

Research Paper

Pilot injection strategies for methane slip reduction in methane-diesel dual-fuel engines

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ABSTRACT

The objective of this study is to investigate the effects of key factors related to pilot injection on the performance and exhaust emission characteristics of a Dual Fuel (DF) engine. To this end, combustion and exhaust data were obtained under variations in pilot injection timing, injection duration, injection pressure, and multi-stage injection strategies. The results showed that as the pilot injection timing was advanced, Indicated Mean Effective Pressure (IMEP) increased. However, when the pilot injection was advanced beyond Before Top Dead Center (BTDC) 16°, IMEP no longer increased, and heat release characteristics caused by end-gas auto-ignition appeared. As the injection duration increased, IMEP increased linearly. When the injection pressure rose above 1000 bar, fuel supply and spray penetration increased, and unlike the case of increased injection duration where only the pilot fuel quantity rose, end-gas auto-ignition occurred. In terms of exhaust emission characteristics, as the pilot injection timing was advanced, the injection duration lengthened, and the injection pressure increased, unburned methane emissions decreased while nitrogen oxide emissions increased. Among the multi-stage pilot injection strategies, the Pre 2nd Pilot showed the best performance, achieving the shortest combustion duration, highest IMEP, lowest Coefficient of Variation (COV) of IMEP, and a significant reduction in unburned methane, thereby recording the lowest greenhouse gas emissions. Although Nitrogen oxides (NO_x) emissions increased due to higher combustion temperature and pressure, they remained relatively low compared to other conditions with similar greenhouse gas levels.

1. Introduction

The maritime industry is a critical sector responsible for over 80% of global trade-related logistics transportation, and it is known that approximately 99% of the energy used in this sector is derived from fossil fuels [1].

Furthermore, the International Maritime Organization (IMO) has projected that, in the absence of additional mitigation measures, carbon dioxide emissions from the shipping industry could increase by up to 50% compared to 2018 levels [2]. The importance of reducing greenhouse gas emissions in the maritime industry continues to grow, and in response, the IMO reinforced its greenhouse gas reduction strategy originally announced in 2018 and reported the updated version in 2023.

The revised strategy outlines specific targets: a minimum 40% reduction in carbon dioxide emissions per transport work by 2030, a 70% reduction in total annual greenhouse gas emissions from international shipping compared to 2008 levels by 2040, and the achievement of carbon neutrality by 2050 [3].

The IMO's greenhouse gas regulations are being strengthened to achieve carbon neutrality, and in order to meet the upcoming 2030 greenhouse gas reduction targets, specific mitigation measures were agreed upon at the 83rd session of the Marine Environment Protection Committee (MEPC), held in April 2025 [4,5].

Greenhouse gas reduction in the maritime sector is recognized as a critical issue not only by the IMO but also by the European Union (EU),

Abbreviations: ATDC, After Top Dead Center; BTDC, Before Top Dead Center; CAXX, Crank Angle at XX% mass fraction burned; CAXX–XX, Combustion interval from CAXX to CAXX; CH₄, Methane; CO₂, Carbon dioxide; DAQ, Data Acquisition; DF, Dual Fuel; EU, European Union; EU ETS, European Union Emissions Trading System; FTIR, Fourier Transform Infrared Spectroscopy; GWP, Global Warming Potential; IMEP, Indicated Mean Effective Pressure; IMEP COV, Indicated Mean Effective Pressure Coefficient of Variation; IMO, International Maritime Organization; LNG, Liquefied Natural Gas; MEPC, Marine Environment Protection Committee; MFB, Mass Fraction Burned; MFC, Mass Flow Controller; NDIR, Non-Dispersive Infrared Absorption; NO_x, Nitrogen oxides; RCCI, Reactivity Controlled Compression Ignition; ROHR, Rate of Heat Release; SO₂, Sulfur Dioxide; TDC, Top Dead Center; UHC, Unburned Hydrocarbon.

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which is one of the leading entities in environmental regulation [6]. According to the implementation plan of the EU Emissions Trading System (EU ETS), only carbon dioxide will be subject to regulation until 2025. However, starting in 2026, unburned methane primarily emitted from Liquefied Natural Gas (LNG)-powered vessels and nitrous oxide mainly emitted from ammonia-fueled ships will also be included in the scope of regulation [7].

As regulations to reduce greenhouse gas emissions from the shipping industry are being strengthened, interest is increasing in technologies that replace diesel fuel with alternative fuels. [8–10] Representative alternative fuels include LNG, methanol, ammonia, and hydrogen, each with distinct advantages and disadvantages.

Methanol has the advantage of being liquid at room temperature, making it easy to transport and store, but it has been reported to cause corrosion or wear of engine components [11,12]. Ammonia and hydrogen are carbon-free fuels that emit no carbon dioxide during combustion, making them essential fuels for achieving carbon neutrality in future internal combustion engines. However, in the short term, due to insufficient infrastructure and technological maturity, they are difficult to use widely.

Lastly, LNG fuel contains carbon, which makes it less favorable than carbon-free fuels such as hydrogen or ammonia in terms of greenhouse gas reduction. Nevertheless, since the related infrastructure and technology are already mature, LNG has attracted significant attention as a short-term means of reducing greenhouse gases. In this respect, LNG fuel is regarded as a practical short-term greenhouse gas reduction option that can be applied in the shipping industry.

However, LNG has a higher autoignition temperature and a slower flame speed than diesel, which can lead to flame quenching. As a result, a portion of the methane supplied to the engine combustion chamber may remain unburned and be released through the exhaust, a phenomenon known as methane slip.

Methane remains in the atmosphere for a relatively short period, but it has a strong greenhouse effect and is known to exert a much more powerful short-term warming impact than carbon dioxide. Based on the 100-year time horizon of Global Warming Potential (GWP100), methane is estimated to have about 28 times the greenhouse effect of carbon dioxide. Consequently, mitigating methane slip in LNG engines is recognized as a key challenge for achieving substantial greenhouse gas

reduction in LNG-powered vessels [13].

Fig. 1 illustrates the mechanisms of methane slip occurrence in LNG engines. Methane slip primarily results from unburned methane generated during the flame quenching process and methane leakage into the exhaust during the scavenging process. The formation of unburned methane caused by flame quenching occurs in crevice volume regions, such as the bulk area inside the combustion chamber and the narrow spaces between the piston rings and the combustion chamber wall.

To reduce methane slip, LNG engine manufacturers are considering various methods, and the application of a methane oxidation catalyst can be regarded as one of the mitigation strategies. A methane oxidation catalyst can be applied not only to newly built ship engines but also to currently operating vessels with minimal engine modifications. However, there are several technical challenges associated with its implementation.

Methane is one of the more difficult substances to oxidize, requiring temperatures above 600 °C in the absence of a catalyst. Therefore, methane oxidation typically employs catalysts containing platinum group metals (PGMs), with platinum and palladium being the most widely used [17–19]. However, ensuring the durability of methane oxidation catalysts under the exhaust gas conditions of marine engines, which contain poisoning substances such as moisture and sulfur oxides, remains an important challenge. [17,20–24].

In addition to exhaust after-treatment devices such as methane oxidation catalysts, it is also possible to reduce methane slip generated during the combustion stage. For methane slip occurring during the scavenging process, reduction can be achieved by optimizing the intake and exhaust valve timing, by optimizing the timing of methane fuel injection into the intake manifold, or by directly injecting gas fuel into the combustion chamber [13,25]. Furthermore, methane slip caused by the crevice volume of the combustion chamber, such as piston rings and intake valve seats, can be reduced by decreasing the crevice volume of the engine combustion chamber. Even without reducing the crevice volume, it is known that optimizing the ignition timing can reduce the unburned mixture caused by the crevice volume [26].

Among the methane slip generated during the combustion stage, methane slip caused by flame quenching of the bulk charge accounts for the majority. Therefore, in order to suppress flame quenching of the bulk

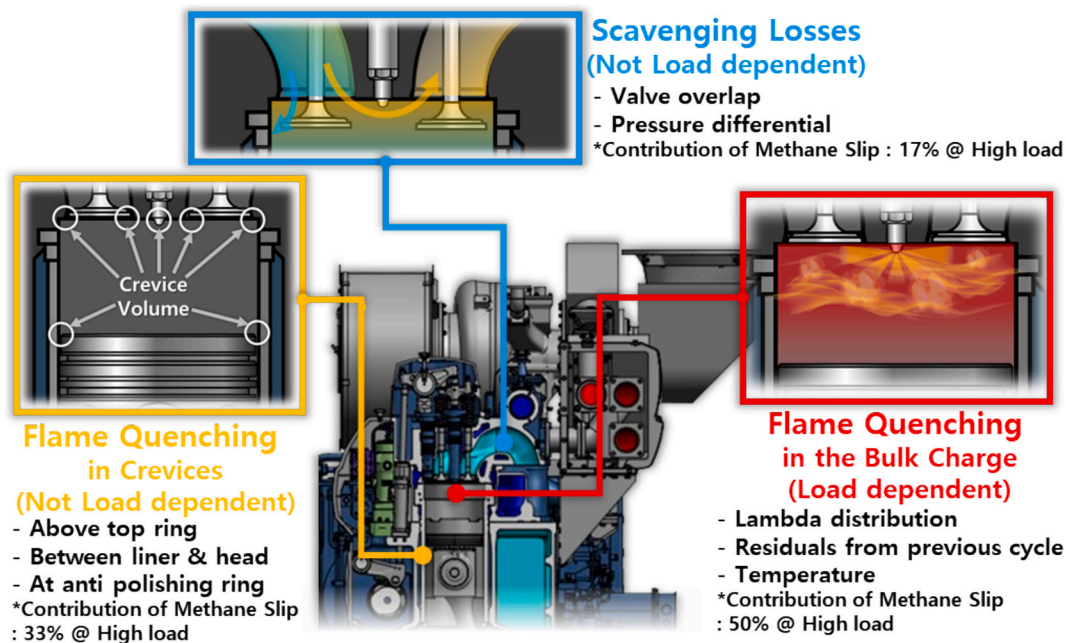


Fig. 1. Sources of methane slip in marine LNG engines [14–16].

mixture, it is necessary to improve the reactivity of the mixture itself or to optimize the pilot fuel injection strategy to enhance flame propagation.

To reduce methane slip caused by flame quenching of the bulk charge, one method is to supply methane fuel mixed with hydrogen, which has a faster flame speed. Another method is to inject pilot fuel during the intake stroke so that it mixes well with the methane–air mixture, thereby enhancing the reactivity of the mixture [13,27–32].

The pilot fuel injection strategy has a significant influence on the combustion of the mixture. For smooth combustion of the mixture, an appropriate pilot injection strategy involving the timing and duration of pilot fuel injection is required. If the pilot injection timing is too early, the combustion pressure increases, and once the pressure exceeds a certain threshold, the fuel–air mixture around the periphery of the combustion chamber may auto ignite. In addition, when the injection timing is excessively advanced, the injected pilot fuel can mix with the surrounding mixture, leading to dilution of its concentration and a reduction in its effectiveness as an ignition source, which may result in misfiring. Conversely, excessively delayed pilot fuel injection timing is also known to cause misfiring [33,34].

It is also possible to reduce methane slip through the Reactivity Controlled Compression Ignition (RCCI) combustion strategy, and in the RCCI combustion mode, when the pilot injection is advanced beyond a certain threshold, the widely distributed diesel within the combustion chamber ignites simultaneously at multiple locations. This contributes to the reduction of methane slip and is also known to lower nitrogen oxide emissions [35–37].

The strategy of split pilot fuel injection has been reported to suppress the formation of unburned methane caused by flame quenching. Regarding the timing of split injection, when pilot fuel is injected early during the compression stroke and partially mixed with the fuel–air mixture, followed by a secondary injection, the resulting pilot flame distribution becomes broader compared to single injection. Increasing the mass fraction of the first injection has been shown to significantly reduce unburned methane near the cylinder walls [38,39].

In LNG marine engines, various methods have been attempted or studied to reduce unburned methane emissions. Approaches that require modifications to the combustion chamber geometry or the application of new devices can be applied to newly built LNG ship engines, but they are difficult to implement in existing LNG vessels already in operation. In contrast, adjustments to pilot injection strategies can be made without mechanical modifications, making them applicable to both newly manufactured engines and those currently in use, and thus represent a cost-effective method for reducing methane slip.

Although many researchers have studied unburned methane emissions under different pilot injection strategies, most analyses have focused on only one or two strategies distinct from conventional pilot injection. Therefore, this study analyzes the engine performance and exhaust emission characteristics, including methane slip, under three single pilot injection strategies and three split pilot injection strategies using a research single-cylinder dual-fuel (DF) engine platform. Through this analysis, the study aims to provide information on the effects of each pilot injection strategy on engine performance and emissions, while proposing pilot injection strategies that can reduce methane slip in LNG–Diesel dual-fuel engines with minimal side effects.

2. Experimental method

2.1. Engine test apparatus

The single-cylinder DF engine used in this study was constructed by modifying a commercial diesel engine. The combustion chamber shape of the original commercial diesel engine featured a “Re-entrant Type” bowl designed for diesel combustion, which presents disadvantages when performing dual-fuel combustion under conditions with a high methane energy ratio. Therefore, the piston bowl shape was redesigned with reference to the combustion chamber geometry of a medium-speed

DF engine, and the compression ratio was adjusted to simulate the pressure and temperature conditions of the medium-speed DF engine's combustion chamber. The manufactured single-cylinder engine has a combustion chamber volume of 1133 cc and a compression ratio of 17.15, with detailed specifications provided in Table 1.

The single-cylinder engine used in this study offers several advantages for combustion analysis compared to multi-cylinder engines. Inter-cylinder interference effects are eliminated, and auxiliary systems required for engine operation, such as cooling, lubrication, and fuel supply, are not driven by the engine. As a result, the operating conditions of these auxiliary systems do not affect engine performance, allowing combustion characteristics to be analyzed with greater precision.

Fig. 2 presents the schematic diagram of the single-cylinder engine used in the experiment, illustrating the equipment employed for engine operation and the interconnections between components. The single-cylinder engine was driven by a 55 kW DC motor dynamometer, which enabled precise control of engine load and rotational speed.

The intake air supplied to the engine was delivered as dry, particle-free air by compressing ambient air using an air compressor, followed by treatment through an air dryer and filter. The flow rate was regulated using a mass flow controller (MFC). An in-line heater was installed to control the intake air temperature. Additionally, a surge tank was incorporated to stabilize the supplied airflow and to minimize the impact of pulsations caused by the opening and closing of the intake valve on the intake pressure.

The gaseous fuel supplied to the engine was compressed methane with a purity of 99.95%. To maintain a stable supply pressure, four 47-l gas cylinders were connected in parallel. The gas was depressurized using a regulator and its temperature was controlled via an in-line heater before being delivered to the MFC. The methane discharged from the MFC was then supplied to a venturi mixer installed in the engine intake manifold, where it was uniformly blended with intake air prior to entering the combustion chamber.

Ultra-low sulfur diesel was used as the pilot fuel supplied to the engine, and a dedicated fuel pump system driven by an electric motor was employed for its delivery. The pressure of the supplied pilot fuel was regulated by controlling the duty cycles of the common rail pressure control valve and the high-pressure pump metering valve using a pulse-width modulation (PWM) method.

A solenoid-type diesel injector manufactured by Denso, equipped with a seven-hole nozzle, was used for pilot injection. The injection timing and duration were controlled using the CompactRIO system from National Instruments.

The in-cylinder combustion pressure was measured using a piezo-electric pressure sensor (model 6052C) from Kistler, while the intake and exhaust pressures were measured using pressure transducers. Detailed information related to the sensors used for combustion data acquisition is summarized in Table 2. Experimental data acquisition and analysis were conducted using a DAQ system provided by National Instruments.

Table 1
Specifications of 1.1 L single cylinder Dual Fuel engine.

