

A Study for Improvement of Performance and Emissions of a Hydrogen Dual Fuel Engine

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Chapter 1 Introduction

1.1 Overview

Anthropogenic activities have contributed to global warming of approximately 1.1 degrees Celsius [1]. The source of global warming is greenhouse gas, which is emitted by the daily activities of humans through various sectors, making it essential to understand these sectors to address the global warming. Fig. 1.1 illustrates global greenhouse gas emissions according to sectors [2]. Most sectors exhibit increasing trends yearly, indicating that global warming could increase further if not restricted, with electricity and heat production contributing the most. This is because the main source of electricity generation is through the combustion of fossil fuels. In 2023, the major source of electricity production is coal which account of about 36% of world total electricity source share, followed by natural gas, which comprise of about 23% of world share [3]. In fact, the fossil fuel is the largest contributor to the global warming, accounting for over 75% of global greenhouse gas emissions and nearly 90% of all carbon dioxide emissions [4], highlighting the importance of switching from fossil fuels to renewable and green energy, to limit the global warming.

Moreover, greenhouse gas emissions from transportations also present a significant contribution to the greenhouse gas emission. In Fig.1.1, transportation is second largest sector to contribute to greenhouse gas emission [2]. This is because despite lower emissions compared electricity and heat generation, the emission has grown steadily owing to the increasing use of vehicles and freight. Worldwide transportation needs are anticipated to surge in the upcoming decades due to population growth, increased wealth, and greater accessibility to automobiles, rail services, and air travel. According to the International Energy Agency (IEA) Energy Technology Perspectives report, global transport, measured in passenger kilometers, is projected to double by 2070. Additionally, the report forecasts a 60% rise in car ownership rates and a threefold increase in both passenger and freight aviation demand within the same time frame until 2070 [5].

Furthermore, in the United States, which currently has the biggest economy in the world, the greenhouse gas was emitted mostly from transportation sectors [6]. Fig. 1.2 shows the greenhouse gas emitted from sectors in the US in 2022[6]. The transportation sector

contributes 28% of total greenhouse gas emissions, which is marginally higher than the electric power sector, underscoring the necessity for decarbonization in transportation. Over 94% of the fuel used in transportation is petroleum based, resulting in direct emissions of greenhouse gas [7].

Internal combustion engine has been a major prime mover in transportation sectors and the main contributor to the greenhouse gas emissions [7]. These engines have played a crucial role in powering automobiles and ships for over a century. With growing concerns about global warming, alternative technologies like electric and hybrid powertrains are gaining traction. In small-scale land transportation, where spark ignition engines have traditionally served as the primary powertrain, the utilization of electric propulsion systems, such as plug-in electric, battery electric and battery hybrid, has emerged as innovative power sources that facilitate emission reduction. This shift towards alternative powertrains has led to significant advancements in battery technology, charging infrastructure, and energy management systems. As a result, electric vehicles are becoming increasingly competitive in terms of range, performance, and cost. However, the usage of electric as propulsion indirectly emits greenhouse gas, and the electric generation sector has been reported to mostly emits the greenhouse gas[3] as the electric is generated by burning the fossil fuel. Therefore, the usage of electric as energy source is not considered fully environmentally friendly, until the generation of electric is done by energy that emits no carbon or carbon neutral energy.

Meanwhile, within heavy duty operations where compression ignition engines have predominantly served as primary energy converters, electrification presents significant challenges [8]. This is primarily due to the limited availability of electric power sources and their current infeasibility for ultra-long distance and extended operations. The compression-ignition engine, i.e., diesel engine, has maintained its dominance in maritime and industrial sectors despite the passage of more than 130 years since the first diesel engine was developed. This is attributed to its high efficiency and ability to operate under high load conditions. Additionally, the easy access to diesel fuel and the well-established network for its distribution support the ongoing use of diesel engines in heavy duty operations. However, diesel engines present serious issue to the emission problems as according to the World Economic Forum, although heavy-duty trucks which is empowered by diesel engine,

constitute merely 1% of all vehicles on the road, they account for 25% of global emissions from road transportation [9]. To tackle this emission problems and limit the global warming, replacing the fossil fuel in diesel engine with alternative resources is a crucial and prompt solution.

1.2 Global carbon neutrality strategy and emission regulations

The global warming issues led to the adoption of the Paris Agreement by 196 Parties at COP21 to limit global warming to below 2 degrees Celsius, preferably 1.5 degrees compared to pre-industrial levels [10]. The adoption of the Paris Agreement in 2015 has resulted in nations announcing their intentions to reduce the greenhouse gas emission through carbon neutrality.

The United States has established its objective to achieve carbon neutrality by 2050 [11]. This goal is to be accomplished through the implementation of the Long Term Strategy [12]. This strategy encompasses decarbonizing the electric sectors, electrifying end uses across transportation, buildings, and industries, and enhancing energy efficiency. The utilization of fossil fuels in electric sectors is gradually phased out in favor of wind and solar energy. In the transportation sector, which is the largest carbon emitter in the United States [6], electrification is prioritized. However, sustainable biofuels and hydrogen produced with low or zero greenhouse gas emission i.e. clean hydrogen are preferred alternatives for modes of transport that are challenging to electrify, such as aviation, marine, and heavy-duty vehicles. In the industrial sector, incentives will be provided for research and development of clean hydrogen to power certain industrial facilities [12].

Japan, home to major producers of light-duty and heavy-duty vehicles globally [13], aims to achieve net-zero carbon emissions by 2050 [14]. Carbon emissions in Japan have been decreasing significantly, with a reduction of 15% in 2022 compared to the year 2000 [15], indicating that Japan is progressing towards carbon neutrality by 2050. As carbon neutrality by 2050 cannot be realized through conventional efforts, it is imperative to significantly accelerate efforts toward structural changes in the energy and industrial sectors and undertake substantial investment for innovation [16]. The Ministry of Economy, Trade and Industry of Japan (METI) has introduced a green growth strategy to address

this matter. The strategy incorporates five cross-sectoral policy tools and detailed action plans for 14 specific sectors [16]. The electricity and heat production sector has been the major carbon emitter in Japan, comprising approximately 48% of emissions in 2022, followed by transportation, which accounts for about 19% [15]. In the electric sector in Japan, decarbonization of electricity is set to be achieved by replacing fossil fuels with renewable energy, nuclear energy, and hydrogen power. In the transport sector, electrification is preferred, with hydrogen and biofuel preferred in hard-to-electrify transportation [16]. Based on the statement, hydrogen has received increased focus as an alternative energy source in Japan. Furthermore, the Sixth Strategic Energy Plan, which was also introduced to realize net-zero carbon emissions by 2050, recharacterized hydrogen and ammonia as future energy fuels and carriers that also play a role in power supply, rather than merely new energy sources [17]. Hydrogen is expected to serve as a source in power generation, mobile power source through fuel cells, and as a heat power source [17]. The focus on hydrogen as an energy source in Japan dated back to the introduction of the Basic Hydrogen Strategy in 2017. The main objectives of this strategy are to expand the usage of hydrogen up to 3 million tons/year by 2030, and approximately 20 million tons/year by 2050, as well as to reduce the supply cost of hydrogen and construct a domestic hydrogen supply in Japan [17].

The focus on hydrogen energy from various countries has highlighted the importance of hydrogen in the future. This encouraged the research and development on hydrogen energy utilization worldwide.

Nevertheless, decarbonization presents significant challenges, including nitrogen oxides (NO_x) emissions from combustion systems. This arises because efforts to lower CO, such as using alternative fuels, can possibly lead to higher combustion temperatures, which in turn promote the formation of NO_x . Similar to any high temperature combustion, hydrogen, despite its potential as a clean fuel, generates NO_x emissions when it burns. Moreover, hydrogen could emit higher NO_x emission compared to fossil fuel during combustion processes, due to its higher heating value [18]. These present challenges to achieve low emission from hydrogen combustion. Furthermore, to prevent the impact of emissions to the environment such as acid rains and air pollutions, and to the human health due to the emission from combustions, responsible bodies have introduced several regulations to limit

emissions, mostly from internal combustion engines.

In Europe, the emission from vehicles has been regulated since the introduction of EURO 1 in 1992. The aim is to improve air quality by reducing air pollution by introducing more severe standards [19]. NO_x emission was limited as a total with HC emission until EURO 3, which requires a separate measurement [19]. This regulation has been increasingly stringent. EURO 4 was implemented in 2005, and to comply with Euro 4 standards, diesel vehicles must not release more than 0.25 g/km of NO_x gases and 0.025 g/km of PM, while gasoline-powered vehicles are limited to 0.08 g/km of NO_x emissions. EURO 5 was implemented in 2009, and to comply with Euro 5 standards, diesel vehicles can only emit no more than 0.18 g/km of NO_x emissions, and gasoline-powered vehicles are limited to 0.06 g/km of NO_x emissions[20]. The Euro 5 standard requires PM to be reduced to a value of no more than 0.005 g/km of PM emission, which essentially forced the widespread adoption of diesel particle filters (DPF). Recent regulation in action is EURO 6 which was introduced in 2014. EURO 6 has limit significantly the NO_x emission from diesel engines by 56% compared to EURO 5, which has become the trigger to the widespread of NO_x exhaust aftertreatment system [21].

In land transport in Japan, to address NO_x pollution from existing vehicles in densely populated urban areas, Japan's Ministry of the Environment implemented "The Motor Vehicle NO_x Law" in 1992. This legislation, formally known as the "Law Concerning Special Measures to Reduce the Total Amount of Nitrogen Oxides Emitted from Motor Vehicles in Specified Areas," targeted 196 cities, wards, towns and villages across Tokyo, Saitama, Kanagawa, Osaka, and Hyogo Prefectures[22]. These areas were identified as having significant air pollution due to vehicle-emitted nitrogen oxides. The law mandated various measures to control NO_x emissions from vehicles in use, including the enforcement of emission standards for certain vehicle types. In June 2001, the regulation underwent modifications to strengthen the current NO_x standards and incorporate PM control measures. The revised legislation is officially titled "Law Concerning Special Measures to Reduce the Total Amount of Nitrogen Oxides and Particulate Matter Emitted from Motor Vehicles in Specified Areas," commonly referred to as the Automotive NO_x and PM Law[23]. In new vehicles, the Ministry of Environment has initiated the regulation of automobile exhaust

emissions as early as 1966, with progressively more stringent standards implemented over subsequent years[24]. For gasoline vehicles, regulations introduced in 2000, 2001, and 2002 (referred to as the "New Short-Term Regulations") strengthened emission standards for CO, HC, and NO_x and mandated onboard diagnostic (OBD) systems. For diesel vehicles, similar regulations implemented in 2002, 2003, and 2004 enforced stricter controls on NO_x and particulate matter (PM). In 2005, both gasoline and diesel vehicles underwent revised emission testing methods, leading to the implementation of the 2005 regulations (known as the "New Long-Term Regulations"). This regulation was also recognized as highly stringent for both light and heavy vehicles, with heavy-duty emission standards of NO_x = 2 g/kWh and PM = 0.027 g/kWh being the most stringent diesel emission regulation globally [25]. Subsequently, in 2008, the 2009 regulations (known as "Post New Long-Term Regulations") were introduced. Effective 2009, the emission limits were further tightened to a level comparable to Euro V requirements through these regulations. In 2015, emission standards for heavy-duty diesel vehicles and motorcycles were further tightened, aiming to achieve greater reductions in automobile exhaust emissions with stringency comparable to the Euro VI standards[26].

In sea transportations, the International Maritime Organization (IMO) has been responsible for NO_x regulations. The regulations were first introduced through the adoption of MARPOL Annex VI, which entered into force in 2005. MARPOL Annex VI establishes limits on sulfur oxide and nitrogen oxide emissions from ship exhausts and prohibits deliberate emissions of ozone-depleting substances. This introduced the regulations on NO_x emissions from ships in 2005 namely TIER I. This legislation governs the NO_x emissions from engines on oceangoing vessels starting January 1, 2000. Subsequently, a revised MARPOL Annex VI, adopted in October 2008, introduced Tier II regulations that mandated a 15% to 22% decrease in NO_x emissions compared to the initial regulations. This amendment became effective on July 1, 2010, and was applicable to engines installed on ships with keel-laying dates of January 1, 2011, or later. The protocol was further revised by the introduction of Tier III regulations. These apply when any ship with a keel-laying date after January 1, 2016, navigates an ECA (Emission Control Area) within 200 nautical miles. Tier III regulations require a further 80% NO_x reduction compared to Tier I. Fig. 1.3 shows

the value and comparison of the three regulations[27].

This increasingly stringent emission regulations and carbon neutral movements over the concern of environmental problems from emissions has propelled towards research on developing new technologies and ideas as solutions for this matter.

1.3 Introduction to hydrogen energy as carbon neutral strategy

Among the carbon neutral fuels available, hydrogen is a recyclable and long term renewable option [28, 29]. Hydrogen is an odorless, invisible gas, and the most abundant gas on Earth. It is also the lightest element consist of 75% from all matter on Earth. However, hydrogen is not readily available in nature in unbound molecular form, but in compounds with other elements [30]. There is a need to produce hydrogen through certain methods. Hydrogen could be produced using steam reforming, gasification, water electrolysis, etc. Most of the hydrogen is produced through steam reforming. Hydrogen could be produced using various sources, but as the common hydrogen production methods nowadays is through steam reforming, the common source for hydrogen production nowadays is natural gas [31]. The hydrogen produced using natural gas is considered as grey hydrogen. While the hydrogen is not green enough as the hydrogen is produced using fossil fuel and emits carbon, efforts are being done to support and increase the production of low carbon hydrogen. For example, China is among the nations actively pursuing renewable energy for hydrogen production. The key to achieving large-scale green hydrogen production in the country lies in reducing the costs associated with renewable energy electricity and electrolyzers[32]. Moreover, the United States have planned the clean hydrogen strategy and roadmap as a strategy to accelerate progress towards carbon neutrality. A combination of methods for hydrogen production is expected to be utilized through this strategy at least until 2050. These methods include extracting hydrogen from water through electrolysis, generating hydrogen from fossil fuels while implementing carbon capture and storage technologies, and deriving hydrogen from biomass and waste materials [33].

To support the carbon neutrality and the expansion of hydrogen energy utilization through Basic Hydrogen Strategy [17], the utilization of hydrogen through various sectors

is expected. In electricity generation, fuel cell is proposed as the generation system. In a fuel cell, hydrogen fuel interacts with oxygen, generating electricity and water through an electrochemical reaction [31]. In transportation sector, the hydrogen could be utilized as energy carrier in EV through fuel cell system (FCEV). FCEVs are more efficient than conventional internal combustion vehicle and the emission from FCEVs are water and warm air [34]. Vehicles powered by fuel cells present notable benefits, such as extended driving distances and quicker refueling periods in comparison to electric vehicles that rely on batteries. Nevertheless, widespread commercial use remains restricted due to various obstacles, including elevated costs, insufficient hydrogen fueling stations, and the requirement for further improvements in fuel cell technology [35].

Hydrogen also possible to be utilized directly in transport sector through combustion. It could be applied as fuel in turbine or internal combustion engines. However, the hydrogen possesses several challenges as combustion fuel in internal combustion engines due to its unique combustion characteristics. Hydrogen has a high ignition temperature, but a low ignition energy [36]. This caused some abnormal combustion in engines. Moreover, the high-speed flame propagation could enhance efficiency through reduced combustion duration, but the short quenching distance of hydrogen can lead to increased heat transfer from the burning gas to the combustion chamber walls [37]. To asses the effectiveness of hydrogen utilization in internal combustion engines, research is being actively conducted to clarify the combustion behavior through various parameters, as well as its effect on performance and emission of an internal combustion engine.

1.3.1 Hydrogen combustion in internal combustion engines

Hydrogen could be converted to energy in internal combustion engines using several methods. Firstly, in spark ignition (SI) engines, hydrogen could be ignited and burned using spark ignition method, similar to gasoline. Moreover, hydrogen fuel can be burned using spark ignition at equivalence ratios below the lean flammability limit of gasoline/air mixtures, offering significant advantages [38, 39]. Most of previous research in the past has focused application of hydrogen in SI engines [18]. Several research reported that utilizing hydrogen in SI engines offer significant advantages in performance and

emissions, despite presenting certain obstacles [40]. Hydrogen-fueled SI engines typically exhibit higher thermal efficiency, particularly in lean mixtures and low equivalence ratios, due to the rapid burning characteristics of hydrogen and the wide flammability limits [40–42]. However, power loss may occur at low speeds; at high speeds, hydrogen competes favorably with gasoline due to its fast combustion properties [41]. In terms of emissions, hydrogen substantially decreases carbon monoxide (CO), hydrocarbon (HC) and carbon dioxide (CO₂), since hydrogen combustion does not involve carbon [43, 44]. Nevertheless, the elevated combustion temperatures associated with hydrogen tend to increase nitrogen oxide (NO_x) emissions, which can be addressed by operating in lean conditions [43, 44]. However, SI engines which are actively used in small scale land transport are likely to be replaced by electric motor, due to the decision of most governments on electrification on electrification-capable transports [12, 16].

Hydrogen utilization is primarily focused on modes of transportation and machinery that present challenges for electrification [12, 16], including ultra long distance and heavy-duty transports, maritime transports, and industrial machineries. The major energy converter for these applications are the compression ignition (CI) engines. In CI engines, utilizing hydrogen as an energy source presents challenges due to its high ignition temperature, which is significantly higher than that of diesel [45]. This characteristic renders hydrogen unsuitable as a sole fuel in CI engines, as its limited ignition ability [45, 46] can result in reduced control over the ignition process, leading to abnormal combustion and restricting the operable conditions of the engine. Consequently, controlling hydrogen ignition is critical for effective application. To control the hydrogen ignition, the dual-fuel combustion method has shown promise in providing fuel flexibility and improving engine performance. In this approach, hydrogen is ignited using diesel fuel or other liquid fuels as the pilot ignition source. This approach has gained considerable interest because it enables the ignition of high-ignition-temperature gaseous fuels, which suits hydrogen. Several studies have investigated hydrogen dual-fuel combustion in CI engines. For example, Szwaja and Rogaliski [47] examined hydrogen-diesel dual-fuel combustion in CI engines and found that incorporating hydrogen as an additional fuel enhanced engine performance. This improvement was attributed to high-speed flame propagation of hydrogen, which

increased the degree of constant volume combustion.

Zhou et al. [48] have investigated the effects of hydrogen addition on combustion and engine performance in a diesel engine. Hydrogen concentration was manipulated by supplying in the range of 10-40% of the total fuel energy. It was reported that improvements in engine performance and reductions in emissions of CO and particulate matters (PM) could be achieved from medium to high loads by adding hydrogen. Hydrogen addition also resulted in reduction of most of the unregulated emissions investigated.

Deb et al. [49] examined the hydrogen-diesel dual-fuel combustion in a direct injection diesel engine. They varied the hydrogen energy content from 0% to 42% while maintaining constant engine speed and power. Results indicate that incorporating hydrogen leads to improved brake thermal efficiency and decreased brake specific energy consumption. Additionally, emissions of CO, CO₂, and smoke were reduced, though NO_x emissions increased. The investigation demonstrates that higher hydrogen concentrations result in elevated peak in-cylinder pressure and heat release rates, thereby enhancing combustion efficiency. From this, it could be concluded that hydrogen addition in smaller quantity in diesel operation could improve efficiency. The reduction of CO, and particulate matters indicated the engine emission is improved except for NO_x emissions.

Hydrogen-diesel dual fuel combustion with hydrogen as primary fuel has been investigated by Tsujimura and Suzuki [50]. The performance and emissions characteristics under varying conditions in a single-cylinder diesel engine were examined. The experiments utilized hydrogen energy content up to a maximum fraction of 90%, demonstrating that higher hydrogen fractions enhance thermal efficiency and reduce carbon emissions. However, at higher loads and hydrogen fractions exceeds approximately 50%, the probability of abnormal combustion, such as pre-ignition, increases, potentially limiting operable conditions [50]. The experiment mainly focused on hydrogen concentration, while factors involving pilot fuel was not clarified. In dual fuel operation generally, gaseous fuel ignition is controlled by pilot fuel, controlling pilot fuel characteristics could influence the combustion behavior. For instance, adjusting the composition of pilot fuel can improve performance and emissions in pilot ignited direct injection natural gas engines. It was reported that adding n-butanol to diesel pilot fuel improve thermal efficiency by up to 6.5% at low and medium

engine loads due to extended ignition delay which resulted in reduced combustion duration [51]. The quantity of pilot fuel also plays a crucial role. Increasing pilot fuel amount at low loads in dual fuel combustion can improve efficiency and reduce emissions, while excessive pilot fuel at high loads may lead to early knocking [52]. Furthermore, the reported abnormal combustion in [50] could be improved, and the operation limit could be further widened through pilot fuel improvements. To ensure hydrogen as sustainable and relevant technology, further improvements on the efficiency and widens the operation range in hydrogen-dual fuel engine is necessary, hence the knowledge regarding the effects of pilot fuel parameters onto combustion behaviors and engine performance is required.

Moreover, as the global community continues to seek sustainable energy solutions, the synergy between hydrogen and biofuels could play a crucial role as a future sustainable fuel. The integration of hydrogen and biofuel could present a promising avenue for sped up carbon neutral achievements. Biofuels have emerged as a prominent alternative fuel for diesel engines, owing to its potential for net zero carbon emissions and renewable sourcing. To utilize the benefits of biofuel renewability and its properties in hydrogen dual fuel, Hydrotreated Vegetable Oil (HVO) has emerged as a viable candidate for pilot fuel. Its significant properties, including a high cetane number, exhibit superior ignition characteristics compared to conventional diesel. Furthermore, unlike ester-type biodiesel fuels, HVO does not exhibit negative effects particular to biodiesel such as oil aging [53] nor does it lead to increased NO_x emissions or deposit buildup [54]. Moreover, the absence of aromatics in HVO has significantly reduced smoke emissions [54, 55].

In neat HVO operation in diesel engine, HVO has demonstrated excellent combustion behaviors. Bortel et al. [54] in their study, have elucidated the effect of HVO and its blends with diesel fuel combustion on the performance and emission of a passenger car size diesel engine. They reported improved power, which slightly increases the efficiency with neat HVO operation. Moreover, PM significantly decreases, with a maximum reduction of 80% with neat HVO, but NO_x emission only demonstrated a slight reduction. Smigins et al [55] compared the performance and emissions of a diesel engine using Hydrotreated Vegetable Oil (HVO), fossil diesel, and a blend of the two (HVO5, with 5% HVO). They reported that HVO improved engine power and torque, reduced specific fuel consumption by 0.86%,

and lowered hydrocarbon emissions by 60% compared to fossil diesel. However, it slightly increased NO_x emissions by 3%.

Regarding previous research on hydrogen-HVO dual fuel in diesel engine, dual fuel operation with only small quantity of hydrogen added to the air during HVO as primary fuel operation, has been investigated by Pinto et al. [56]. Their study examined the effect of hydrogen addition, up to 28% of energy fraction, on an engine operated with HVO and farnesane biodiesel. Emission results indicated that neat HVO produced lower NO_x, particulate matter (PM), CO, and CO₂ emissions compared to diesel. With hydrogen enrichment, a significant reduction in PM, CO, and CO₂ emissions was observed for HVO, although NO_x emissions increased by up to 62%. Hydrogen enrichment marginally reduced brake thermal efficiency (BTE); however, both HVO single- and dual-fuel modes demonstrated higher BTE and lower greenhouse gas emissions compared to diesel-hydrogen dual-fuel operation, suggesting that HVO with hydrogen enrichment is a promising alternative to diesel in terms of both efficiency and emissions reduction [56]. However, the emissions and performance of an HVO-hydrogen dual-fuel engine were elucidated with hydrogen as a secondary fuel (maximum hydrogen energy fraction of only approximately 28%) [56]. However, to effectively utilize hydrogen energy and achieve carbon neutrality, hydrogen is expected to be used as the primary fuel source. Moreover, previous studies have reported varying results for ignition delay behavior as the hydrogen fraction is manipulated in hydrogen dual-fuel combustion [47–50]. Pinto et al. suggested that hydrogen enrichment may either reduce or increase ignition delay, depending on the hydrogen fraction and temperature [56]. As Pinto et al. employed a low hydrogen concentration with HVO [56], the knowledge of the ignition delay of the hydrogen-HVO dual fuel with a high hydrogen concentration could not be obtained from the reported results. As combustion may manifest differently under high hydrogen concentrations, investigating the combustion behavior of hydrogen-HVO dual fuel with high hydrogen concentration is necessary. Moreover, Pinto et al. elucidated the increase in ignition delay with increasing hydrogen, and the peak of heat release rate increases. However, the thermal efficiency of the brake is reduced due to a reduction in the maximum pressure [56]. Karagoz suggested that pilot fuel injection should be modified to reduce this inefficient behavior under H₂ enrichment [53]. To improve the

efficiency of hydrogen HVO dual fuel combustion, determining operating standards with optimum pilot injection strategies is necessary.

The effect of combustion behavior of hydrogen dual fuel can be assessed through detailed studies of engine emissions and performance. Studies on engine performance, mostly regarding hydrogen-diesel dual fuel have been well established[47–50]; however, the systematic correlation with combustion parameters derived from optical diagnosis in internal combustion engine has not been thoroughly investigated. To achieve a high efficiency and safe operation, developing a concrete strategy for controlling the combustion process in dual fuel is necessary. This involved understanding the complex relationship between combustion behavior and engine performance. However, dual fuel combustion is a complicated process, especially in pilot-ignited systems, which involve a complex transition between pilot fuel autoignition and the premixed-like phase of combustion [57]. This complexity increases when using hydrogen as a fuel, given its tendency to autoignition because of its low ignition energy. The distribution of the pilot fuel can significantly affect autoignition, likely due to variations in the local equivalence ratio. To gain a thorough understanding of engine performance behavior, it is important to clarify key aspects of the combustion process, including pilot fuel ignition location, the progression of the pilot flame through the hydrogen-air premixture, and the spatial distribution of the pilot fuel spray.

For example, in previously stated reports [47–50], utilization of hydrogen combustion in diesel engine could exhibit improved efficiency due to high-speed flame propagation of hydrogen which increase the degree of constant volume combustion. However, thermal efficiency may also be adversely affected due to the higher flame speed and short quenching distance of hydrogen, as well as combustion occurring near the piston and cylinder wall [37]. Combustion visualization can elucidate this phenomenon through the observation of combustion location and velocity, whereas combustion close to piston wall and squish could indicate more energy loss to cooling. Moreover, the fast flame propagation close to TDC and reduced combustion duration could exhibit improved thermal efficiency through enhanced degree of constant volume combustion. Utilizing OH* chemiluminescence, the luminescence could indicate the high-temperature region due to combustion, thereby enabling visualization of the combustion process.

In hydrogen dual fuel combustion visualization field, only few reports could be found. Recently, Rajasegar et al. [58] have investigated the impact of adding hydrogen on the ignition and combustion behavior of a pilot diesel-spray in the context of dual-fuel combustion for lean-premixed hydrogen-internal combustion engines. Through experiments using a heavy-duty, single-cylinder, optical engine and zero-dimensional chemical-kinetics simulations, the researchers studied the spatial and temporal evolution of two-stage autoignition of n-heptane, injected into a premixed charge of hydrogen and air. The study also included high-speed cool-flame and OH* chemiluminescence imaging techniques to understand the ignition behavior and provided a comparative analysis with natural gas dual-fuel combustion [58]. However, the impact of hydrogen and NG concentration has been the focus. OH* chemiluminescence results indicated that introducing the hydrogen-air mixture resulted in fewer flame kernels, and flame formation was not strictly confined to the pilot fuel jet due to the presence of premixed H₂-air outside the periphery of the pilot fuel jet. Increasing the concentration enhanced the intensity of this behavior and increased ignition delay, which consequently increased the entrainment of ambient air into the pilot fuel, jointly affecting the location of ignition and subsequent combustion. Through chemical kinetics simulation results, Rajasegar et al. suggested that longer pilots that would ensure the availability of fuel-rich zones would be required to achieve reliable ignition in hydrogen dual fuel combustion[58].

Dual fuel combustion allows better control of the combustion process[59, 60]. While the ignition process of the hydrogen-dual fuel is mostly controlled by pilot fuel, there is a need to find effective pilot fuel injection strategies to enhance thermal efficiency. This could be achieved by comparing the engine performance results and the combustion process. Therefore, visualization of combustion process is necessary. Furthermore, visualization could confirm the combustion area and progression, which could help to identify a high likelihood of abnormal combustion, such as knocking, and facilitate its prevention.

