kHz	≈ 160	≈ 80
Linearity, all ranges (at 23 °C)	%/FSO	≤ ±0.3	≤ ±0,1
Acceleration sensitivity axial	bar/g	<0.0002	
radial	bar/g	<0.0005	
Operating temperature range	°C	–20 ... 350	
Temperature min./max.	°C	–50 ... 400	
Sensitivity change 200 °C ±50 °C	%	≤±0.5	≤1,5
23 ... 350 °C	%	≤±2	
Thermal shock error (at 1 500 1/min, IMEP = 9 bar)			
Δp (short-term drift)	bar	≤±0.5	
Δpmi	%	≤±2	
Δpmax	%	≤±1	

Table 3. 3 Pressure sensor specification

3.6 Emissions measurement

The NOx, CO, $CO_2$and THC emissions were analyzed using an exhaust gas analyzer (MK 9000). Before tests, all the gas sensors of the exhaust analyzer were calibrated with standard gases. Table 3.3 summarized the specifications of the exhaust gas analyzer. Figure 3.4 shows the (MK 9000) exhaust gas analyzer.

Parameter	Sensor	Range
O_2	Electrochemical	0-25.00 / 0-100.00 vol.%
CO	Electrochemical	0 – 30000 ppm
NO	Electrochemical	0 – 5000 ppm
NO_2	Electrochemical	0 – 2000 ppm
NOx	Calculated	0 – 6000 ppm
SO_2	Electrochemical	0 – 5000 ppm
CO_2	Calculated	0 – 100%
Tair	Pt100	-40 - 100
Tgas	Tc K/N	-40 – 1200/1600
Excess air	Calculated	1.00 - infinity
Efficiency	Calculated	1 – 120%
Smoke index	Paper filter method	0 – 9 Bacharach

Table 3. 4 Exhaust gas analyzer specification

Figure 3. 4 Exhaust gas analyzer

3.7 Fuel consumption measurement

Diesel fuel consumption was measured by CUW-6200H electronic scale Figure 3.5 shows the display of electronic scale was monitored for about 5 minutes and 30 seconds by using a camera and then save as a video file on the PC. Table 3.5 shows the specifications of the electronic scale used in this experiment.

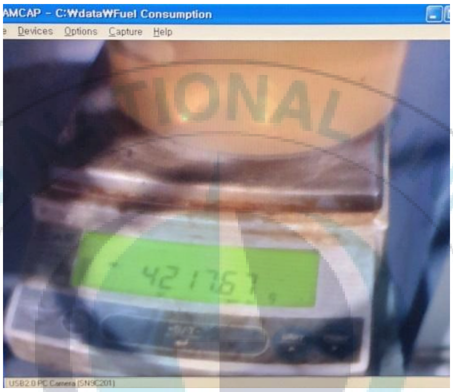

Figure 3. 5 Electronic scale

Model	CUW6200H
Maximum display (g)	6200 g
Minimum display (g)	0.01 g
Tare range (g)	1000-6200 g
Repeatability (g)	≤ 0.01 g
Linearity (g)	±0.01 g
Response time(s)	1.5s-2.5
Operation temperature (°C)	5-40
Pan size(mm) (W × D)	170 × 180
Dimension(mm) (W × D × H)	190W × 317D × 78H
Weight (kg)	4.6
Display	LCD
Power requirements	DC12 v
DATA I/O	RS-232C

Table 3. 5 Specification of Electronic Scale

3.8 Test conditions

Experiments were carried out at different common-rail pressure at the constant engine speed of 2000 rev/min with the output torque of 5.7 kgf.m and 11.2 kgf.m, corresponding to engine loads of 11.7 kW and 23.0 kW respectively. Two experiments were carried out in this research. Experiment conditions are shown in table 3.6. Experiments were firstly carried out with diesel mode and then carried out with dual fuel mode. The common-rail pressure of diesel fuel and natural gas percentages was controlled respectively by their own dedicated ECU. In all the whole test process, the intake air temperature was controlled at 40 ± 3°C and the cooling water temperature was maintained at75 ± 3°C.

To analyze the combustion and emission characteristics of the diesel mode and dual fuel mode according to the different common-rail pressure and natural gas mass ratio, all experiments were carried out with the same injection timing of 13°CA BTDC. In dual fuel mode, diesel fuel injection quantity is reduced and at the same time, the natural gas mass ratio was increased to keep the engine output loads constant. In this study, the natural gas mass ratio replacement of diesel fuel consumption was from 20 percent to about 90 percent. The natural gas substitution ratio is calculated using the following equation.

$$s_t = \frac{q_{md}}{q_{dd}+q_{md}} \times 100 \quad (9)$$

Where, q_{dd} is the mass consumption rate of diesel fuel and q_{md} is the mass consumption rate of natural gas.