Item	Specification
Bore (mm)	107
Stroke (mm)	126
Connecting Rod (mm)	200
Displacement Volume (cc)	1133
Compression Ratio	17.15
Intake Valve Open (degree)	343
Intake Valve Close (degree)	567
Exhaust Valve Open (degree)	136
Exhaust Valve Close (degree)	372
Injector hole Number	7
Injector hole diameter (mm)	0.174

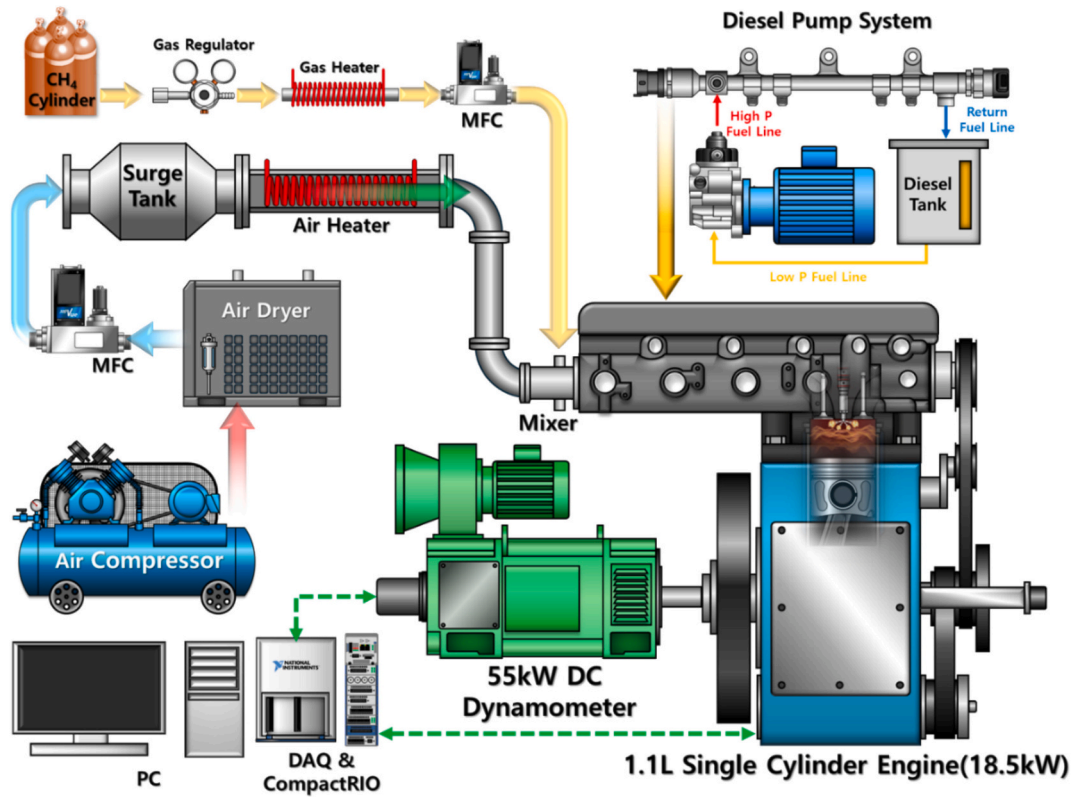


Fig. 2. Schematic diagram of engine experiment apparatus.

Table 2

Specifications of the sensor used to measure engine combustion characteristics.

Instruments	Item	Specification
Piezoelectric pressure sensor (Kistler, 6052C)	Measuring range	0–300 bar.a
	Linearity, all ranges (23°)	$\leq \pm 0.5\%$ of full scale
Intake air pressure sensor (Sensys, PTD)	Measuring range	–1–2 bar.g
	Accuracy	$\pm 0.5\%$ of full scale
Exhaust gas pressure sensor (Sensys, PTD)	Measuring range	0–10 bar.g
	Accuracy	$\pm 0.5\%$ of full scale
Intake air Mass Flow Controller (Teledyne Hasting, HFC-307B)	Measuring range	0–2000 SLPM
	Accuracy	$\pm 0.5\%$ of reading + 0.2% of full scale
Methane Mass Flow Controller (Teledyne Hasting, HFC-303B)	Measuring range	0–100 SLPM
	Accuracy	$\pm 0.5\%$ of reading + 0.2% of full scale
Exhaust gas temperature sensor (K-type thermocouple)	Measuring range	0–1000 °C
	Accuracy	$\pm 1.5^\circ\text{C}$ or $\pm 0.4\%$ of reading
Engine Speed Sensor (Autonics, E50)	Maximum response frequency	300 kHz
	Maximum response speed	50,000 RPM

For exhaust gas analysis, two types of exhaust gas analyzers were used: the HORIBA VA-5000 and the GASMET GT-6000. The primary analysis of exhaust gas concentrations was based on data obtained from the GASMET GT-6000, which utilizes Fourier-transform infrared (FTIR) spectroscopy. The HORIBA VA-5000, which operates on the non-dispersive infrared (NDIR) principle, was used exclusively for real-time methane concentration monitoring at 0.5-s intervals and served to cross-validate the methane measurements from the GASMET GT-6000. Detailed specifications of both analyzers are provided in Table 3.

Table 3

Specifications of exhaust gas analyzer (HORIBA VA-5000 & GASMET GT-6000).

Item	Specification
	Measurement principle
	Non-dispersive infrared rays (NDIR)
	Species
	CH ₄
HORIBA VA-5000	Measurement range
	0–3150 ppm
	Linearity
	$\pm 1.0\%$ of full scale
	Repeatability
	$\pm 0.5\%$ of full scale
	Response time
	T90: 30 s
	Drift (Zero & Span)
	$\pm 2.0\%$ /week of F-S
	Measurement principle
	Infrared spectroscopy (FTIR)
	Species
	CH ₄ , CO, CO ₂ , NO, NO ₂ , SO ₂ , H ₂ O
	CH ₄ : 0–20,000 ppm
	CO: 0–2000 ppm
	CO ₂ : 0–10 vol%
	NO: 0–1000 ppm
	NO ₂ : 0–1000 ppm
	SO ₂ : 0–50 ppm
	H ₂ O: 0–30 vol%
	Zero-point drift
	< 2% of measuring range per zero-point calibration interval
	Sensitivity drift
	None
	Linearity deviation
	< 2% of measuring range
	Temperature drift
	< 2% of measuring range across long term operating temperature range

2.2. Combustion and exhaust data analysis method

Combustion results were calculated using in-cylinder pressure data measured by a piezoelectric pressure sensor. Since the piezoelectric pressure sensor cannot detect absolute pressure, the values from the intake pressure sensor, which can measure absolute pressure, were used to correct the zero reference of the in-cylinder pressure data. From the combustion chamber pressure data, combustion characteristics such as

the rate of heat release (ROHR), mass fraction burned (MFB), and indicated mean effective pressure (IMEP) were calculated, using the average in-cylinder pressure data over 100 cycles. In addition, to evaluate combustion stability, IMEP was calculated for each of the 100 individual cycles of in-cylinder pressure data, and the coefficient of variation of IMEP (IMEP COV) was determined. The uncertainty of the previously calculated average IMEP value was represented by the standard deviation of the 100 individual IMEP data points.

Emission results were calculated using data measured by an exhaust gas analyzer employing the Fourier Transform Infrared Spectroscopy (FTIR) method. The results were obtained as the average of exhaust gas data continuously measured at 30-s intervals for about 10 min, starting from the point when the concentration of emitted gases stabilized. Furthermore, the uncertainty of the measured exhaust gas data was represented by calculating the standard deviation of the exhaust gas data measured during the 10-min period.

In-cylinder temperature is one of the key factors influencing the formation of nitrogen oxides (NO_x) and unburned hydrocarbons (UHC). However, direct measurement of this temperature is difficult. When thermocouples are used, only the temperature at a specific point inside the combustion chamber can be measured, and due to the time delay associated with heat transfer from the combustion gases to the thermocouple, it is challenging to capture the rapid temperature fluctuations caused by combustion.

Therefore, in this study, the bulk gas temperature inside the cylinder was calculated by constructing a 1D model of the experimental engine, as shown in Fig. 3, using the Three-Pressure Analysis (TPA) method provided by Gamma Technologies' GT-Power software.

The Three-Pressure Analysis (TPA) method uses measured intake pressure, exhaust pressure, and in-cylinder pressure from engine experiments to perform non-predictive combustion simulations based on the two-zone combustion concept, Woschni's heat transfer correlation, and chemical equilibrium calculations, thereby enabling the calculation of physical quantities that are difficult to measure experimentally. Model validation was performed by comparing the actual in-cylinder pressure with the pressure calculated through the Three-Pressure Analysis and confirming that the two results match. Fig. 4 shows that the experimental and analytical results match closely, confirming the accuracy of the constructed TPA model.

According to the Intergovernmental Panel on Climate Change (IPCC) report, carbon dioxide (CO_2) is the primary greenhouse gas contributing most to global warming. However, methane (CH_4), despite its shorter atmospheric lifetime, exhibits much stronger instantaneous radiative forcing per unit mass compared to CO_2 . Based on the 100-year Global Warming Potential (GWP100), methane is known to have a greenhouse effect equivalent to about 28 times that of CO_2 [40].

To quantitatively assess the greenhouse effect of exhaust emissions under different engine operating conditions, the impact of unburned methane was converted into an equivalent CO_2 emission amount. This conversion was incorporated into the total greenhouse gas emission analysis. The calculation method for total greenhouse gas emissions, based on the exhaust gas flow rate $\dot{m}_{\text{exh}}$, is presented in Eq. 2.1.

$$\dot{m}_{\text{GHG}} = \dot{m}_{\text{CO}_2} + 28\dot{m}_{\text{CH}_4} = \dot{m}_{\text{exh}}[\text{CO}_2] + 28\dot{m}_{\text{exh}}[\text{CH}_4] \quad (2.1)$$

2.3. Engine experimental conditions

To analyze the combustion and exhaust emission characteristics of a methane–diesel dual-fuel engine under applicable pilot injection strategies, the controlled variables listed in Table 4 were selected. As mentioned earlier, the main cause of methane slip is flame quenching. To explore pilot injection strategies that can reduce methane slip by suppressing flame quenching, the following pilot injection strategies were included in the experimental conditions.

First, changes in pilot injection timing delay or advance the start of combustion, thereby altering the combustion phase, and the resulting combustion and exhaust characteristics were analyzed. In addition to changes in the combustion phase, the amount and distribution of injected pilot fuel can affect the distribution and intensity of the high temperature regions generated by pilot fuel ignition, which in turn can influence the combustion and ignition of the methane–air mixture. Therefore, to evaluate the combustion and exhaust characteristics according to the amount and distribution of injected pilot fuel, experiments were conducted under conditions of varying pilot injection duration, injection pressure, and multi-stage pilot injection. The specific manipulated values for each pilot injection strategy are presented in Table 5.

Pilot injection timing was based on a reference condition of injection at Before Top Dead Center (BTDC) 8° , and a total of seven pilot injection timing conditions were tested. The experimental conditions for pilot injection duration were based on a reference condition of 0.33 ms, and a total of seven pilot injection duration conditions were tested.

Pilot injection pressure was set to a reference value of 600 bar, with five additional pressure levels tested, resulting in six pilot injection pressure conditions overall.

To analyze the combustion and exhaust gas characteristics of the DF engine when pilot fuel was injected in multiple stages, three types of multi-stage pilot injection strategies were applied: Post 2nd Pilot, Early 2nd Pilot, and Pre 2nd Pilot. To make the supplied diesel amount similar between the multi-pilot injection condition and the single injection condition with a duration of 0.33 ms, the injection duration of the first and second injections in the multi-pilot injection was set equally to 0.27 ms.

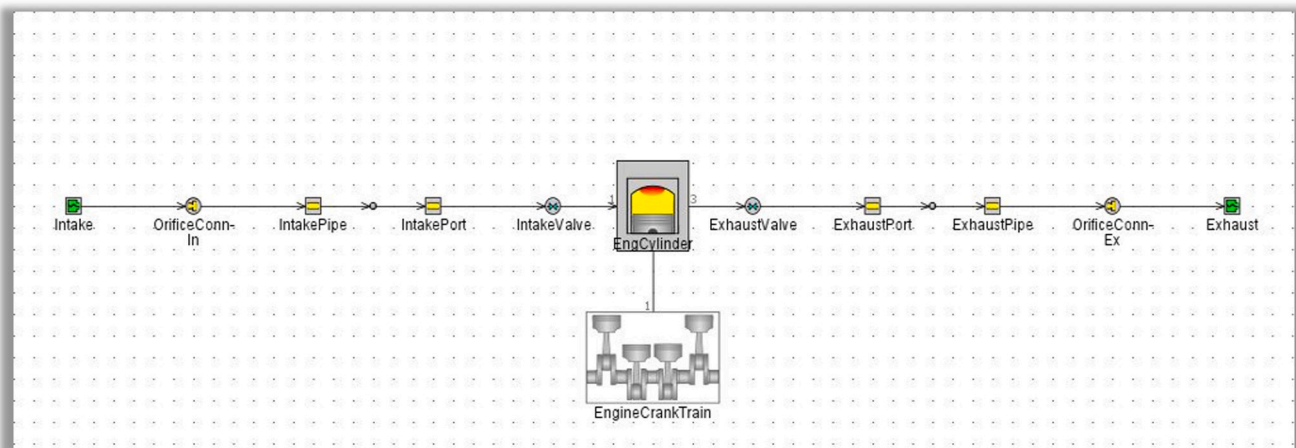


Fig. 3. Three pressure analysis (TPA) model of the single-cylinder DF in GT-Power.

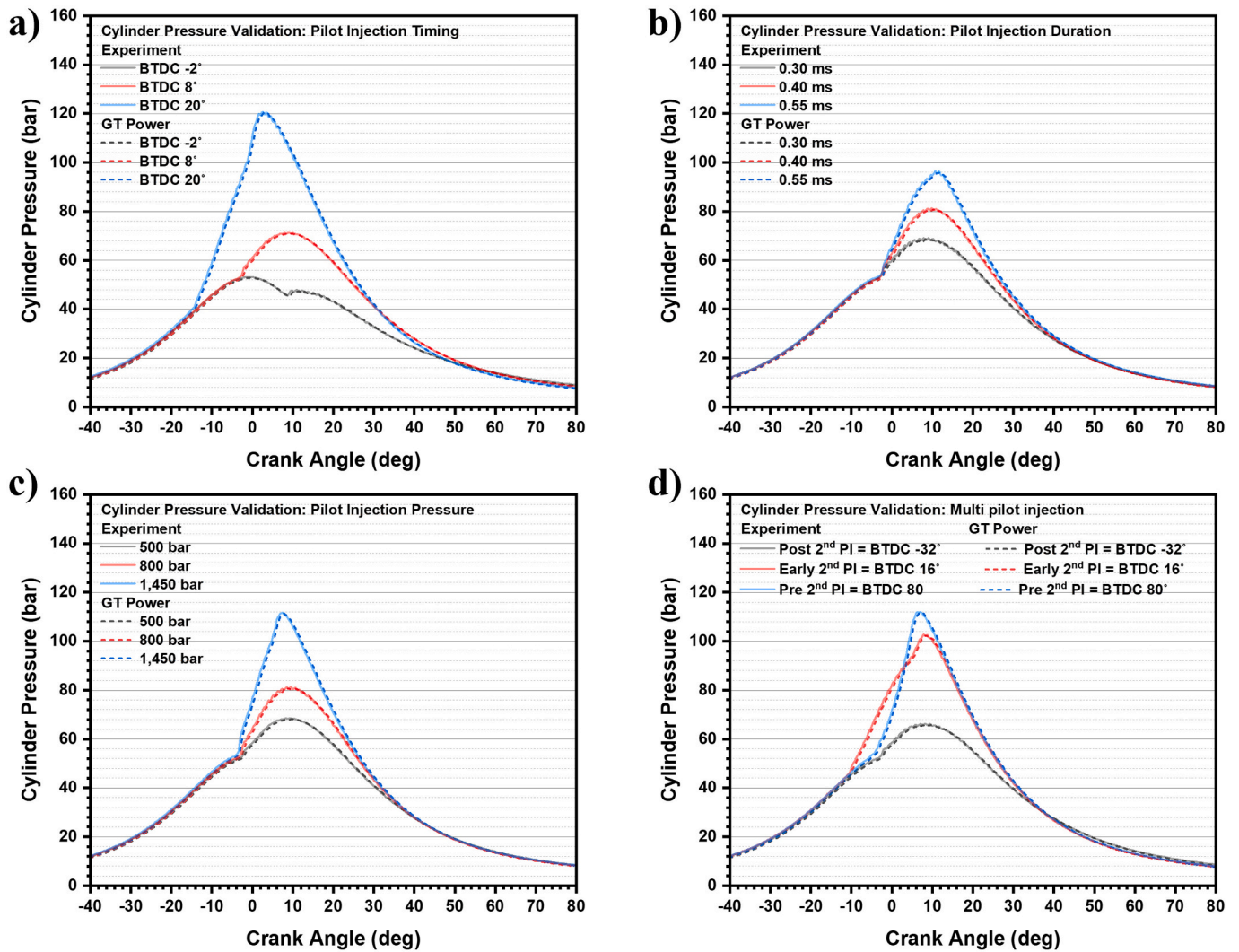


Fig. 4. Comparison of averaged in-cylinder pressure obtained from simulation (TPA) and experimental measurements under varying pilot injection strategy: (a) pilot injection timing, (b) pilot injection duration, (c) pilot injection pressure, and (d) multiple pilot injections.