Through the visualization of the combustion process, hydrogen dual-fuel engine performance can be optimized by developing an effective injection strategy through the visualization of combustion process. This can enhance thermal efficiency and minimize energy losses. However, regarding emission, despite the low carbon emission achieved

through hydrogen combustion, one problem remains. Similar to any high temperature combustion behavior, the combustion of hydrogen-dual fuel emits NO_x . Moreover, hydrogen dual-fuel combustion emits relatively high NO_x at high hydrogen concentrations, primarily due to hydrogen high flame temperature and energy content[18]. To solve the problems regarding emissions, a few strategies have been developed, and some of them are currently employed in existing transports.

1.4 Strategies to achieve low emissions in internal combustion engines

To reduce NO_x emission, which is emitted due to high temperature combustion, following solutions exist. Among the most promising technologies for managing NO_x emissions, particularly in diesel-powered heavy-duty vehicles, is Selective Catalytic Reduction (SCR). In this technology, urea solution is introduced into the exhaust gas stream. At high temperatures, this solution undergoes hydrolysis. The resulting ammonia then interacts with NO_x in the presence of a catalyst. This reaction produces harmless nitrogen and water, which are then released into the atmosphere [61].

Another viable method for controlling NO_x emissions is Exhaust Gas Recirculation (EGR). The fundamental principle of EGR involves redirecting a segment of the exhaust gases back into the combustion chamber. This technique decreases the flame temperature and lowers the oxygen levels within the combustion chamber, thereby inhibiting the production of NO_x [62, 63]. However, NO_x emission exhibits a trade-off relationship with carbon emission, particularly PM, and a similar phenomenon occurs when utilizing EGR. The low oxygen concentration when using EGR increase the tendency of increased PM emissions[64]. Moreover, EGR tends to decrease engine thermal efficiency due to reduced indicated work and increased pumping work, particularly at higher loads and low speeds[65].

To reduce PM emissions, the proven technology is the diesel particulate filter(DPF). DPF is an aftertreatment device designed to reduce PM emissions output from diesel engine. It works by trapping soot particles from the exhaust gas as the soot passes through the filter channels. As the exhaust flows through the porous walls of the filter, soot is deposited, forming a soot cake that enhances filtration efficiency over time [66].

However, above mentioned technologies necessitate modifications to the engine. While EGR has been reported to decrease thermal efficiency under certain conditions[65], Selective Catalytic Reduction (SCR) has been observed to degrade over time [67, 68]. The increasingly stringent emission regulations are rendering both aforementioned technologies less effective, and periodic hardware modifications are required to meet these regulations. To address this problem, an alternative approach has gained attention. Reducing emissions through fuel modifications does not require hardware alterations and could result in significant cost and time savings. Water-emulsified fuels have been gaining attractions as solutions, as it is reported to simultaneously reduce both NO_x and soot emissions[69, 70]. The incorporation of hydrogen with emulsified fuel through the emulsification of the pilot fuel could potentially achieve the objectives of carbon neutrality and low emissions.

1.4.1 Introduction to water emulsified fuel as low emission strategy

Water-emulsified fuel is an approach to introduce water into the combustion chamber through the fuel injection system by mixing water with the fuel itself. Conventional fuel is combined with water through an emulsification process, ensuring a thermodynamically stable fuel. When injected into the combustion chamber, the water in the fuel absorbs heat through the latent heat of evaporation, resulting in a lower temperature inside the combustion chamber. This consequently restricts the formation of NO_x .

The usage of water emulsified fuel in engines has been studied by researchers for over two decades[71]. Extensive research has been conducted on emulsified fuels in diesel engines, particularly exploring the impact of varying water content. Abu-Zaid [72] investigated the influence of water content in emulsified fuels on the performance of a single-cylinder, direct-injection engine. Water content in the emulsified fuels ranged from 5vol% to 20vol% with 5vol% increments. Higher water content resulted in increased brake thermal efficiency. A significant average thermal efficiency increases of approximately 3.5% was achieved with 20vol% emulsified fuel compared to diesel across the engine speed range investigated. Increasing the water content also led to a reduction in brake-specific fuel consumption and exhaust gas temperature. Similarly, Alahmer et al. [73] examined the performance and emissions of a water-cooled, four-cylinder diesel engine fueled with

emulsified diesel containing water content ranging from 5vol% to 30vol%. The engine speed was varied from 1000 rpm to 3000 rpm. It was reported that usage of emulsified fuels resulted in reduced NO_x emissions and maintained or improved thermal efficiency under certain conditions compared to diesel. The highest average thermal efficiency increase was observed at a 5% water content due to increased brake power. NO_x emissions decreased with increasing water content at all engine speeds, and exhaust gas temperature decreased as well.

Fahd et al. [74] conducted an experimental study on the emissions and performance of a 4-cylinder turbocharged diesel engine fueled with 10% water-emulsified diesel under different load conditions. They observed improved NO emissions throughout all load conditions, but higher CO emissions over diesel at low load conditions due to lower combustion temperature. The thermal efficiency of emulsified fuel was lower than that of diesel due to its lower calorific value.

While these reports provide insights into engine emissions and performance with emulsified fuel at varying water content, there have been inconsistencies in the results, particularly concerning thermal efficiency improvements [75]. Different studies [72–74, 76] report varying optimal water contents for efficiency enhancements. Factors such as engine load condition, engine speed, and base fuel type may contribute to these inconsistencies. One crucial factor is the water droplet size in emulsified fuel, which has been actively studied in spray flame and hot plate experiments, demonstrating its influence on micro explosion occurrence and intensity [77, 78]. However, the influence of water droplet size on emulsified fuels in diesel engine studies remains limited. The occurrence of micro explosions in diesel engines is yet to be definitively proven, with differing views [79, 80] but the water droplet size can affect the degree of water dispersion in emulsified fuels. Therefore, there is a need to investigate the influence of different water droplet sizes in emulsified fuels with various water contents to elucidate their effects on diesel engines. Furthermore, the optimal properties of emulsified fuel (water content and droplet size) that improve both engine emissions and performance have not yet been reported. Hence, there is a need to clarify the optimal water content and droplet diameter in emulsified fuel that improve engine emissions and performance.

1.5 Research objectives and outline of this thesis

To achieve reliable net zero carbon emissions and low emission diesel engine operation through hydrogen dual fuel, ensuring high efficiency and low emission operation from hydrogen dual fuel engine is necessary. Therefore, the purpose of this study is to clarify the operation strategies that improve performance and emission of a hydrogen dual fuel engine. Previous research on hydrogen addition in HVO-fueled engines has elucidated that engine performance and emissions differ compared to when hydrogen is added to diesel-fueled engines[56], although the difference in combustion processes remained unclear. To ensure optimal operation of hydrogen-HVO dual fuel engines, it is imperative to expand the knowledge base regarding the effects of pilot fuel on engine performance and combustion processes in hydrogen dual fuel mode. Moreover, to ensure highly efficient hydrogen dual fuel operation, it is necessary to acquire knowledge regarding flame position and flame propagation in hydrogen-air surroundings, thereby clarifying the optimal combustion strategies which improves thermal efficiency. Furthermore, while previous operations utilized hydrogen as a secondary fuel, to fully harness hydrogen energy in diesel engines, hydrogen is anticipated to serve as the primary fuel in dual fuel engines. As low hydrogen concentrations were employed in previous studies[47–49], it is hypothesized that different engine performance and emission characteristics could be observed when higher hydrogen concentrations are utilized, due to the increased probability of variations in ignition delay and combustion behaviors.

For this purpose, hydrogen-HVO dual fuel operations are initially conducted using a single-cylinder diesel engine to elucidate the effects of pilot fuel and pilot injection strategies onto engine performance and emission characteristics. Subsequently, to determine effective hydrogen intake standards and injection strategies in hydrogen dual fuel, the combustion process is studied. To achieve this, the hydrogen-diesel and hydrogen-HVO dual fuel combustion processes are visualized when the injection parameters are altered in a Rapid Compression and Expansion Machine (RCEM).

To achieve low emission, the problems regarding NO_x emissions from hydrogen dual fuel engines should be addressed. The water emulsified fuel is assessed as a possible solution

as it reduces NO_x emission without compromising soot emissions. However, to employ water emulsified fuel, determining the optimum emulsification strategy such as water content and water droplet diameter is necessary to ensure low emission operation without deteriorating engine performance. The effect of water content in emulsified fuel is actively reported, however there are contradicting reports regarding the optimum water content that provide improved efficiency and emission. Water droplet diameter is among the factors that could influence this matter, but the effect of water droplet diameter is not yet well documented. Therefore, an engine emission and performance assessment test of emulsified fuels was conducted to identify the water content and water droplet diameter that can improve both emission and engine performance.

In this dissertation, Chapter 1 discusses the background, motivations and purpose of this study. Chapter 2 examines the engine performance and emission in hydrogen-HVO dual fuel mode and hydrogen-diesel dual fuel mode. Chapter 3 analyzes the combustion processes in hydrogen-diesel dual fuel mode, and the hydrogen-HVO dual fuel combustion processes. Chapter 4 will examine the engine performance and emission from water emulsified fuels while manipulating water contents and droplet diameter. Lastly, Chapter 5 will summarize the findings from this study.

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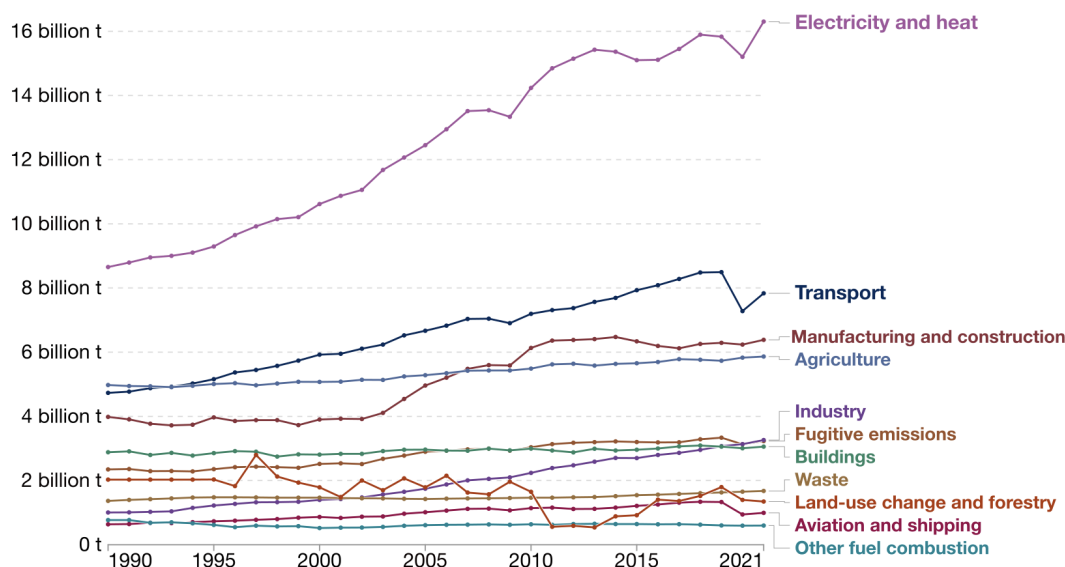
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Figures

Greenhouse gas emissions by sector, World

Our World
in Data

Greenhouse gas emissions¹ are measured in tonnes of carbon dioxide-equivalents² over a 100-year timescale.



Data source: Climate Watch (2024)

OurWorldinData.org/co2-and-greenhouse-gas-emissions | CC BY

Note: Land-use change emissions can be negative.

1.Greenhouse gas emissions: A greenhouse gas (GHG) is a gas that causes the atmosphere to warm by absorbing and emitting radiant energy. Greenhouse gases absorb radiation that is radiated by Earth, preventing this heat from escaping to space. Carbon dioxide (CO₂) is the most well-known greenhouse gas, but there are others including methane, nitrous oxide, and in fact, water vapor. Human-made emissions of greenhouse gases from fossil fuels, industry, and agriculture are the leading cause of global climate change. Greenhouse gas emissions measure the total amount of all greenhouse gases that are emitted. These are often quantified in carbon dioxide equivalents (CO₂eq) which take account of the amount of warming that each molecule of different gases creates.

2. Carbon dioxide equivalents (CO₂eq): Carbon dioxide is the most important greenhouse gas, but not the only one. To capture all greenhouse gas emissions, researchers express them in "carbon dioxide equivalents" (CO₂eq). This takes all greenhouse gases into account, not just CO₂. To express all greenhouse gases in carbon dioxide equivalents (CO₂eq), each one is weighted by its global warming potential (GWP) value. GWP measures the amount of warming a gas creates compared to CO₂. CO₂ is given a GWP value of one. If a gas had a GWP of 10 then one kilogram of that gas would generate ten times the warming effect as one kilogram of CO₂. Carbon dioxide equivalents are calculated for each gas by multiplying the mass of emissions of a specific greenhouse gas by its GWP factor. This warming can be stated over different timescales. To calculate CO₂eq over 100 years, we'd multiply each gas by its GWP over a 100-year timescale (GWP100). Total greenhouse gas emissions – measured in CO₂eq – are then calculated by summing each gas' CO₂eq value.

Figure 1.1: Global greenhouse gas emissions by sector (year 1990-year 2021) [2].

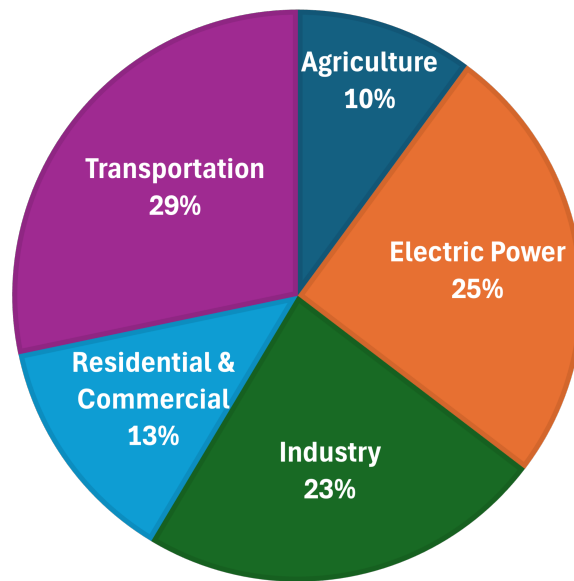


Figure 1.2: Greenhouse gas emitted from sectors in the US in 2022 [6].

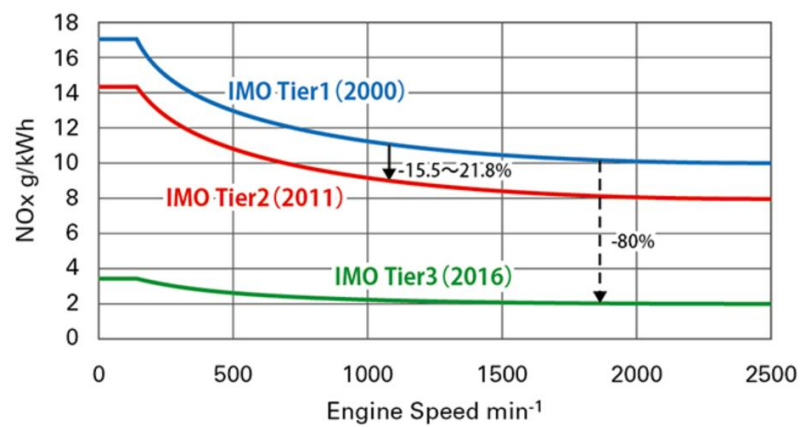


Figure 1.3: NO_x regulations by International Maritime Organizations [27].

Chapter 2 Performance and emissions of a hydrogen dual fuel engine using diesel and HVO as pilot fuels

2.1 Introduction

Studies on hydrogen-diesel dual-fuel engines have demonstrated improved thermal efficiency and reductions in CO, CO₂ and smoke emissions. However, these benefits come at the cost of increased NO_x emissions with higher hydrogen concentrations [1–3]. In contrast, research on hydrogen-HVO dual-fuel combustion remains limited, resulting in gaps in understanding its performance and emissions characteristics.

Hydrogen dual-fuel combustion can be categorized into two primary methods: (1) hydrogen as a secondary fuel, supplementing the combustion of diesel or other liquid fuels, and (2) hydrogen as the primary fuel, where a liquid fuel such as diesel or HVO acts as a pilot fuel to initiate combustion. The study by Pinto et al. examined hydrogen-HVO dual-fuel operation, but their research was limited to low hydrogen concentrations, with a maximum hydrogen energy fraction of 28% [4]. At higher hydrogen concentrations, the risk of abnormal combustion, such as pre-ignition and knock increases, which can limit operational conditions [2, 5].

Moreover, the ignition delay behavior in hydrogen dual-fuel combustion exhibits complex dependencies on hydrogen concentration and temperature, potentially leading to either longer or shorter ignition delays [6, 7]. Higher hydrogen concentrations could alter combustion characteristics, requiring further investigation. Additionally, Pinto et al. observed a decline in brake thermal efficiency (BTE) with increasing hydrogen content, which has also been reported for other pilot fuels [4, 8–11]. To mitigate this efficiency loss, pilot fuel injection strategies must be optimized [7].

To facilitate the effective utilization of hydrogen as a primary fuel, a comprehensive understanding of the optimum operating conditions is essential to achieve high efficiency and low emissions while avoiding abnormal combustion. Therefore, this study aims to evaluate the performance and emission characteristics of a hydrogen dual-fuel engine, utilizing HVO and diesel as pilot fuels, and to identify optimal operating strategies that maximize efficiency

while minimizing emissions.

This chapter investigates the performance, emissions, and combustion characteristics of hydrogen-HVO dual-fuel in a single cylinder diesel engine. Utilizing high hydrogen concentration, the engine performance and emissions in the hydrogen-HVO dual-fuel mode are elucidated. To determine the optimal operational range with high efficiency and lower emissions without abnormal combustion under various load conditions, this study involved varying the hydrogen excess air ratio, pilot quantity, and intake pressure. Furthermore, to ascertain the suitability and effectiveness of HVO as a pilot fuel in the hydrogen dual-fuel mode, hydrogen-diesel dual-fuel combustion under identical conditions was examined and compared to hydrogen-HVO dual fuel combustion.

2.2 Experimental setup and methods

Fig. 2.1 shows a schematic diagram of the experimental setup. The experimental setup comprises a single-cylinder diesel engine with a common rail fuel injection system, an AC electric dynamometer, various measuring devices, and an externally driven supercharger (S/C) with an intercooler. The intake air passes through an air filter, laminar flowmeter, and surge tube and is pressurized by the S/C before being introduced into the cylinders. The exhaust line is equipped with an exhaust chamber to prevent exhaust pulsation. Commercially available hydrogen gas ($> 99.99\%$ purity) was used as the main fuel and was injected into the intake pipe through a gas injector located immediately before the swirl control valve. The pilot fuel was supplied by a common rail injection system and fuel injector (Denso, G3P). Table 2.1 lists the test engine specifications. A single-cylinder diesel engine with a bore of 85 mm and a stroke of 96.9 mm was used, which had a toroidal combustion chamber and compression ratio of 14.3. The knocking intensity could be increased at a higher compression ratio; hence, a compression ratio slightly lower than that of a conventional diesel engine was employed. Table 2.2 lists the properties of the pilot fuels used in this study; HVO and JIS No. 2 diesel fuel were utilized for comparison. The density of HVO is lower than that of diesel fuel because HVO mostly comprises normal hydrocarbons and has no aromatic content. Although the cetane index is applicable only to petroleum-derived diesel fuel, it was applied to HVO in this study for comparative purposes. It must be stated here that calculated

value of the cetane index is an assumed value; however, it can be used for comparison with the cetane index of diesel fuel. Consequently, HVO has a higher cetane index than diesel fuel, suggesting that HVO is expected to have higher ignitability than diesel.

Table 2.3 summarizes the experimental conditions used in the study. The engine was operated at a constant speed of 1,800 rpm. The experiments were conducted at a fixed intake temperature and engine oil filter temperature under all conditions. To analyze the effects of the intake pressure, the intake pressure was varied from 100 to 220 kPa in increments of 20 kPa by adjusting the rotation speed of the S/C. The exhaust pressure was simultaneously adjusted to match the intake pressure using a backpressure valve. To examine the effects of pilot fuel types and quantity, the engine was operated with pilot injection quantities of 3.0, 6.0, and 13.0 mm³/cycle during HVO as pilot fuel, while injection quantities of 2.9, 5.7, and 12.4 mm³/cycle were used during diesel as pilot fuel to achieve same injected energy of the pilot fuels. The injection timing was adjusted such that ignition occurred at TDC under all conditions. Ignition is defined as the crank angle at which the tangent line to the maximum differential of the heat release rate $dQ/d\theta$ intersects the crank angle axis. Hydrogen was injected to the intake pipe using a gas injector at -380° ATDC under all conditions, and the hydrogen excess air ratio (λ_{H_2}) was changed in the range between 4.0 and 2.5 by 0.5.

The in-cylinder pressure was subsequently measured to study the combustion performance, which was accomplished by obtaining signals from a piezoelectric pressure transducer (Kistler, Type 6052C) installed in the cylinder and a charge amplifier (Kistler, Type 5018A). The heat release rates were calculated from the in-cylinder pressure history (50-cycle average), and calculated using equation below.

$$\frac{dQ}{d\theta} = \frac{1}{\kappa - 1} \left(V \frac{dp}{d\theta} + \kappa p \frac{dV}{d\theta} \right) + \frac{dQ_c}{d\theta} \quad (2.1)$$

Here, p stands for the in-cylinder pressure, and V represents the volume of the combustion chamber. κ represents the mean specific heat ratio of the in-cylinder gas. Also, $\frac{dQ_c}{d\theta}$ is the rate of heat loss due to cooling, determined using Newton's law of cooling and Woschni's heat transfer coefficient.

To evaluate the extent of energy replaced by hydrogen in this study, the hydrogen mixing ratio (r_{H_2}) was calculated using the following equation:

$$r_{H_2} = \frac{m_{H_2} \cdot LHV_{H_2}}{m_{pilot} \cdot LHV_{pilot} + m_{H_2} \cdot LHV_{H_2}} \times 100 \quad (2.2)$$

where, m_{H_2} : hydrogen mass [kg/cycle], LHV_{H_2} lower heating value of hydrogen [MJ/kg], m_{pilot} : pilot fuel mass [kg/cycle], LHV_{pilot} lower heating value of the pilot fuel [MJ/kg].

The following equation was used to determine the cooling loss ratio (X_c):

$$X_c = 100 - X_{W_i} - X_u - X_e \quad (2.3)$$

where, X_{W_i} indicated work fraction [%], X_u combustion loss fraction [%], X_e exhaust loss fraction [%], respectively. The exhaust loss fraction was calculated based on the exhaust flow rate and temperature.

The DCV, an indicator of how close the actual cycle is to the ideal constant volume cycle (Otto cycle), was calculated using the following formula:

$$DCV = \frac{1}{Q} \int \frac{dQ}{d\theta} \left\{ \frac{1 - 1/\epsilon_{\theta}^{\kappa-1}}{1 - 1/\epsilon^{\kappa-1}} \right\} d\theta \quad (2.4)$$

where, Q denotes the input energy, θ denotes the crank angle, ϵ denotes the compression ratio, ϵ_{θ} denotes the compression ratio at a specific crank angle, and κ denotes the specific heat ratio.

An engine exhaust gas measuring system (Horiba, MEXA-1600DEGR) was used to measure NO_x , total hydrocarbon (THC), CO, CO_2 , and O_2 concentrations. The number of filtered smoke particles was measured using a reflective smoke meter (AVL, 415S).

In conducting the experiments, the operating range was subject to the following constraints:

1. Restriction based on the maximum in-cylinder pressure (p_{max}): The maximum cylinder pressure was limited to 12 MPa at an engine speed of 1,800 rpm.

2. Limitation by maximum pressure rise rate (dp_{max}): The maximum pressure rise rate $dp/d\theta_{max}$ was constrained to 1.0 MPa/deg.
3. Constraints owing to abnormal combustion: Operations were discontinued in the event of a misfire, pre-ignition before pilot injection, or knocking. The knock onset is defined as the point at which the maximum cylinder pressure oscillation exceeds 0.4 MPa, following the application of a high-pass filter at 3 kHz to the in-cylinder pressure.

2.3 Results and discussion

2.3.1 Combustion behavior in hydrogen dual fuel

To clarify the combustion behavior in hydrogen dual fuel, Fig. 2.2 presents the in-cylinder pressure, p and heat release rates, $dQ/d\theta$ under conditions of diesel fuel as pilot fuel, with a pilot fuel injection quantity of 2.9 mm³/cycle and an intake pressure of 140 kPa. The black line represents neat diesel fuel combustion (without hydrogen), while the red and blue lines represent hydrogen-diesel dual-fuel combustion at $\lambda_{H_2} = 2.6$ (red) and $\lambda_{H_2} = 4.0$ (blue). For neat diesel fuel, the heat release rate curve shows a single peak, indicating combustion dominated by diesel fuel. When hydrogen is introduced, the lines exhibit multiple peaks: the first peak corresponds to rapid diesel pilot fuel combustion, the second peak represents premixed hydrogen combustion, and a third peak appears at higher hydrogen concentrations ($\lambda_{H_2} = 2.6$). Under this high hydrogen concentration, autoignition of hydrogen in the end-gas region occurs, as suggested by Azimov et al. [12]. These peaks are labeled as ①, ②, and ③ in the figure. As λ_{H_2} decreases (higher hydrogen concentration), the second peak overlaps with the first peak due to increased hydrogen near the pilot flame, accelerating combustion. The peak of $dQ/d\theta$ increases with lower λ_{H_2} , indicating higher heat input and faster flame propagation, resulting in shorter combustion duration and higher in-cylinder pressure.

2.3.2 Effects of pilot fuel onto engine performance and emissions

To clarify the impact of using diesel fuel and HVO as pilot fuels in a hydrogen dual-fuel engine on the engine performance, emissions, and combustion behavior, a series

of tests were performed with various pilot fuel quantities and intake pressures for both fuels. The pilot fuel quantity of 3.0 mm³/cycle for HVO and 2.9 mm³/cycle for diesel injected under intake pressure of 140 kPa were selected as representative to show the effects of pilot fuel types, in which hydrogen concentration could be increased until a λ_{H_2} of approximately 2.5 without abnormal combustion.

To elucidate the variations in combustion behavior between different pilot fuel types, the present study examines and illustrates the in-cylinder pressure, p and heat release rates, $dQ/d\theta$ under conditions involving neat diesel fuel, neat HVO, hydrogen-diesel, and hydrogen-HVO fuels, as depicted in Fig. 2.3. Under the neat diesel fuel and neat HVO conditions, the pilot fuels were combusted without hydrogen, and the total excess air ratios were 22.3 and 22.4, respectively, and indicated with black colored lines. The distinctive characteristic of neat diesel fuel and neat HVO combustion is the presence of a single peak in the heat release rate curve. However, when hydrogen is introduced at approximately $\lambda_{H_2} = 4.0$, a second peak emerges, and at higher hydrogen concentrations, a third peak becomes evident. These peaks are labeled as ①, ②, and ③ in Fig. 2.3. The peaks are more pronounced during HVO combustion. Despite the later start of pilot injection (SOI) for HVO, HVO ignites first, indicating that HVO is a highly ignitable fuel. The short ignition delay of HVO is assumed to result in a smaller initial $dQ/d\theta$ peak and a smaller pilot flame distribution. Consequently, the peak of $dQ/d\theta$ is lower with HVO, and the combustion duration tends to be longer. Furthermore, a decrease in λ_{H_2} leads to an increase in the maximum peak of $dQ/d\theta$, accompanied by a shorter combustion duration. This occurs because of the increased heat input and high flame propagation speed of hydrogen, which accelerates combustion and enables more fuel to be burned within a shorter period of time [10]. This results in a higher peak in-cylinder pressure.

Fig. 2.4 illustrates the engine performance and emission characteristics for diesel fuel and HVO under the conditions of an intake pressure of 140 kPa, pilot fuel quantity of 3.0 mm³/cycle (HVO) and 2.9 mm³/cycle (diesel fuel), versus λ_{H_2} . λ_{H_2} was gradually decreased until approximately $\lambda_{H_2} = 2.5$ under all conditions; however, knocking occurred with diesel fuel, limiting the operation just before knocking occurred at $\lambda_{H_2} = 2.6$. Pre-ignition occurred at λ_{H_2} slightly lower than 2.5 when HVO was used as the pilot fuel. Under

knocking conditions, it is highly possible that pre-ignition also occurred, which may have caused knocking. To prevent damage to the test apparatus, the operating conditions were limited immediately after abnormal combustion occurred; thus, pre-ignition and knocking occurrences could not be distinguished in this case. For the diesel fuel case, knocking was detected first; therefore, it was reported that operation was limited owing to knocking. However, the possibility that pre-ignition occurred simultaneously could not be ruled out.