Fuel	RPM	Output (KW)	CR.P (MPa)	Injection Timing (BTDC)	EGR	VGT	NG (L/M)	NG%
DO	2000	11.7	50	-13	5	5	0	0
NG	2000	11.7	50	-13	5	5	10	26.5
NG	2000	11.7	50	-13	5	5	16	52.4
NG	2000	11.7	50	-13	5	5	20	70.4
DO	2000	11.7	60	-13	5	5	0	0
NG	2000	11.7	60	-13	5	5	10	26.5
NG	2000	11.7	60	-13	5	5	15	46.3
NG	2000	11.7	60	-13	5	5	17	63.2
DO	2000	11.7	70	-13	5	5	0	0
NG	2000	11.7	70	-13	5	5	10	25.8
NG	2000	11.7	70	-13	5	5	15	46.6
NG	2000	11.7	70	-13	5	5	18	64.5

Table 3.6 (a) First Experimental conditions

Fuel	RPM	Output (KW)	CR.P (MPa)	Injection Timing (BTDC)	EGR	VGT	NG (L/M)	NG%
DO	2000	23.0	60	-13	5	5	0	0
NG	2000	23.0	60	-13	5	5	15	27.2
NG	2000	23.0	60	-13	5	5	23	54.0
NG	2000	23.0	60	-13	5	5	30	86.6
DO	2000	23.0	70	-13	5	5	0	0
NG	2000	23.0	70	-13	5	5	14	23.7
NG	2000	23.0	70	-13	5	5	20	38.3
NG	2000	23.0	70	-13	5	5	26	74.9
DO	2000	23.0	80	-13	5	5	0	0
NG	2000	23.0	80	-13	5	5	13	21.9
NG	2000	23.0	80	-13	5	5	21	43.3
NG	2000	23.0	80	-13	5	5	29	74.0

Table 3.6 (b) Second Experimental conditions

Table 3. 6 Experimental conditions

4 Results and Discussion

4.1 Analysis of Combustion Characteristics

4.1.1 Coefficient of Variance

The stability of engine combustion is considered by the coefficient of variance (COV) computed from the in-cylinder pressure data. The COV derived from the standard deviation of the cylinder pressure as a percentage of its mean value. Shown in equation 10.

$$COV_{IMEP} = \frac{\sigma_{IMEP}}{\overline{IMEP}} \times 100 \tag{10}$$

Where, $\overline{\text{IMEP}}$ is the mean of indicated mean effective pressure and σ_{IMEP} is the standard deviation of indicated mean effective pressure. Figure 4.1 shows COV_{IMEP} variables with a mass friction ratio of natural gas on each load. At 11.7 kW, as the mass fraction of natural gas increased, COV_{IMEP} also increased, but at load 23 kW, it can be seen COV_{IMEP} is almost the same as the diesel mode. This is related to diffusion combustion at 11.7 kW in combustion characteristics. In general, below 10% COV_{IMEP} is treated as a stable operation, so there was no unstable operation in this experiment.

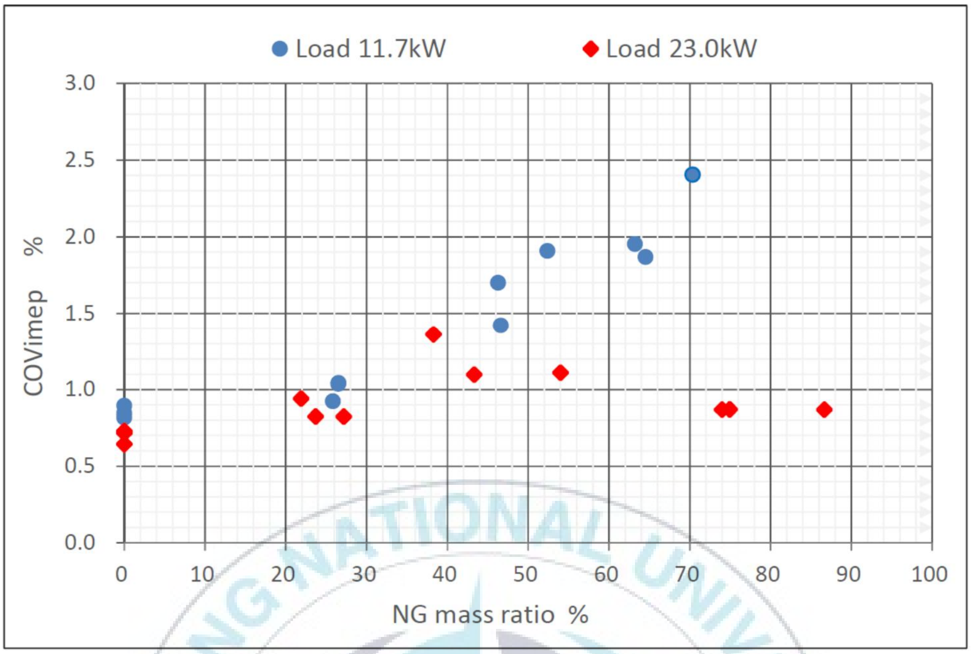

Figure 4. 1 COV_{IMEP} variables with mass friction ratio of natural gas on each load