Table 4

Controlled variables among the engine experimental conditions.

Experimental Conditions	Description
Engine Speed (RPM)	950
Intake Methane Flow Rate (LPM)	29.5
Intake Air Temperature (°C)	45.0
Coolant Temperature (°C)	70.0
Oil Temperature (°C)	70.0

The Post 2nd Pilot strategy involves an additional pilot injection after the reference injection timing. The Early 2nd Pilot strategy introduces an additional pilot injection immediately before the reference injection timing. The Pre 2nd Pilot strategy advances the additional pilot injection by more than 50 crank angles (CA°) prior to the reference timing, injecting pilot fuel into the combustion chamber during the compression stroke well in advance.

The proportion of pilot fuel energy relative to the total fuel supplied to the combustion chamber under each pilot injection strategy is shown in Table 6 below.

Table 5

Manipulated variables among the engine experimental conditions.

Experimental Conditions		Description						
Pilot Injection Timing (CA°, BTDC)		−2	0	4	8 (Ref.)	12	16	20
Pilot Injection Duration (ms)		0.30	0.33 (Ref.)	0.36	0.40	0.45	0.50	0.55
Pilot Injection Pressure (bar)		500	600 (Ref.)	700	800	1000	1450	
Multi Pilot Injection (CA°, BTDC)	Ref. Injection	8						
	Post 2nd Injection	−32	−16		−8		0	
	Early 2nd Injection	12		16		20		
	Pre 2nd Injection	60		80		100		

Table 6

Proportion of pilot fuel energy relative to the total fuel supplied to the combustion chamber under each pilot injection strategy.

Experimental Conditions		Ratio of pilot fuel energy to total energy						
Pilot Injection Timing (CA°, BTDC)		5.1%						
Pilot Injection Duration (ms)		0.30	0.33	0.36	0.40	0.45	0.50	0.55
		3.7%	5.1%	6.3%	7.9%	10.8%	13.5%	16.3%
Pilot Injection Pressure (bar)		500	600	700	800	1000	1450	
		4.2%	5.1%	6.0%	7.2%	8.9%	13.1%	
Multi Pilot Injection (CA°, BTDC)	Ref. Injection	5.1%						
	Post 2nd Injection							
	Early 2nd Injection	4.9%						
	Pre 2nd Injection							

3. Results and discussion

3.1. Effect of pilot injection timing

In this study, to investigate the combustion and emission characteristics associated with pilot injection timing, experiments were conducted under a fixed lambda condition of 1.9. The pilot fuel quantity and injection pressure were held constant, while the injection timing was varied within a range of BTDC -2° to 20°. Combustion behavior and exhaust emissions were measured across these conditions.

Fig. 5 presents the combustion pressure curves and heat release rates measured by varying the pilot fuel injection timing from BTDC -2° to 20°, illustrating the influence of injection timing on combustion pressure and heat release characteristics. As shown in Fig. 5, advancing the pilot injection timing leads to an increase in peak heat release rate and combustion pressure.

As the pilot injection timing was advanced, notable changes were observed in the shape of the ROHR curve. When the pilot fuel was injected between BTDC -2° and 12°, the ROHR curve exhibited a rapid rise immediately after pilot fuel ignition, followed by a gradual increase and a slow decline. This behavior indicates that the mixture surrounding the pilot flame combusted rapidly, resulting in a sharp increase in ROHR, and the subsequent combustion proceeded in a premixed mode, producing a gently sloping ROHR curve.

However, at more advanced injection timings of BTDC 16° and 20°, the heat release rate (HRR) curve exhibited a clear two-stage rise. The first sharp increase was due to the ignition of the pilot fuel, while the second sharp increase occurred near TDC, which can be attributed to the autoignition of the methane-air mixture in the outer regions of the combustion chamber under high temperature and pressure conditions.

Fig. 6 illustrates the MFB profiles corresponding to various pilot fuel injection timings, providing a quantitative assessment of how injection

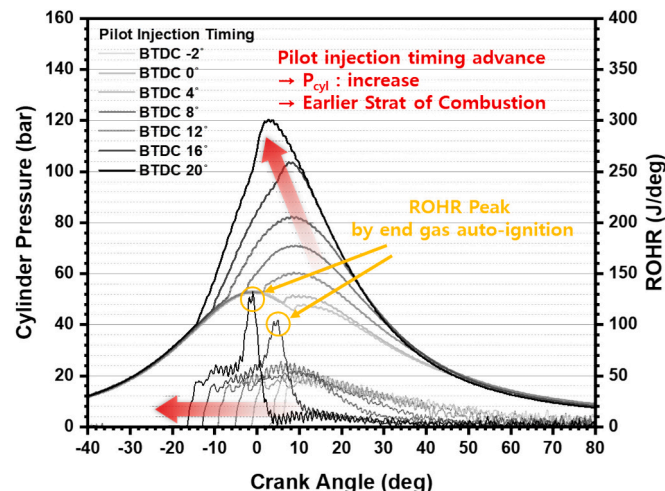


Fig. 5. Cylinder pressure and ROHR according to pilot injection timing.

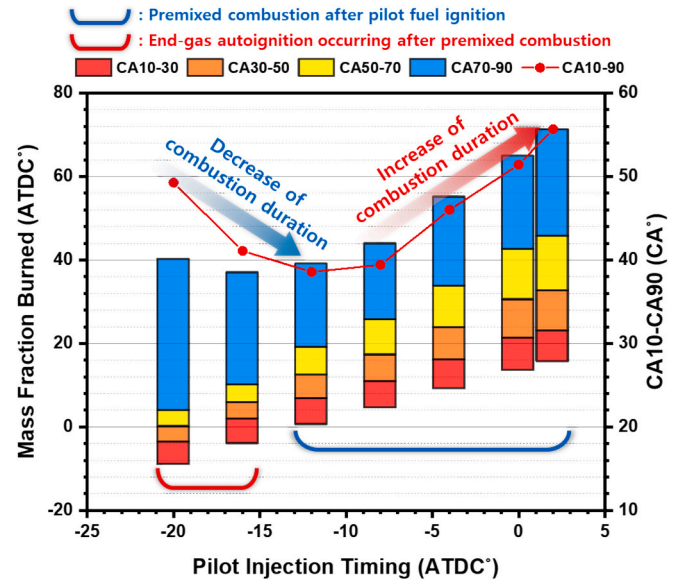


Fig. 6. Comparison of Mass Fraction Burned according to pilot injection timing.

timing influences the combustion progression rate. In the results for injection timings ranging from BTDC -2° to 12°, combustion initiation occurs after TDC, and as the injection timing is advanced, the duration of CA10–90 shortens. This indicates that the combustion phase shifts closer to TDC, and compared to later injection conditions, flame propagation occurs under higher temperature and pressure conditions, thereby shortening the overall combustion duration.

From the point when the pilot injection timing reaches BTDC 16°, combustion begins before TDC, and due to the autoignition of the mixture in the end-gas region, the combustion speed of the mixture is accelerated. As a result, the duration of the CA10–70 interval becomes significantly shorter than that of the CA70–90 interval.

Fig. 7 shows the IMEP and the IMEP COV as functions of pilot injection timing. In general, the crank angle at which 50% of the fuel mass is burned (CA50) is known to be an important parameter for converting the thermal energy generated from combustion into mechanical work [41].

According to the results in Fig. 7, as the pilot injection timing is advanced from BTDC -2° to 16°, IMEP increases. However, when the injection timing is further advanced to BTDC 20°, CA50 deviates from the optimal range, resulting in a decrease in IMEP.

In the case of IMEP COV, it shows a decreasing trend as the pilot injection timing is advanced. Yet, once the injection timing reaches BTDC 8°, IMEP COV approaches about 2%, and further advancement does not lead to a noticeable reduction in combustion variability.

Fig. 8 shows the CO₂ emissions corresponding to various pilot injection timings. Fig. 9 presents the CH₄ emissions under the same conditions. Fig. 10 illustrates the results of converting methane emissions into CO₂-equivalent values based on their global warming potential (GWP).

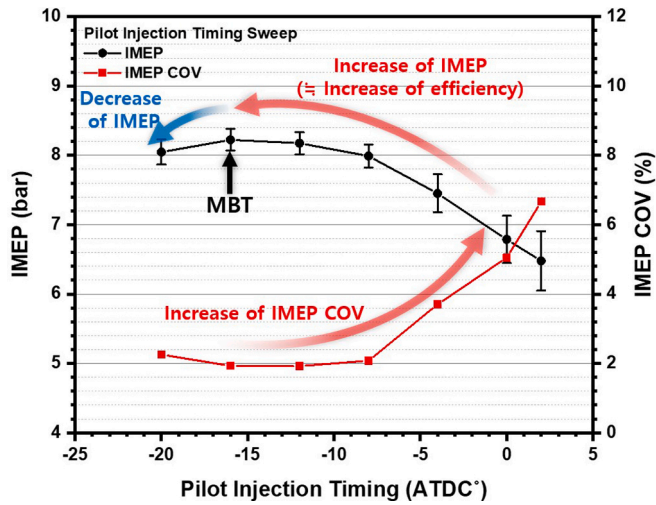


Fig. 7. IMEP & IMEP COV according to pilot injection timing.

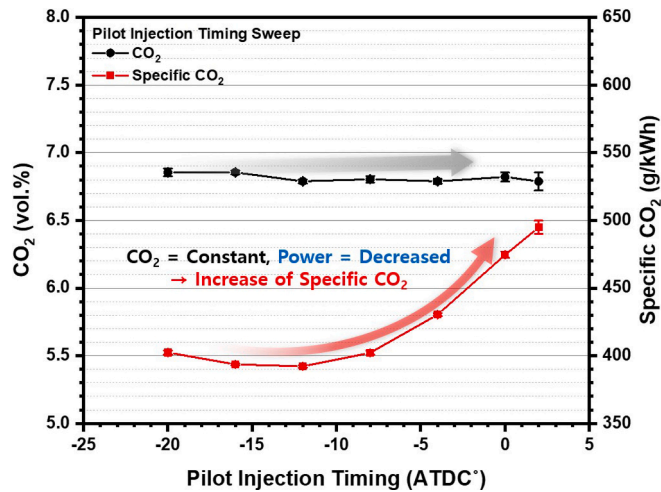


Fig. 8. Carbon dioxide emission according to pilot injection timing.

According to the results shown in Fig. 8 and Fig. 9, the concentrations of CO₂ and CH₄ in the exhaust gas remained nearly constant up to the condition where the pilot injection timing was BTDC 12°. When the timing was advanced beyond this point, autoignition of the mixture in the end-gas region caused the CH₄ concentration to decrease significantly, while oxidation of CH₄ led to a slight increase in CO₂ concentration.

On the other hand, the specific CO₂ emission and specific CH₄ emission, defined as the amount emitted per unit of engine output, decreased markedly as the pilot injection timing was advanced up to BTDC 12°. In this range (BTDC -2° to 12°), the concentrations of CO₂ and CH₄ in the exhaust gas remained nearly unchanged, but the increase in engine output reduced the specific emissions.

When the pilot injection timing was BTDC 20°, the specific CO₂ emission increased due to the decrease in engine output, while the specific CH₄ emission decreased because the reduction in CH₄ concentration caused by autoignition of the mixture in the end-gas region was greater than the effect of the reduced engine output.

According to the results shown in Fig. 10, both the total greenhouse gas emissions and the specific emissions were lowest when the pilot injection timing was set to BTDC 20°, with the specific emission recorded at 957 g/kWh. The trend in greenhouse gas emissions with respect to pilot injection timing closely resembled the variation in methane emissions presented in Fig. 8. The reason the greenhouse gas emission trend is similar to the methane emission trend is that methane has a

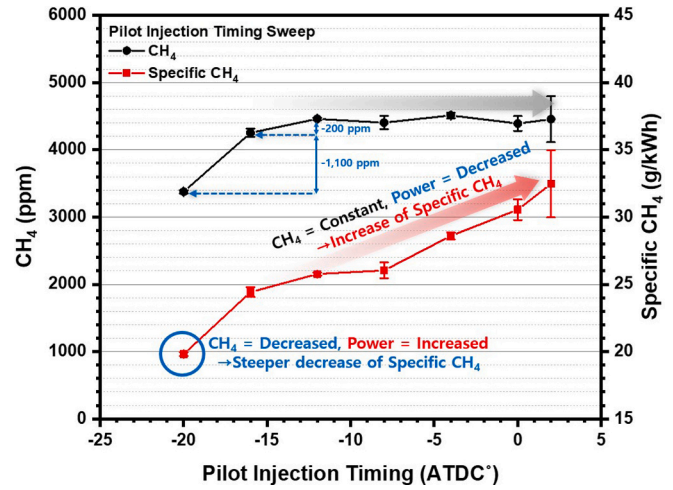


Fig. 9. Methane emission according to pilot injection timing.

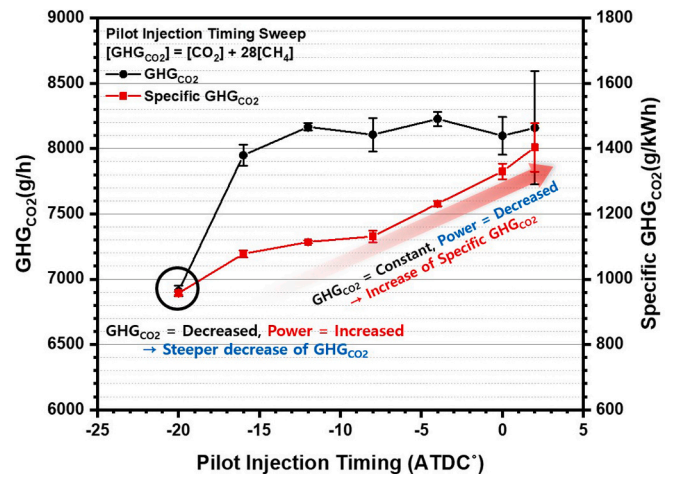


Fig. 10. Green House Gas emission according to pilot injection timing. (For interpretation of the references to colour in this figure legend, the reader is referred to the web version of this article.)

global warming potential approximately 28 times greater than that of carbon dioxide.

Fig. 11 shows the calculated average combustion chamber temperature according to pilot injection timing. The results indicate that as the pilot injection timing is advanced, the ignition timing of the air-fuel mixture also advances, leading to an earlier onset of temperature rise. When the temperature rises earlier in the cycle, the heat generated from combustion combined with the upward motion of the piston increases the pressure and further compresses the in-cylinder gases. This synergistic effect contributes to a higher average combustion chamber temperature.

Fig. 12 illustrates the concentration and specific emission of NO_x in the exhaust gas as a function of pilot injection timing. The results show that both NO_x concentration and specific emissions increase as the pilot injection timing is advanced. This trend is attributed to the rise in peak pressure and temperature within the combustion chamber as the injection timing is advanced, which enhances thermal NO_x formation reactions and consequently increases NO_x production.