Information on the hydrogen mixing ratio, injection timing, gross indicated effective pressure (IMEPg) under these conditions could be obtained from Figure 2.4(a). Moreover, the injection timing is denoted as “Inj. timing” and the indicated thermal efficiency is presented as η_i . HVO was injected later than diesel fuel, as observed in “Inj. timing” plots in Figure 2.4(a). This is owing to the higher cetane index of HVO than that of diesel fuel, which provides a shorter ignition delay. When the hydrogen excess air ratio was reduced, the injection timing was slightly retarded owing to the shorter ignition delay. Previous studies have suggested that an increased hydrogen fraction makes retarded ignition possible under high hydrogen concentration but with a low residual heat effect (several motored cycles before each fired cycle) condition[13] and low hydrogen concentration condition[14]. However, under these operating conditions and high hydrogen concentration, a higher hydrogen fraction likely resulted in higher residual heat and temperature, providing more favorable conditions for pilot fuel ignition.

In terms of engine performance, in Fig. 2.4(a), diesel fuel had a slightly higher indicated thermal efficiency than HVO. The longer ignition delay, which resulted in a higher peak heat release rate and reduced combustion duration, is the reason for the indicated thermal efficiency trend, as discussed in Fig. 2.3. Moreover, a higher indicated thermal efficiency was observed when λ_{H_2} was lowered to approximately 3.0. This can be attributed to the high-speed flame propagation of hydrogen, which reduced the combustion duration. However, the increase of $dQ/d\theta$ and in-cylinder pressure peaks, as well as shorter combustion duration could not deduce the increment of indicated thermal efficiency alone, because the maximum peak timing is slightly retarded at a lower λ_{H_2} . Moreover, HVO also exhibited retardments on the peaks compared to diesel fuel. Moreover, the extremely high $dQ/d\theta$ could deteriorate the indicated thermal efficiency through increased cooling loss.

Furthermore, from λ_{H_2} of 3.0 to 2.5, the indicated thermal efficiency exhibits decreasing trends under both pilot fuel conditions. To deduce these trends of the indicated thermal efficiency, the DCV and cooling loss ratio were studied, which will be discussed in Figure 2.5 later.

The emission behavior is illustrated in Fig. 2.4(b). No significant changes were observed in exhaust emissions for the different pilot fuel types, except for smoke emissions. Diesel fuel and HVO both exhibit only low smoke value, with diesel fuel exhibiting slightly higher smoke than HVO at low λ_{H_2} . This is because HVO has no aromatic content. The smoke from both HVO and diesel fuel increased as λ_{H_2} was decreased, and at approximately $\lambda_{H_2} = 2.5$, the smoke from diesel fuel doubled that from HVO. This was also because of the shorter ignition delay, which reduced the mixing time between the pilot fuels and air, resulting in locally rich conditions of the pilot fuels. Moreover, hydrogen possesses low density and high diffusivity properties. When hydrogen is injected into the intake port from the injection system, it blocks the entry of O_2 due to the volume limitations of the intake port[15], resulting in low O_2 intake per volume of air. This results in restricted O_2 entrainment into pilot fuel. Consequently, the oxidation of pilot fuel is restricted, leading to elevated smoke emissions. When λ_{H_2} was lowered, both HVO and diesel fuel emitted higher CO, but higher NO_x was also emitted, with significant increase (maximum increase of NO_x approximately 3000 ppm). Hydrogen is delivered into the cylinder with heterogeneity, although it is injected into the intake pipe. Therefore, the local equivalence ratio is expected to be high in certain areas, causing elevated NO_x emission. Furthermore, the elevated combustion temperature at higher hydrogen concentrations also contributes to increased NO_x emissions.

Fig. 2.5 presents the plots of DCV and cooling loss ratio (X_c) against λ_{H_2} under an intake pressure of 140 kPa, with an HVO pilot quantity of 3.0 mm³/cycle and a diesel fuel pilot quantity of 2.9 mm³/cycle. In the case of diesel fuel, the DCV is higher compared to HVO, owing to a higher $dQ/d\theta$ peak and shorter combustion duration. When λ_{H_2} is decreased, an increase in DCV is noted. This improves indicated thermal efficiency as λ_{H_2} decreases until 3.0. X_c represents the fraction of energy lost to wall heat transfer, which reduces indicated thermal efficiency. While the cooling loss ratio exhibits minimal variation

between diesel fuel and HVO, a slight increasing trend is observed as λ_{H_2} decreases from 3.0 to approximately 2.5. This deteriorates indicated thermal efficiency, leading to a decrease in indicated thermal efficiency with λ_{H_2} from approximately 3.0, owing to a higher fraction of energy loss.

2.3.3 Effects of pilot quantity onto engine performance and emissions

Previous study has highlighted the crucial role of pilot fuel quantity in influencing efficiency, emissions, and abnormal combustion in dual-fuel engines [16]. Therefore, this study further investigates its impact on hydrogen dual-fuel engine. The performance, and emission characteristics in hydrogen dual-fuel engines, along with the combustion and ignition characteristics were compared by altering the injection quantity q_j of HVO (3.0, 6.0, and 13.0 mm³/cycle) at an intake pressure p_s of 140 kPa. However, pre-ignition and knocking occurred, which restricts the operating condition of λ_{H_2} to 2.9 for 13.0 mm³/cycle and approximately 2.5 for 3 and 6.0 mm³/cycle.

Fig. 2.6 presents the 50-cycle averages of the heat release rate $dQ/d\theta$ and the in-cylinder pressure p , under the conditions where the HVO injection quantity were varied as described above. The start of pilot injection (SOI) is indicated on the curve of the $dQ/d\theta$, with the corresponding SOI symbol being colored next to the legend. An increased pilot quantity resulted in a higher $dQ/d\theta$ peak. Moreover, a greater tendency for the first and second peaks to overlap could be observed. The increased pilot quantity increases pilot fuel dispersion, in which more pilot fuel was premixed with hydrogen. This resulted in more hydrogen being combusted simultaneously with the pilot fuels; hence, the first and second peaks overlapped, and a shorter combustion duration was obtained. Here, the shorter combustion duration and increasing peak of $dQ/d\theta$ with pilot quantity is assumed to affect the thermal efficiency, and NO_x emission. Hence the engine performance is studied in Fig. 2.7.

The engine performance and emissions results are depicted in Fig. 2.7(a). r_{H_2} decreases as the pilot quantity increases because a higher fraction of hydrogen is replaced by pilot fuel energy. The pilot quantity hardly affected the ignition delay as can be observed from “Inj. timing”. IMEPg increased with the pilot quantity owing to the increased heat input

from the pilot fuel. However, η_i exhibits no change when the pilot quantity is increased to 6.0 mm³/cycle, and it deteriorates when the pilot quantity is increased further to 13.0 mm³/cycle. To deduce this indicated thermal efficiency trend, DCV and X_c are plotted and discussed later in Fig. 2.8.

Regarding emissions, CO₂ and CO increased with the pilot quantity owing to the increase in carbon input while THC displayed minimal change when the pilot quantity was increased to 6.0 mm³/cycle, as can be observed in Fig. 2.7(b). However, when the pilot quantity was further increased to 13.0 mm³/cycle, THC decreased because more pilot fuel increases local combustion temperatures, enhancing the oxidation of unburned hydrocarbons, thereby reducing THC emissions. Moreover, the higher pilot quantity increases the pilot spray velocity, thereby enhancing the mixing rate of the pilot fuel. NO_x increased with the pilot quantity because of the increase in heat input. Smoke displayed minimal changes between pilot quantities except at 6.0 mm³/cycle.

Fig. 2.8 exhibited the DCV and X_c results when the pilot quantity is manipulated. The DCV exhibited increasing trend until a pilot quantity of 6.0 mm³/cycle; however, at 13 mm³/cycle, almost no DCV improvement was observed. In contrast, X_c displayed less of an effect when the pilot quantity increased to 6.0 mm³/cycle. When the pilot quantity was increased to 13.0 mm³/cycle, a significant increase in X_c was observed, which deteriorated the indicated thermal efficiency in Fig. 2.7(a). The high cooling loss ratio at a pilot quantity of 13.0 mm³/cycle was owing to the higher momentum of the spray impinging onto the cylinder wall at a high pilot quantity.

2.3.4 Effects of intake pressure onto engine performance and emissions

To investigate the effects of different load conditions and pilot spray penetration behaviors on combustion behaviors, the intake pressures were varied in this study. The injection quantity of HVO was set to 3.0 mm³/cycle and λ_{H_2} was varied from 2.5 to 4 at intake pressures of 100, 140, and 200 kPa. To cancel out the pumping work produced by increasing the intake pressure, the exhaust pressure was increased, matching the at approximately the same value as intake pressure. The hydrogen excess air ratio was altered in increments of approximately 0.50. However, pre-ignition limited the operation with λ_{H_2}

down to 2.5 for intake pressure of 100 and 140 kPa owing to pre-ignition. For the 200 kPa intake pressure, the operational limit of the engine was achieved at $\lambda_{H_2} = 3.0$.

Fig. 2.9 illustrates the 50-cycle averages of the heat generation rate $dQ/d\theta$ and cylinder pressure p at intake pressures of 100, 140, and 200 kPa. The ignition delay is shorter at higher intake pressures because of the easier ignition of the pilot fuel caused by the higher cylinder pressure near the TDC. The trend indicates that the maximum $dQ/d\theta$ increases, but the combustion duration decreases with decreasing λ_{H_2} at all intake pressures. At an intake pressure of 100 kPa, the ignition delay was longer, resulting in more premixed hydrogen and pilot fuel; hence, a single peak was observed under most conditions. Three combustion stages were observed at an intake pressure of 140 kPa and 200 kPa. Increasing the intake pressure elevates the heat release rate (HRR) peak; however, the combustion duration tends to be prolonged, with heat is still generated even at 30° CA, under intake pressure of 200 kPa. This is attributed to the high intake pressure, which restricts pilot fuel spray development, leading to a more gradual progression of the pilot flame toward the premixed hydrogen. As this combustion behavior could influence engine performance and emission the engine performance and emission results are plotted in Fig. 2.10.

Fig. 2.10(a) shows the engine performance and exhaust gas emissions at intake pressures of 100, 140, and 200 kPa using HVO. IMEPg increased as intake pressure is increased indicating that higher load is achieved at high intake pressure due to higher pressure in the cylinder. As the amount of air increases at higher intake pressure, higher hydrogen amount is required to achieve the same λ_{H_2} hence r_{H_2} increased with intake pressure. “Inj. timing” is delayed when the intake pressure is increased, due to shorter ignition delay as stated previously. The indicated thermal efficiency exhibit improvements when the intake pressure increased. The heat release rate exhibit higher peak of heat release rate, could possibly explain the indicated thermal efficiency trend, but it was also observed that higher intake pressure lengthens the combustion duration. Hence, the plot of DCV and X_c are necessary to explain this indicated thermal efficiency trend. This will be discussed later in Fig. 2.11.

Regarding emissions in Fig. 2.10(b), almost no effect was observed when the intake pressure was increased, except that a lower THC was emitted at a higher intake pressure,

probably because of the higher oxygen available to oxidize the hydrocarbon. When λ_{H_2} was decreased from 4.0 to 3.0, NO_x increases with maximum increment of about 1000 ppm, and almost no change is observed from smoke emission. However, when λ_{H_2} was decreased further from 3.0 to 2.5, significant NO_x increment was observed with about 2500 ppm increment. Again, this trend is probably due to the heterogeneous supply of hydrogen into the cylinder.

Fig. 2.11 shows the DCV and X_c values under various intake pressures. The intake pressure exhibited only a minimal effect on the DCV. At λ_{H_2} higher than approximately 3.0, the DCV decreases at small magnitudes when the intake pressure is increased, owing to longer combustion duration observed in Figure 2.9. When λ_{H_2} is decreased, the DCV is increased, and the increasing intensity increases with intake pressure, owing to higher hydrogen concentration that increases the $dQ/d\theta$ peak near TDC. Therefore, at $\lambda_{H_2} = 3.0$ and lower, almost no effect in DCV is observed when the intake pressure is increased. However, the cooling loss ratio is reduced when the intake pressure is increased. This is believed to be owing to the lower spray penetration of the pilot fuel at higher intake pressures, resulting in ignition occurring upstream and away from the wall. This led to improvements in indicated thermal efficiency at high intake pressures, as shown in Figure 2.10(a).

2.3.5 Operation range comparing diesel fuel and HVO

In the previous section, the effects of the intake pressure were elucidated using HVO with $q_j = 3 \text{ mm}^3/\text{cycle}$, and the effects of pilot fuel were clarified using HVO and diesel fuel with $q_j = 3 \text{ mm}^3/\text{cycle}$ as a representative. While increasing the intake pressure exhibited higher indicated thermal efficiency owing to lower spray penetration, we assume that higher pilot quantities at various intake pressures could exhibit different behaviors onto the effects on engine performance and emission and onto the operable range because a higher pilot quantity could result in higher spray momentum and penetration, and the spray is more distributed. To study this case briefly, the contour plots are shown in Fig. 2.12. Moreover, to provide a comprehensive discussion, the results of using diesel fuel as a pilot fuel are presented. The injection quantity, q_j of HVO was set to 6.0 and 13.0 mm^3/cycle and that of diesel fuel to 5.7 and 12.4 mm^3/cycle , and the intake pressure p_s was varied from 100

to 220 kPa in 20 kPa increments. For each p_s , λ_{H_2} was varied from 2.5 to 4 in steps of approximately 0.50. Certain conditions could not be applied owing to abnormal combustion and safety limits of our equipment.

Fig. 2.12 presents contour plots of indicated thermal efficiency (η_i), cooling loss ratio (X_c), indicated specific NO_x ($IsNO_x$), and smoke with λ_{H_2} on the horizontal axis and gross indicated mean effective pressure (IMEPg) on the vertical axis for each pilot fuel type and injection quantity. The red line in the diagrams represents the limit owing to the maximum cylinder pressure (p_{max}), the yellow line represents the limit owing to knocking, and the green line represents the limit owing to the pressure increase rate (dp_{max}).

When HVO is injected at 6.0 mm³/cycle and diesel fuel at 5.7 mm³/cycle, the operable range for both HVO and diesel fuel are almost similar. Regardless of the pilot fuel type, abnormal combustion is likely to occur in the region of approximately $\lambda_{H_2} < 3.0$. The higher amount of smoke observed for both pilot fuel types near the region where abnormal combustion occurred was believed to be owing to the occurrence of abnormal combustion in some cycles during the measurement and uneven hydrogen intake. Furthermore, the indicated thermal efficiency increased with the intake pressure owing to the improved cooling loss ratio at higher intake pressures. Based on the plots, it can be concluded that using HVO as pilot fuel, the engine can be operated at richer hydrogen concentration than when using diesel, to achieve highly efficient operations. Regardless of the pilot fuel type and injection quantity, NO_x tended to increase with increasing hydrogen concentration (lower λ_{H_2}). When comparing pilot fuel type, a higher percentage of areas with lower soot was observed for HVO compared to diesel fuel. HVO has slightly smaller areas with high indicated thermal efficiency but comparable IMEPg and lower NO_x emission. As a strategy to operate with high efficiency and low emission, when utilizing diesel fuel as pilot fuel, the engine should be operated at high load and lean hydrogen condition. Meanwhile, when utilizing HVO as pilot fuel, the engine need to be operated at slightly richer hydrogen concentration condition and high load.

When operating under the conditions of 13.0 mm³/cycle HVO and 12.4 mm³/cycle diesel fuel injection, diesel fuel exhibited a narrower operable range. This is owing to the longer ignition delay of diesel fuel compared to that of HVO. This led to a rapid increase in

heat release rate. Moreover, the wider distribution of diesel fuel compared to that of HVO accelerates chemical reactions in the premixture, leading to autoignition. These factors, in turn, limit the operation owing to the in-cylinder maximum pressure and pressure increase rate limits. Conversely, the operable range was wider when HVO was used as the pilot fuel.

2.4 Conclusions

To clarify the effect of fuel type on the performance and emissions of a hydrogen dual-fuel engine, the type of pilot fuel, hydrogen excess air ratio, pilot quantity, and intake pressure were varied for a single-cylinder diesel engine.

1. HVO reduces smoke while providing combustion and performance characteristics equivalent to those of diesel fuel.
2. Under a constant intake pressure, NO_x and smoke increase with a lower hydrogen excess air ratio.
3. NO_x and smoke decrease with decreasing pilot injection quantity.
4. Increasing intake pressure increases indicated thermal efficiency owing to lower cooling losses.
5. Under conditions where the hydrogen excess air ratio is below a certain value, abnormal combustion, such as knocking or pre-ignition, is likely to occur; however, HVO can be operated at lower excess air ratios than when using diesel fuel.
6. When HVO is used as the pilot fuel, the engine can be operated at a lower load range than when diesel fuel is used.
7. As a strategy to operate with high efficiency and low emission, when utilizing diesel fuel as pilot fuel, the engine should be operated at high load and lean hydrogen condition. Meanwhile, when utilizing HVO as pilot fuel, the engine need to be operated at slightly richer hydrogen concentration condition and high load.

Based on contour plots, it could be concluded that the distribution of pilot fuel spray appears to be the primary factor influencing the differing combustion behaviors of HVO and diesel, and under varying pilot fuel quantities. HVO is assumed to have a narrower pilot fuel distribution due to its shorter ignition delay, reducing hydrogen autoignition tendencies and enabling operation at higher hydrogen concentrations, and low load conditions. At higher

pilot fuel quantities, the distribution is estimated to widen, narrowing the operable range and causing high-velocity sprays to impinge on the piston bowl wall, increasing cooling losses, therefore reducing indicated thermal efficiency. However, as this hypothesis is based on a complex combustion process that may exhibit contradictory behavior, visualization of the combustion process is essential for validation.

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Figures

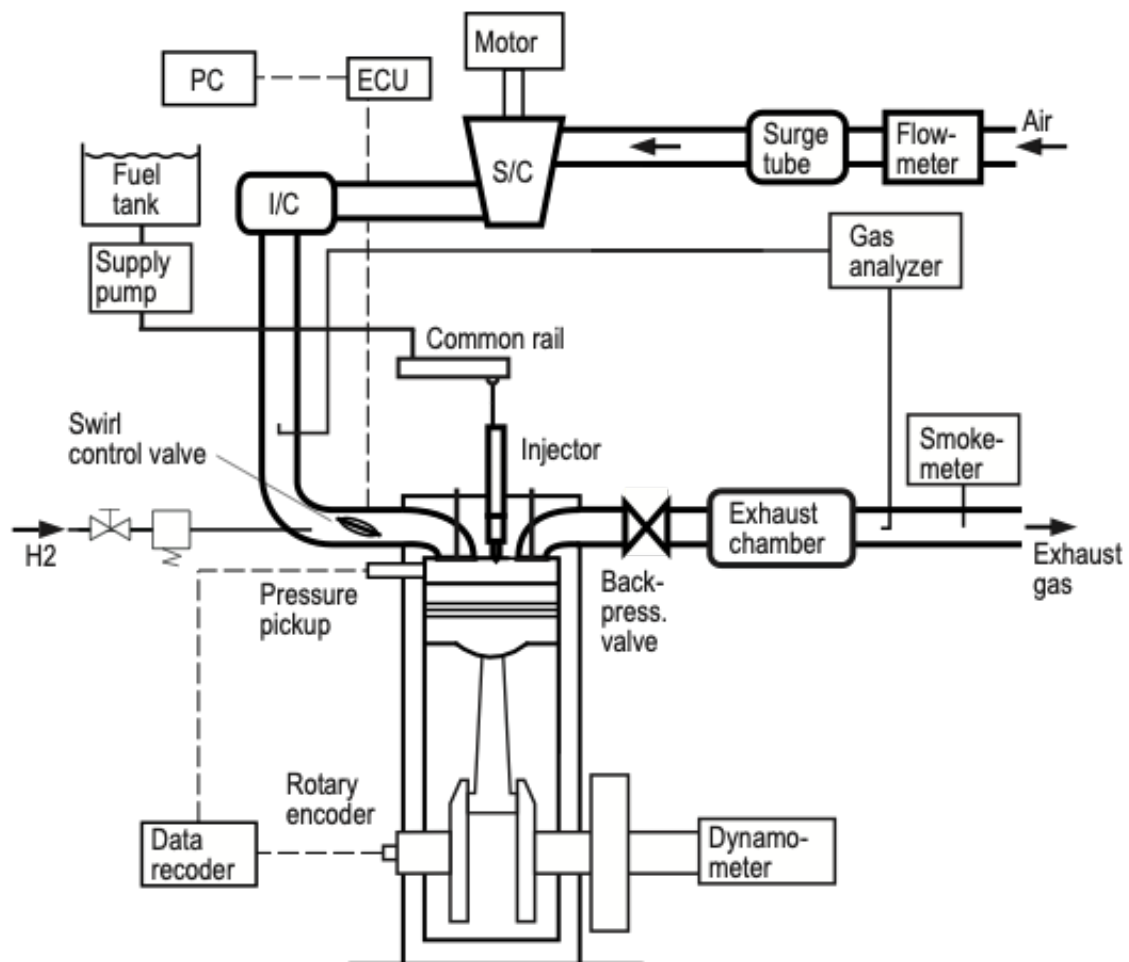
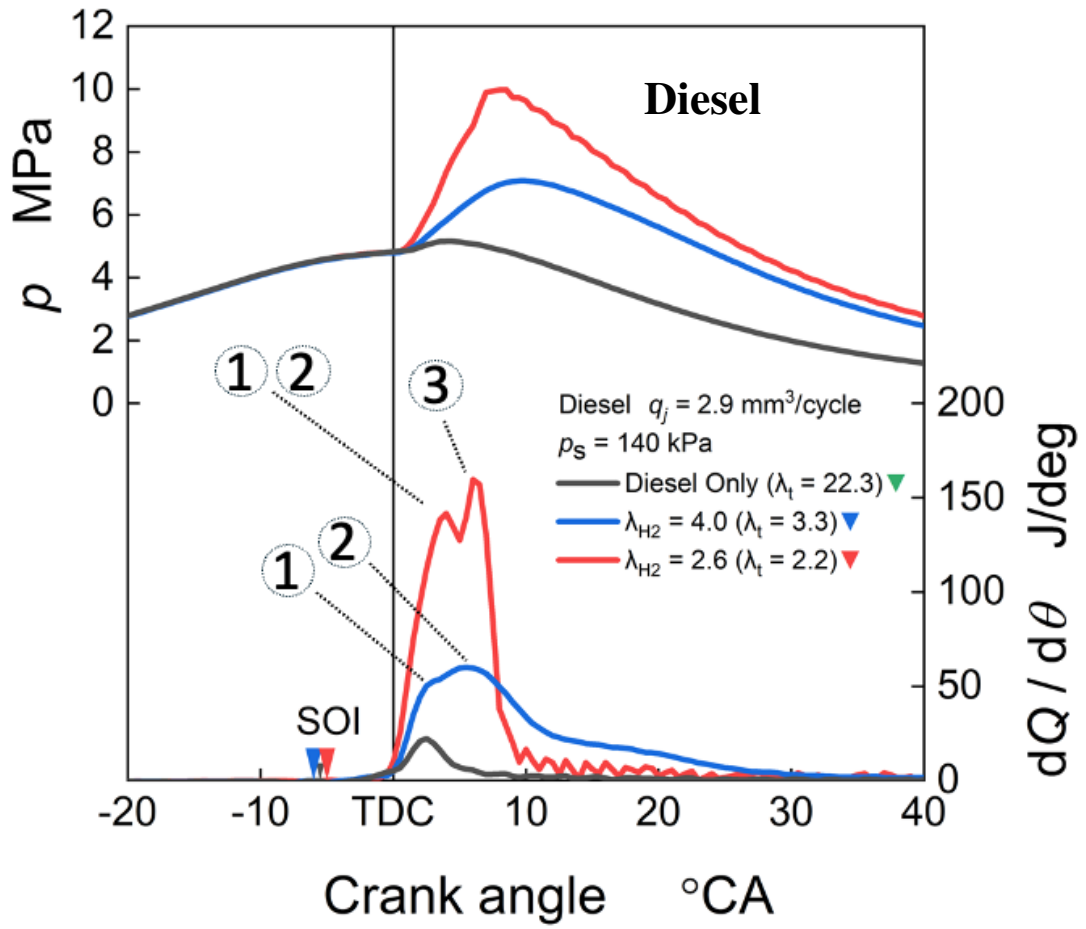
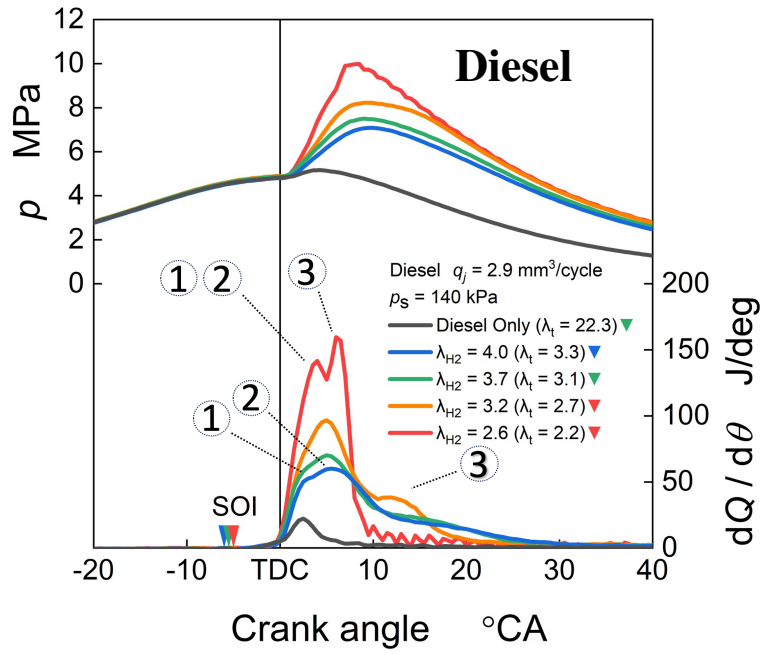


Figure 2.1: Schematic diagram of the experimental setup.

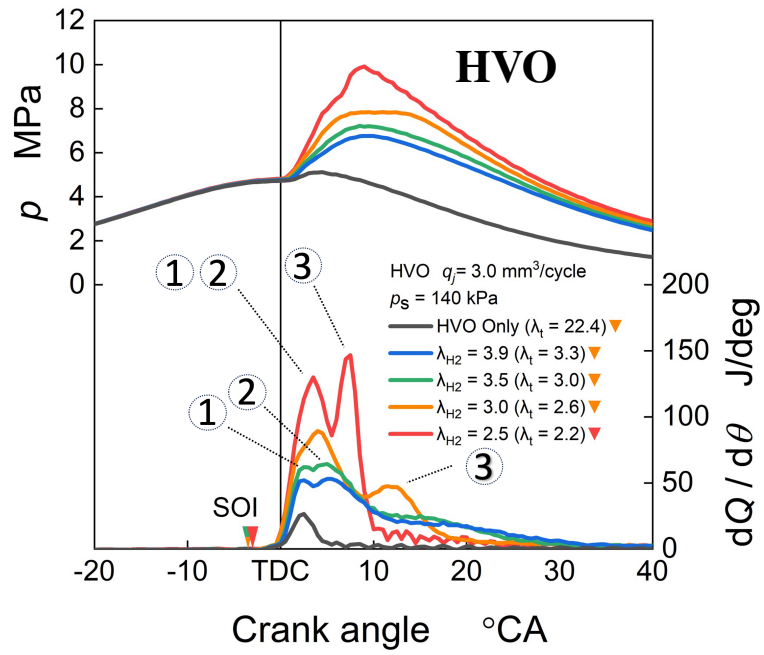


(a) Diesel fuel

Figure 2.2: In-cylinder pressure, p and heat release rate, $dQ/d\theta$ of hydrogen-diesel fuel under intake pressure, $p_s = 140 \text{ kPa}$, and pilot quantity of $2.9 \text{ mm}^3/\text{cycle}$.