4.1.2 Inside Cylinder Pressure

When analyzing the engine performance, the inside cylinder pressure is a very important source and needs to understand very well about this. Under a dual fuel mode, natural gas is introduced at the intake manifold by the injector and mix with fresh air and then compressed at the compression stroke. Figures 4.2 and 4.3 show the peak cylinder pressure and the heat releasing rate at each load. Equation 11 shows the equation for heat release rate calculation. In this experiment, the result showed that the inside pressure of dual fuel mode is slightly lower than that of diesel mode at compression stroke and during the initial stage of combustion. This is because the specific heat capacity ratio of natural gas is higher than the air. This caused the inside cylinder temperature of dual fuel mode to lower at the compression stroke. As a consequence, compression pressure of dual fuel mode is lower than the diesel mode. At the initial stage of combustion, the slow burning of natural gas can cause the lower inside cylinder pressure under dual fuel mode.

$$\frac{dQ}{dt} = \frac{\gamma}{\gamma-1}\,\mathrm{p}\frac{dV}{dT} + \frac{1}{\gamma-1}\,\mathrm{V}\frac{dp}{dt} \qquad (11)$$

Where γ is the ratio of specific heat ${C_p}/{C_v}$,

P is in-cylinder pressure,

V is the volume in the cylinder.

4.1.3 Peak Inside Cylinder Pressure

At engine loads 11.7 kW and common-rail pressure of 50 MPa, 60 MPa, 70 Mpa, the results showed that the peak cylinder pressure and heat release rate of dual fuel mode is lower than that of diesel mode with natural gas ratio 50% and 70% and it was almost same at 25% of the natural gas mass ratio. There are no peak cylinder pressure and heat release rate results in dual fuel mode that shows higher tendency than the diesel mode at this low load condition. This is because low load condition can access the lean condition of diesel fuel supply. So, the natural gas mass ratio is increased the peak cylinder pressure is decreased at the low load condition.

At engine loads 23 kW and common-rail pressure of 60 MPa, 70 MPa and 80 Mpa, the results showed that all the test conditions under dual fuel mode, peak in-cylinder pressure and heat release rate are higher than normal diesel mode. As the natural gas mass ratio increased, peak in-cylinder pressure and peak heat release rate also increased. This is because of increasing the load with the constant speed that results in an increasing pilot injection diesel fuel and also increasing the natural gas mass admit to the engine.

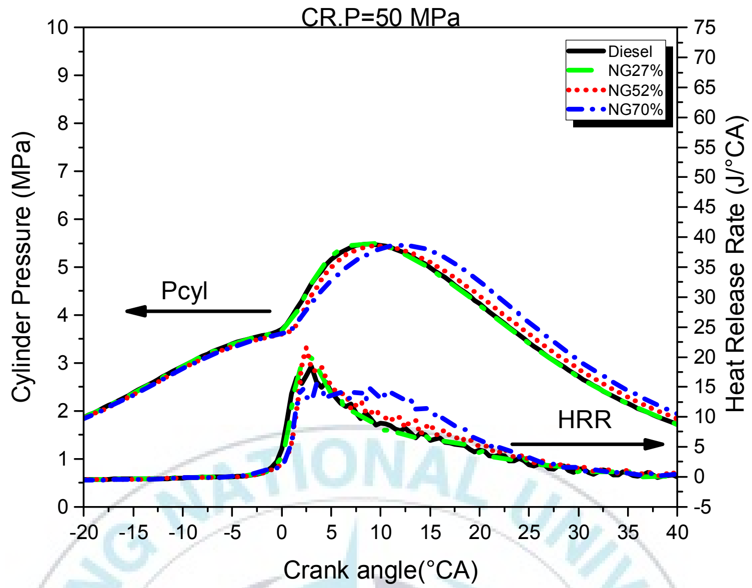

Figure 4.2 (a)