3.2. Effect of pilot injection duration

In DF engines, combustion characteristics are known to vary depending on the mixing ratio of diesel and methane [42,43]. In this study, the effects of varying diesel pilot injection quantities, limited to no more than 10% of the total energy input, on the combustion and emission

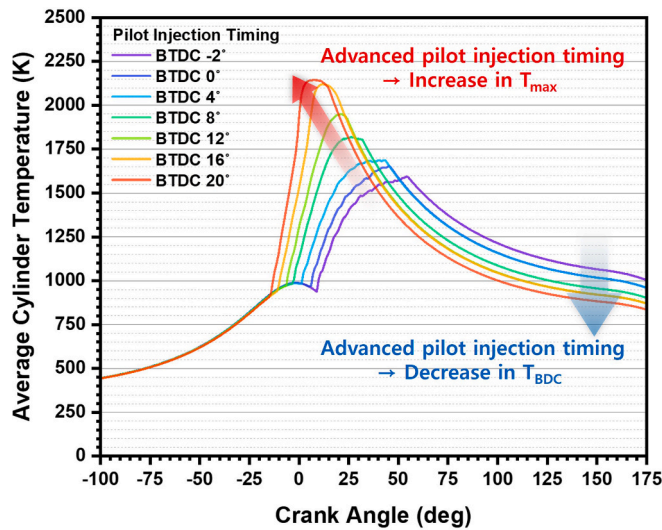


Fig. 11. Calculation results of average in-cylinder temperature according to pilot injection timing.

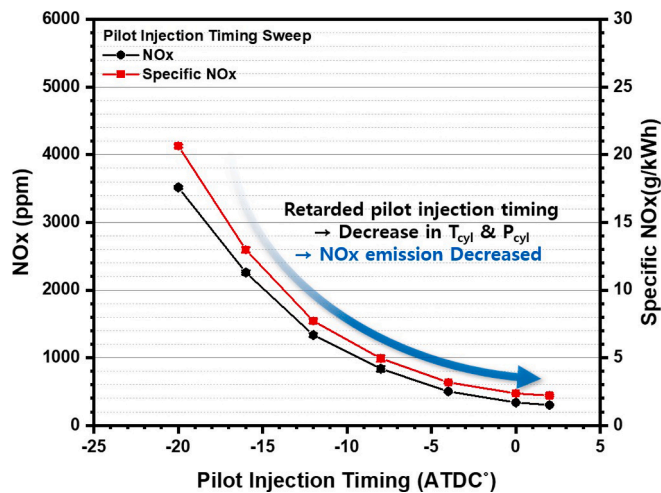


Fig. 12. NO_x emission according to pilot injection timing.

characteristics of a DF engine were investigated. The supply of methane fuel was kept constant, while the pilot injection timing was fixed at BTDC 8° and the injection pressure was maintained constant. The pilot injection duration was varied from 0.30 ms to 0.55 ms, and the corresponding combustion behavior and exhaust emissions were measured.

Fig. 13 presents the measured in-cylinder pressure and heat release rate according to pilot injection duration, illustrating the influence of increased pilot fuel quantity on combustion pressure and heat release behavior. As the pilot injection duration increases from 0.30 ms to 0.55 ms, both the peak combustion pressure and the heat release rate rise accordingly.

This increase is attributed to the enhanced ignition energy resulting from the larger pilot fuel quantity, as well as the greater total energy supplied to the combustion chamber. The rise in ignition energy is evidenced by the magnitude of the initial heat release peak associated with premixed combustion of the pilot diesel, as observed in the early phase of the heat release rate curve. However, no secondary peak in the heat release rate caused by the autoignition of the end gas was observed.

Fig. 14 illustrates the combustion mass fraction as a function of pilot injection duration. As the pilot fuel injection duration increases, the crank angle corresponding to CA10 indicative of ignition delay is observed to advance. A comparison between the conditions with a pilot

injection duration of 0.30 ms and 0.55 ms reveals that the ignition timing advances by approximately 3 CA°, indicating a reduction in ignition delay.

The CA10–70 interval, which represents the main combustion duration, was found to decrease as the pilot injection duration increased. A comparison between the conditions with a pilot injection duration of 0.30 ms and 0.55 ms revealed that the CA10–70 interval was reduced by approximately 10 CA°. This decrease was attributed to the increased pilot fuel quantity, which enhanced the ignition energy supplied to the methane–air mixture.

Fig. 15 presents the indicated IMEP and its IMEP COV as a function of pilot injection duration, providing insight into the impact of pilot fuel quantity on engine performance and combustion stability.

As the pilot injection duration increases, IMEP was observed to rise proportionally due to the increase in ignition energy supplied to the combustion chamber and the greater total fuel energy resulting from the increased pilot fuel quantity. A comparison between the conditions with a pilot injection duration of 0.30 ms and 0.55 ms revealed that IMEP increased by approximately 1.6 bar. In addition, as the pilot injection duration increased, ignition stability improved, resulting in a decrease in IMEP COV.

However, when the pilot injection duration exceeded 0.45 ms, no further reduction in IMEP COV was observed.

In Fig. 16, CO₂ emissions are shown to increase linearly with pilot injection duration. This trend is attributed to the rising proportion of diesel, which has a higher carbon content, among the total fuel supplied to the combustion chamber as the pilot fuel quantity increases. Specific CO₂ emissions decrease by approximately 10 g/kWh when the pilot injection duration increases from 0.30 ms to 0.36 ms, but from 0.36 ms to 0.55 ms, specific CO₂ emissions are observed to rise gradually. However, since both engine output and CO₂ emissions increase simultaneously with longer pilot injection durations, the variation in specific CO₂ emissions remains relatively small.

In Fig. 17, the concentration of unburned methane in the exhaust gas shows little change when the pilot injection duration increases from 0.30 ms to 0.36 ms. However, within the range of 0.40 ms to 0.55 ms, methane concentration decreases proportionally with increasing pilot injection duration. This trend is attributed to the enhanced ignition energy supplied to the methane–air mixture due to the increased diesel quantity, which reduces the occurrence of unburned methane caused by flame quenching. Moreover, the increase in pilot injection duration not only contributes to higher engine output but also mitigates flame quenching effects, thereby reducing specific methane emissions.

According to the results shown in Fig. 18, when the pilot injection duration increases from 0.30 ms to 0.40 ms, the effect of increased CO₂ emissions outweighs the reduction in unburned CH₄, leading to a rise in total greenhouse gas emissions. However, when the pilot injection duration is further extended from 0.40 ms to 0.55 ms, a significant reduction in unburned CH₄ emissions is observed, resulting in a decrease in total greenhouse gas emissions despite the continued rise in CO₂ emissions.

Fig. 19 presents the calculated average in-cylinder temperature as a function of pilot injection duration. The results indicate that while the timing of temperature rise remains unchanged with increasing pilot injection duration, the overall in-cylinder temperature increases. This phenomenon is attributed to both the greater total energy supplied to the combustion chamber and the accelerated combustion rate, which leads to a larger portion of the air–fuel mixture being burned when the piston is near TDC.

Fig. 20 shows the NO_x concentration and specific emissions in the exhaust gas as a function of pilot injection duration. As the pilot fuel quantity increases, both NO_x concentration and specific emissions were observed to rise simultaneously. This increase in NO_x emissions is due to the elevated in-cylinder pressure and temperature resulting from the greater pilot fuel quantity, which promotes thermal NO_x formation.

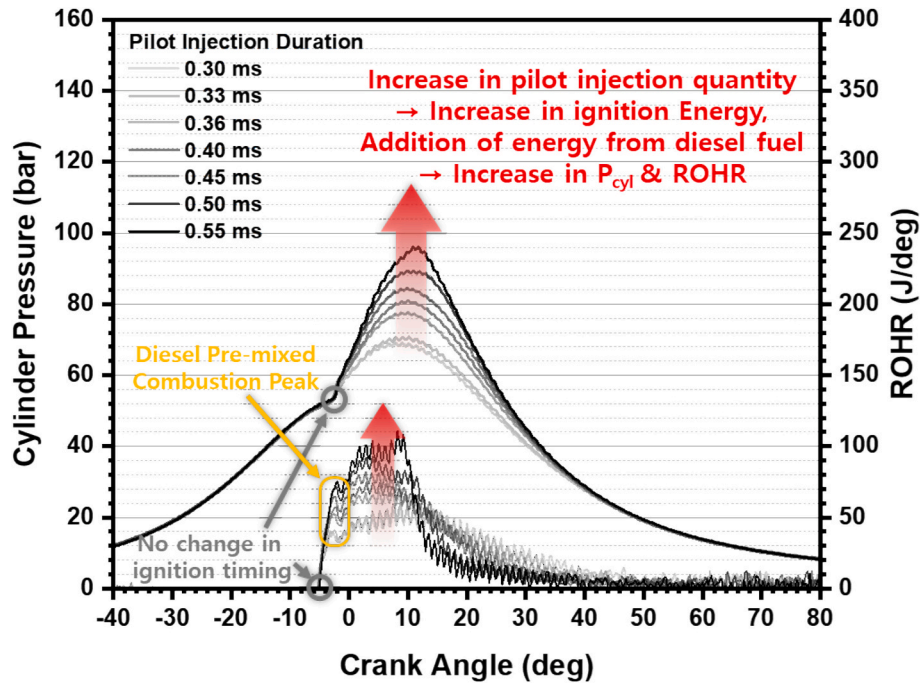


Fig. 13. Cylinder pressure and ROHR according to pilot injection duration.

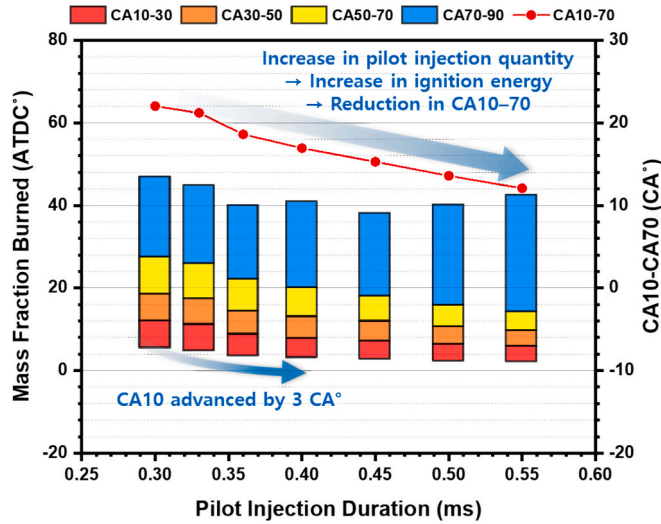


Fig. 14. Mass Fraction Burned according to pilot injection duration.

3.3. Effect of pilot injection pressure

In DF engines, pilot fuel injection pressure has been reported to influence combustion performance and exhaust emissions. According to Liu et al., increasing the pilot injection pressure enhances the penetration of diesel fuel within the combustion chamber, thereby improving the mixing between diesel and the in-cylinder gases. This leads to an advancement of CA50 and a reduction in ignition delay, along with a decrease in unburned hydrocarbon emissions [44].

In this study, the effects of pilot injection pressure on combustion and exhaust emission characteristics were analyzed. To independently identify the influence of pilot injection pressure, the methane fuel supply, pilot fuel quantity, injection duration, and injection timing were kept constant, while only the injection pressure was varied from 500 bar to 1450 bar.

Fig. 21 shows the effects of varying the pilot injection pressure from 500 bar to 1450 bar on in-cylinder pressure and HRR. The results indicate that as

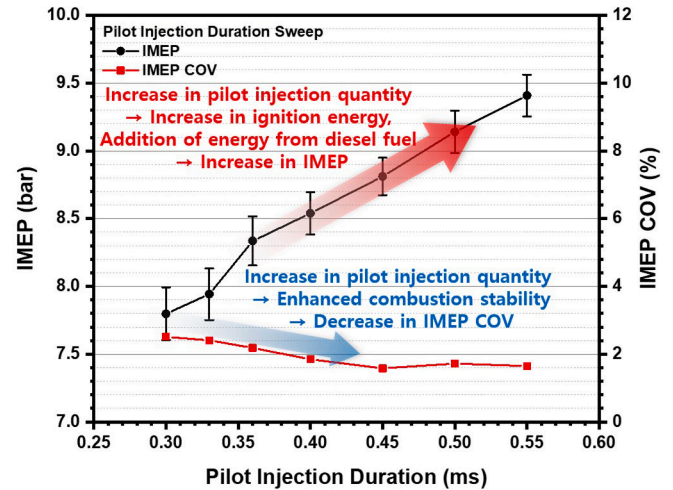


Fig. 15. IMEP & IMEP COV according to pilot injection duration.

the pilot injection pressure increases, both combustion pressure and heat release rate. In addition, under conditions where the pilot injection pressure exceeds 1000 bar, a secondary peak in the heat release rate is observed.

Both an increase in pilot injection pressure and an increase in pilot injection duration result in a greater pilot fuel quantity. However, while the increase in pilot injection duration led to higher combustion pressure and heat release rate, it did not produce the second heat release peak associated with end-gas autoignition. Therefore, the second heat release peak observed under increased pilot injection pressure is attributable not merely to the increase in fuel quantity, but to the extended spray penetration caused by the higher injection pressure.

Fig. 22 was reconstructed based on the in-cylinder diesel fuel distribution images obtained from the study conducted by Federico et al., and this figure schematically illustrates how the distribution of injected fuel expands as pilot injection pressure increases [45]. As shown in the figure, the extension of pilot fuel penetration caused by higher injection pressure is presumed to promote the autoignition of end-gas.

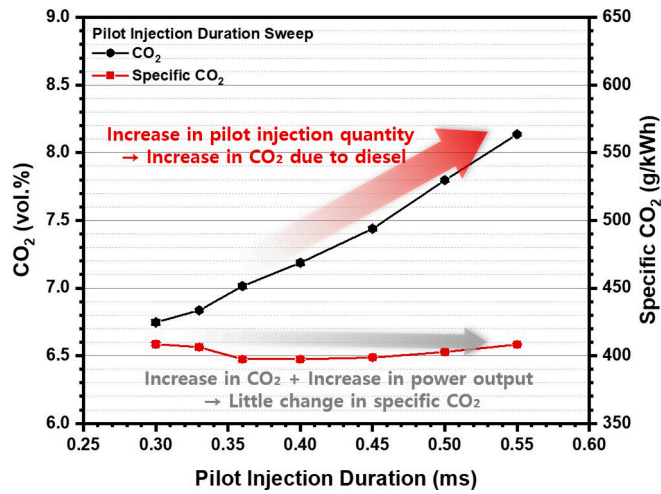


Fig. 16. Carbon dioxide emission according to pilot injection duration.

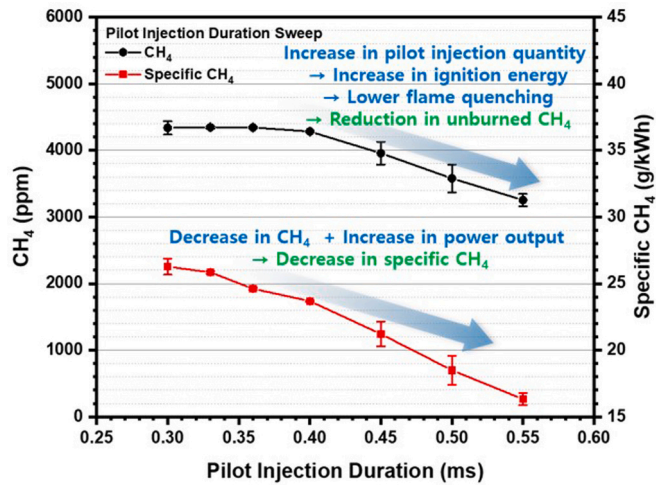


Fig. 17. Methane emission according to pilot injection duration.