(a) Diesel fuel



(b) HVO

Figure 2.3: In-cylinder pressure, p and heat release rate, $dQ/d\theta$ of HVO and diesel fuel under intake pressure, $p_s = 140 \text{ kPa}$, and pilot quantity of 3.0 and 2.9 mm^3/cycle , respectively.

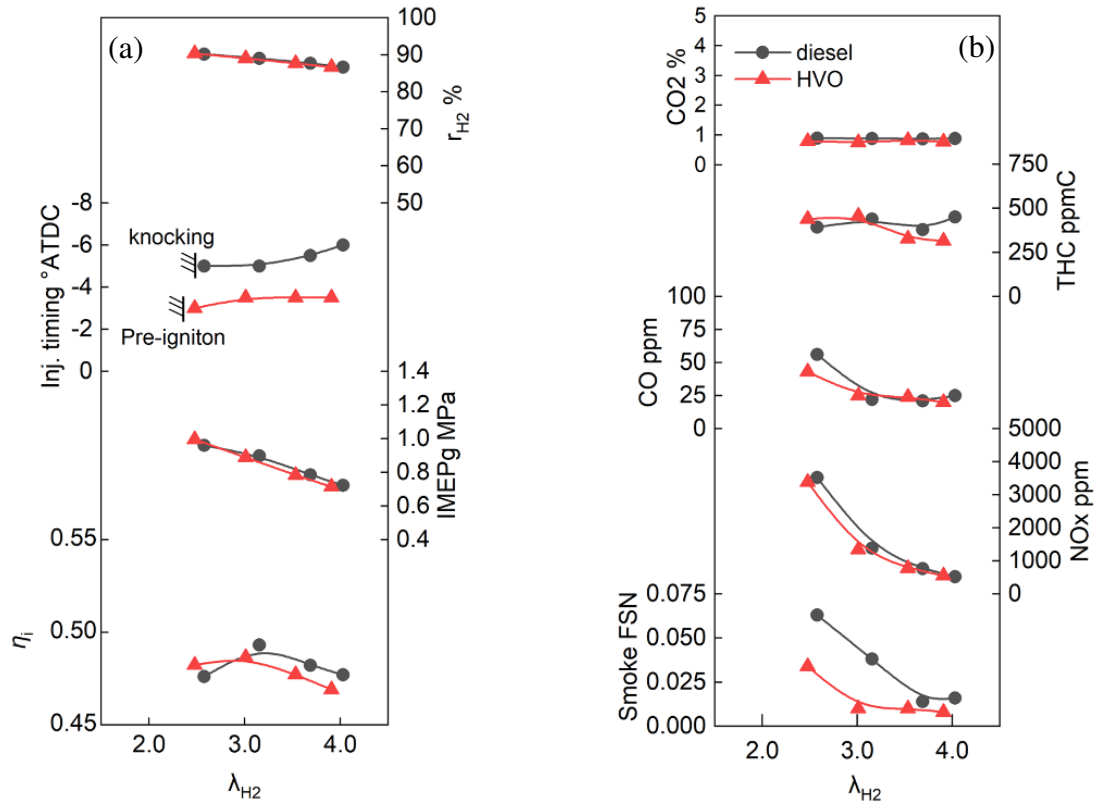


Figure 2.4: Engine performance (a) and emission (b) of HVO and diesel fuel as pilot fuels under intake pressure, $p_s = 140$ kPa and pilot quantity of 3.0 and 2.9 mm³/cycle, respectively.

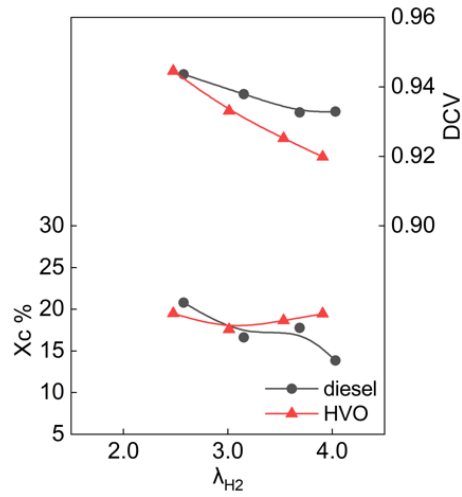


Figure 2.5: DCV and X_c under conditions of HVO and diesel fuel, under intake pressure, $p_s = 140$ kPa, and pilot quantity of 3.0 and 2.9 mm³/cycle, respectively.

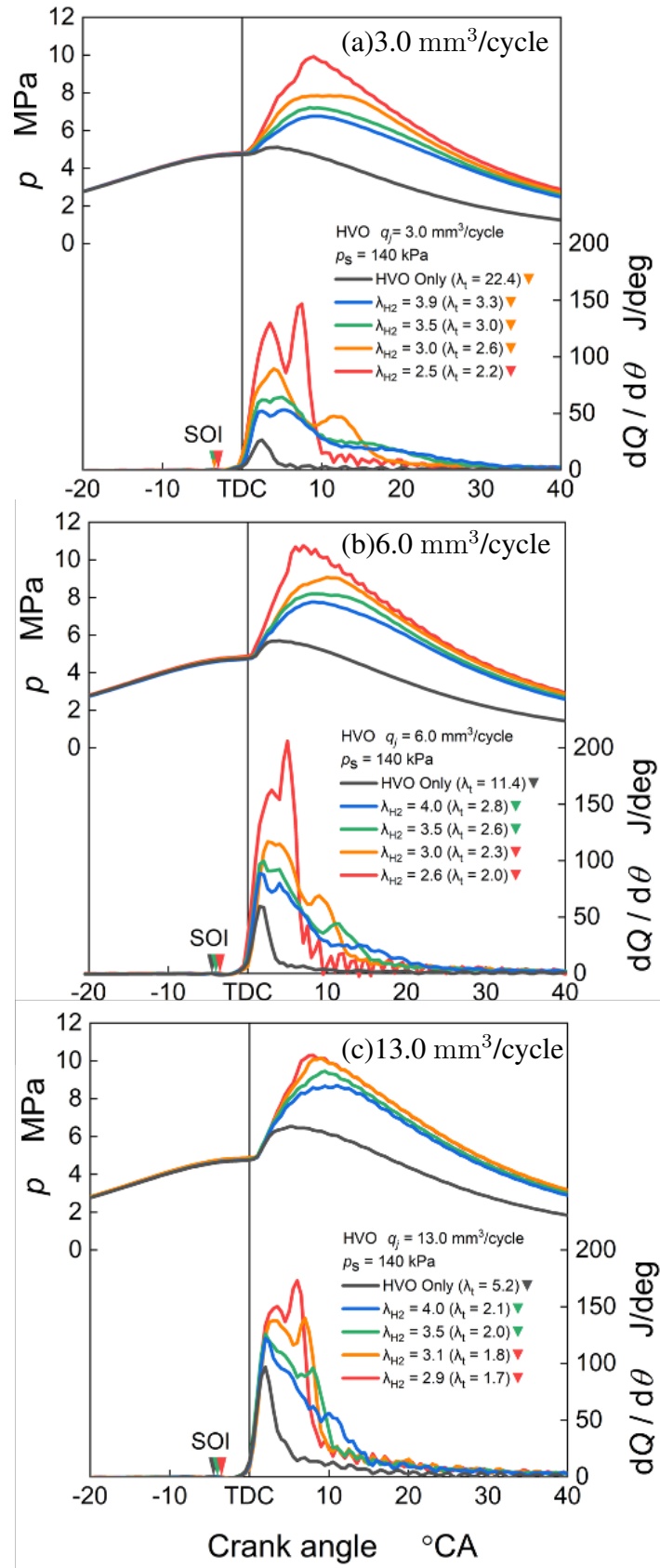


Figure 2.6: In-cylinder pressure, p , and heat release rate, $dQ/d\theta$ of HVO under intake pressure, $p_s = 140$ kPa, and pilot quantity, q_i of 3.0, 6.0, and 13.0 mm³/cycle.

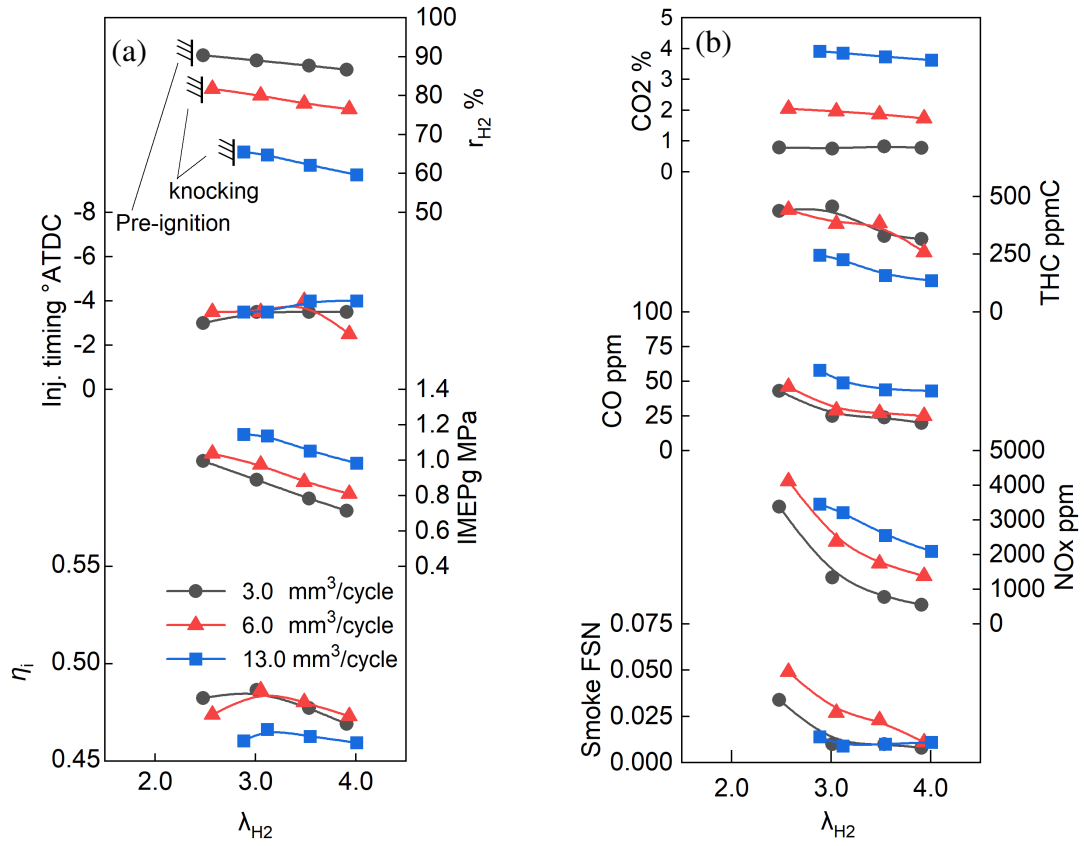


Figure 2.7: Engine performance (a) and emission (b) of HVO under intake pressure, $p_s = 140$ kPa and pilot quantity of 3.0, 6.0, and 13.0 mm³/cycle.

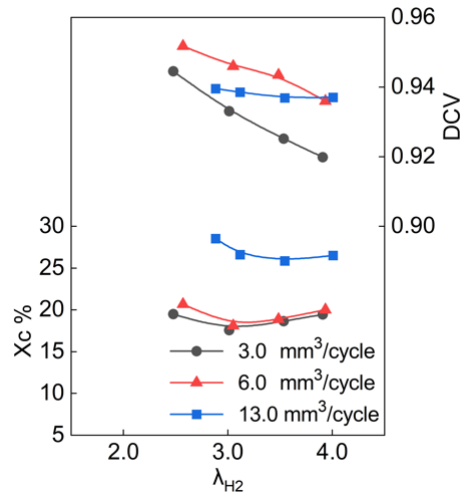


Figure 2.8: DCV and X_c of HVO under intake pressure, $p_s = 140$ kPa and pilot quantity, q_j of 3.0, 6.0 and 13.0 mm³/cycle.

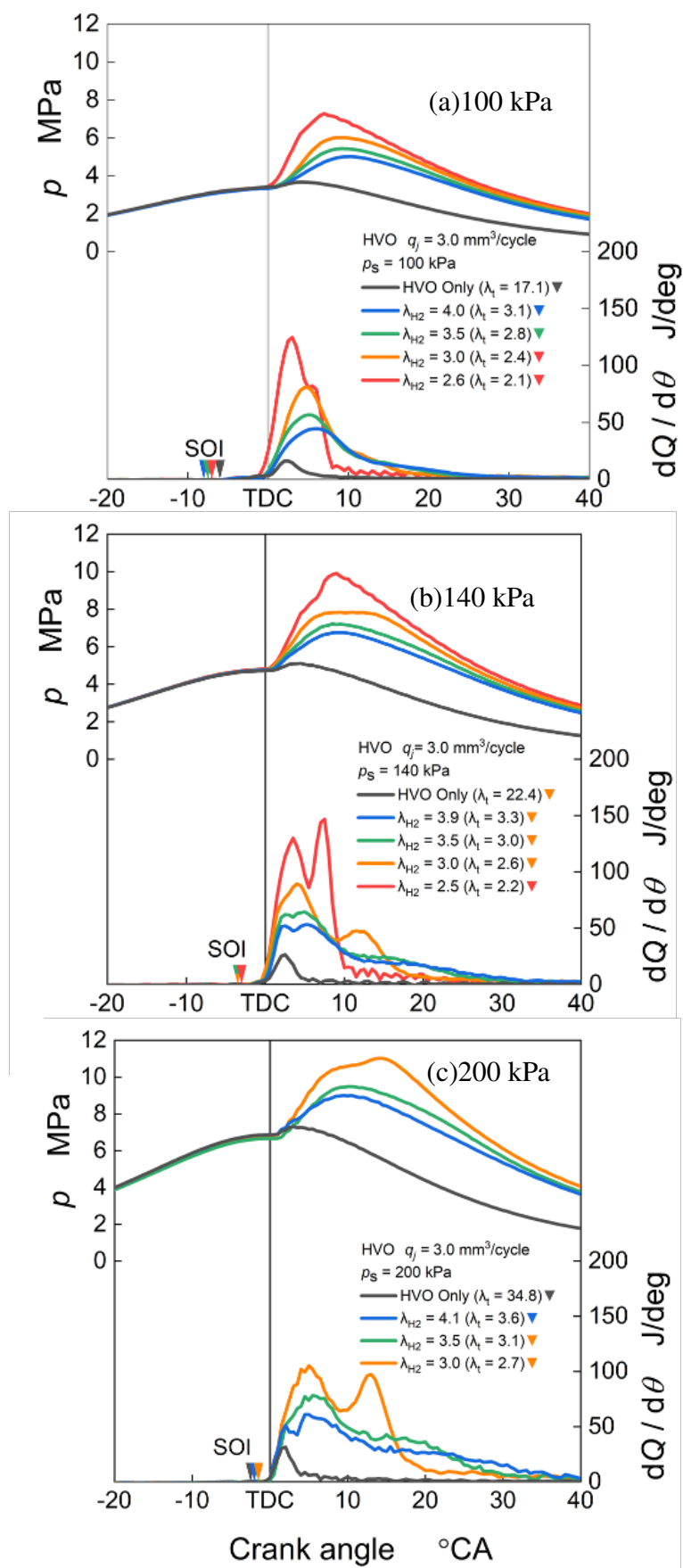


Figure 2.9: In-cylinder pressure, p and heat release rate, $dQ/d\theta$ of HVO pilot quantity of $3.0 \text{ mm}^3/\text{cycle}$, intake pressure, $p_s = 100, 140$, and 200 kPa .

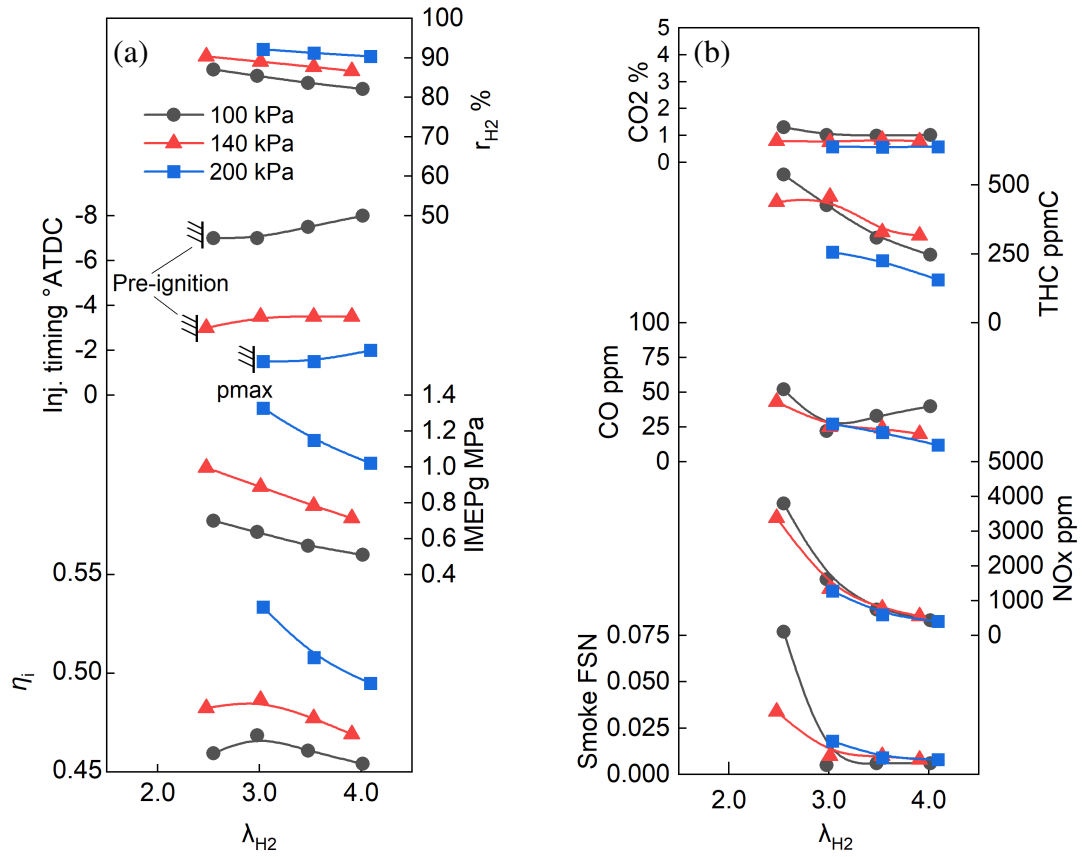


Figure 2.10: Engine performance (a) and emission (b) of HVO pilot quantity of 3.0 mm³/cycle, intake pressure, $p_s = 100, 140,$ and 200 kPa.

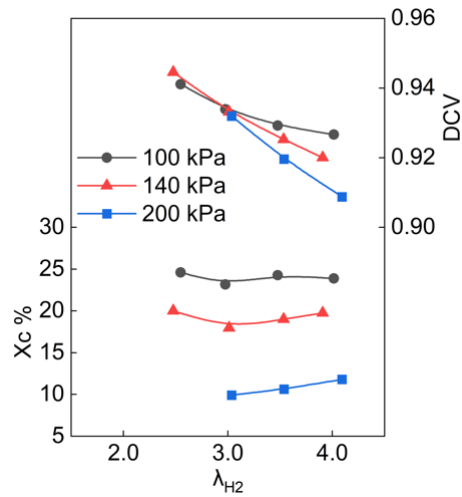
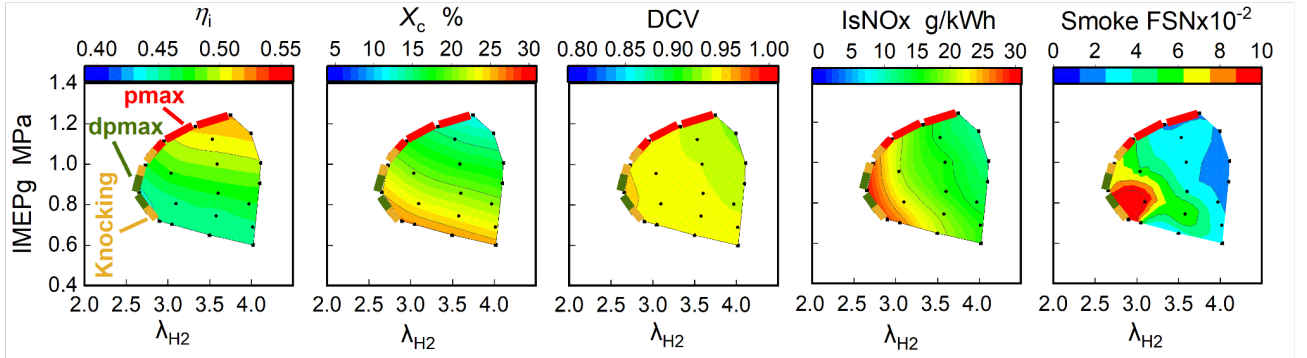
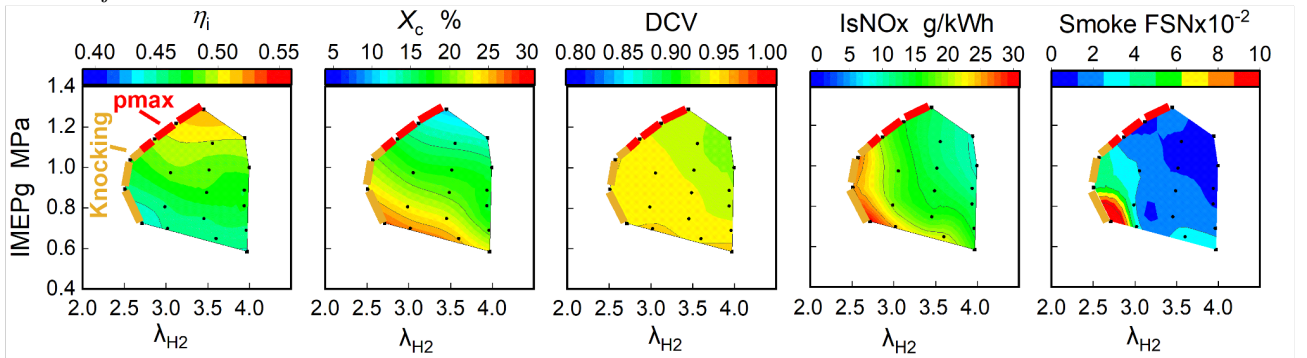


Figure 2.11: DCV and X_c of HVO pilot quantity of 3.0 mm³/cycle, intake pressure, $p_s = 100, 140$ and 200 kPa.

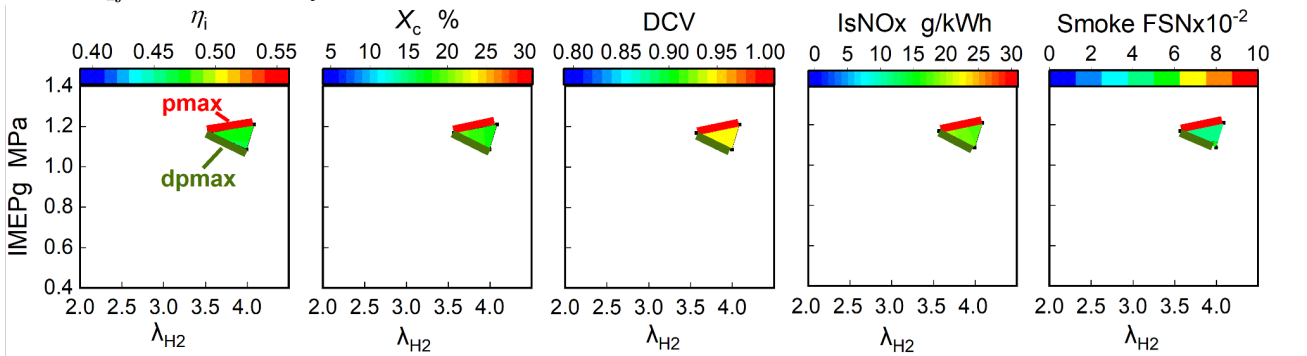
Diesel, $q_j = 5.7 \text{ mm}^3/\text{cycle}$



HVO, $q_j = 6.0 \text{ mm}^3/\text{cycle}$



Diesel, $q_j = 12.4 \text{ mm}^3/\text{cycle}$



HVO, $q_j = 13.0 \text{ mm}^3/\text{cycle}$

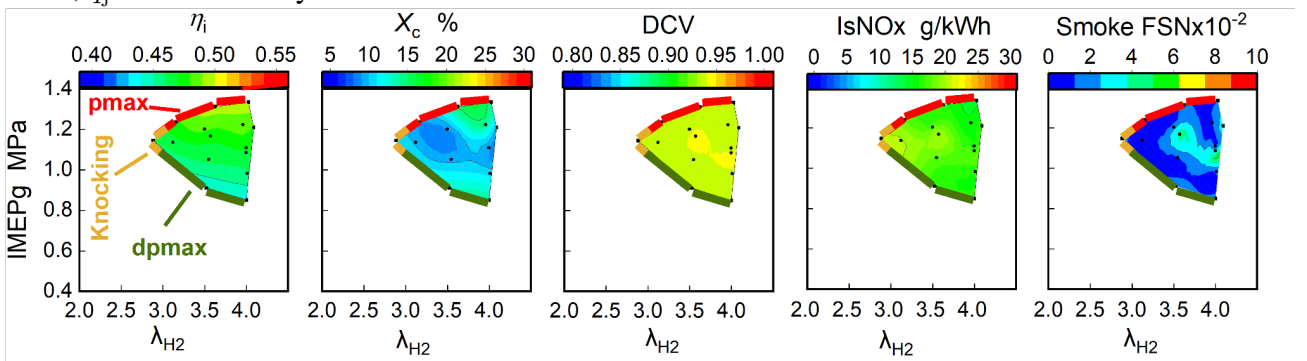


Figure 2.12: Engine performance and emission characteristics for HVO and diesel fuel under varying load and hydrogen excess air ratio.

Tables

Table 2.1: Technical specifications of the test engine.

Engine Type	Direct injection, water cooled, single cylinder diesel engine
Bore × stroke	85 × 96.9 mm
Displacement	550 cc
Compression ratio	14.3
Supercharging	External supercharging
Number of valves	2 intake, 2 exhaust
Pilot fuel injection system	Common rail injection system
Pilot fuel injector nozzle	0.123 mm × 7 (156° spray cone angle)

Table 2.2: Properties of pilot fuels.

	Test method	Unit	HVO	Diesel fuel
Density (15 °C)	JIS K2249-1	kg/m ³	780.4	822.5
Kinetic viscosity (30°C)	JIS K 2283	mm ² s	3.848	3.564
Lower heating value	JPI-5S-65-11	MJ/kg	43.67	42.99
95% distillation point	JIS K 2254	°C	295	358
Cetane number	JIS K 2280-5	-	-	54.5
Cetane index	JIS K 2280-5	-	95	60

*The cetane index of HVO is beyond the scope of this application.

Table 2.3: Details of experimental conditions.

Engine speed	rpm	1800
Intake and exhaust pressure	kPa (a)	100 220
Intake temperature	°C	40
Pilot injection timing	°ATDC	(TDC ignition)
Pilot injection quantity	mm ³ /cycle	HVO: 3.0, 6.0, 13.0 Diesel fuel: 2.9, 5.7, 12.4
H ₂ injection signal timing	°ATDC	-380
λ_{H_2}	-	2.5 4
Temperature at oil filter	°C	80

Chapter 3 Combustion analysis of hydrogen dual fuel mode based on OH* chemiluminescence images

3.1 Introduction

To ensure a reliable operation under hydrogen dual fuel mode, controlling the combustion process and injection strategy is necessary. This could be achieved by acquiring knowledge regarding pilot fuel ignition location, pilot flame progression towards hydrogen, flame propagation location, and pilot fuel spray distribution.

However, combustion process in dual fuel engine is a complicated process; pilot-ignited dual fuel combustion involves a complex transition between pilot fuel autoignition and the premixed-like phase of combustion [1]. This highlights the necessity of optical study of the combustion process in hydrogen dual fuel mode. Therefore, in this Chapter, the visualization of the combustion process in hydrogen dual fuel mode is conducted and discussed to clarify the characteristics of ignition and combustion process in hydrogen dual fuel combustion, aiming to understand the combustion phenomenon in hydrogen dual fuel mode.