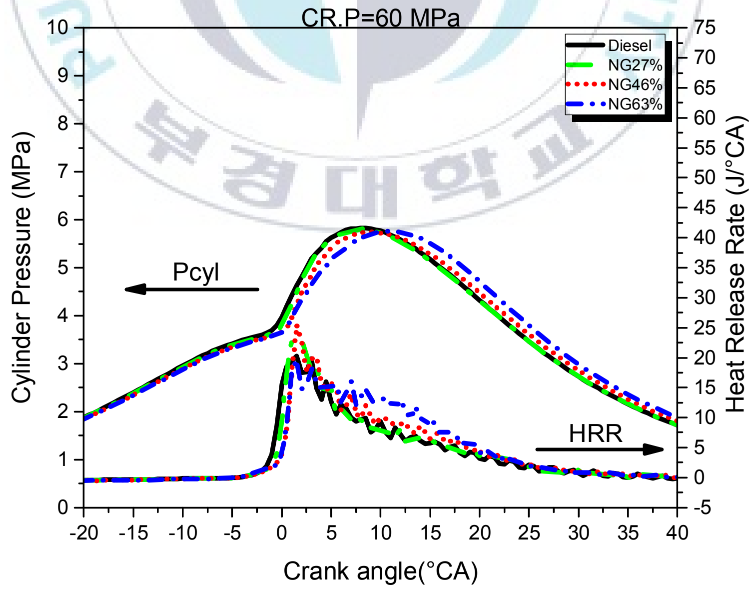

Figure 4.2 (b)

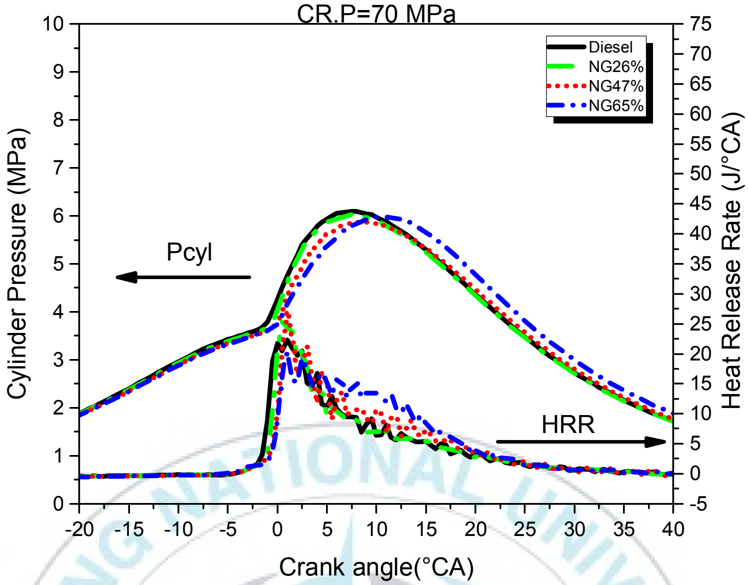

Figure 4.2 (c)

Figure 4. 2 Peak inside cylinder pressure and heat release rate at 11.7 kW

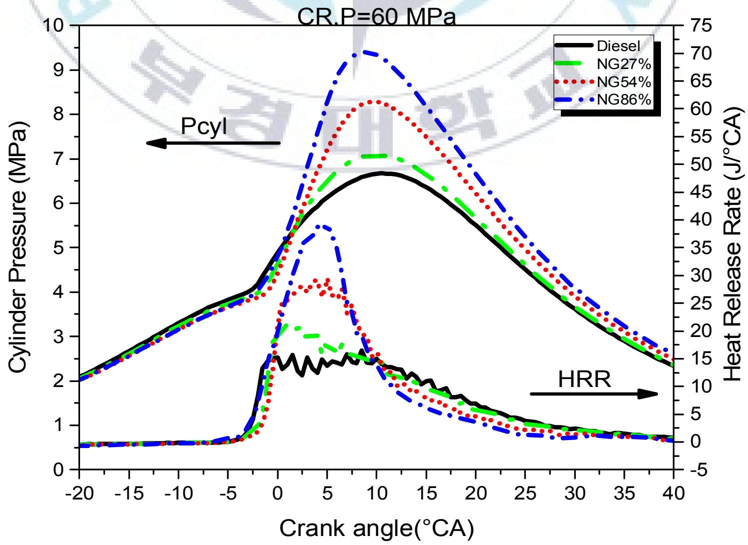

Figure 4.3 (a)

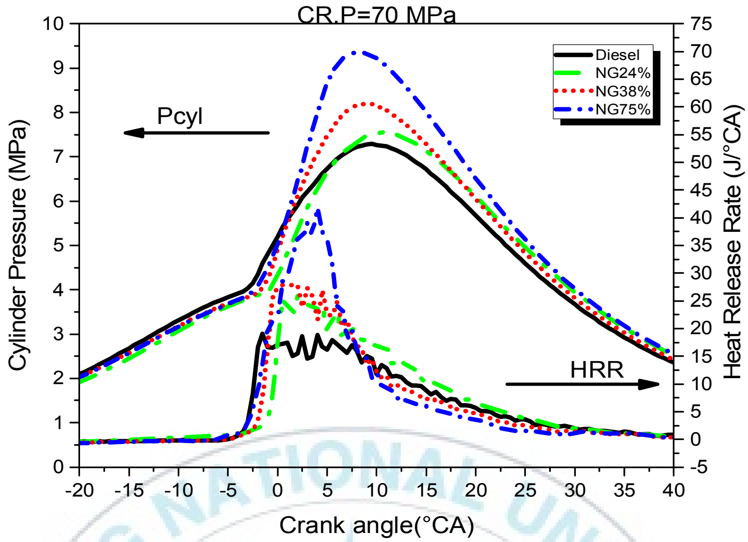

Figure 4.3 (b)