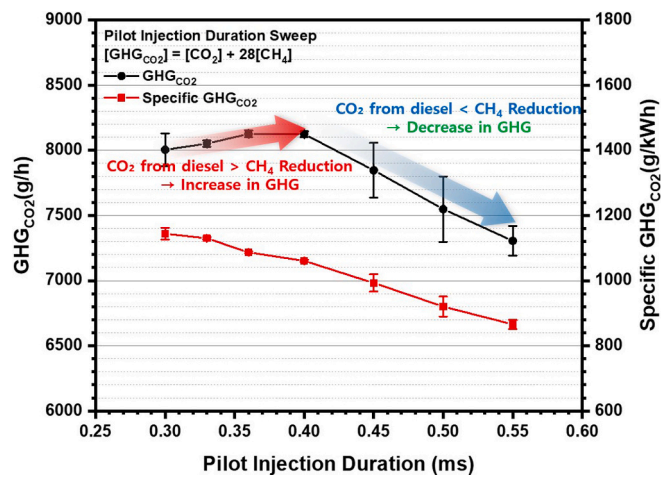


Fig. 18. Green House Gas emission according to pilot injection duration. (For interpretation of the references to colour in this figure legend, the reader is referred to the web version of this article.)

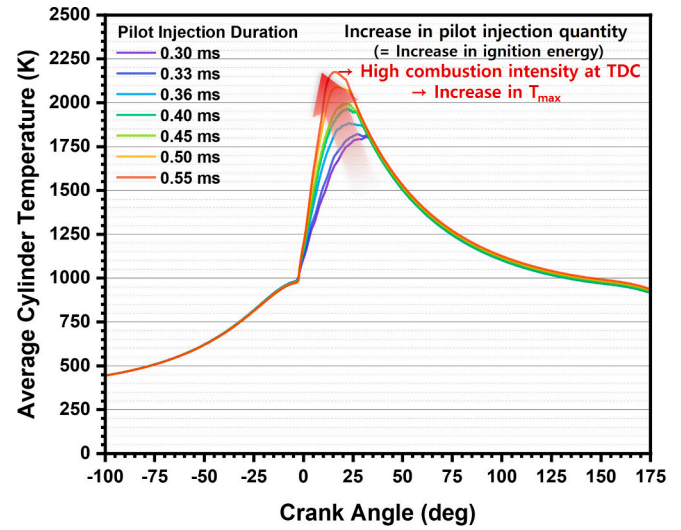


Fig. 19. Calculation results of average engine cylinder temperature according to pilot injection duration.

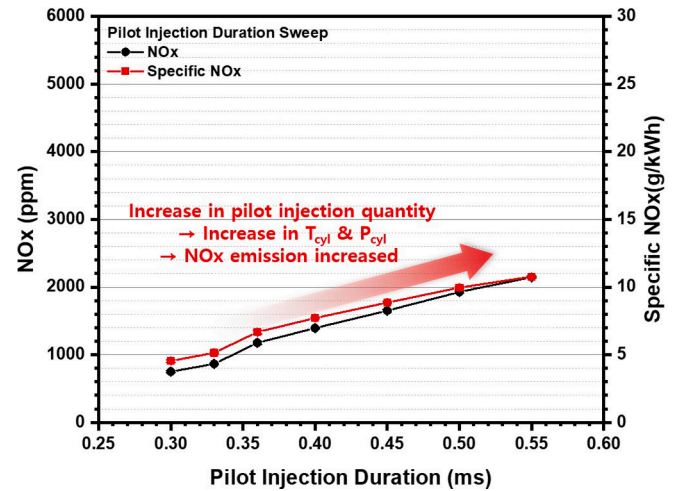
Fig. 20. NO_x emission according to pilot injection duration.

Fig. 23 shows the MFB profiles corresponding to various pilot injection pressures, and it was observed that as injection pressure increases, ignition delay decreases and the combustion rate accelerates.

As pilot fuel injection pressure increases, CA₁₀ advanced. Specifically, when the injection pressure was raised from 500 bar to 1450 bar, CA₁₀ advanced by approximately 5 CA°. In addition, the combustion duration between CA₁₀ and CA₇₀ shortened with increasing injection pressure. When the pilot injection pressure increased from 500 bar to 1450 bar, the CA₁₀–70 interval decreased by about 11.5 CA°, indicating a faster combustion rate under higher injection pressure conditions. This is presumed to be due to the increase in pilot fuel quantity and the deeper penetration of pilot fuel.

Fig. 24 shows the increase in IMEP and the decrease in IMEP COV with rising pilot injection pressure. This is presumed to be due to the increase in total fuel quantity and the faster combustion rate resulting from the higher pilot injection pressure.

Fig. 25 shows that the CO₂ concentration in the exhaust gas increases linearly with rising pilot injection pressure. This increase is due to the greater pilot fuel quantity resulting from higher injection pressure under a constant injection duration, which raises the proportion of diesel with

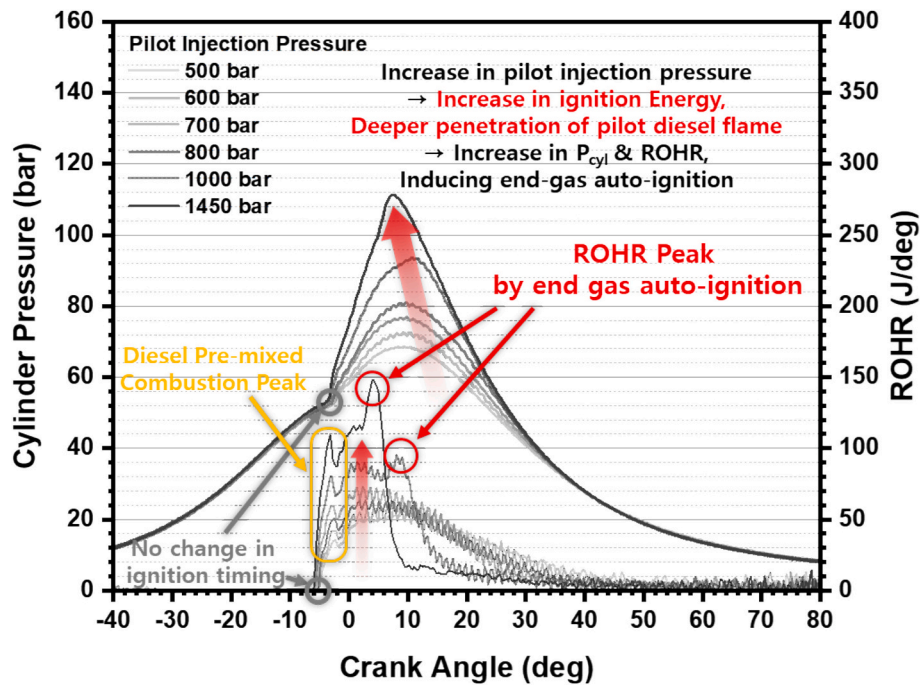


Fig. 21. Cylinder pressure and ROHR according to pilot injection pressure.

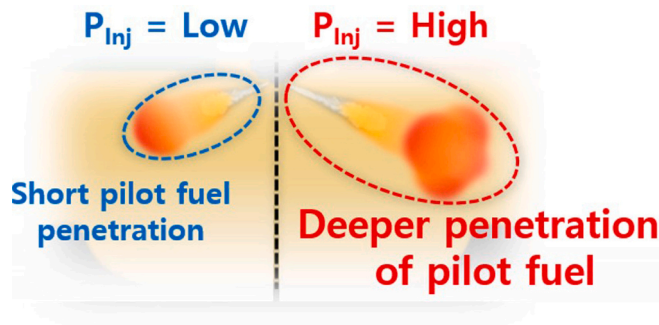


Fig. 22. Schematic diagram of diesel fuel distribution according to pilot injection pressure.

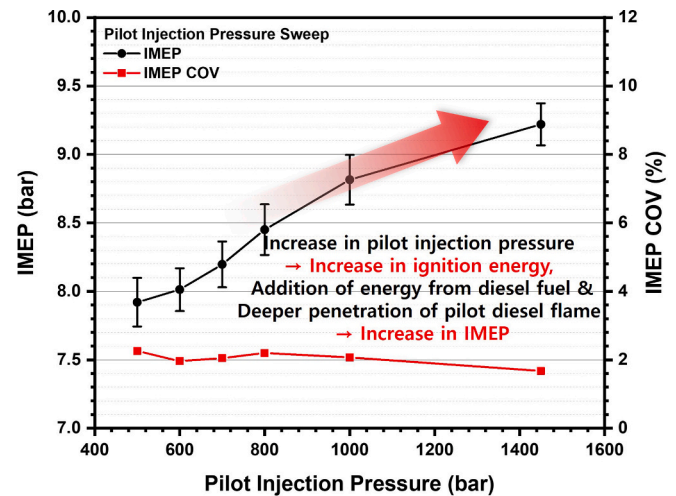


Fig. 24. IMEP & IMEP COV according to pilot injection pressure.

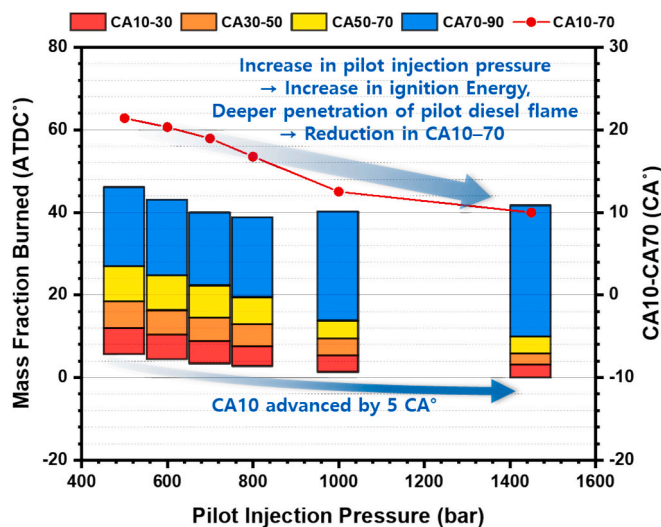


Fig. 23. Mass Fraction Burned according to pilot injection pressure.

high carbon content in the total fuel supplied to the combustion chamber, thereby increasing CO_2 emissions. In addition, the specific CO_2 emissions did not change significantly as pilot injection pressure increased, since both CO_2 emissions and engine output rose simultaneously, balancing the emission intensity.

Fig. 26 illustrates that the concentration of unburned CH_4 in the exhaust gas generally decreases with increasing pilot injection pressure. However, the rate of reduction varies depending on the pressure range. When the injection pressure increases from 500 bar to 800 bar, the decrease in methane concentration is relatively modest, whereas a more pronounced reduction is observed when the injection pressure exceeds 1000 bar.

Under conditions where the pilot injection pressure exceeds 1000 bar, the significant reduction in unburned CH_4 concentration appears to result from end-gas autoignition induced by the increased spray penetration length. This phenomenon suppresses the generation of unburned CH_4 near the cylinder wall region caused by flame quenching.

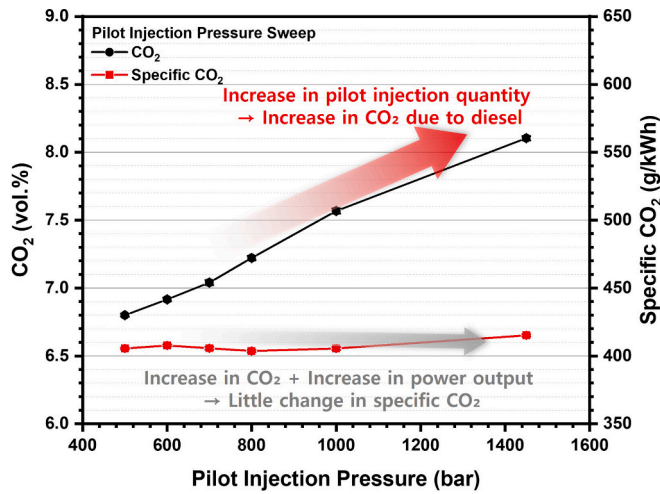


Fig. 25. Carbon dioxide emission according to pilot injection pressure.

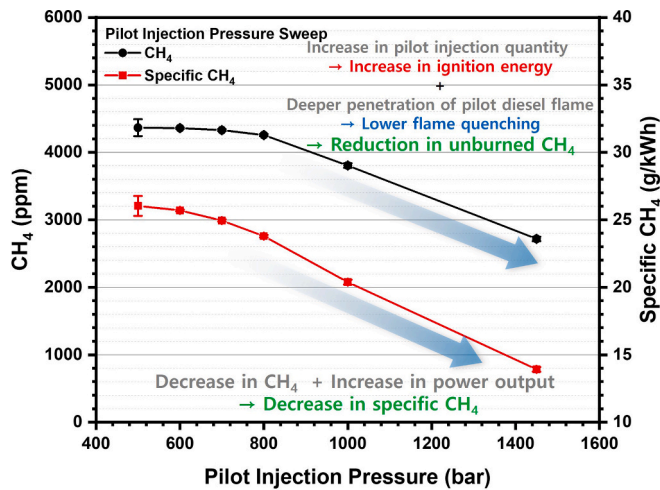


Fig. 26. Methane emission according to pilot injection pressure.

Fig. 27 presents the greenhouse gas emissions as a function of pilot injection pressure, showing that total greenhouse gas emissions first increase and then decrease as injection pressure rises. When the pressure increases from 500 bar to 800 bar, the rise in CO₂ emissions due to increased pilot fuel quantity outweighs the reduction in unburned methane, leading to higher greenhouse gas emissions.

However, when the pressure exceeds 1000 bar, the reduction in CH₄ emissions has a greater impact on greenhouse gas emissions than the increase in CO₂ emissions, resulting in a decrease in total greenhouse gas output. In terms of specific greenhouse gas emissions, a consistently decreasing trend was observed with increasing pilot injection pressure.

In this study, the injection duration was fixed while the injection pressure was increased, and the performance and greenhouse gas emissions were analyzed according to the injection pressure.

Therefore, since higher injection pressure increases the penetration depth of the pilot fuel while simultaneously changing the pilot injection quantity, it is necessary to distinguish whether the achievement of lower unburned methane emissions and greenhouse gas emissions at higher pilot injection pressure is due to the change in penetration depth of the pilot fuel or due to the increase in pilot injection quantity.

Fig. 28 shows the comparison of unburned methane emission concentration and greenhouse gas reduction per injection quantity resulting from increased pilot injection duration and increased injection pressure. According to Fig. 28 (a), Increasing injection pressure reduces more

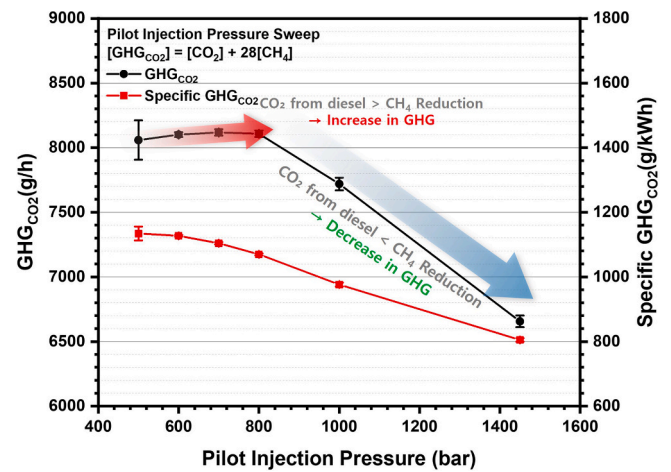


Fig. 27. Green House Gas emission according to pilot injection pressure. (For interpretation of the references to colour in this figure legend, the reader is referred to the web version of this article.)

unburned methane per unit increase in injection quantity compared to extending the injection duration. Furthermore, according to Fig. 28 (b), even when considering the IMEP increase caused by the rise in pilot injection quantity, it is still confirmed that increasing injection pressure achieves a greater reduction in greenhouse gases relative to the increase in injection quantity.

These results suggest that, in terms of greenhouse gas reduction through unburned methane reduction, the penetration depth of pilot fuel is more important than simply increasing the pilot injection quantity.