In a previous study by Rajasegar et al. [2] on optical analysis in hydrogen-diesel dual-fuel mode, it was observed that the pilot flame extended beyond the pilot fuel jet due to the presence of premixed hydrogen-air mixtures around the jet periphery. Higher hydrogen concentrations intensified this effect, shifting the ignition location and altering the subsequent flame behavior. The study emphasized the need to control pilot injection to establish fuel-rich zones crucial for reliable ignition in hydrogen combustion[2]. Therefore, following this findings, this study examines the effects of varying injection parameters under different hydrogen concentrations. Since pilot injection pressure could influence spray mixing rates and penetration, and the pilot injection quantity could affect the pilot spray momentum and duration, both injection parameters were employed and varied in this study.

Since HVO differs from diesel in properties such as density, kinematic viscosity, and auto-ignition characteristics, it is expected that spray development, auto-ignition, and flame propagation in hydrogen-air premixed mixtures will differ from cases where diesel is used

as the pilot fuel. This was also proven when studying the engine performance and emission results when employing HVO as pilot fuel in Chapter 2. It was observed that HVO exhibited different engine performance, and a wider operable range compared to diesel fuel as pilot fuel, which was attributed to shorter ignition delay properties.

Therefore, to ensure a reliable operation strategy under HVO as pilot fuel in hydrogen dual fuel, the visualization of the combustion process was then conducted under hydrogen-HVO dual fuel mode and compared with hydrogen-diesel dual fuel, under fixed total excess air ratio was also conducted. Initially, to study the combustion process under different hydrogen concentration, visualization of combustion process utilizing diesel as pilot fuel was conducted. The pilot injection parameters such as pilot injection quantities and pilot injection pressure were varied to clarify the injection strategies required to control the combustion. Then, utilizing HVO as pilot fuel the combustion process was clarified while comparing with diesel fuel. Subsequently, the effects of varying pilot injection pressure and pilot quantity were investigated. To achieve this objective, an optically accessible Rapid Compression and Expansion Machine (RCEM) was employed.

3.2 Experimental setup and methods

Fig. 3.1 presents a schematic diagram of the experimental setup. In this study, the combustion was visualized using an RCEM modified from a naturally aspirated four- stroke-cycle single-cylinder diesel engine (Yanmar NFD170). The cylinder head of the RCEM was designed and manufactured independently such that the quartz window and intake/exhaust valve could be attached to it. A driving motor (Mitsubishi Electric SF-PR18.5KW 4P 200V) connected to the flywheel was driven externally, and the crank angle and TDC position were detected using a rotary encoder (MTL MEH-130-1024P) attached to the shaft coupling. By controlling the opening and closing timings of the intake valve and the timing of the fuel injection signal using a timing controller (LabSmith LC880), it was possible to produce a single compression and expansion stroke for a diesel engine. An inverter (Mitsubishi Electric, FR-A820-18.5K-E1) was used to adjust the engine speed while monitoring the crank angle detected by a rotary encoder. A mixture of air and hydrogen was introduced into the combustion chamber through an intake valve connected to a mixing

tank. An absolute pressure transducer (KYOWA Electronic Instruments PABA-500KP) and a dynamic strain measuring instrument (KYOWA Electronic Instruments DPM-711B) were used to measure the pressure in the mixing tank. The pilot fuel was supplied using a common-rail injection system. HVO and commercially available diesel fuel (JIS No. 2) are used as pilot fuels in this study.

The RCEM is a modified diesel engine in which the original intake-exhaust valve and fuel injection systems have been removed. The introduction of intake air/hydrogen was facilitated through an independent intake/exhaust valve and a mixing tank, while fuel injection was achieved via a common rail system, thus ensuring experimentally flexible conditions. When operating the RCEM, these systems were controlled via a timing controller that synchronized with the rotary encoder. During operation, the intake and exhaust valves remained open, and no compression or expansion stroke occurred until the intake temperature reached the desired level. At this point, the timing controller triggered the closure of the intake valve and the injection of fuel at a predetermined crank angle, producing a complete compression and expansion stroke without residual gas effects. The RCEM specifications are summarized in Table 3.1.

To study the combustion behavior of the diesel-hydrogen dual fuel mode, the total excess air ratio, pilot injection pressure, and pilot injection quantity were manipulated. The experimental conditions are listed in Table 3.2. The cranking speed was 900 rpm, and the in-cylinder pressure and temperature were 5 MPa and 700 K, respectively. The pilot injection timing was adjusted to fix the ignition timing at 2°CA for all the conditions. Detailed experimental parameters are provided in Table 3.3. During this study, the injection pressure $p_j = 80$ MPa and injection quantity $q_j = 6$ mm³ were referred to as standard conditions. First, the effect of the hydrogen concentration was studied by manipulating the total excess air ratio λ_{total} at 3 and 4 with the standard pilot injection. Additionally, the influence of pilot injection pressure was studied by reducing the injection pressure to 40 MPa, and increasing the injection pressure to 120 MPa. Finally, the effect of pilot injection quantity was observed by adjusting the pilot fuel injection volume to 3, 6, and 13 mm³. Two photography methods were employed in this study: a direct imaging to clarify the liquid phase of pilot spray and luminous flame region and an OH* chemiluminescence imaging to clarify the

development and propagation of high temperature region (flame). For direct imaging, a continuous green solid-state laser (Civil Laser LSR532N-5W) with a wavelength of 532 nm was used to capture Mie-scattered light from the diesel spray, recorded using a high-speed color camera (Photron Fastcam SA-Z). For OH* chemiluminescence imaging, which focuses on wavelengths around 310 nm, a bandpass filter (ASAHI SPECTRA UZ0310) and an ultraviolet transmission lens (Nikon UV-Nikkor) were used. Due to the low intensity of OH* chemiluminescence, an image intensifier (HAMAMATSU C10880-03F) was attached to amplify the signal. Imaging conditions are detailed in Table 3.3. The visualization area captured three pilot spray plumes, as shown in Fig. 3.2. The visualization window was aligned so that the spray axis of one of the seven injector nozzles was centered, allowing the capture of three pilot fuel spray plumes, including two adjacent pilot spray plumes. The middle pilot spray plume could be observed being injected in an upward direction. In conducting the visualization of combustion process in hydrogen-diesel dual fuel mode, the engine oil has contacted the piston under certain condition, and this has been a problem, since the engine oil autoignite and ignite the hydrogen-air mixture under certain condition. This interrupt the combustion process, and the experiment which were interrupted by oil ignition were excluded. As a measure, this time when visualizing the hydrogen-HVO dual fuel combustion process, using the same RCEM, with the RCEM has been positioned vertically as a measure to prevent engine oil from contacting the piston. To facilitate the photography process, a mirror was employed to reflect the visualization window, which is facing upwards. No change was made onto the RCEM specifications, hence the specifications of the RCEM used in this experiment could be obtained from Table 3.1. HVO mainly served as the pilot fuels in this study. To compare the HVO combustion process with conventional diesel (JIS no. 2) as pilot fuel combustion process, hydrogen-diesel dual fuel combustion process was also visualized as reference. To compare the effects of different pilot fuels onto combustion, the injection quantity was adjusted so that the heat input for HVO and diesel was the same. The experimental conditions for this study are shown in Table 3.4. The in-cylinder pressure at TDC was set to 5 MPa, the temperature at TDC was fixed at 700 K, and the total excess air ratio was fixed at 3.0. The standard condition used diesel as the pilot fuel, with an injection pressure $p_j = 80$ MPa and a pilot injection quantity $q_j = 6.0$ mm³. For the corresponding HVO

condition, the pilot injection quantity was set to $q_j = 6.27 \text{ mm}^3$ to match the injected energy of the diesel fuel. Additionally, to examine the effects of injection pressure, experiments were conducted at a reduced injection pressure of 40 MPa. Furthermore, to analyze the combustion process at lower injection quantity, experiments were also conducted with half the HVO injection quantity at $q_j = 3.14 \text{ mm}^3$. Under all conditions, the pilot injection timing was adjusted to maintain a constant ignition timing of 2°CA . In this study, the ignition timing is defined as the crank angle at which the tangent line to the rapid increase of heat release rate intersects the crank angle axis. Three experiments were conducted under each condition. All the results presented below are the selected representative single-cycle data, which are the closest results to the average of the three combustion trials under each condition.

Regarding combustion visualization, as it was found in hydrogen-diesel dual fuel visualization study that direct imaging did not exhibit any luminous flame, only OH^* chemiluminescence imaging has been employed as visualization method in this study. The OH^* chemiluminescence emitted from high temperature regions during combustion was captured through a mirror reflecting the quartz window installed in the cylinder head. The imaging condition for this study is almost similar as OH^* chemiluminescence imaging conditions in Table 3.3, except that the image intensifier (I.I.) gain has been reduced from 900 to 880 to reduce overexposure under high luminosity conditions and noises, ensuring better image quality. As the RCEM has been vertically positioned, the outline of the visualization range used to capture combustion and spray behavior in the cylinder is shown in Fig.3.3. The middle pilot spray plume could be observed being injected in a direction from left to right. The left half of the visualization range covers the combustion chamber cavity, while the right half corresponds to the squish area.

3.3 Visualization results of combustion process in hydrogen-diesel dual fuel mode comparing hydrogen concentration

3.3.1 Visualization of combustion process using direct photography

As expected, for the results of direct photography, only the diesel spray could be visualized because the flame was almost non-luminous. It is thought that under the present

conditions, the ignition delay is very long, thus providing sufficient mixing time for the pilot fuel with air and making the flame more premixed and non-luminous. However, the development process of the pilot spray was captured because the laser irradiated the region where the pilot fuel was injected. To clarify the relationship between the spray development and combustion, direct images under each condition were investigated. Fig. 3.4 presents the effect of the pilot injection pressure on the development of the pilot spray obtained from direct photography at a total excess air ratio $\lambda_{\text{total}} = 4.0$. The bright green region visible before -4°CA was the dense region of the pilot spray. The spray tip penetration increased with the pilot injection pressure, and the injection duration decreased. The ignition timing was set at 2°CA under all conditions, thus indicating that ignition occurred after the end of the injection. No luminous flames were observed after the ignition.

Fig. 3.5 presents the images obtained using direct photography with varying pilot-injection quantities. The spray development process was similar in the early stage of injection; however, the spray tip penetration was greater for $q_j = 13 \text{ mm}^3$ corresponding to a longer injection duration. However, the injection ended before ignition occurred under both conditions.

3.3.2 Effects of H_2 concentration onto combustion process

To understand the combustion process when using diesel fuel as a pilot fuel, experiments were conducted under standard condition of $p_j = 80 \text{ MPa}$ and an injection volume of $q_j = 6 \text{ mm}^3$. Fig. 3.6 presents the in-cylinder pressure and heat release rate (HRR) obtained. The start of injection (SOI) was adjusted such that the ignition timing occurred at 2°CA for all the conditions. The duration of injection (DOI) for this work is approximately $2\text{--}3^\circ\text{CA}$, depending on the injection conditions. Ignition occurred after the end of the injection for all the conditions. As observed, the pilot injection started earlier for $\lambda_{\text{total}} = 3$. This is owing to the lower oxygen concentration in the premixture for a lower λ_{total} and the chemical inhibition effect of hydrogen, which causes an elongated ignition delay of the pilot spray [2]. In this study, the ignition delay is defined as the time from the SOI to the ignition timing. The HRR peak—either a single peak or dual peak—indicates significant information in the DDF study. A dual-peak heat release rate (HRR) indicates that

the combustion of the airhydrogen mixture (second peak) occurs after the ignition of the pilothydrogen mixture(first peak) [3]. This behavior was observed at an HRR of $\lambda_{\text{total}} = 4$. As λ_{total} decreased to 3 with the addition of more hydrogen, the HRR shifted toward a single peak with a higher maximum heat release and shorter combustion duration. At $\lambda_{\text{total}} = 3$ and 4, the excess air ratios for the hydrogen fuel alone were 3.6 and 5.1, respectively. Therefore, the richer hydrogen was ascribed to a shorter combustion duration, in which the high flame propagation speed of hydrogen increased the HRR [3]. Regarding the dual peaks observed, generally, under continuous cycle engine operation, the heat release rate can exhibit three heat release rate peaks as reported by Zhou et al. [4] and Akcay et al. [5], the hydrogen dual fuel combustion in diesel engine can exhibit three heat release rate peaks. The first peak is the heat release induced by pilot ignition, and the second peak is attributed to the combustion of premixed air-hydrogen. The third peak attributed to autoignition at end gas region i.e. PREMIER combustion [6] or knocking could not be observed under this study, due to lack of residual heat effect and low combustion temperature under this RCEM operation.

Fig. 3.7 presents the combustion process with a standard pilot injection for $\lambda_{\text{total}} = 3$ and 4. In the images, the red dashed curve marks the edge of visualization window, and the white curve represents the edge of piston bowl. Under the condition of $\lambda_{\text{total}} = 4$ a bright region rapidly appears near the edge of the piston bowl after the ignition begins and then spreads alongside the wall of the piston bowl until approximately 3.5°CA which is followed by the expansion of the low-brightness area from these regions. It should be noted that it was impossible to distinguish between the combustion regions of diesel fuel and hydrogen in this work. However, when estimating the distribution of diesel fuel owing to spray, it can be assumed that the high-brightness area extending up to approximately 3.5°CA is a region with a high local equivalence ratio wherein the diesel fuel mainly burns, and there is a high possibility that the low-brightness area that spreads beyond that region is a high-temperature area owing to the hydrogen premixed combustion. Compared to the condition of $\lambda_{\text{total}} = 3$, the intensity of OH^* is weaker at $\lambda_{\text{total}} = 4$ and hardly spreads beyond 11.5°CA . As can be observed from the HRR behavior in Fig. 3.6, the heat release continued even at subsequent crank angles, suggesting that the flame temperature at that time was low; hence, the OH^* chemiluminescence intensity was low. Meanwhile, at $\lambda_{\text{total}} = 3$ the combustion regions of

the diesel fuel and hydrogen appear almost simultaneously, corresponding to the single-stage HRR in Fig. 3.6. In addition, a region with a high OH* intensity can be observed even at the latter stage of combustion, which suggests that the flame temperature is higher.

3.3.3 Effects of pilot injection quantity onto combustion process

The effect of the pilot injection quantity under hydrogen-diesel dual fuel combustion was examined. The HRR under each condition is compared with the injection amount in Fig. 3.8. It can be observed from the conditions of $\lambda_{\text{total}} = 3.0$ that the shape of the HRR changes significantly depending on the injection quantity. Compared to the condition of $q_j = 6 \text{ mm}^3$, when the injection quantity was increased to 13 mm^3 the HRR increased sharply, the peak was high, and the combustion duration was shortened. However, when the injection quantity was reduced to 3 mm^3 , the HRR of the initial stage, which was mainly caused by the combustion of diesel fuel, could not be observed, and the HRR decreased with a longer combustion duration. This is because, as the quantity of pilot injection decreased, the ignition delay increased, resulting in an extended mixing period of air and pilot fuel within the cylinder, consequently leading to slower combustion and lower peak of heat release rate. The longer ignition delay is attributed to higher spray dilution rate of pilot fuel at lower pilot quantity. A similar tendency was observed under the conditions of $\lambda_{\text{total}} = 4.0$ and $p_j = 80 \text{ MPa}$.

Fig. 3.9 presents OH* images under the condition of $\lambda_{\text{total}} = 3.0$. When the injection amount was increased to 13 mm^3 the bright region rapidly spread from approximately 3.5°CA along the edge of the piston bowl. The expansion of the high-brightness region corresponds to the rapid increase in the HRR in Fig. 3.10, and it is considered that this combustion mainly consists of diesel fuel. In addition, under these conditions, the burnt gas with momentum spreads to the squish region owing to its long injection duration.

When the injection quantity was decreased to 3 mm^3 a high-temperature region appeared at the lower left, which probably originated from the left spray at 1.5°CA and the region with low brightness propagated slowly outwards to the edge of the piston bowl until approximately 13.5°CA . Under this condition, with a hydrogen mixture and a low excess air ratio, the propagation of the region with low brightness is thought to be mainly

composed of hydrogen; hence, the temperature is low. At 12.5°CA, another bright region appeared at the edge of the piston and slowly spread outwards. It is believed that the unburned hydrogen mixture at the end of the piston bowl was ignited owing to the high temperature and pressure during the expansion of the combustion gas. Compared to the standard condition under constant $\lambda_{\text{total}} = 3.0$, it could be observed that combustion occurs mainly along the piston bowl wall under standard condition, while the combustion under reduced pilot fuel quantity condition occurs upstream close to injector. These findings could validate the efficiency results under engine operation condition, which exhibit lower thermal efficiency for higher pilot quantity. At high pilot quantity, the pilot fuel has high spray momentum, and this resulted in combustion occurring the downstream area, close to piston bowl wall.

3.3.4 Effects of pilot injection pressure onto combustion process

Based on the above results regarding varying pilot quantities, it can be concluded that lower pilot quantities resulted in the pilot fuel becoming more diluted with air. This led to decreased OH* luminosity due to lower combustion temperatures, consequently causing the combustion to progress more slowly. As lower pilot injection pressure is hypothesized to enhance the dilution rate of the pilot fuel, the effect of pilot injection pressure is then examined in this section. Fig. 3.10 presents the in-cylinder pressure and HRR at the pilot injection pressures of 40, 80, and 120 MPa for both λ_{total} . The pilot injection pressure hardly exhibits any effect on the ignition delay, which agrees with the findings of Lyn regarding the effect of the injection pressure on the ignition delay [7]. As the injection pressure decreased, the diesel mixing rate decreased and the local equivalence ratio increased considerably, which provides favorable conditions for the ignition of the pilot spray. This resulted in a rapid increase and a higher peak for initial combustion. As the pilot injection pressure increased, the dual-peak HRR became more significant at $\lambda_{\text{total}} = 4$.

Fig. 3.11 presents the OH* images with varying pilot injection pressures under the conditions of $\lambda_{\text{total}} = 3.0$ and $q_j = 6 \text{ mm}^3$. When the pilot injection pressure was 40 MPa, the bright region rapidly spread from upstream of the diesel spray in the direction of the spray immediately after ignition. Subsequently, the diesel spray shape was maintained for a long time until 9.5°CA. Under low-injection-pressure conditions, the momentum of

the spray was small, and the mixing rate was low. Consequently, ignition occurred in the middle of the spray development, and a flame was formed corresponding to the diesel-air mixture distribution. When the pilot injection pressure was increased to 120 MPa, ignition occurred near the edge of the piston bowl, from where the low-luminosity region spread to the entire visualization range by 9.5°CA. Compared with the low-luminosity-region expansion at 80 MPa in Fig. 3.3, the low-luminosity region expansion at 120 MPa was slower, and the expansion continues until 9.5°CA. Owing to the increased spray tip penetration and momentum, ignition occurred at the edge of the piston bowl. This was followed by the expansion of the ignition region with low luminosity until 9.5°CA, which is thought to be the expansion of the diesel-hydrogen premixed combustion. This is because when the pilot injection pressure is higher, the injection duration is shorter, and ignition occurs after a certain time has passed since the end of injection. This resulted in enhanced homogeneity of the diesel mixture in the combustion chamber, which consequently led to a leaner local flame, thereby causing a gradual expansion of the region with low luminosity. This corresponds to a single low HRR peak and a long combustion period at $p_j = 120$ MPa, as observed in Fig. 3.10(b).

3.4 Visualization results of combustion process utilizing HVO as pilot fuel

3.4.1 Effects of pilot fuel types onto combustion process

To investigate differences in the combustion process based on the type of pilot fuel, HVO was then used as a pilot fuel. The pilot injection pressure was set at 80 MPa, and the injection quantity of HVO was adjusted to 6.27 mm³ to match the injected energy of 6 mm³ of diesel. Fig. 3.12 shows the in-cylinder pressure and heat release rate versus crank angle, comparing HVO and diesel fuel as pilot fuels in hydrogen dual fuel mode. To achieve an ignition timing of 2°CA, HVO is injected at approximately -5°CA, around 5°CA, later than when diesel fuel as pilot fuel. This is due to higher cetane properties of HVO compared to diesel fuel. Similar to diesel, a weak heat release attributed to low-temperature oxidation was observed around 0°CA to 2°CA. However, from 2°CA, rapid heat release rate with

sharper slope was observed for HVO until 5°CA. The peak heat release rate was higher with HVO, but it declined rapidly after 8°CA, whereas for diesel, the decline was more gradual, extending to 20°CA. The shorter ignition delay of HVO resulted in a higher concentration of the fuel-air mixture at the point of ignition, leading to a higher heat release rate. Furthermore, HVO exhibits dual peak heat release rate compared to diesel fuel. To investigate further regarding this matter, the combustion process is studied.

Fig. 3.13 provides insight into the combustion process by showing the visualization images of combustion in hydrogen-HVO dual fuel mode within the visualization range using OH* chemiluminescence at different crank angles. At 2°CA, luminescence began near the base of the spray on the left side of the image. The high-intensity luminescence expanded downstream along the spray axis, reaching the piston bowl wall by 5°CA. This luminescence was primarily caused by the autoignition of the pilot fuel. After 2°CA, weaker luminescence spread from the bright regions toward the squish area, indicating flame propagation into the hydrogen-air mixture surrounding the spray. This corresponds with the appearance of the second peak of heat release rate from 6°CA in Fig. 3.12.

Compared with visualization images when diesel was used as the pilot fuel in Fig. 3.7(b), HVO combustion exhibited ignition kernels with spray plume shapes attributed to pilot fuel combustion on the pilot fuel spray axis. This is owing to the relatively short ignition delay of HVO, which results in a lower leaning out of the pilot fuel, creating locally rich regions in the spray regions. This consequently increases the luminescence intensity during HVO combustion. Thus, the broader distribution and high-intensity region near the spray base in HVO combustion acts as an ignition source, accelerating hydrogen combustion and shortening the overall combustion duration.

3.4.2 Effects of low HVO pilot injection quantity onto combustion process

Based on the results in subsection 3.3.3, as the pilot injection quantity was small, ignition occurred near the injection nozzle. As this could improve engine efficiency through reduced cooling loss, under HVO as pilot fuel condition, the HVO injection quantity is reduced to 3.14 mm³ at a pilot injection pressure of 80 MPa. Fig. 3.14 illustrates the

in-cylinder pressure and heat release rate, with the results for 6.27 mm^3 injection provided for comparison. The hydrogen mixing ratio was increased roughly from 86% to 93% when the injection quantity was reduced. This information can be obtained from Table 3.4. However, reducing the pilot injection quantity did not significantly affect the ignition delay but resulted in a slower increase in the heat release rate and a lower peak. A single peak of heat release rate is observed, with delayed maximum peak timing occurring at approximately 8°CA .

Fig. 3.15 shows the combustion images obtained with 3.14 mm^3 of HVO injected as pilot fuel. It is demonstrated here that between 2.5°CA and 6°CA , luminescence along the spray axis was attributable to the ignition of the pilot fuel. However, the luminescence intensity was lower and luminescence region were smaller than with 6.27 mm^3 pilot quantity in Fig. 3.13. The reduced pilot quantity led to a smaller spray region, resulting in a higher spray dilution rate. This resulted in a leaner local pilot flame, which consequently led to a lower flame temperature. This explains the low heat release rate and extended combustion duration in Fig. 3.14.

After 5°CA , weak luminescence, attributed to hydrogen combustion, propagated around the pilot flame. When comparing visualization images at 6.27 mm^3 pilot quantity in Fig. 3.13, the luminescence from hydrogen premixed luminesce intensity was smaller than when a larger quantity of pilot fuel was utilized. The smaller autoignition flame in the pilot fuel resulted in slower continuous autoignition through the hydrogen-air mixture, despite the lower excess air ratio. Consequently, hydrogen combustion required additional time to propagate throughout the combustion chamber. Meanwhile, at 6.27 mm^3 pilot quantity, the flame with high momentum propagated rapidly and reached the piston bowl wall at approximately 5°CA , subsequently proceeding to the squish area.

3.4.3 Effects of low HVO pilot injection pressure onto combustion process

In subsection 3.3.4, when the pilot injection pressure was decreased, the high-temperature region expanded conically immediately after ignition. This resulted in thick and dense flame formed in the shape of the spray jet. This could possibly inhibit the pilot fuel dilution observed when the pilot quantity is decreased above. Moreover, the

shorter ignition delay of the HVO resulted in ignition kernels between the piston bowl wall and nozzle, leading to a rapid heat release rate. To investigate the effects of further reducing ignition delay, the pilot injection pressure was decreased to 40 MPa using HVO as pilot fuel. Fig. 3.16 illustrates the in-cylinder pressure and heat release rate. However, only slight change is observed in the ignition delay and heat release peak. The reduced pilot injection pressure also exhibits dual peak of heat release rate which indicates dual stage of combustion. The slightly delayed injection timing (elongated ignition delay) resulted in slightly higher peak of heat release rate, particularly during the first stage of combustion in which pilot fuel is mainly combusted. The second stage of combustion exhibits slightly reduced maximum peak when the pilot injection pressure is reduced.

Fig. 3.17 shows the combustion images obtained with HVO at 40 MPa. From 2°CA to 5°CA, the intense luminescence region propagated along the spray development with a broader distribution, comparable to the 80 MPa case in Fig. 4.6. The high intensity suggests that a lower pilot injection pressure prevented significant leaning of the pilot fuel, resulting in relatively high-temperature combustion. This resulted in rapid heat release during the pilot fuel combustion; hence, a slight increase in the peak was observed during the single-stage combustion. The broader pilot distribution resulted in the combustion of the pilot close to the premixed hydrogen; therefore, the dual peak of the heat release rate tended to overlap with each other; hence, a lower peak was observed from the second peak of the heat release rate.

3.5 Conclusion

To clarify the combustion process, focusing on combustion location and flame formation in hydrogen dual-fuel mode, the visualization of combustion process were performed using OH* chemiluminescence imaging and an RCEM. Using diesel as pilot fuel, the effect of pilot injection pressure, and pilot injection quantity were varied, under different hydrogen concentration. Due to different ignition delay behavior, the combustion process using HVO as pilot fuel was also studied by varying pilot injection pressure and pilot quantity at constant hydrogen concentration. The following conclusions could be drawn from this study.

1. The high-temperature region caused by the combustion first appeared at the periphery

of the piston bowl and spread along its wall. The ignition delay was relatively long, and no luminous flames were observed in the direct imaging. Therefore, the combustion zones of the diesel and hydrogen could not be distinguished. However, based on the estimated distribution of the diesel fuel, it is assumed that the high-temperature region from the late combustion stage spread mainly owing to premixed hydrogen combustion.

2. A decreased total excess air ratio with a constant pilot injection reduced the combustion duration, and the HRR tended to shift from a dual peak to a single peak.
3. When diesel fuel is employed as the pilot fuel, the pilot spray ignites after impacting the piston bowl, due to long ignition delay, while HVO exhibits a shorter ignition delay, resulting in the ignition of the pilot spray prior to wall impact, with flame propagation proceeding from the developing pilot spray flame to the hydrogen premix.
4. When the pilot injection quantity was small, ignition with small pilot flame occurred near the injection nozzle, due to lower spray momentum but the dilution rate increases, and combustion progress slowly, with low peak heat release rate. When the pilot injection quantity is increased, the injection duration increased, and the burned gas with momentum spread to the squish area.
5. When the pilot injection pressure was decreased, higher heat release rate peak with shorter combustion duration was observed. The pilot fuel dilution is inhibited, resulting in high-temperature region expanded conically in the shape of spray jet immediately after ignition. This resulted in thick and dense flame formation. With HVO, the slight decrease in ignition delay resulted in broader flame formation. However, when the pilot injection pressure was high, the high-temperature region spread along the bowl wall, and the intensity of OH* was relatively weak because ignition occurred well after the end of the injection.
6. Visualization studies of the combustion process highlight a strategy for high efficiency: concentrate combustion at the center of combustion chamber and form a thick, dense flame.

When comparing the engine performance results with the visualization findings,

it is evident that combustion occurring near the cylinder wall during diesel pilot combustion, and the higher spray momentum which extends spray into the squish area at higher pilot quantities, contributes to increased cooling losses, thereby negatively impacting thermal efficiency.