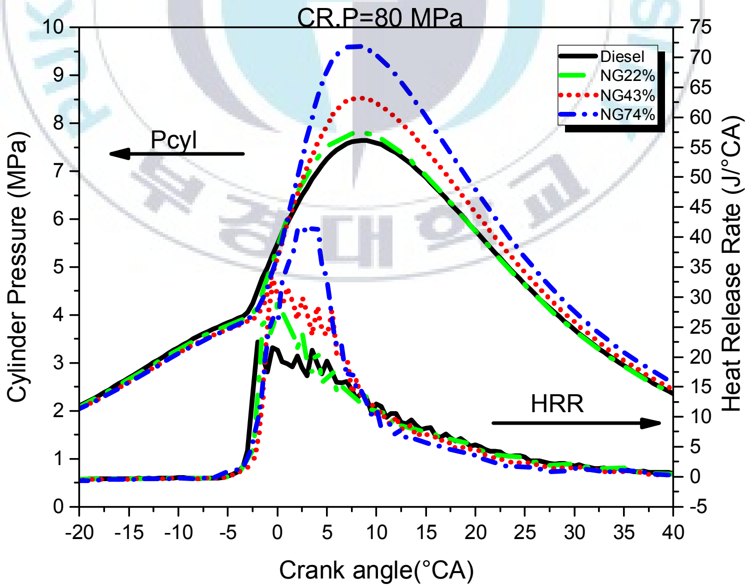

Figure 4.3 (c)

Figure 4. 3 Peak inside cylinder pressure and heat release rate at 23.0 kW

4.2 Emission Characteristics

4.2.1 NOx emission

NOx emission is strongly dependent on the oxygen concentration, combustion chamber temperature and, high temperature. Figure 4.4 shows the NOx emission of diesel and dual fuel modes with different common-rail pressure and the different natural gas mass ratio of 11.7 kW. It can be seen that NOx emission increases as the common-rail pressure increases because of increases of common-rail pressure caused by increases in-cylinder pressure and heat release rate, thus NOx emission increases as the common-rail pressure increases. However, compared to diesel mode NOx reduced about 0.3 g/kWh in all different common-rail pressure.

Figure 4.5 shows NOx emission of diesel and dual fuel modes with different common-rail pressure and the different natural gas mass ratio of 23.0 kW. The results showed that NOx emission of dual fuel mode increased compared to diesel mode in all the test conditions at this load. NOx emission increased as the common-rail pressure increased and also NOx emission increased as the natural gas mass ratio increased. This is because of the combustion chamber temperature increase. At the same RPM, increase of the load caused the increase of the pilot diesel injection amount and also a natural gas mass ratio.