Fig. 29 presents the calculated average temperature within the combustion chamber as a function of pilot injection pressure. The results indicate that while the timing of temperature rise remains largely unchanged with increasing injection pressure, the overall chamber temperature increases. This temperature elevation is attributed to the greater total energy supplied to the combustion chamber due to increased pilot fuel quantity, as well as the advancement of the combustion phasing, which causes the mixture to ignite closer to TDC, thereby raising in-cylinder temperatures.

Fig. 30 shows the NO_x concentration and specific emissions in the exhaust gas as a function of pilot injection pressure, and it was observed that both NO_x concentration and specific NO_x emissions increase as injection pressure rises. This trend is attributed to the increase in in-cylinder temperature and pressure caused by higher pilot injection pressure, which promotes thermal NO_x formation.

3.4. Effect of Multi-stage pilot injection

In this study, Multi pilot injection was implemented by introducing an additional pilot injection before or after the baseline injection timing, which was set at BTDC 8°. When the additional injection was performed after the baseline timing, it was designated as "Post 2nd pilot injection." If the additional injection occurred earlier than the baseline, specifically between BTDC 12° and BTDC 20°, it was referred to as "Early 2nd pilot injection." Likewise, when the additional injection was introduced significantly earlier, between BTDC 60° and BTDC 100°, it was termed "Pre 2nd pilot injection." The combustion and exhaust emission results obtained from the dual pilot injection configurations were compared with those from the "Single Injection" case, in which only one pilot injection was performed at BTDC 8°.

Fig. 31 compares the in-cylinder pressure and heat release rate between the single pilot injection strategy and the post pilot injection strategy. The results show that both combustion pressure and heat release rate under the post 2nd pilot injection condition are lower than

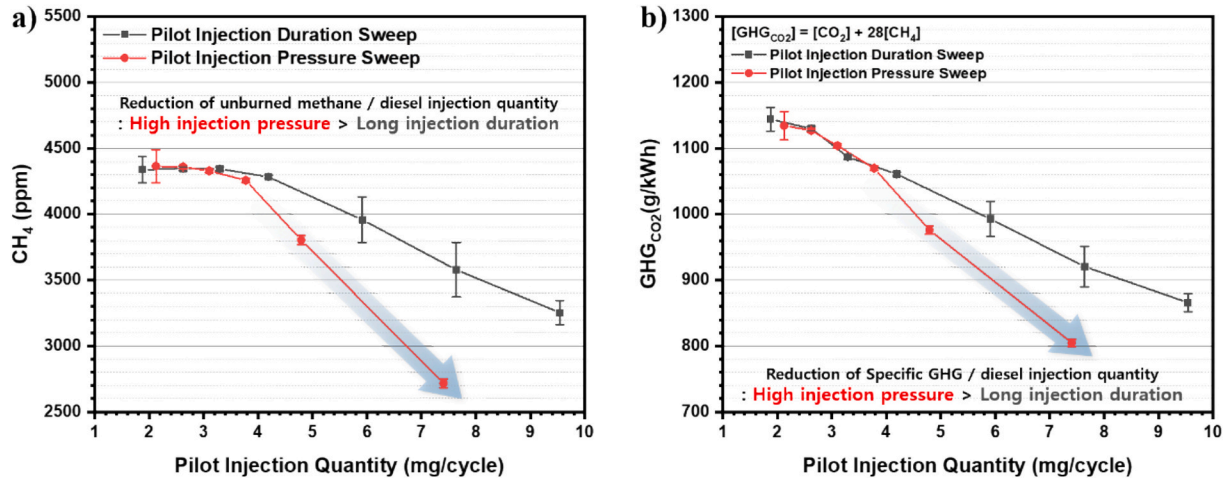


Fig. 28. Comparison of (a) unburned methane emission concentration and (b) greenhouse gas reduction relative to pilot fuel injection quantity under strategies of increased injection duration and increased injection pressure.

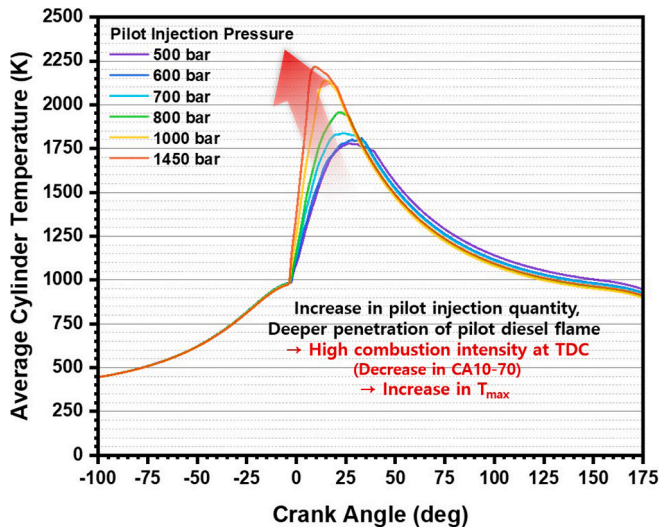


Fig. 29. Calculation results of average engine cylinder temperature according to pilot injection pressure.

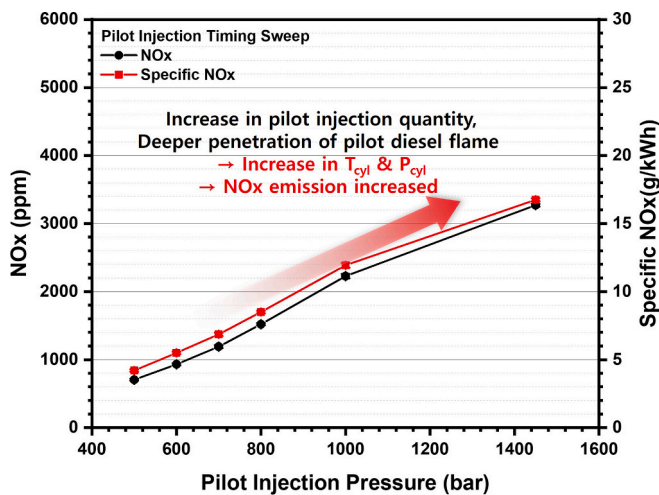


Fig. 30. NO_x concentration according to pilot injection pressure.

those observed with the single pilot injection at the Ref timing, and the negative combustion characteristics of the “Post 2nd Injection” strategy appear to be caused mainly by two factors.

First, the pilot fuel quantity is divided into two injections, reducing the amount of fuel in the initial pilot responsible for ignition. As a result, the number of diesel flame kernels initiating the combustion of the methane–air mixture decreases, leading to a slower pressure rise and reduced heat release rate.

Second, the pilot fuel injected after the first pilot injection is introduced into a combustion mixture that is hot but has a low oxygen concentration, and therefore fails to combust properly, resulting in little meaningful contribution to the overall combustion process.

Fig. 32 compares the in-cylinder pressure and heat release rate between the single pilot injection strategy and the early 2nd pilot injection strategy, in which an additional pilot injection is introduced prior to the reference injection timing (Ref). When the early 2nd pilot injection is performed ahead of the Ref timing, ignition of the methane–air mixture is initiated by the preceding pilot. It was observed that advancing the early 2nd pilot injection timing leads to an earlier onset of combustion.

As the ignition timing of the methane–air mixture advances, both the combustion pressure and heat release rate increase. Notably, when the early 2nd pilot injection is performed at BTDC 16° and 20°, a distinct

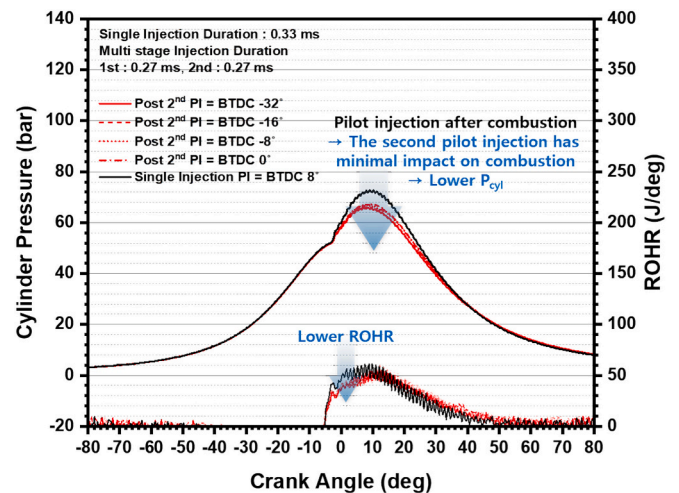


Fig. 31. Cylinder pressure and ROHR according to the multi-stage pilot injection strategy (Post 2nd Pilot Injection).

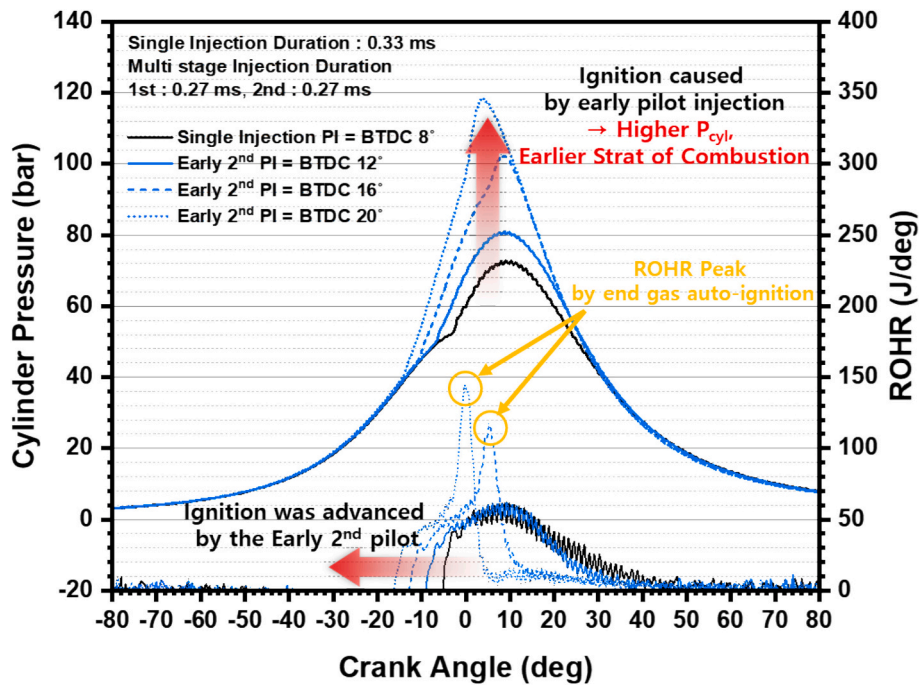


Fig. 32. Cylinder pressure and ROHR according to the multi-stage pilot injection strategy (Early 2nd pilot Injection).

heat release peak caused by end-gas autoignition appears in the heat release rate profile, consistent with the behavior observed in Fig. 5 under the same injection timings.

Fig. 33 compares the in-cylinder pressure and heat release rate between the single pilot injection strategy and the pre 2nd pilot injection strategy. In terms of heat release characteristics, the single pilot injection strategy shows a broad and low-intensity heat release profile. In contrast, the pre 2nd pilot injection strategy shows a narrow and high-intensity heat release profile.

In addition, a distinct difference was observed in the timing of heat release onset between the reference injection condition and the strategy involving pre 2nd pilot injection followed by ignition through pilot injection at BTDC 8°. Under the single pilot injection strategy, a heat release peak appears due to pilot diesel ignition, followed by a heat release curve associated with the combustion of the methane–air mixture.

However, when the pre 2nd pilot injection is performed and ignition occurs with the subsequent injection at BTDC 8°, the heat release rate rises sharply immediately after the BTDC 8° injection, and the onset of heat release is advanced.

Fig. 34 is a reconstructed image based on the CFD-calculated in-cylinder temperature distribution results from studies conducted by Mustafa et al. and Amin et al. on situations where pilot fuel is divided into two injections [38,46].

When pilot fuel is pre-injected and subsequently ignited by the later pilot injection, as shown in the figure, the pre-injected fuel does not ignite immediately but reaches the cylinder wall, and is then presumed to combust following the ignition of the later pilot fuel.

As a result, a wide high-temperature region is formed from the cylinder center to the wall, which is estimated to accelerate the combustion of the methane–air mixture and reduce unburned methane.

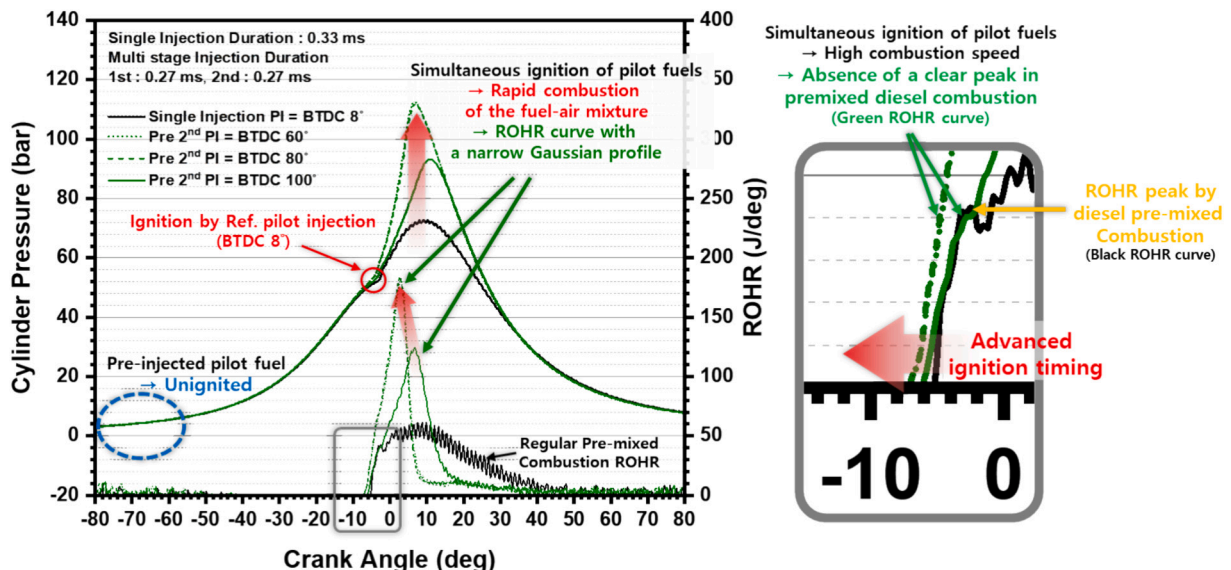


Fig. 33. Cylinder pressure and ROHR according to the multi-stage pilot injection strategy (Pre 2nd Pilot Injection).

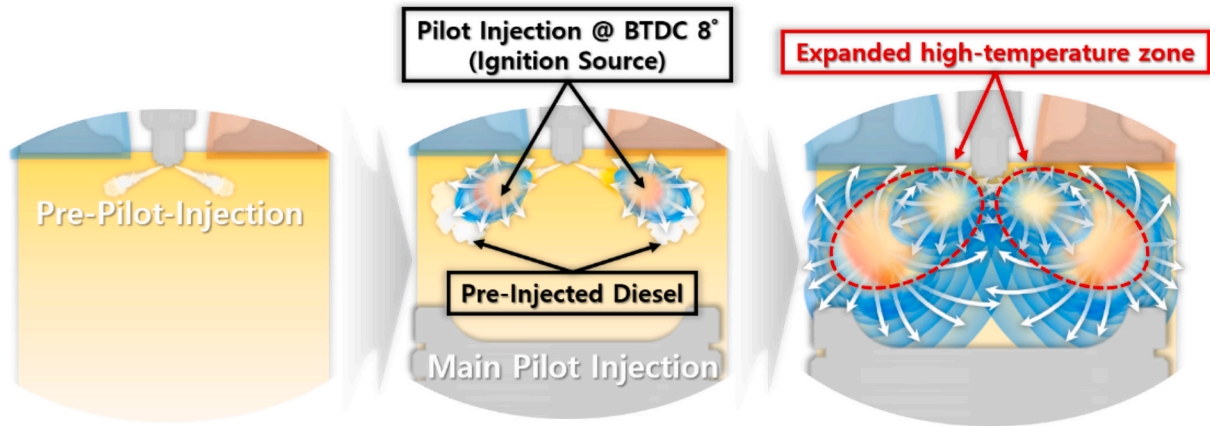


Fig. 34. Conceptual diagram of ignition process: Multi-stage pilot injection (pre 2nd pilot & ref-pilot).