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Figures

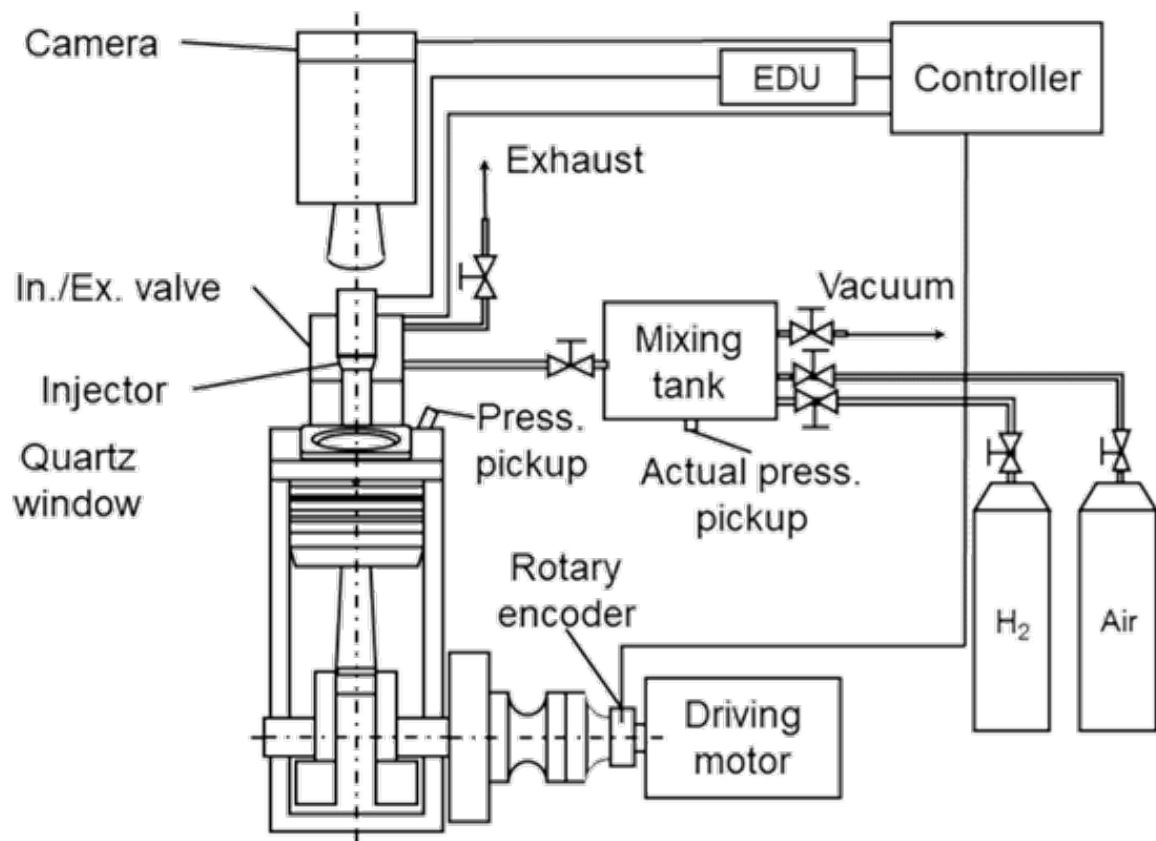


Figure 3.1: Schematic diagram of the experimental setup.

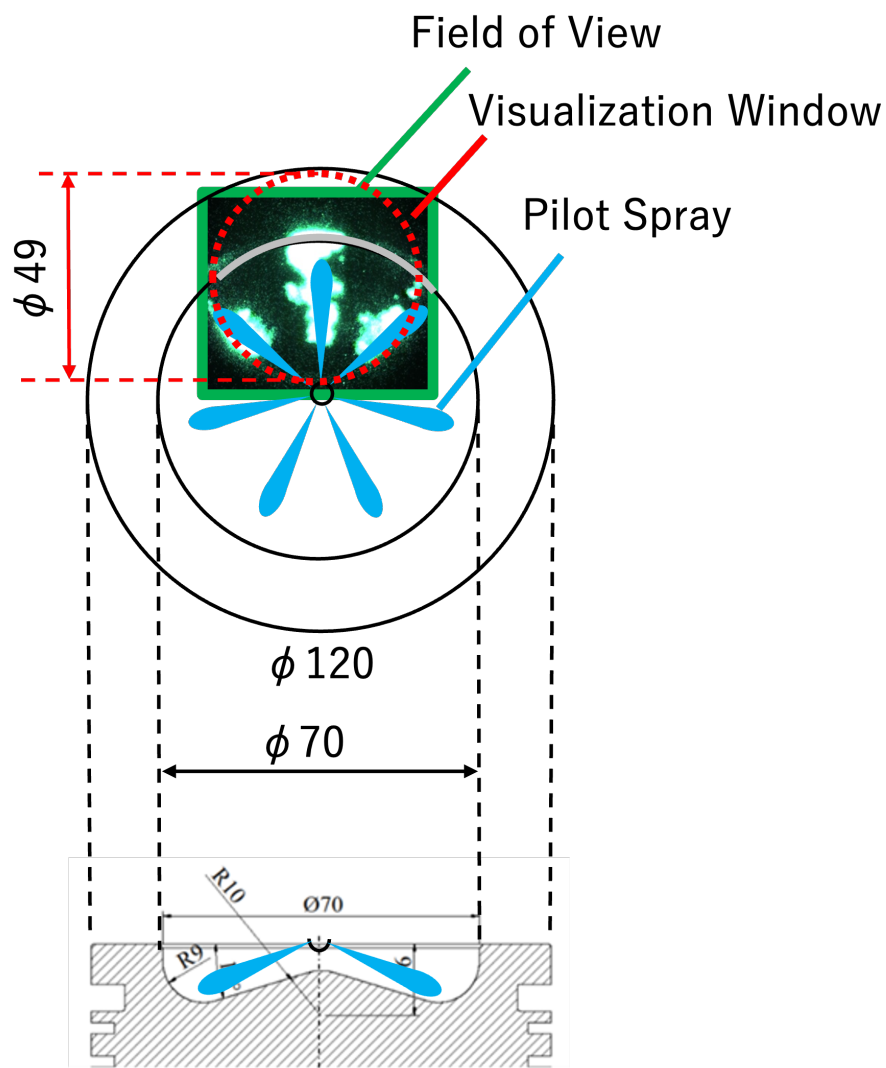


Figure 3.2: Schematic of visualization area before when studying hydrogen diesel dual fuel mode.

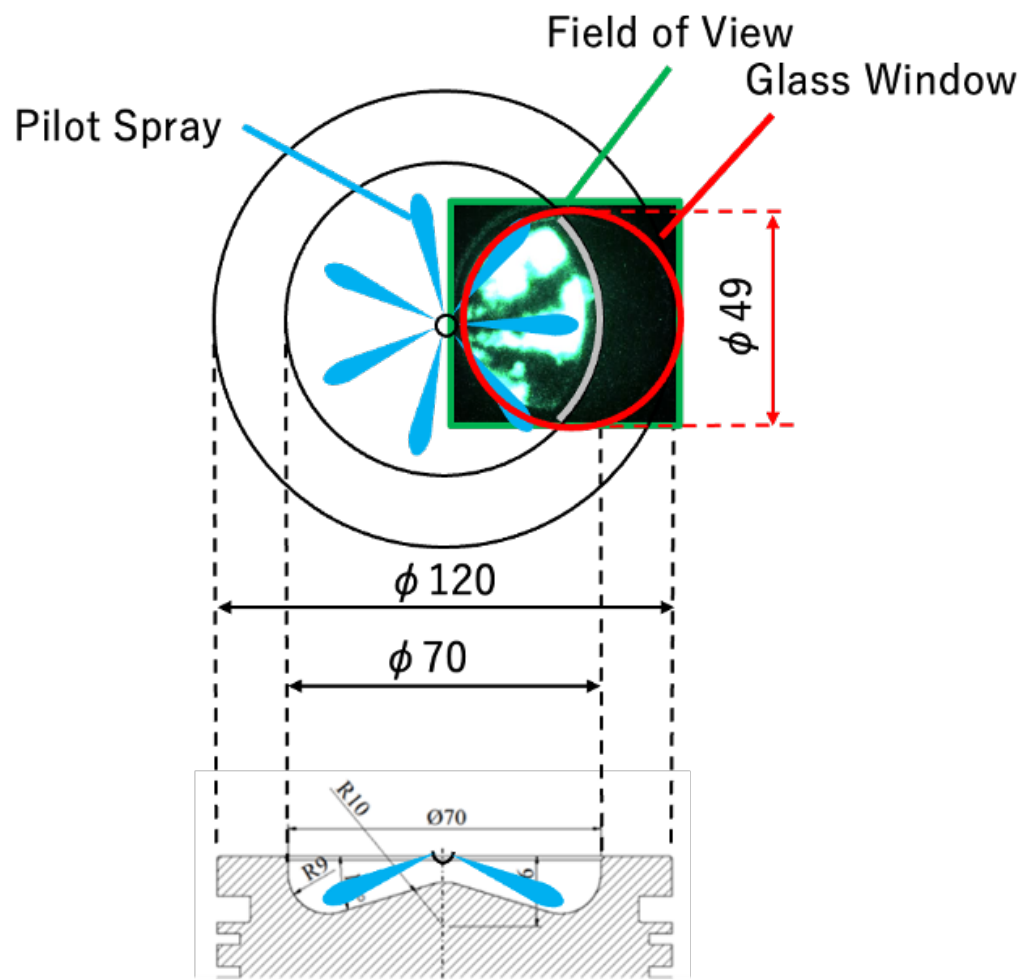
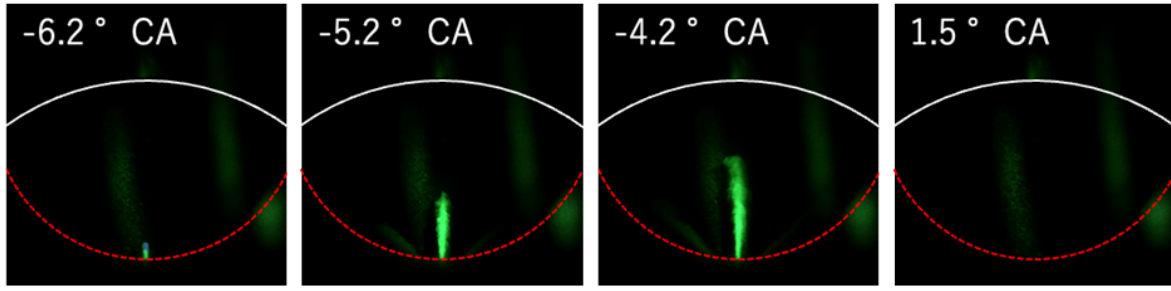
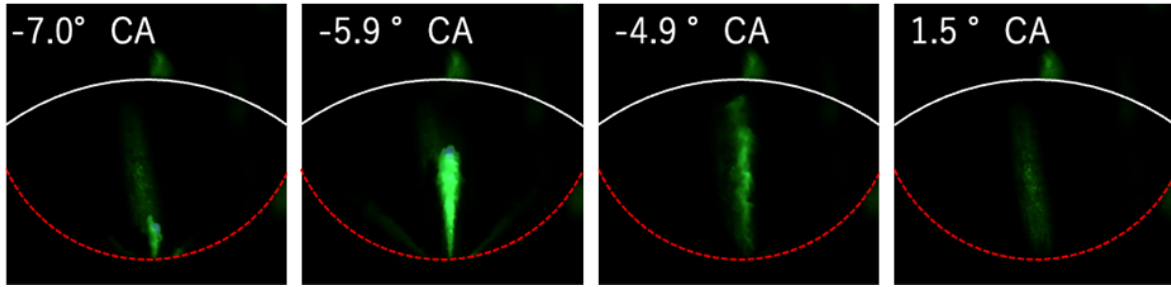


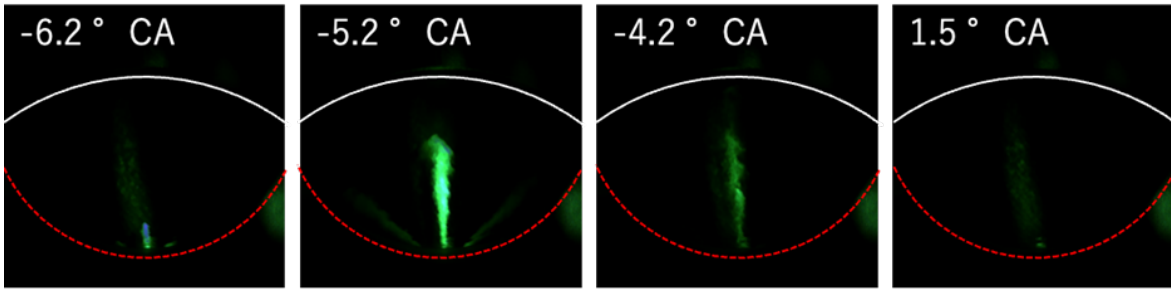
Figure 3.3: Schematic of visualization area after the RCEM was vertically positioned.



(a) $\lambda_{\text{total}} = 4.0$, $p_j = 40 \text{ MPa}$, $q_j = 6 \text{ mm}^3$



(b) $\lambda_{\text{total}} = 4.0$, $p_j = 80 \text{ MPa}$, $q_j = 6 \text{ mm}^3$



(c) $\lambda_{\text{total}} = 4.0$, $p_j = 120 \text{ MPa}$, $q_j = 6 \text{ mm}^3$

Figure 3.4: Visualized images using direct photography with respect to pilot injection pressure.

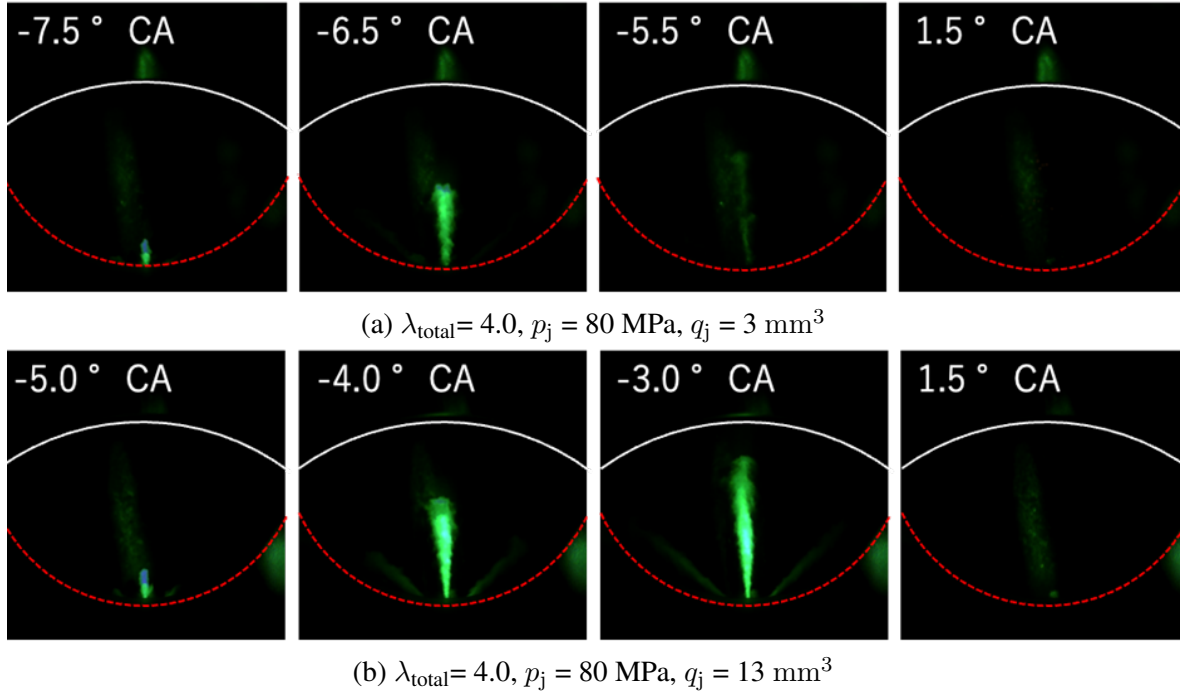


Figure 3.5: Visualized images using direct photography with respect to pilot injection quantity.

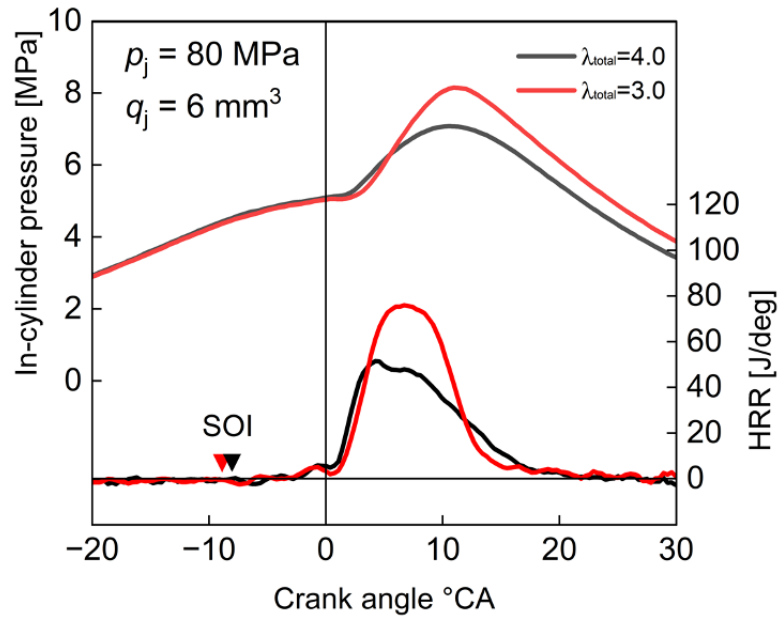
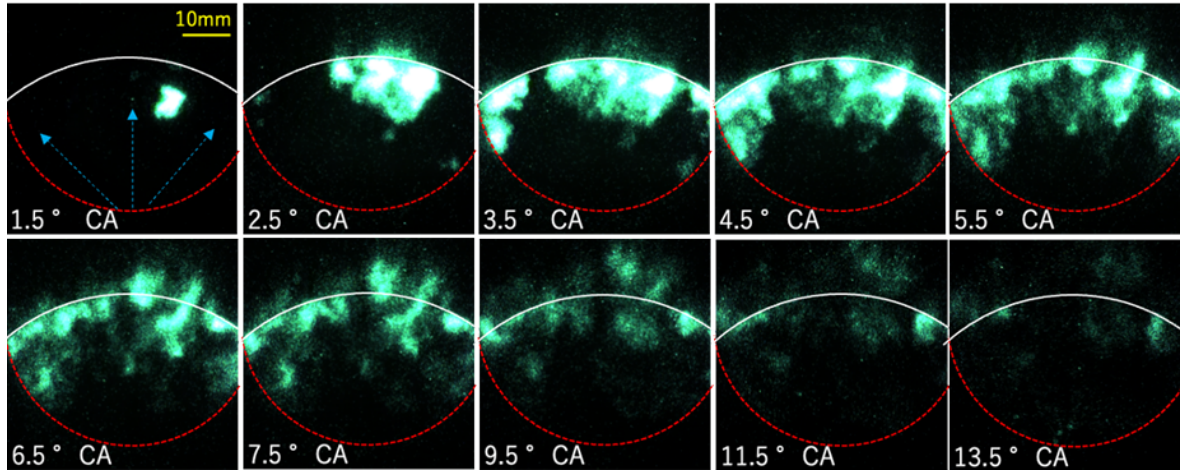
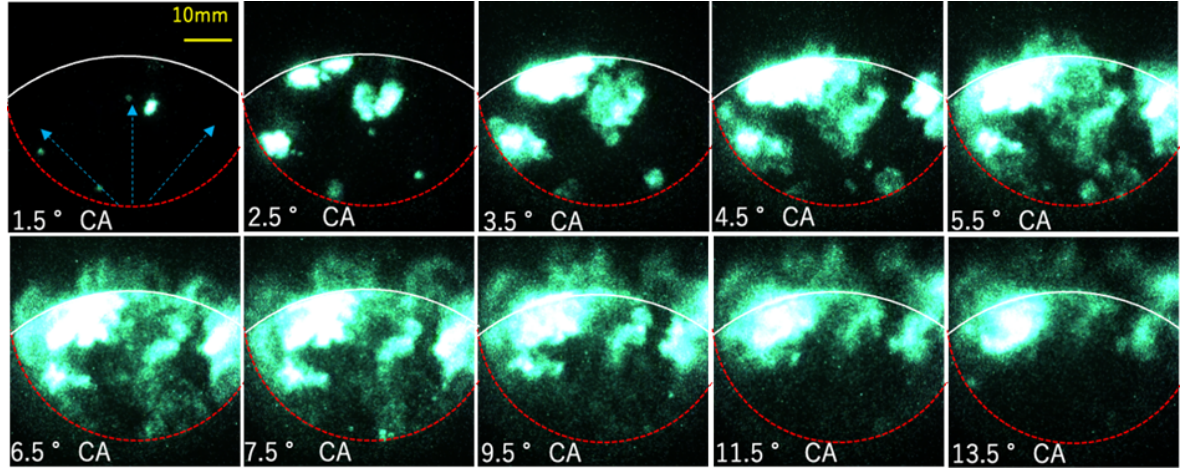


Figure 3.6: The effects of λ_{total} on in-cylinder pressure and heat release rate under standard condition.

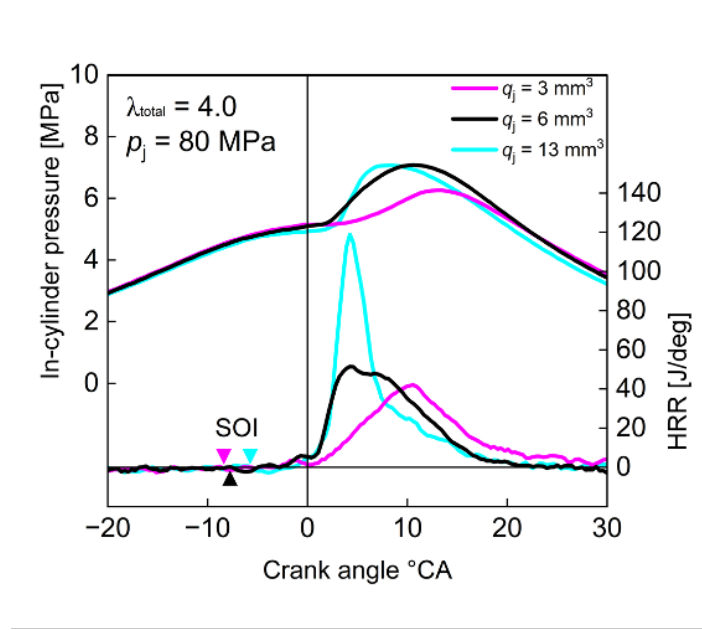


(a) $\lambda_{\text{total}} = 4.0$ ($p_j = 80$ MPa, $q_j = 6$ mm³)

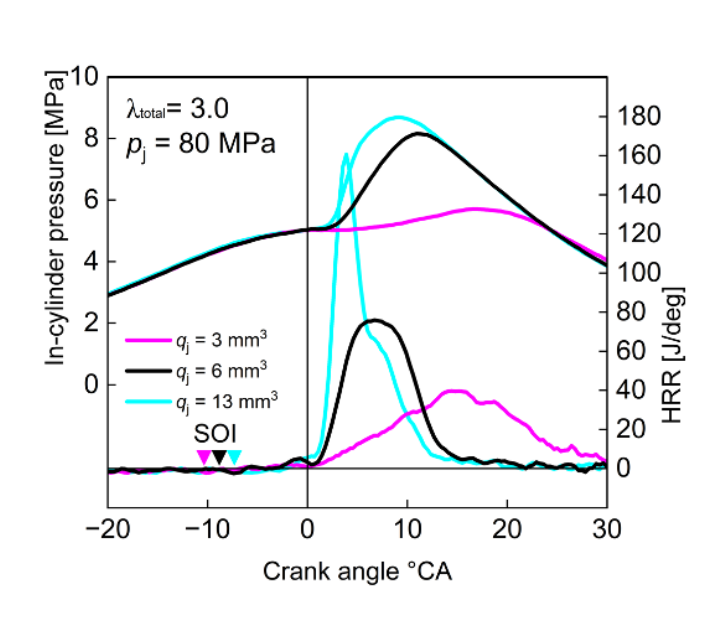


(b) $\lambda_{\text{total}} = 3.0$ ($p_j = 80$ MPa, $q_j = 6$ mm³)

Figure 3.7: Visualized images using OH* photography with respect to λ_{total} .

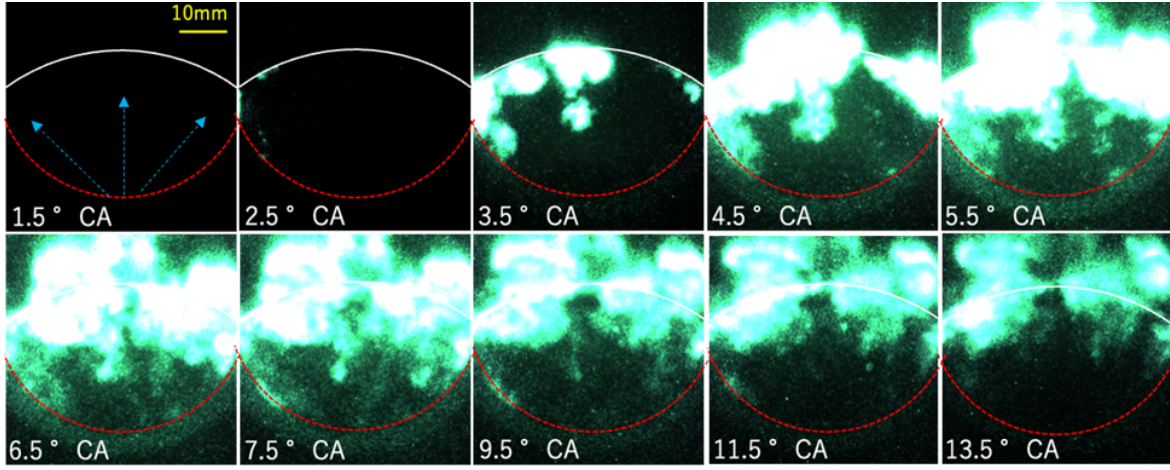


(a) $\lambda_{\text{total}} = 4.0$ ($p_j = 80$ MPa)

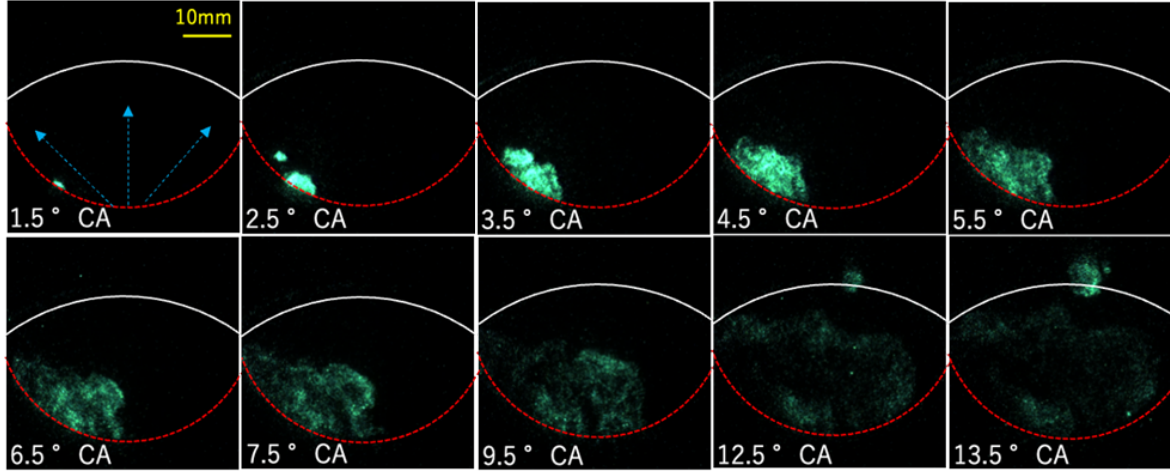


(b) $\lambda_{\text{total}} = 3.0$ ($p_j = 80$ MPa)

Figure 3.8: The effects of pilot injection quantity on in-cylinder pressure and heat release rate.