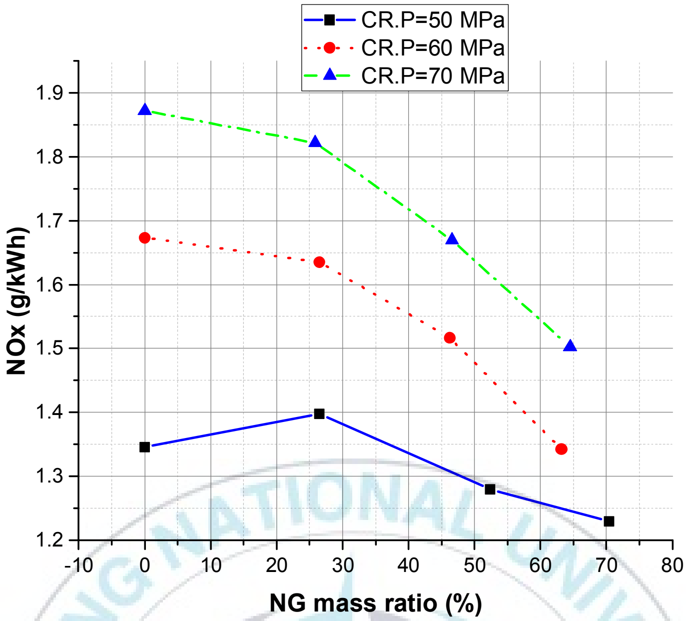

Figure 4. 4 NOx emissions at 11.7 kW

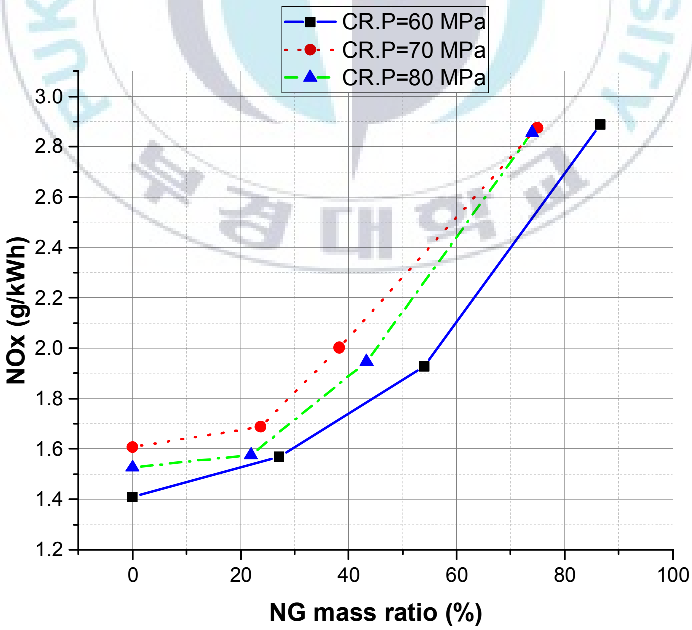

Figure 4. 5 NOx emissions at 23.0 kW

4.2.2 CO emission

Figure 4.6 shows CO emission at 11.7 kW and Figure 4.7 shows CO emission at 23.0 kW. The results showed that CO emission of dual fuel mode is much greater than the diesel mode at 11.7 kW load. However, the result showed the opposite effect in 23.0 kW. More CO is produced as the natural gas mass ratio is increased at 11.7 kW but at 23.0 kW it is found to be decreased in CO emission. Moreover, increased the common-rail pressure, CO emission has significantly decreased from 30% of the natural gas mass ratio of 11.7 kW. This caused due to the following reasons. Firstly, CO is generally produced in the natural gas rich region duel to the lack of oxygen. Secondly, CO is produced combustion temperature is less than 1450 K.