Fig. 35 presents the combustion mass fraction profiles under various pilot injection strategies, enabling a more quantitative assessment of ignition delay and combustion rate. In the case of the post 2nd pilot strategy, both the ignition delay and the CA10–70 duration was the longest among the tested conditions. Notably, the CA10–70 phase accounted for approximately 50% of the total combustion duration (CA10–90), representing the highest proportion observed.

Under the post 2nd pilot strategy, the methane–air mixture is ignited by the pilot fuel injected at BTDC 8°, while the subsequently injected pilot fuel does not contribute meaningfully to the combustion process, as previously discussed in Fig. 31. Consequently, the effective ignition energy supplied to the mixture is lower than that of the single injection strategy, resulting in a delayed CA10 and an extended CA10–70 duration.

Conversely, the pre 2nd Pilot strategy in Fig. 35 exhibited a significantly shorter CA10–70 duration less than CA10° indicating a much faster combustion rate. The ignition delay following the pilot injection at BTDC 8° was also reduced to below 8 CA°.

Fig. 36 presents a comparison of the IMEP and its IMEP COV between the multi-stage pilot injection strategy and the single pilot injection strategy.

In Fig. 36, generally exhibited lower IMEP and higher IMEP COV compared to the single pilot injection strategy. Furthermore, as the timing of the additional pilot injection was delayed, IMEP decreased while IMEP COV increased. This is because, after the methane–air mixture is ignited by

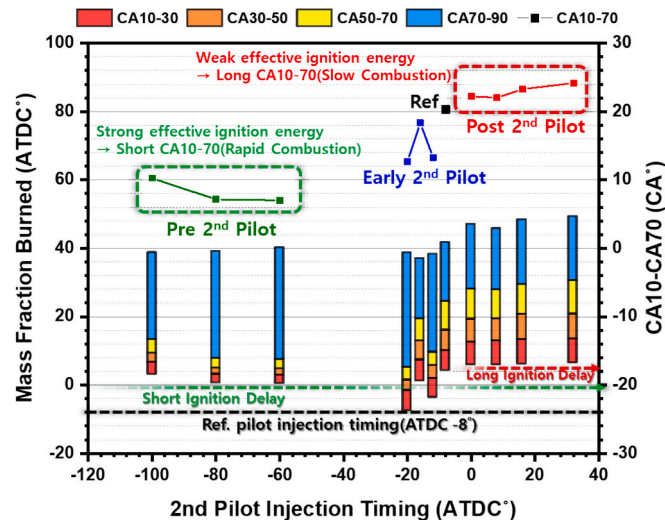


Fig. 35. Mass Fraction Burned according to the multi-stage pilot injection strategy.

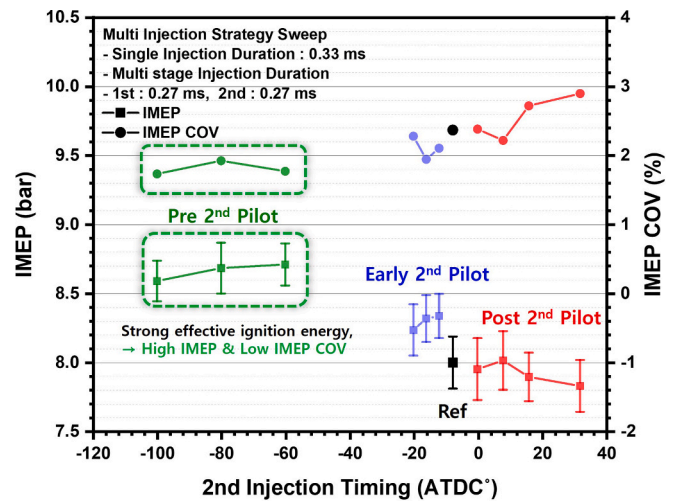


Fig. 36. IMEP & IMEP COV according to the multi-stage pilot injection strategy.

the pilot fuel injected at BTDC 8°, the combustion of the subsequently injected diesel fuel becomes incomplete or inefficient.

The early 2nd pilot strategy showed higher IMEP and lower IMEP COV compared to the single pilot injection strategy. This is because the ignition of the mixture was initiated by the pilot fuel injected earlier among the two pilot injections, causing the combustion phase to advance. However, it was found that as the timing of the additional pilot fuel injection was further advanced, the IMEP decreased.

The pre 2nd pilot strategy recorded the highest IMEP and the lowest IMEP COV among all injection strategies. This is attributed to the simultaneous ignition of the pre-injected pilot fuel and the pilot fuel injected at BTDC 8°, which enhances combustion stability and increases combustion speed.

Additionally, when the pre 2nd pilot strategy was applied, a decreasing trend in IMEP was observed as the timing of the additional pilot injection was further advanced. This is presumed to result from the fact that as the pre 2nd pilot injection timing is advanced, the pre-injected pilot fuel becomes diluted with the mixture before ignition occurs by the pilot injection at BTDC 8°, thereby forming a weaker ignition kernel.

In Fig. 37 and Fig. 38, for the post 2nd pilot strategy, the concentrations of CO₂ and CH₄ hardly changed depending on the timing of the additional pilot injection. However, as the timing of the additional pilot injection was delayed, the specific CO₂ emissions and specific CH₄ emissions tended to increase. This appears to be because the concentration of the two components in the exhaust gas remained relatively

constant, but the output decreased due to the delayed injection timing, thereby increasing the specific emissions. As a result, in Fig. 36, the post 2nd pilot strategy generally recorded higher greenhouse gas emissions among the three multi-stage pilot injection strategies.

In Fig. 37 and Fig. 38, for the early 2nd pilot strategy, as the timing of the additional pilot injection was advanced, the CO_2 concentration increased and the CH_4 concentration decreased. This is because, as the timing of the additional pilot injection was advanced, the pressure and temperature in the combustion chamber increased, and autoignition occurred in the end-gas region, thereby oxidizing more unburned CH_4 into CO_2 .

Moreover, since the magnitude of increase and decrease in the concentrations of the two components according to the timing of the additional pilot injection was greater than the change in engine output, the specific CO_2 emissions increased and the specific CH_4 emissions decreased. As a result, in Fig. 39, the early 2nd pilot strategy recorded 950 g/kWh of greenhouse gas emissions when the additional pilot injection was performed at BTDC 20° .

In Fig. 37 and Fig. 38, for the pre 2nd pilot strategy, the CO_2 concentration in the exhaust gas was the highest and the CH_4 concentration was the lowest among all multi-stage pilot injection strategies. This is presumed to be because the pilot fuel injected in advance and the pilot fuel injected at BTDC 8° ignited simultaneously, forming widely distributed ignition kernels in the combustion chamber, which effectively oxidized the unburned CH_4 .

Furthermore, in the case of applying the pre 2nd pilot strategy, when the additional pilot injection was introduced at BTDC 60° and BTDC 80° , both the specific CO_2 emissions and specific CH_4 emissions were lower than those of other multi-stage pilot injection strategies. These results are because the pre 2nd pilot strategy produced higher output and lower specific unburned CH_4 emissions compared to other multi-stage pilot injection strategies. As a result, in Fig. 39, the pre 2nd pilot strategy recorded 765 g/kWh of greenhouse gas emissions when the additional pilot injection was performed at BTDC 60° , which is the lowest greenhouse gas emission value among all pilot injection strategies.

Fig. 40 presents the calculated average in-cylinder temperature under the post 2nd pilot multi-stage pilot injection strategy. As shown, the average temperature begins to rise following the onset of combustion of the methane–air mixture at BTDC 8° . However, as the timing of the additional pilot injection is further retarded, the average temperature tends to decrease.

This trend is attributed to the fact that delayed pilot injections occur under conditions of reduced oxygen concentration within the combustion chamber. Consequently, the combustion of the additionally injected diesel fuel becomes less effective, leading to a lower overall temperature rise.

Fig. 41 presents the calculated average in-cylinder temperature

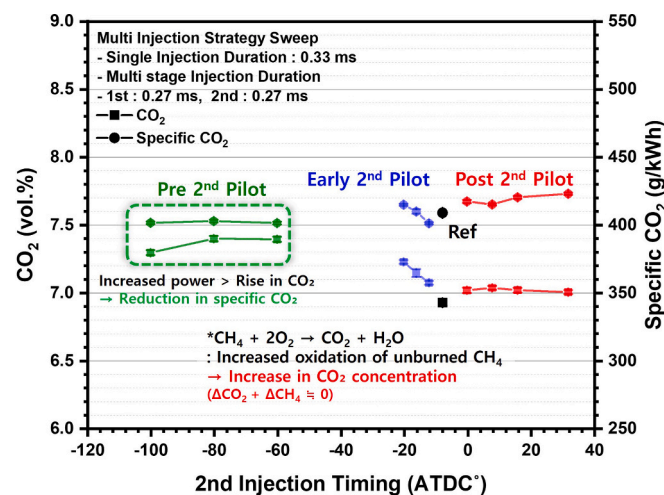


Fig. 37. Carbon dioxide emission according to the multi-stage pilot injection strategy.

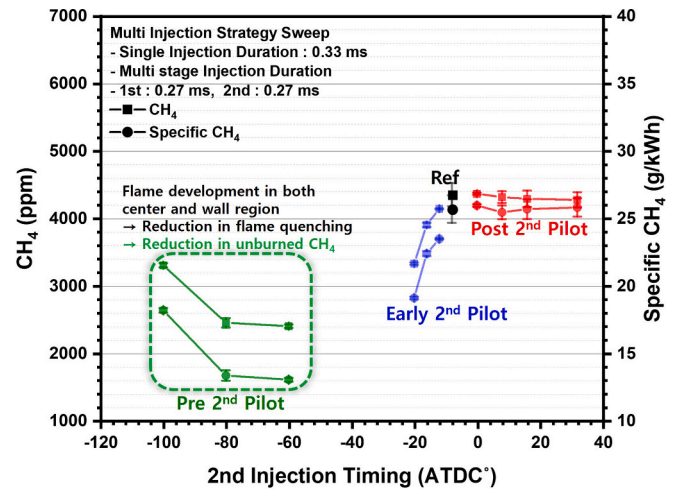


Fig. 38. Methane emission according to the multi-stage pilot injection strategy.

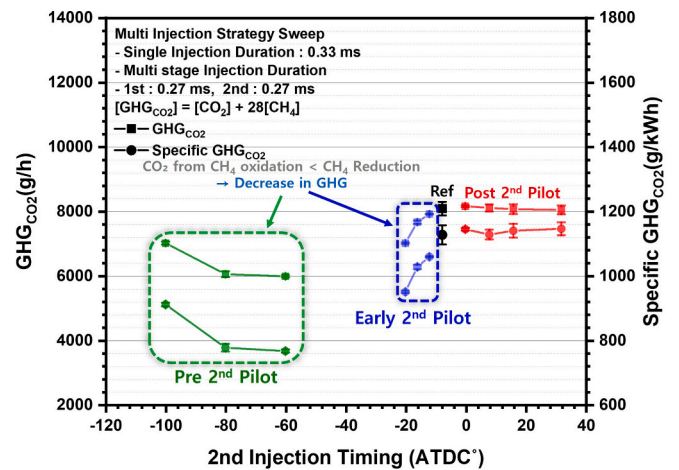


Fig. 39. Green House Gas emission according to the multi-stage pilot injection strategy. (For interpretation of the references to colour in this figure legend, the reader is referred to the web version of this article.)

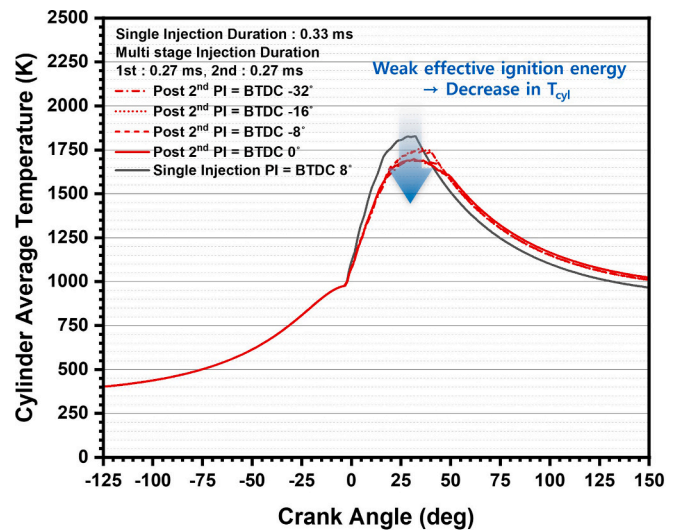


Fig. 40. Calculation results of average engine cylinder temperature according to the multi-stage pilot injection strategy (Post 2nd Injection).

under the early 2nd pilot strategy. The results indicate that as the timing of the additional pilot injection is advanced, the onset of temperature rise occurs earlier, and the peak in-cylinder temperature increases correspondingly.

Fig. 42 presents the calculated average in-cylinder temperature under the pre 2nd pilot strategy. The results indicate that advancing the timing of the additional pilot injection leads to a reduction in peak in-cylinder temperature.

This phenomenon is presumed to occur because when the pilot fuel is injected as early as BTDC 100°, it undergoes significant diffusion before being ignited by the pilot fuel injected at BTDC 8°.

Fig. 43 presents the concentration and specific emissions of NO_x in the exhaust gas under various multi-stage pilot injection strategies. When the post 2nd pilot strategy was applied, both the NO_x concentration and specific emissions were lower compared to other pilot injection strategies. This reduction is attributed to the relatively weak ignition energy provided by the pilot injection, which leads to slower combustion and consequently lower in-cylinder temperatures conditions that suppress the formation of thermal NO_x.

Under the early 2nd pilot strategy, Fig. 43 shows that both the concentration and specific emissions of NO_x in the exhaust increase sharply as the timing of the additional pilot injection is further advanced. This trend corresponds with the results in Fig. 41, which indicate that advancing the pilot injection timing leads to a rise in peak in-cylinder temperature and prolongs the duration of elevated combustion chamber temperatures. These thermal conditions promote the formation of thermal NO_x, resulting in higher NO_x emissions.

Under the pre 2nd pilot strategy, Fig. 43 shows that both the concentration and specific emissions of NO_x in the exhaust decrease as the timing of the additional pilot injection is further advanced. This trend corresponds with the results in Fig. 40, which indicate that advancing the pilot injection timing leads to a reduction in peak in-cylinder temperature, thereby suppressing the formation of thermal NO_x.

Notably, in the pre 2nd pilot strategy, when the additional injection was carried out at BTDC 60°, the peak in-cylinder temperature was approximately 1160 °C, which is similar to that observed in the early 2nd pilot strategy with injection at BTDC 20°. However, in the latter case, the NO_x concentration and specific emissions were significantly higher. This is presumed to be because, although the peak in-cylinder temperatures under both conditions are similar, in the early 2nd pilot strategy with additional injection at BTDC 20°, the combustion chamber gases remained at high temperature for a longer duration.

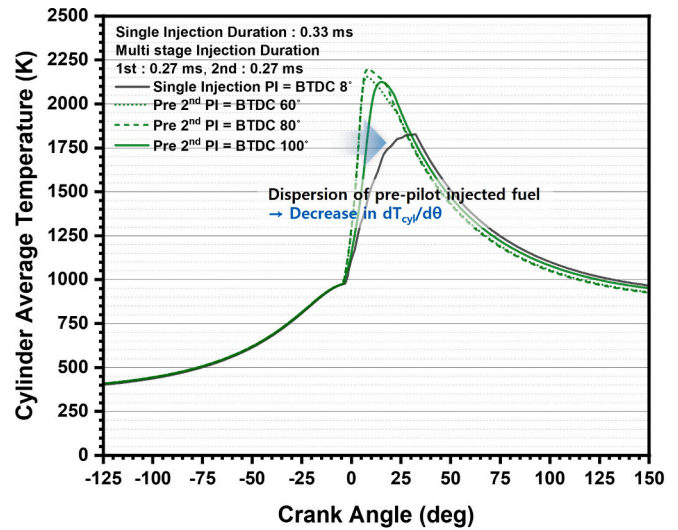


Fig. 42. Calculation results of average engine cylinder temperature according to the multi-stage pilot injection strategy (Pre 2nd Injection).