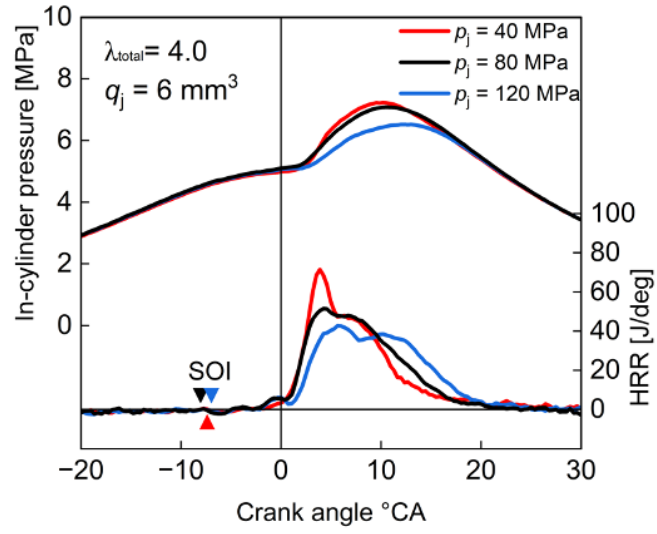


(a) $q_j = 13 \text{ mm}^3$ ($p_j = 80 \text{ MPa}$, $\lambda_{\text{total}} = 3.0$)

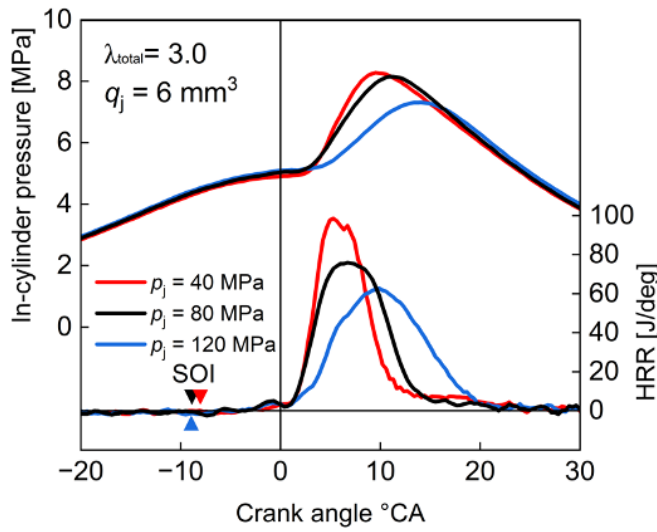


(b) $q_j = 3 \text{ mm}^3$ ($p_j = 80 \text{ MPa}$, $\lambda_{\text{total}} = 3.0$)

Figure 3.9: Visualized images using OH* photography with respect to pilot injection quantity.

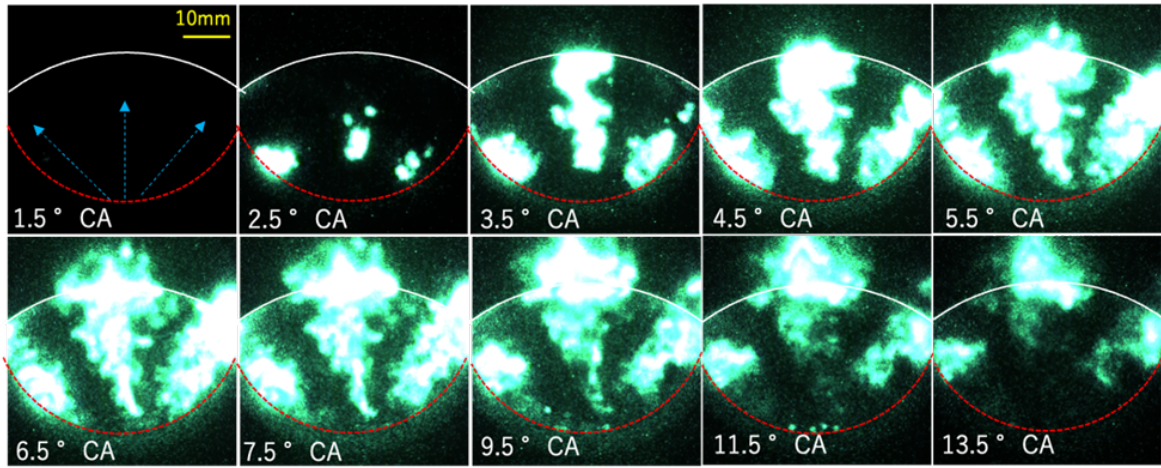


(a) $\lambda_{\text{total}} = 4.0$ ($q_j = 6 \text{ mm}^3$)

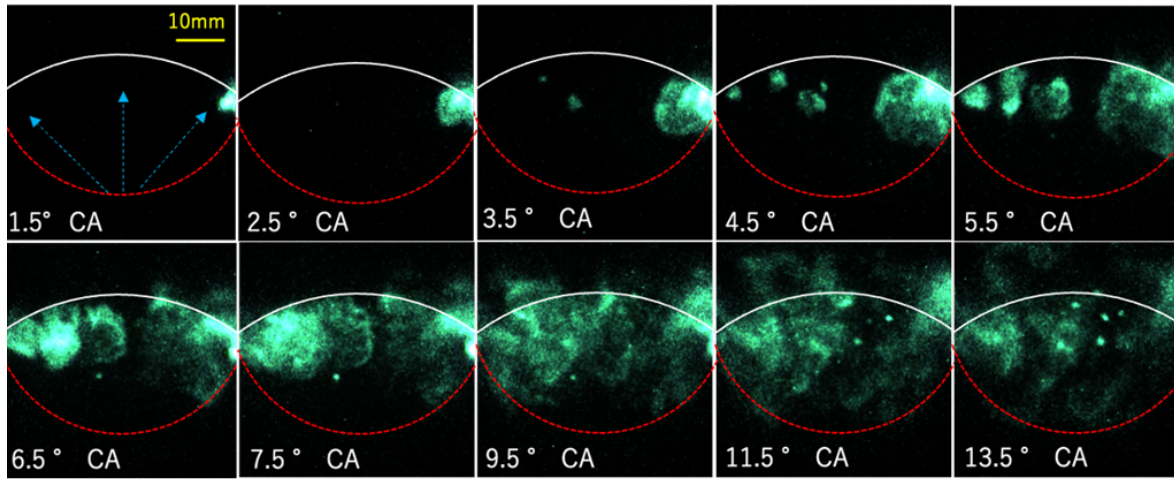


(b) $\lambda_{\text{total}} = 3.0$ ($q_j = 6 \text{ mm}^3$)

Figure 3.10: The effects of pilot injection pressure on in-cylinder pressure and heat release rate with the standard pilot injection for (a) $\lambda_{\text{total}} = 4.0$ and (b) $\lambda_{\text{total}} = 3.0$.



(a) $p_j = 40$ MPa ($q_j = 6 \text{ mm}^3$, $\lambda_{\text{total}} = 3.0$)



(b) $p_j = 120$ MPa ($q_j = 6 \text{ mm}^3$, $\lambda_{\text{total}} = 3.0$)

Figure 3.11: Visualized images using OH* photography with respect to pilot injection pressure.

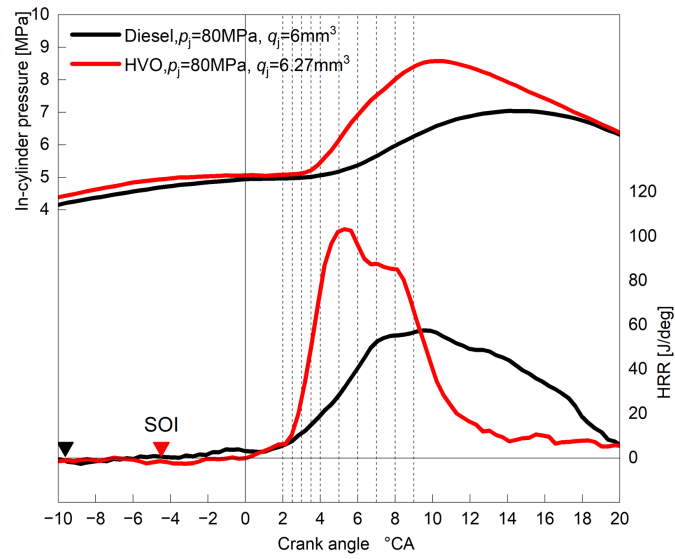


Figure 3.12: In-cylinder pressure and heat-release rate versus crank angle (comparison of diesel fuel and HVO)

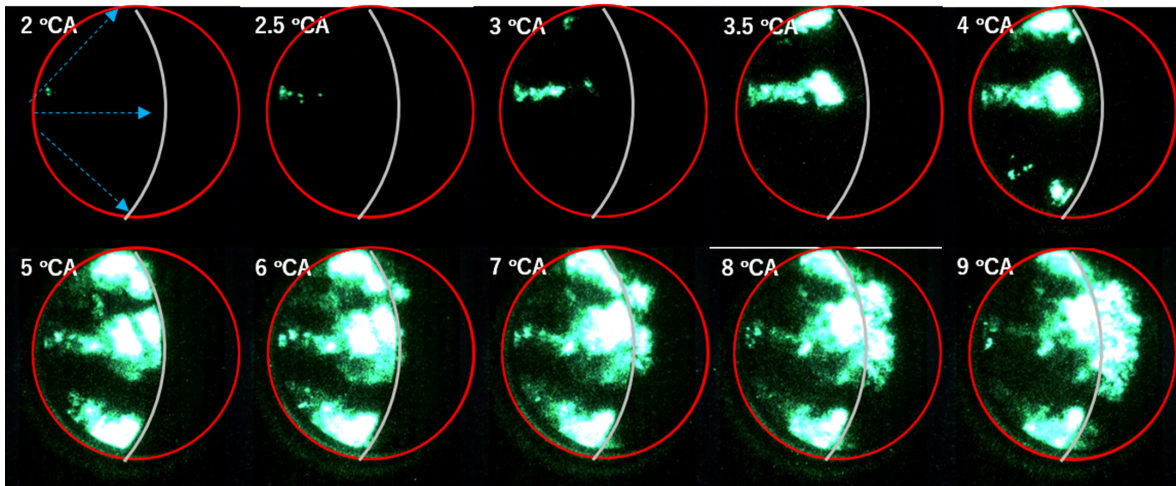


Figure 3.13: Images of OH* chemiluminescence (pilot fuel: HVO, $p_j = 80$ MPa, $q_j = 6.27$ mm³)

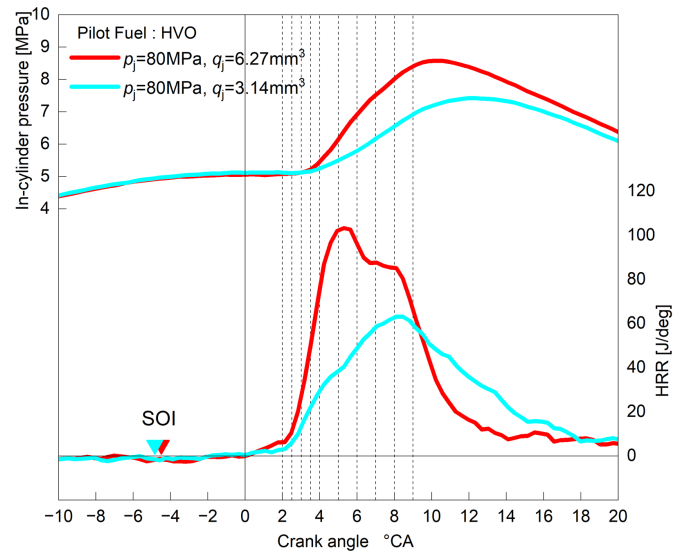


Figure 3.14: Effect of pilot injection quantity under fixed total excess air ratio condition on in-cylinder pressure and heat-release rate
(pilot fuel: HVO, $p_j = 80 \text{ MPa}$)

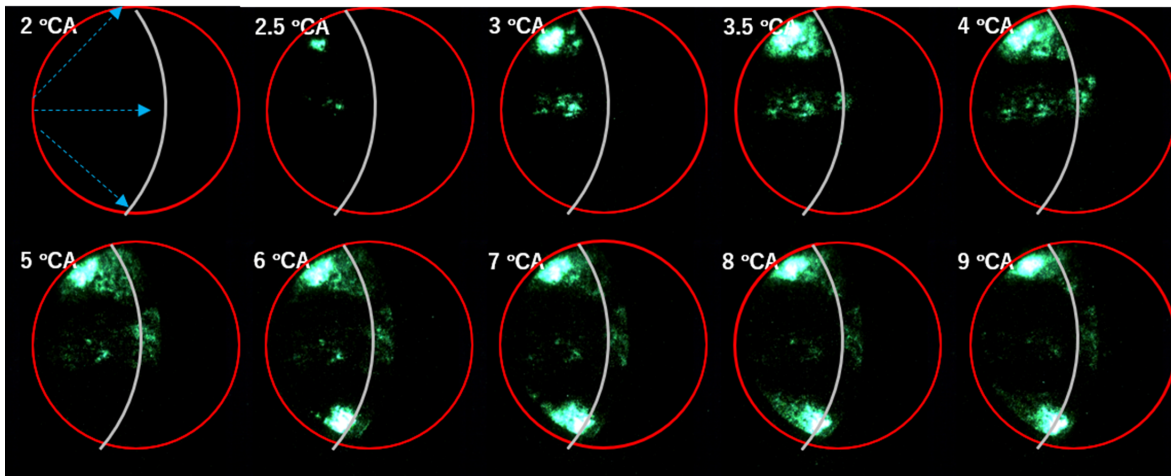


Figure 3.15: Images of OH* chemiluminescence
(pilot fuel : HVO, $p_j = 80 \text{ MPa}$, $q_j = 3.14 \text{ mm}^3$)

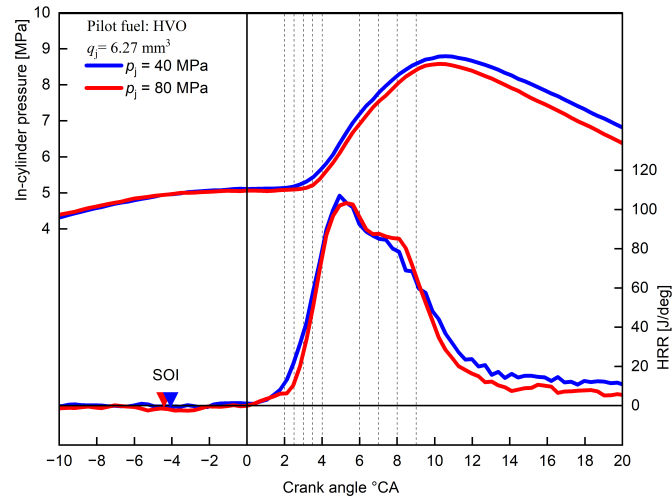


Figure 3.16: Effect of pilot injection pressure on in-cylinder pressure and heat-release rate (pilot fuel : diesel fuel and HVO)

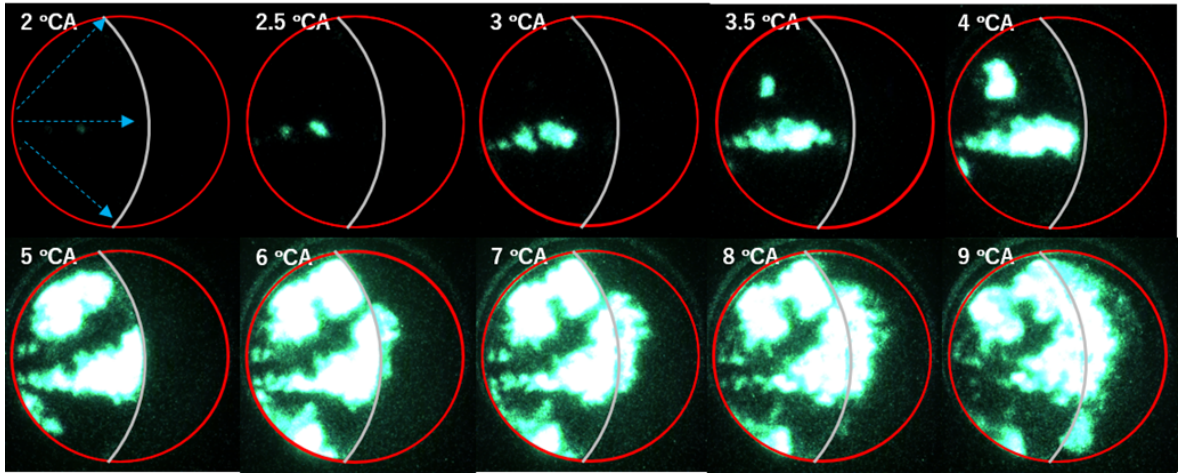


Figure 3.17: Images of OH* chemiluminescence (pilot fuel : HVO, $p_i = 40$ MPa, $q_i = 6.27$ mm³)

Table

Table 3.1: Technical specifications of the RCEM.

Bore $\times$ Stroke	102 $\times$ 105 mm
Displacement	857 cc
Compression ratio	15.4
Diesel injection system	Common rail injection system
Diesel injector nozzle	0.123 mm $\times$ 7

Table 3.2: Standard experimental conditions.

Cranking speed	[rpm]	900
Pressure at TDC	[MPa]	5.0 $\pm$ 0.1
Temperature at TDC	[K]	700
Ignition timing	[$^{\circ}$ CA]	2.0 $\pm$ 1
Total excess air ratio λ_{total}	-	4.0, 3.0
Injection quantity q_j	[mm ³]	3.0, 6.0, 13.0
Injection pressure p_j	[MPa]	40, 80, 120

Table 3.3: Imaging conditions for the photography methods.

	Direct imaging	OH* chemiluminescence
Frame rate [fps]	50400	50400
Range [pixel $\times$ pixel]	512 $\times$ 512	512 $\times$ 512
Resolution [mm/pixel]	0.0933	0.0951
Exposure time [ms]	0.16	18.23
I.I. gain	-	900
I.I. gate	-	open

Table 3.4: Operation conditions using HVO and diesel fuel as pilot fuel.

	Unit	HVO		Diesel	
Cranking speed	[rpm]	900			
In-cylinder pressure at TDC	[MPa]	5.0			
In-cylinder temperature at TDC	[K]	700			
Total excess air ratio λ_{total}	-	3.0			
Injection quantity q_j	[mm ³]	3.14	6.27	3.00	6.00
H ₂ excess air ratio	-	3.23	3.49	3.29	3.63
H ₂ energy ratio	[%]	93.1	86.1	92.7	85.3
Ignition timing	[°CA]	2.0 ± 1.0			
Injection pressure p_j	[MPa]	40,80			

Chapter 4 Analysis of emission and combustion behaviors of water-emulsified diesel with various water contents and droplet diameters

4.1 Introduction

In addressing carbon emission challenges using hydrogen dual fuel, a new concern has arisen: NO_x emissions from the combustion. Emission results from Chapter 2 exhibited relatively high NO_x emission despite low PM emissions. One strategy to simultaneously decrease both NO_x and PM emissions is the use of water-emulsified fuel. This method incorporates water into the combustion chamber via the fuel injection system by blending it with the fuel. Conventional fuel undergoes an emulsification process with water, creating a thermodynamically stable mixture. Upon injection into the combustion chamber, the water absorbs heat through the latent heat of vaporization, leading to a reduction in combustion chamber temperature. This resulted in a decrease of NO_x emissions.

Emulsified fuels exhibit several notable characteristics, including improved brake thermal efficiency at higher water content levels, reduced brake-specific fuel consumption, and lower exhaust gas temperatures[1–3], besides significantly reduce NO_x emissions due to the cooling effect of water evaporation. The thermal efficiency generally improves with water content, though some studies report reduced efficiency due to the lower calorific value of emulsified fuels[1, 2]. This demonstrated that the results have shown discrepancies, especially regarding enhancements in thermal efficiency [4].

The variability in results regarding the performance and emissions of emulsified fuels can be attributed to multiple factors, such as engine load, speed, and the type of base fuel used. A critical yet underexplored aspect is the role of water droplet diameter in emulsified fuels. While studies on spray flames and hot plates have highlighted the influence of droplet size on micro-explosion behavior and its intensity [5, 6], research within diesel engine applications remains scarce. The occurrence and significance of micro explosions in diesel engines are still a topic of debate, with divergent views in the literature [7, 8]. Nonetheless, water droplet size directly affects the dispersion of water within the

fuel, potentially influencing combustion efficiency and emissions. Investigating how varying water droplet sizes and water contents impact diesel engine performance is essential to address these gaps. Furthermore, identifying the suitable water content and droplet size combination for maximizing engine efficiency and minimizing emissions remains a critical objective.

Hence, Chapter 4 aims to examine the effect of water content and water droplet size onto the combustion and emission characteristics in a diesel engine. This is to determine the optimum water content and water droplet diameter that could both reduce emissions and improve engine performance. From this, emulsification strategy of pilot fuel in hydrogen dual fuel could be clarified.

4.2 Experimental method

4.2.1 Surfactant selection

To achieve stability of an emulsion, which is thermodynamically unstable, the addition of a surfactant becomes crucial in the emulsification process. Surfactants play a vital role in reducing the interfacial tension between water and oil. The hydrophilic-lipophilic balance (HLB) value of surfactants holds significant importance in achieving a stable emulsion. The HLB value needs to be adjusted based on the intended application of the emulsion, as a higher HLB value indicates a greater hydrophilic tendency. To produce a water-in-oil (W/O) emulsion, an HLB value of approximately 3 to 6 is recommended [9]. This study employs a combination of two surfactants, Emasol O-10(F) and Emanon 4110 to achieve an HLB value of 6. The properties of the surfactants are tabulated in Table 4.1. The selection of surfactants and their respective volumes are determined based on the work of Mizutani et al. [10, 11], which investigated the effect of surfactant volume on the separation time and emulsification rate of the emulsified fuel.

4.2.2 Emulsified fuel preparation

In this study, emulsification of a commercially available diesel with purified water in varying volumes was conducted to produce water-in-oil (W/O) emulsified fuels with different

water contents and droplet diameters. A total volume of 2 vol% surfactants indicated in Table 4.1 was added to the diesel and stirred for a certain period. Subsequently, the desired volume of water was added and stirred for approximately 3 minutes as the final preparation step before emulsification. Two emulsification methods were employed:

1. The strong stirring method (STR) utilizing a magnetic stirrer, which involved vigorous stirring (2500 rpm) of the premix for over 6 hours, resulting in the creation of very fine water droplets.
2. The permeate membrane emulsification method, as illustrated in Fig. 4.1. A tube pump (EYELA RP-1000) was employed to force the mixture to flow through a porous glass membrane known as Shirasu Porous Glass (SPG). This method ensured the formation of emulsified fuels with uniform water droplets within the desired diameter range [12]. SPG membranes with pore diameters of 30 μm and 50 μm , referred to as SPG30 and SPG50, respectively, were employed in this research. The dispersed water droplet diameters of the emulsified fuels are varied using the STR method and permeate membrane emulsification method using SPG30 and SPG50. The water contents are varied from 2 vol% to 24 vol% with the increment of 2 vol%, as shown in Table 4.2. It must be mentioned in this context that the emulsified fuels produced using STR methods, SPG30 membrane and SPG50 membrane will be indicated as STR, SPG30 and SPG50, accordingly.

4.2.3 Engine and experiment details

The experiments were carried out using a YANMAR L70V-MEJ2, which is a single-cylinder, air-cooled direct-injection diesel engine. The engine was operated at a constant speed of 2300 rpm, and prior to the start of the experiment, engine heating-up procedures were performed until the engine oil reached a temperature of approximately 353 K. The specifications of the test engine can be found in Table 4.3[13]. To ensure consistent fuel spray conditions throughout the experiments, the governor lever of the engine was fixed at an opening position of 7/8. For power measurement, a hydraulic dynamometer (Tokyo Plant PTB-30R-B) was connected to the engine crankshaft. The fuel consumption rate was examined using a ball-shaped burette connected to the fuel supply system. Two gas analyzers

(HODAKA HT-1300Z and testo 330-LL) were employed to measure NO_x and CO emissions while the soot emission was measured using a smoke tester (BACHARACH True Spot). Fig. 4.2 shows the experimental setup for this study.

4.2.4 Heat release rate

In this study, the combustion behaviors are analyzed by mainly examining the heat release rate, specifically focusing on the apparent rate of heat release (aRoHR). The aRoHR is determined based on the principles of the first law of thermodynamics and could be expressed as Eq. (4.1).

$$\frac{dQ}{d\theta} = \frac{\kappa}{\kappa - 1} \rho \left(\frac{dV}{d\theta} \right) + \frac{1}{\kappa - 1} V \left(\frac{dP}{d\theta} \right) \quad (4.1)$$

where p stands for in-cylinder pressure, and V is defined as the volume of the combustion chamber. The temperature and composition-dependent specific heat ratio, κ , is calculated for each crank angle timing using the NIST-JANAF polynomials [14] while considering the global gas temperature and average gas composition.

4.2.5 Ignition delay and burn ratio calculation

Fig. 4.3 indicates the method used to measure the ignition delay. The ignition delay is measured by calculating the crank angle differences, $\Delta\theta$ from the start of injection (SOI) to the rising point of the heat release rate. Here, SOI is defined as the crank angle timing for the first pressure peak in the fuel injector, which should be constant for all conditions. $\Delta\theta$ is converted to ignition delay time using the engine speed.

Willems et al. introduced the burn ratio (Rb) to characterize this feature as the heat release rate can exhibit either single stage or dual stages during reactivity stratification [15]. Rb is defined as the ratio of the phase-specific combustion durations after and before CA50 and calculated by using the following equation. It was concluded that Rb correlates well with the gross indicated thermal efficiency (GIE) and combustion loss [16]. Combustion loss refers to energy lost due to incomplete fuel conversion into CO_2 and water, indicated by

emissions of CO and unburned hydrocarbons [16].

$$Rb = \frac{CA90 - CA50}{CA50 - CA10} \quad (4.2)$$

where CA90, CA50 and CA10 are crank angles at 90%, 50% and 10% of total heat release, respectively.

4.3 Results and discussion

4.3.1 Fuel formation results

Fig. 4.4 shows the images of water droplets in diesel fuel. The images were taken using a digital camera (Nikon, Nikon 1 J3) connected to a microscope (Kenko, KML-1500V). The microscope used was an optical microscope with a maximum magnification ratio of 1500. Regarding STR fuels, the water droplets are so fine that they cannot be seen clearly using a microscope. As the smallest pixel of water droplet diameter that was measured is 3 pixels (equivalent to $3.3 \mu\text{m}$), $3 \mu\text{m}$ is chosen as the representative SMD for STR fuels. The water droplet diameter distribution and SMD for the emulsified fuels are as indicated in Fig. 4.5.

The distribution of water droplet sizes in emulsified fuels varies based on the emulsification method as can be observed in Fig. 4.5. The strong stirring method (STR) produces very small droplets, mostly under $3 \mu\text{m}$, as stated previously. The porous glass membrane methods (SPG30 and SPG50) generate more uniform droplet sizes. SPG30, with $30 \mu\text{m}$ pores, produces droplets with an average size (SMD) of $26.5 \mu\text{m}$ and a tighter distribution, while SPG50, with $50 \mu\text{m}$ pores, results in larger droplets (SMD = $39.8 \mu\text{m}$) and a wider diameter range. Membrane emulsification offers better control compared to stirring, with smaller pores (SPG30) providing more consistent droplet sizes.

4.3.2 Influences of water emulsification onto engine performance

Low combustion temperature, long ignition delay and lower heat input of emulsified fuels can affect the engine performance. To better understand the relationship between

water droplet size, water content, and these combustion behaviors, the engine performance was investigated while manipulating the water content and droplet diameters. The engine operating speed was set at a fixed value of 2300 rpm, while the governor opening fixed at 7/8. This approach ensured consistent fuel injection conditions across all fuel variations in the jerk pump direct injection system. The fuel injection quantity including water was $12.6 \pm 0.5 \text{ mm}^3/\text{cycle}$. This study focuses on the output power and thermal efficiency as parameters to evaluate the engine performance. The results are illustrated in Fig. 4.6. It could be observed that the output power decreases continuously for all the emulsified fuels when the water content is increased, with the most rapid decrement obtained for STR emulsified fuels, and retarded decrement is observed for the output power of SPG50 emulsified fuels. The decrement of power is due to the latent heat of water, which reduce the heat released and mainly due to the decrease of heat input due to lower diesel fraction.

Regarding thermal efficiency, an increasing trend was observed for all SPG30 and SPG50 emulsified fuels as the water content increased up to 20vol%. In contrast, STR-emulsified fuels exhibited lower thermal efficiency compared to diesel, although an improvement was noted up to 14vol% water content. Among the fuels tested, STR fuels showed the slowest increase in thermal efficiency, while SPG50 fuels displayed the most significant improvement. To have better understanding of these results, the in-cylinder temperature, pressure, and heat release rate are illustrated in Fig. 4.7.