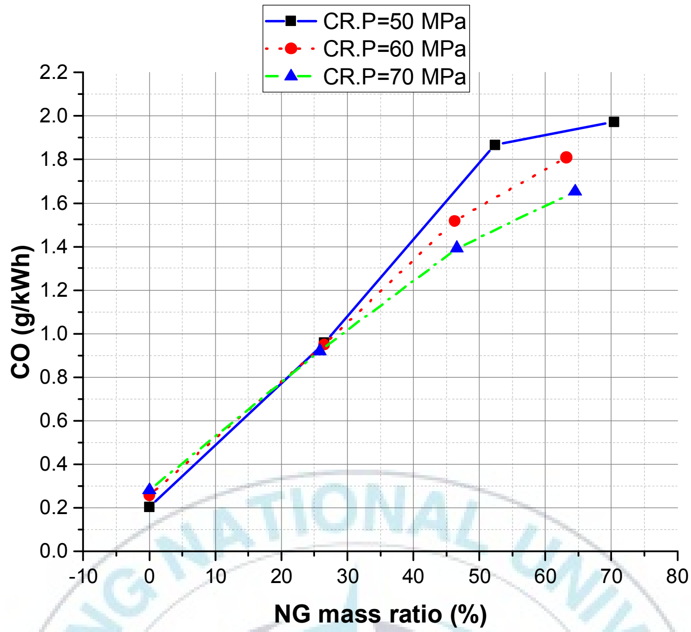

Figure 4. 6 CO emissions at 11.7 kW

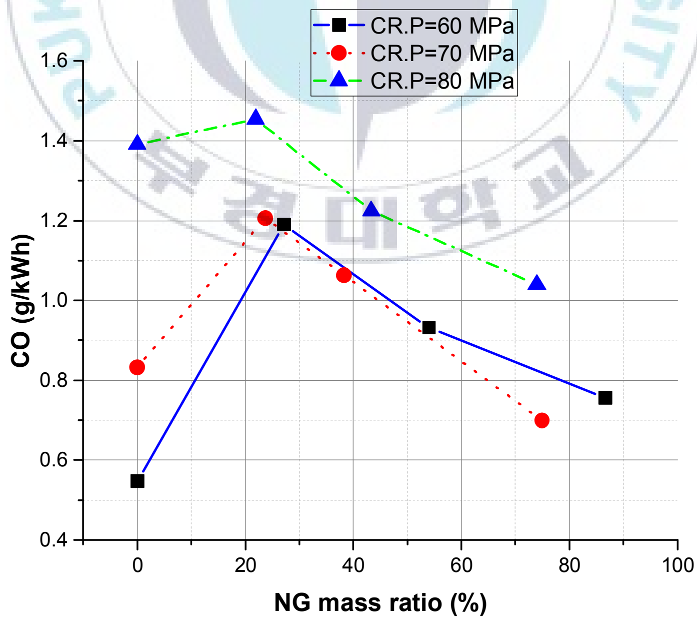

Figure 4. 7 CO emissions at 23.0 kW

4.2.3 THC emission

Figure 4.8 indicates THC emissions at 11.7 kW load with different common-rail pressure and natural gas mass ratio, the results showed that the difference of THC emission according to the common-rail pressure is not seen and THC emission shows an increasing trend as the natural gas mass friction increased. There are two main reasons for this phenomenon. Firstly, flame quenching. When the flame approach near the combustion chamber wall region, the temperature of the mixture is very low to complete the combustion generating unburned HC mixture. Secondly, the injected diesel mixture is too lean for combustion.

Figure 4.9 indicates 23.0 kW load condition, at this load condition with the increased of the injected diesel quantity, THC emissions reduced significantly which caused due to some reasons. First, the ignition diesel amount is increased. Second, the fuel spray atomization is improved. Third, the number of ignition center is increased. Fourth, the heat transfer to the unburned HC mixture is increased. Furthermore, at this load condition, the pressure and temperature are higher, which caused to increase in-cylinder compression temperature. All of these effects reduce the combustion limits of the lean premixed CNG fuel and increase the flame propagation speed. So, more premixed CNG participates in the combustion process.

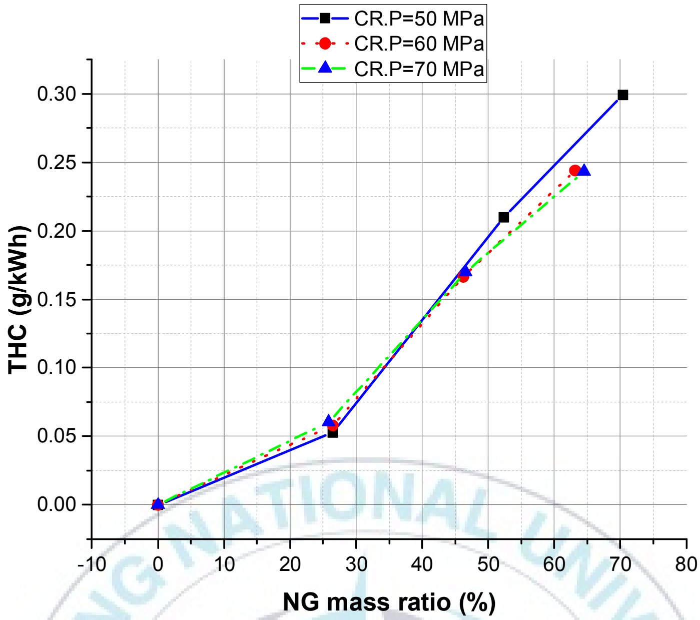

Figure 4. 8 THC emissions at 11.7 kW

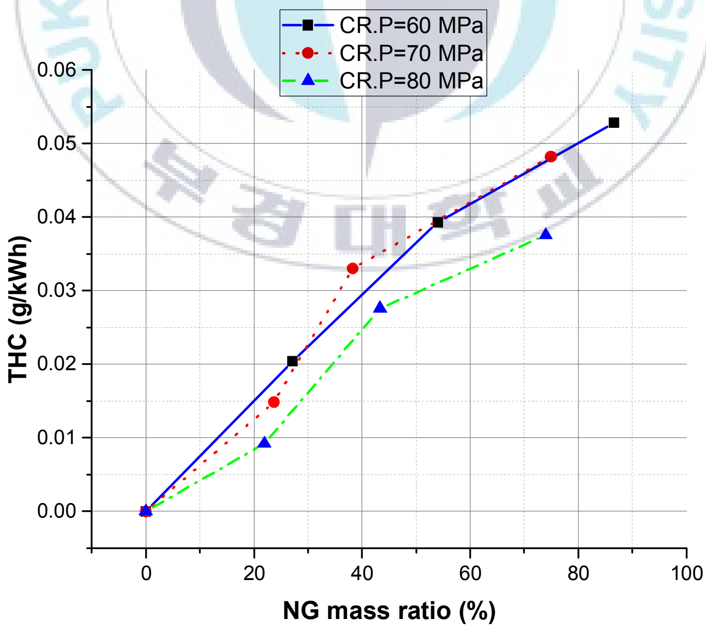

Figure 4. 9 THC emissions at 23.0 kW

4.2.4 CO_2 emission

Figures 4.10 and 4.11 shows the CO_2 emission at 11.7 kW load and 23.0 kW loads with different common-rail pressure and natural gas mass ratio, it can be seen that CO_2 emission decreased with increasing natural gas mass ratio in both tests. At 11.7 kW load, as increasing 10 MPa of common- rail pressure, it was generated 5 g/kWh of CO_2 more. However, similar results were shown regardless of common-rail pressure at 23.0 kW. In comparison to diesel mode, dual fuel mode produced significantly less CO_2 emission in all the test conditions. This is because of the following reasons. First, natural gas is composed of 80 percentages to over 90 percentages of methane, which contains the lowest carbon contents. So, the combustion of natural gas produced lower CO_2 emissions than diesel. Second, under dual fuel mode, incomplete combustion seriously takes place and the combustion chamber temperature is low. That caused some of the fuel is incomplete oxidize to CO which can decrease the CO_2 emission.