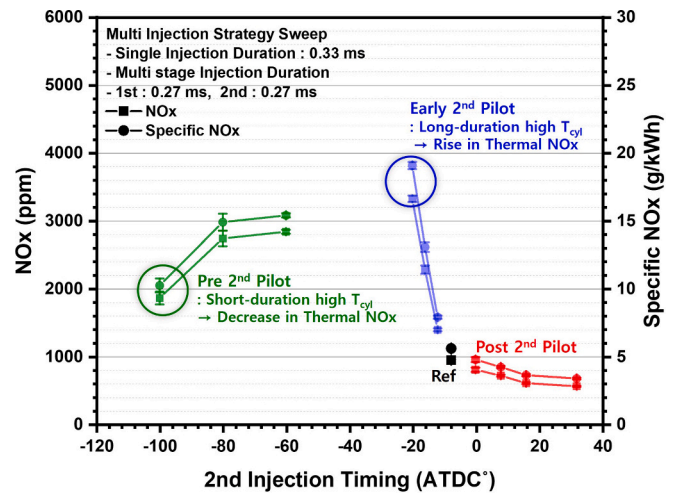


Fig. 43. NO_x emission according to the multi-stage pilot injection strategy.

4. Conclusions

The objective of this study is to investigate the effects of factors related to pilot injection on the performance and exhaust emissions of a DF engine. To this end, combustion and exhaust data were obtained under variations in pilot injection timing, injection duration, injection pressure, and multi-stage injection strategies. The main conclusions derived from these results are summarized as follows:

1. Within the experimental conditions, when the pilot injection timing was advanced, output increased and unburned methane decreased, while nitrogen oxides tended to rise. When the pilot injection timing was advanced beyond BTDC 16°, end-gas autoignition occurred. The pilot injection timing conditions at which end-gas autoignition occurred showed lower greenhouse gas emissions compared to other pilot injection timing conditions. End-gas autoignition has a positive effect in terms of unburned methane oxidation, but it is generally known as an abnormal combustion phenomenon. Therefore, for application to actual engines, considerations regarding engine durability as well as noise and vibration caused by end-gas autoignition are required.

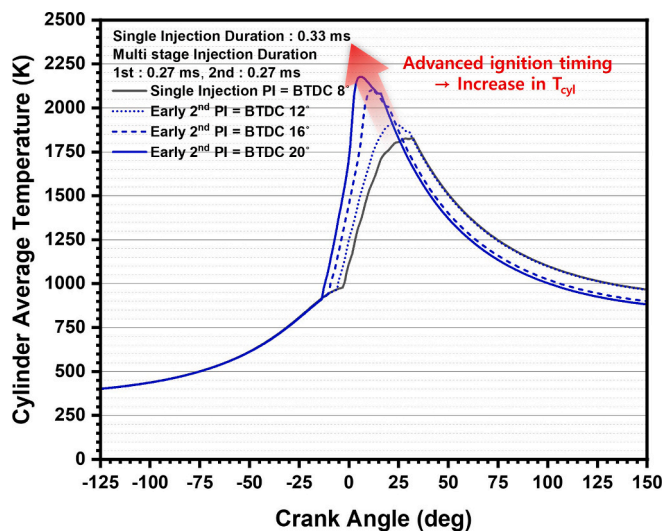


Fig. 41. Calculation results of average engine cylinder temperature according to the multi-stage pilot injection strategy (Early 2nd Injection).

2. When the pilot injection duration increased, the amount of pilot fuel supplied to the combustion chamber also increased, resulting in faster combustion speed, but end-gas autoignition did not occur. When the pilot injection duration was between 0.30 ms and 0.40 ms, greenhouse gas emissions increased in proportion to the injection duration, but when the duration exceeded 0.40 ms, greenhouse gas emissions decreased.
3. When the pilot injection pressure increased, not only did the amount of fuel supplied increase, but the spray penetration length was also extended. This induced combustion behavior is different from simply increasing the pilot injection quantity, and end-gas autoignition was observed under pilot injection pressure conditions above 1000 bar. Under these conditions, unburned methane was significantly reduced.
4. Multi-stage pilot injection tests were conducted under three categorized conditions, and the greenhouse gas reduction effects of each strategy are as follows:

Post 2nd Pilot: Since injection occurred after combustion had already started, the combustion of the additional fuel was incomplete, and no meaningful improvement in methane reduction was observed.

Early 2nd Pilot: Combustion was initiated by the earlier of the two pilot injections, resembling the case of advancing a single pilot injection timing. As the timing of the earlier pilot injection was advanced, greenhouse gas emissions decreased.

Pre 2nd Pilot: Among all pilot injection strategies, it generally recorded relatively low greenhouse gas emissions and high output. This is presumed to be because the pre-injected pilot fuel did not ignite immediately but mixed with the air–methane mixture and ignited almost simultaneously with the pilot fuel injected at BTDC 8°, forming widely distributed ignition kernels inside the combustion chamber, thereby reducing flame quenching and increasing combustion speed.

Credit author statement

Minuk Jeong: Methodology, Investigation, Writing–original draft.

Sungwook Park: Conceptualization, Supervision, Methodology, Resources, Writing – review & editing, Project administration.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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Data availability

Data will be made available on request.

References

- [1] UNCTAD, Review of Maritime Transport 2023, 2023.
- [2] I. Fourth, Greenhouse Gas Study, International Maritime Organization, London, 2020.
- [3] R. MEPC, 2023 IMO Strategy on Reduction of GHG Emissions from Ships, 2023.
- [4] G. Dominioni, Revenue distribution in the IMO net-zero framework, , Available at SSRN 5347655,, 2025.
- [5] J.-Y. Lee, International trend for maritime DECARBONIZATION, Korea Maritime Cooperation Center 15 (2025) 1–18.
- [6] C. Fetting, The European green deal, , ESDN report, December, 2, 2020, 53..
- [7] P. UPDATE, The Maritime Sector in the European Union Emissions Trading System, , POLICY,, 2023.
- [8] A. Ingwersen, A.J.H. Menacho, S. Pfister, J.N. Peel, R. Sacchi, C. Moretti, Prospective life cycle assessment of cost-effective pathways for achieving the FuelEU maritime regulation targets, Sci. Total Environ. 958 (2025) 177880.
- [9] C. Wang, Z. Peng, J. Yang, N. Zhang, K. Li, X. Li, Optimal alternative fuel selection for dual-fuel ships under FuelEU maritime regulations: environmental and economic assessment, J. Mar. Sci. Eng. 13 (2025) 1105.
- [10] H. Lee, J. Lee, G. Roh, S. Lee, C. Choung, H. Kang, Comparative life cycle assessments and economic analyses of alternative marine fuels: insights for practical strategies, Sustainability 16 (2024) 2114.
- [11] Y. Cai, Z. Wang, Q. Li, Y. Song, K. Dong, Corrosion behavior of marine engine piston 16CrMo alloy steel in the artificial methanol fuel combustion product environment, Mater. Today Commun. 49 (2025) 114417.
- [12] W. Zhang, H. Gao, Q. Wang, D. Liu, E. Zhang, Study on the corrosion and wear mechanism of a core friction pair in methanol-fueled internal combustion engines, Materials (Basel, Switzerland) 18 (2025) 1966.
- [13] C.J. Yong-Seok Lee, Kunhwa Jung, Cheolyoung Lim, Hoyojeong Lee, Methane emission reduction technologies for medium-speed dualfuel engines, , CIMAC Congress 2023,, 2023.
- [14] M. Glauner, K. Schleef, J. Dinwoodie, P. Seipel, S. Cepelak, M. Thiele, Lowest GHG emissions on medium-speed engines by higher epsilon and H2 admission to the fuel gas, , CIMAC Congress 2025,, 2025.
- [15] W. CIMAC, Methane and Formaldehyde Emissions of Gas Engines, CIMAC Position Paper, 2014.
- [16] H.K. Diego Delneri, Enrico Orlandini, Sören Hedvik, Jesper Engström, Vitaly Boguslavsky, Jan Torrkulla, Joost van de Venne, Wassim Klink, Advancing sustainable shipping: strategies for methane slip reduction in Wärtsilä four-stroke engines, , CIMAC Congress 2025,, 2025.
- [17] P. Gélin, M. Primet, Complete oxidation of methane at low temperature over noble metal based catalysts: a review, Appl. Catal. Environ. 39 (2002) 1–37.
- [18] A. Gremminger, J. Pihl, M. Casapu, J.-D. Grunwaldt, T.J. Toops, O. Deutschmann, PGM based catalysts for exhaust-gas after-treatment under typical diesel, gasoline and gas engine conditions with focus on methane and formaldehyde oxidation, Appl. Catal. Environ. 265 (2020) 118571.
- [19] W.R. Schwartz, L.D. Pfefferle, Combustion of methane over palladium-based catalysts: support interactions, J. Phys. Chem. C 116 (2012) 8571–8578.
- [20] N. Sadokhina, F. Ghasempour, X. Auvray, G. Smedler, U. Nylén, M. Olofsson, L. Olsson, An experimental and kinetic modelling study for methane oxidation over Pd-based catalyst: inhibition by water, Catal. Lett. 147 (2017) 2360–2371.
- [21] N. Sadokhina, G. Smedler, U. Nylén, M. Olofsson, L. Olsson, The influence of gas composition on Pd-based catalyst activity in methane oxidation– inhibition and promotion by NO, Appl. Catal. Environ. 200 (2017) 351–360.
- [22] A. Gremminger, P. Lott, M. Merts, M. Casapu, J.-D. Grunwaldt, O. Deutschmann, Sulfur poisoning and regeneration of bimetallic Pd-Pt methane oxidation catalysts, Appl. Catal. Environ. 218 (2017) 833–843.
- [23] P. Lott, M. Eck, D.E. Doronkin, A. Zimina, S. Tischer, R. Popescu, S. Belin, V. Briois, M. Casapu, J.-D. Grunwaldt, Understanding sulfur poisoning of bimetallic Pd-Pt methane oxidation catalysts and their regeneration, Appl. Catal. Environ. 278 (2020) 119244.
- [24] A. Russell, C. Henry, N.W. Currier, A. Yezerets, W.S. Epling, Spatially resolved temperature and gas species concentration changes during C3H6 oxidation over a Pt/Al2O3 catalyst following sulfur exposure, Appl. Catal. Gen. 397 (2011) 272–284.
- [25] M. Theile, Numerical Analysis of a Medium Pressure Gaseous Direct Injector on a Marine Medium Speed Dual-Fuel Engine, Heavy-Duty-, On-und Off-Highway-Motoren 2024: CO2-Neutralität und Nachhaltigkeit, 2025, p. 125.
- [26] M.C. Jespersen, T.Ø.H. Rasmussen, A. Ivarsson, Widening the operation limits of a SI engine running on neat ammonia, Fuel 358 (2024) 130159.
- [27] C. Mavrellos, G. Theotokatos, Numerical investigation of a premixed combustion large marine two-stroke dual fuel engine for optimising engine settings via parametric runs, Energ. Conver. Manage. 160 (2018) 48–59.
- [28] S. Bhargava, N.N. Clark, M.W. Hildebrand, Exhaust Gas Recirculation in a Lean-Burn Natural Gas Engine, SAE Technical Paper, 1998.
- [29] A. Sofianopoulos, D. Assanis, S. Mamalis, Effects of hydrogen addition on automotive lean-burn natural gas engines: critical review, J. Energy Eng. 142 (2016) E4015010.
- [30] M. Xin, H. Gan, Y. Cong, H. Wang, Numerical simulation of methane slip from marine dual-fuel engine based on hydrogen-blended natural gas strategy, Fuel 358 (2024) 130132.
- [31] W. Wei, G. Li, Z. Zhang, Y. Long, Y. Huang, H. Zhang, M. Zhou, Fuel consumption and NOx emissions optimization of a marine natural gas engine with lean-burn and reformed exhaust gas recirculation strategies, Fuel 357 (2024) 129970.
- [32] G. Li, Y. Long, Z. Zhang, J. Liang, X. Zhang, X. Zhang, Z. Wang, Performance and emissions characteristics of a lean-burn marine natural gas engine with the addition of hydrogen-rich reformat, Int. J. Hydrogen Energy 44 (2019) 31544–31556.
- [33] M. Choi, S. Park, Effects of pre-mixed gas composition on combustion characteristics of micro pilot dual fuel (MPDF) in a heavy-duty dual fuel engine, Sci. Rep. 12 (2022) 7871.

- [34] M. Choi, S. Park, Effects of intake air conditions and micro-pilot (MP) injection timing on micro-pilot dual fuel (MPDF) combustion characteristics in a single cylinder optical engine, *Energ. Convers. Manage.* 254 (2022) 115281.
- [35] M.T. Zarrinkolah, V. Hosseini, Methane slip reduction of conventional dual-fuel natural gas diesel engine using direct fuel injection management and alternative combustion modes, *Fuel* 331 (2023) 125775.
- [36] A. Kakoei, M. Mikulski, A. Vasudev, M. Axelsson, J. Hyvönen, M.M. Salahi, A. Mahmoudzadeh Andwari, Start of injection influence on in-cylinder fuel distribution, engine performance and emission characteristic in a RCCI marine engine, *Energies* 17 (2024) 2370.
- [37] M. Merts, S. Derafshzan, J. Hyvönen, M. Richter, M. Lundgren, S. Verhelst, An optical investigation of dual fuel and RCCI pilot ignition in a medium speed engine, *Fuel communications* 9 (2021) 100037.
- [38] M.D. Altinkurt, G. Coskun, M. Tunér, A. Turkcan, A comprehensive investigation of early pilot (e-pilot) mode split injection variations for improving NG-diesel dual-fuel combustion in a medium-speed marine engine: experiments and CFD study, *Case Stud. Therm. Eng.* 68 (2025) 105881.
- [39] M.D. Altinkurt, M. Merts, M. Tuner, A. Turkcan, Effects of split diesel injection strategies on combustion, knocking, cyclic variations and emissions of a natural gas-diesel dual fuel medium speed engine, *Fuel* 347 (2023) 128517.
- [40] I.P.o.C. Change, *Climate Change 2014: Mitigation of Climate Change: Working Group III Contribution to the Fifth Assessment Report of the Intergovernmental Panel on Climate Change*, Cambridge University Press, 2014.
- [41] J.A. Caton, Combustion phasing for maximum efficiency for conventional and high efficiency engines, *Energ. Convers. Manage.* 77 (2014) 564–576.
- [42] J. Liu, Q. Guo, J. Guo, F. Wang, Optimization of a diesel/natural gas dual fuel engine under different diesel substitution ratios, *Fuel* 305 (2021) 121522.
- [43] C.-F. Lee, Y. Pang, H. Wu, K. Nithyanandan, F. Liu, An optical investigation of substitution rates on natural gas/diesel dual-fuel combustion in a diesel engine, *Appl. Energy* 261 (2020) 114455.
- [44] J. Liu, W. Zhao, X. Zhang, Q. Ji, H. Ma, P. Sun, P. Wang, Optimizing combustion and emissions in natural gas/diesel dual-fuel engine with pilot injection strategy, *Thermal Science and Engineering Progress* 48 (2024) 102418.
- [45] F. Perini, D. Sahoo, P.C. Miles, R.D. Reitz, Modeling the ignitability of a pilot injection for a diesel primary reference fuel: impact of injection pressure, ambient temperature and injected mass, *SAE Int. J. Fuels Lubr.* (2014) 48–64, 211773, 7.
- [46] A. Yousefi, H. Guo, M. Birouk, An experimental and numerical study on diesel injection split of a natural gas/diesel dual-fuel engine at a low engine load, *Fuel* 212 (2018) 332–346.