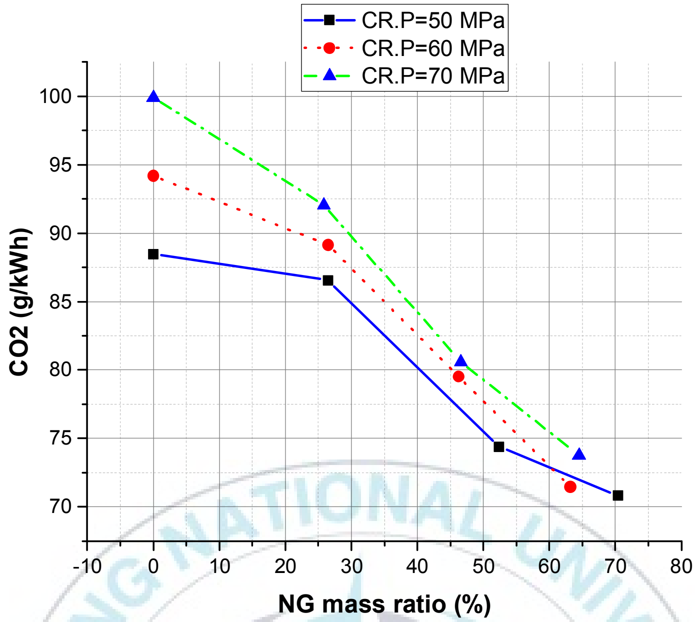

Figure 4. 10 CO_2 emissions at 11.7 kW

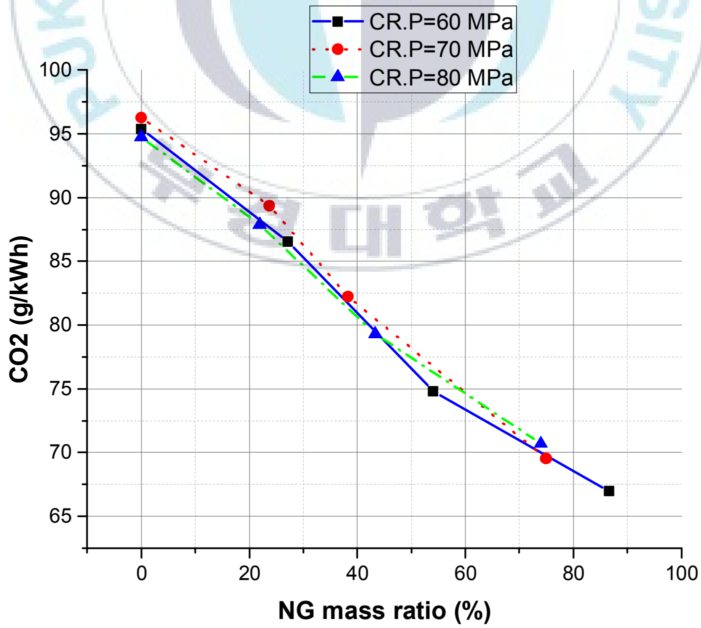

Figure 4. 11 CO_2 emissions at 23.0 kW

5 Conclusion

The CNG/diesel dual fuel engine combustion and emission characteristics have been experimentally investigated in this paper with different common-rail pressure and natural gas mass ratio at optimized injection timing. The investigation results indicate the following:

1) As the mass fraction of natural gas increased, Diffusion combustion is seen at 11.7 kW and the amount of premixed combustion increased and the pressure in the combustion chamber increased significantly at 23.0 kW.

2) The COV_{IMEP} increased as the mass fraction of natural gas increased at a low load of 11.7 kW, but stable driving characteristics are shown without increasing at a middle load of 23.0 kW. This is related to diffusion combustion at low loads in combustion characteristics.

3) There was no significant decrease in NOx with increasing natural gas mass fraction at 11.7 kW but marked an increase in NOx with increasing natural gas mass fraction at 23.0 kW.

4) The difference of THC emission according to the common-rail pressure is not seen and THC emission showed an increasing trend as the mass fraction increased.

5) As the NG mass fraction decreased, the CO_2 emissions decreased, regardless of load.

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