4.3.3 Influences of water emulsification onto combustion behavior

Fig. 4.7 shows in-cylinder temperature, pressure, and heat release rate during combustion for different emulsified fuels compared to pure diesel. From left, the combustion behaviors of STR, SPG30 and SPG50 emulsified fuels are illustrated respectively. The x-axis represents the crank angle (CA). The upper part of the graph displays the in-cylinder temperature, the middle part shows the in-cylinder pressure, and the lower part represents the heat release rate. These parameters help illustrate the combustion characteristics of each fuel type as the engine operates. The heat release rate exhibits a dual stage for all fuels, in which the first peak is the heat release from premixed-like combustion, and the second peak is the diffusion-controlled combustion. Heat release rate rises earlier during diesel combustion

than emulsified fuels due to higher net diesel and local temperature. The heat release rate for emulsified fuels with higher water content rises later because of the latent heat of the water and lower diesel content, which retard the ignition timing. It is difficult to evaluate the retarded ignition from Fig. 4.7; the ignition delay value is measured and plotted in Fig. 4.8 later. As the ignition is retarded, more fuel vaporizes and mixes with the surrounding air, ensuring more gaseous fuel during premixed-like combustion [15, 17]. Therefore, a higher peak is observed during premixed-like combustion of emulsified fuels with higher water content.

In general, the delayed timing of the HRR peak as water content increases leads to a reduction in peak pressure. This delayed peak pressure reduces the power output, especially as water content rises. In the case of 24 vol% water content, although the premixed combustion produces a significantly higher HRR peak, the ignition timing is too delayed, occurring as the piston is already entering the expansion stroke. As a result, despite the high peak pressure, the power output is reduced due to the late combustion. Among the emulsified fuels, STR demonstrates the highest peak pressures across all water content levels, which corresponds to its higher HRR peaks. This could be attributed to the slightly significant ignition delay of the STR fuels as will be explained later under the discussion for Fig. 4.8.

Increasing water content decreases the average in-cylinder temperature, with a significant decrease between 6% and 12%, especially in SPG30 and SPG50 fuels. Based on the decreasing magnitude, it could be concluded that the main reason for the temperature decrement is the reduction of diesel fraction, which significantly decreases the heat input. At the same volume, the latent heat of water only contributes to 6% of reduction from the calorific value of diesel. This decreasing temperature trend due to lower heat input describes the decrement of NO_x of all emulsified fuels. The high peak heat release rate during premixed combustion affects the in-cylinder temperature during the first half. This trend can be seen at a water content of 24vol% for SPG50 and at several water content conditions for STR fuels. However, the low heat input and latent heat of the water as well as a lower peak during diffusion-controlled combustion eventually decrease the combustion temperature; hence the temperature shows a decreasing trend during the second half of combustion, in which higher water content emulsified fuels exhibit lower temperatures.

The increment of ignition delay due to the increment amount of water which results in more heat is absorbed during water vaporization is shown in Fig. 4.8. The effect of water droplet diameter is not significant until 12vol% water content. Beyond that point, STR fuels show drastic increment, while SPG50 exhibit the slowest increment of ignition delay. The higher water droplet content in a single spray of smaller water droplet emulsified fuels is ascribed to this matter. The longer ignition delay contributes to a higher peak of heat release rate, which contributes to higher thermal efficiency, but the decrement of thermal efficiency for emulsified fuels after 20vol% water content for SPG30 and SPG50 emulsified fuels, and after 14vol% for STR emulsified fuels could not be explained by ignition delay. The trend is contrary to the previous report, in which the elongated ignition timing helps to improve thermal efficiency [18]. Therefore, it is assumed that the extremely long ignition delay, which retard the timing for the peak heat release rate, could affect the thermal efficiency. To clarify this matter, the burn ratios of the fuels are plotted together in Fig. 4.8. Burn ratio mainly divides the combustion phase, which is the ratio of diffusion-controlled combustion (CA90-CA50) to the premixed-like combustion period.

Burn ratio increases with water content, and different water droplet diameters of fuels exhibit different increasing rates, with SPG50 emulsified fuel showing the slowest increasing rates. The burn ratio exhibits a trend similar to the ignition delay but with a gentler slope. During the combustion of SPG30 and SPG50 emulsified fuels, the burn ratio seems constant until 20vol% water content, while a significant increase can be seen at 22vol% and 24vol%. water. For STR, the burn ratio increases from 14vol%. It can be concluded that both ignition delay and burn ratio correlates with thermal efficiency. The longer ignition delay increases the thermal efficiency by increasing the peak heat release rate, while higher burn ratio resulted in lower thermal efficiency. The behavior of the burn ratio depends on the increment magnitude of ignition delay, which regards the combustion phasing of heat released rates. Therefore, the combustion phases are studied, in which the CA10, CA50 and CA90 of every combustion are determined and analyzed in Fig. 4.9.

Combustion phases of the heat release rates are shown in Fig. 4.9. CA10 is considered the combustion starting point. It is noted that CA10 water content is increased with linear retardment for STR fuels from the water content of 14vol% due to longer ignition delay as

water content increases. Almost no retardment is observed at lower water content for SPG30 and SPG50 fuels from low water content until 20 vol%, and retardments are observed from 20vol% to 24vol% for SPG30 and SPG50 emulsified fuels. Comparing the minimum water content that regards the CA10, as in Fig. 4.9 and the water content which increases the burn ratio, as in Fig. 4.8, it can be considered that CA10 dominates the role of controlling the burn ratio in this study.

CA50 is considered the center of the heat release rate. Small advancements in CA50 are observed during STR fuel combustion, while almost no change can be seen for SPG30 and SPG50. Therefore, it can be concluded that the retardment of CA10 mainly causes the premixed combustion duration (CA50-CA10) to decline.

CA90 reflects the duration of total combustion. As the water content increases in all emulsified fuels, the combustion characteristics exhibit an advancement in CA90. Comparing with CA10 position, in which it exhibit constant and some retardment when the water content is increased, it can be observed that higher water content resulted in reduced combustion duration (CA10-CA90). This can be attributed to the decrease in the net diesel content and an increase in ignition delay, resulting in higher peak of heat release rate. The decrement of the combustion duration improves the degree of constant volume combustion. This explains the improved thermal efficiency at certain water contents. For STR fuels from 14vol% onwards, SPG30 and SPG50 fuels with 22~24vol% the decreasing magnitude of CA90 is smaller than the retardment of CA10, as well as the advanced CA50 in certain points which causes the premixed-like combustion duration (CA50-CA10) to progress faster and the diffusion-controlled combustion (CA50- CA90) duration to be longer than the conventional diesel situation. Consequently, the ratio of these combustion phases, referred to as the burn ratio increases. This means that more heat is produced after the piston begins its power stroke, which diminishes the efficiency of constant volume combustion. Additionally, the lower temperatures caused by the latent heat of the water and reduced heat input significantly decrease output power, further contributing to this issue.

4.3.4 Influences of water emulsification onto emissions

Previous studies have highlighted the significant impact of introducing water into combustion through emulsification on combustion behavior, including the generation of pollutants such as NO_x , soot, and CO [2]. To better understand the relationship between water droplet size, water content, and these combustion behaviors, the emission characteristics of these pollutants were investigated while manipulating the water content and droplet diameters.

As water content is increased until 24vol%, the emission of NO_x shows an almost linear decreasing trend from the combustion of all emulsified fuels as shown in Fig. 4.10. A little difference due to water droplet diameter effects is observed at low water droplets. However, rather significant differences were found in different water droplet diameter emulsified fuels at higher water content. The maximum reduction is obtained through STR emulsified fuel combustion at 24vol%, and about 40% reduction of NO_x is achieved through diesel. The decreasing trend of NO_x for these emulsified fuels as the water content increases is due to the latent heat of water during vaporization in the combustion chamber and the decrements of diesel fraction, which produce up to 30% less heat input, hence reducing the NO_x which is produced by thermal NO mechanism. The low temperature directly influences NO formation by suppressing the chemical reaction rates, as explained in Zeldovich mechanism reactions [2].

Increasing trends are observed for CO emissions as the water content is increased. The increasing trends are due to the increment of water fraction, which reduce the heat produced and lower the temperature. Overall, the slowest increment can be seen during SPG50 emulsified fuel combustion. Larger water droplet diameter leads to a decrease in the number density of water droplets, and more spray droplets without water are present in the fuel spray. This phenomenon has been reported by Watanabe and Okazaki in their study on emulsified fuel spray flow using shadow imaging. They observed that if the size of dispersed water droplets increases significantly, certain spray droplets, which do not contain dispersed water droplets, or water spray droplets may be produced from the injector. Therefore, it is assumed that droplets without water ignite and burn similarly to diesel fuel, leading to

combustion behaviors of emulsified fuels with larger water droplet diameters that closely resemble those of diesel. The highest emission of CO is from STR emulsified fuel, during water content of 24vol%, which emits about 380 ppm, four times the emission of CO over diesel combustion.

Soot is emitted lower for all emulsified fuel combustion than diesel combustion, as shown in Fig. 4.10. This is due to the lower diesel input as the source of the soot itself. Soot emission is reduced significantly for all emulsified fuels with STR fuels exhibit the most drastic soot decrement. This trend is probably due to the dispersion of small water droplets in all the spray droplets, in which the water vapor dispersed throughout the fuel vapor region. The soot emissions of emulsified fuels decreased as the water content increased up to 24vol%, and the soot emissions of all the emulsified fuels were almost the same at the water content of 24vol%.

As the output power is also reduced when the water content is increased, the brake specific emission, i.e. brake specific NO_x (BsNO_x) and brake specific CO (BsCO) is plotted in Fig. 4.11. Despite the decrease in output power, BsNO_x emissions decrease for all emulsified fuels when the water content is increased. Comparing with NO_x emission, BsNO_x is reduced gentler, with the maximum reduction of BsNO_x is about 28%. Among the emulsified fuels, SPG30 and SPG50 show more stable decreasing trends with lower NO_x emission compared to STR at lower water content. The STR emulsified fuels indicate unstable BsNO_x decrement, at low water content. This is attributed to the lower output power, and slightly higher NO_x emission, likely due to poorer droplet uniformity and inconsistent combustion. BsCO emissions increase significantly with the rise in water content for all emulsified fuels. Comparing with CO emission in Fig. 4.10, almost no change is observed in the increasing magnitude. Despite the decrease in carbon input through the reduction of neat diesel in the emulsified fuels when the water content is increased, the incomplete combustion due to lower temperature in the combustion chamber, which lead to incomplete oxidation of carbon to CO_2 [19], results in a significant increase in CO emission at higher water contents.

4.4 Conclusion

The influences of water content and water droplet diameter in W/O emulsified fuels were experimentally investigated in an air-cooled direct-injection single-cylinder diesel engine with a constant speed of 2300 rpm. By manipulating the water content up to 24vol% with a 2vol% increase and the water droplet size, the following conclusions can be reached:

1. Increasing water content improves pollutant emission, in which a significant reduction of NO_x and soot is achieved due to low combustion temperature and carbon input. However, CO emission displays a trade-off relationship.
2. Lower heat input at higher water content deteriorates the output power for all fuels, but this behavior could be slightly improved by increasing the water droplet size.
3. Water content of 20vol% is a favorable condition to improve thermal efficiency when larger water droplet emulsified fuel is used due to increment of ignition delay, which increases the peak rate of heat release, and this trend agrees with some previous studies in which the maximum efficiency is achieved at 20 vol% [1, 20]. For small water droplet fuel (STR), maximum thermal efficiency is reached at 14 vol%. The thermal efficiency decreases after those water content conditions due to the increment of burn ratio, which increases the effective energy loss (late premixed combustion timing).
4. The burn ratio increases with both water content and droplet size because these factors extend the ignition delay, shifting more combustion into the diffusion-controlled phase. However, there is an optimal range for both water content and droplet size where the burn ratio stays low, maintaining a balance between premixed and diffusion combustion for higher thermal efficiency.
5. Larger water droplet diameter leads to a decrease in the number density of water droplets, and more droplets without water are present in the fuel spray. It is assumed that droplets without water ignite and burn similarly to diesel fuel, leading to combustion behaviors of emulsified fuels with larger water droplet diameters resemble those of neat diesel.
6. Considering the effects of water on engine emission and performance, it can be concluded that the maximum water content of 20 vol%, with the largest water droplet

diameter, is the best fuel property from this research. Hence SPG50 fuel with a water content of 20 vol% is regarded to have the most potential.

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Figures

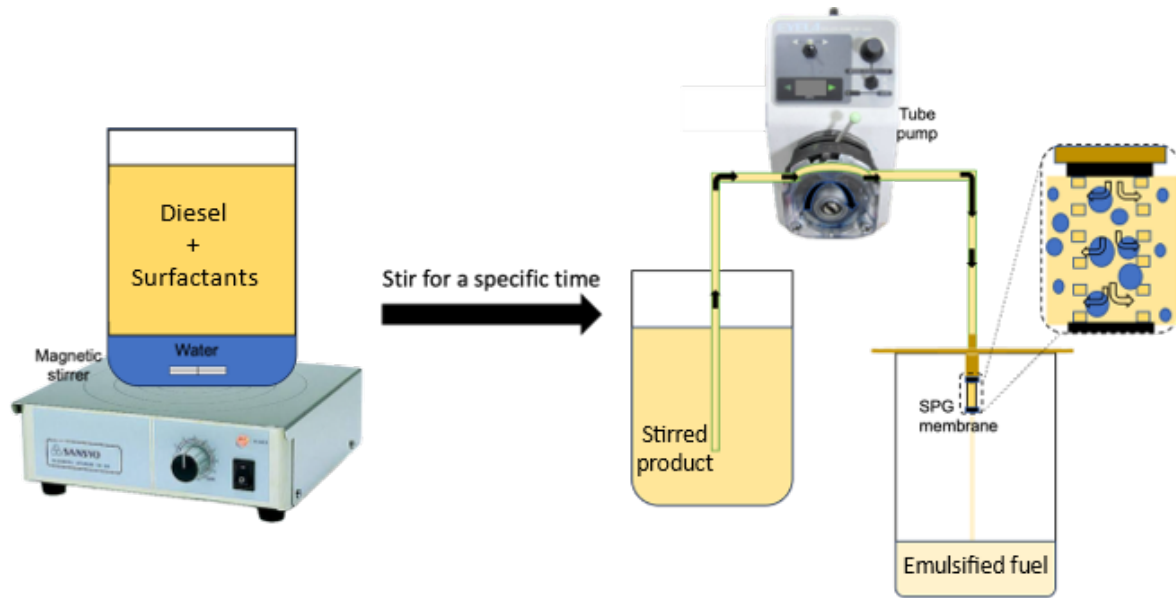


Figure 4.1: Emulsified fuel production using permeate membrane emulsification method.

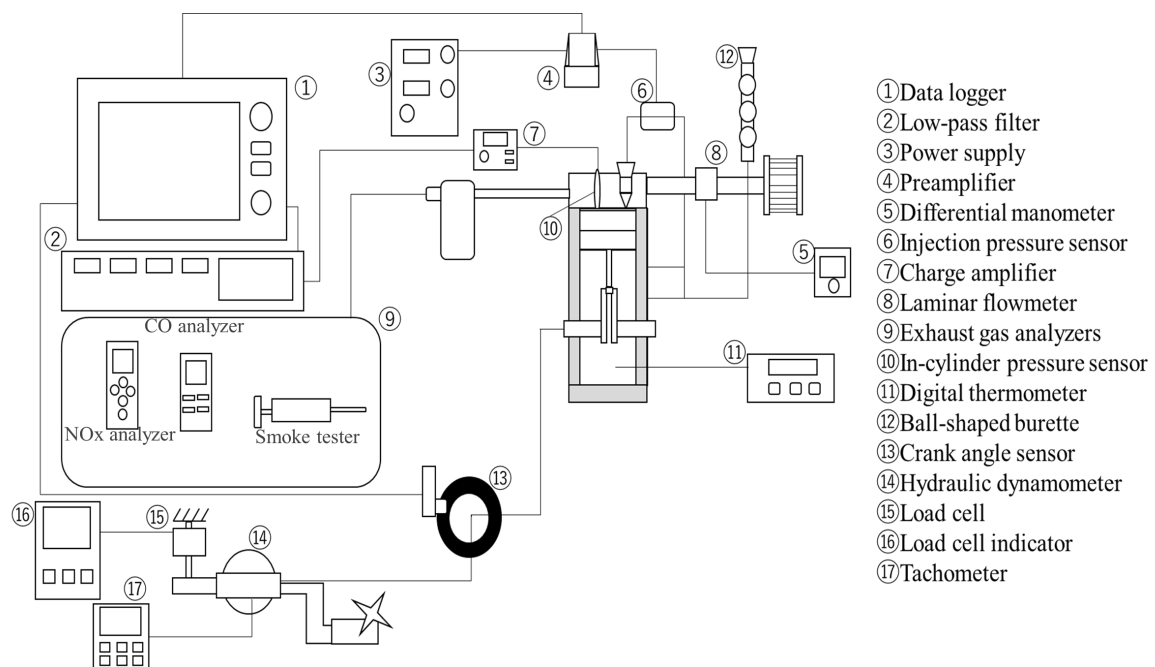


Figure 4.2: Experimental setup.

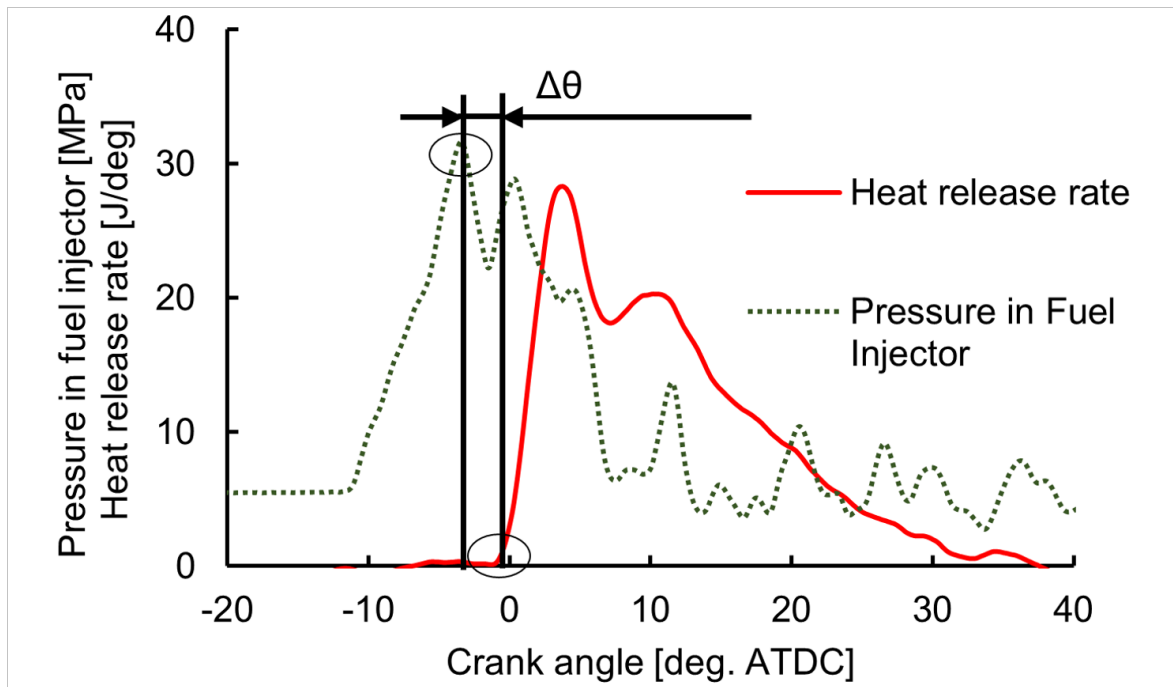


Figure 4.3: Ignition delay measurement method.

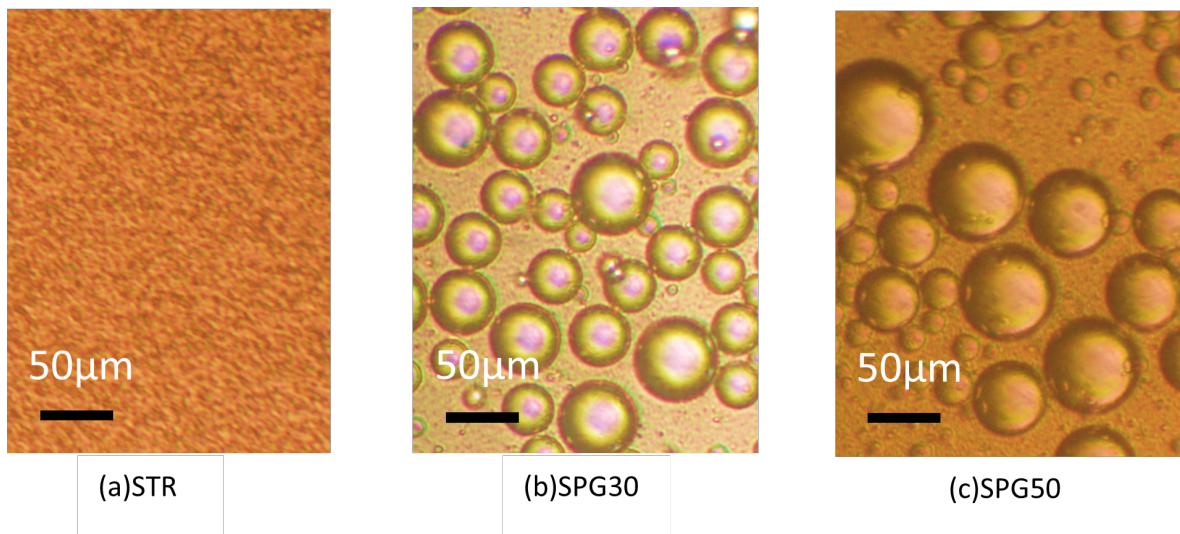


Figure 4.4: Images of water droplets by optical microscope.

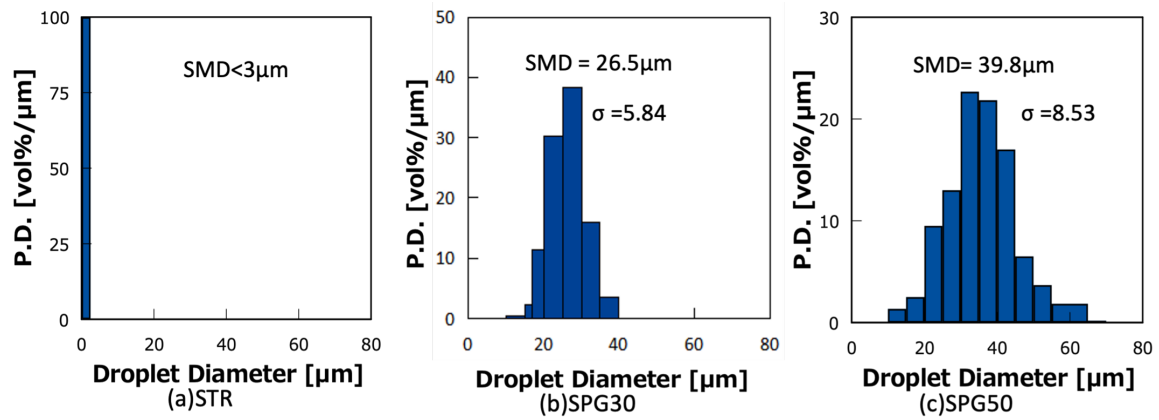


Figure 4.5: The distribution and SMD of water droplet sizes in emulsified fuels based on the emulsification method water droplets.

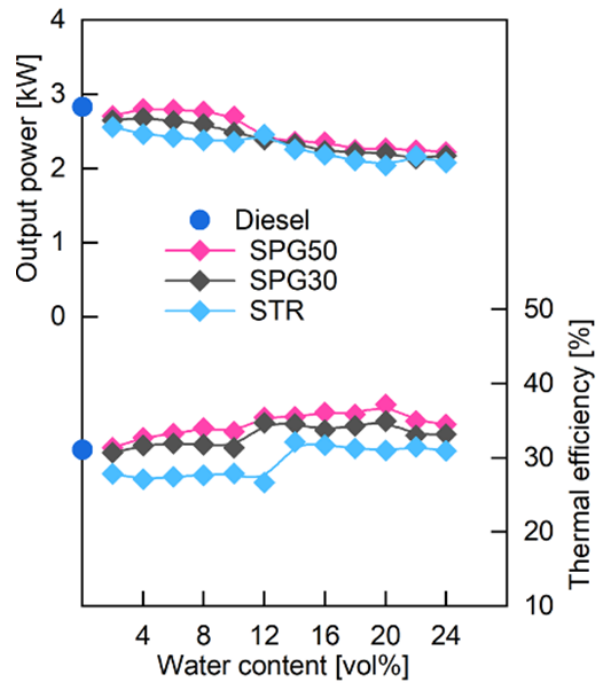


Figure 4.6: Output power and thermal efficiency of emulsified fuels and diesel.

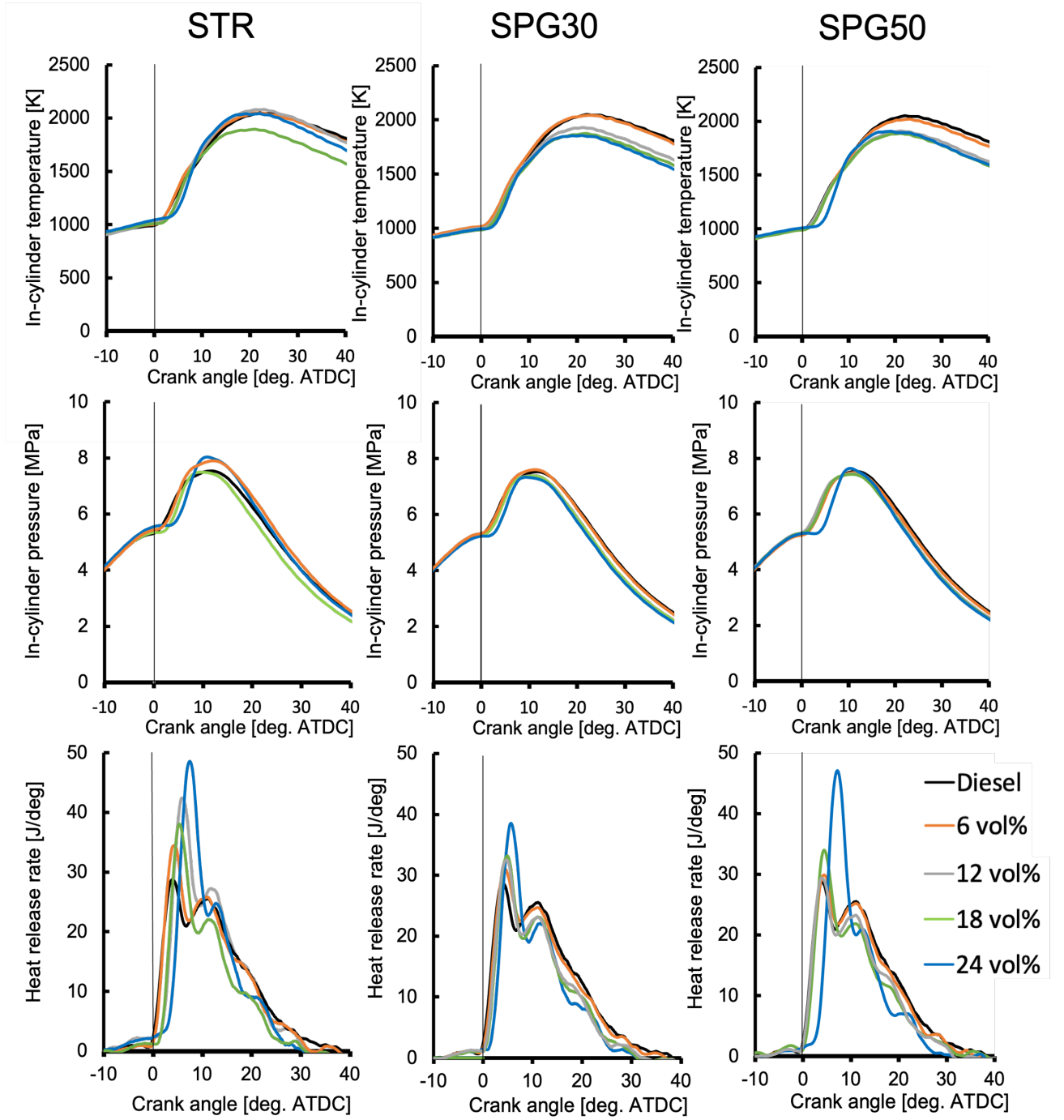


Figure 4.7: Variations of heat release rate and average in-cylinder temperature versus crank angle of emulsified fuels.

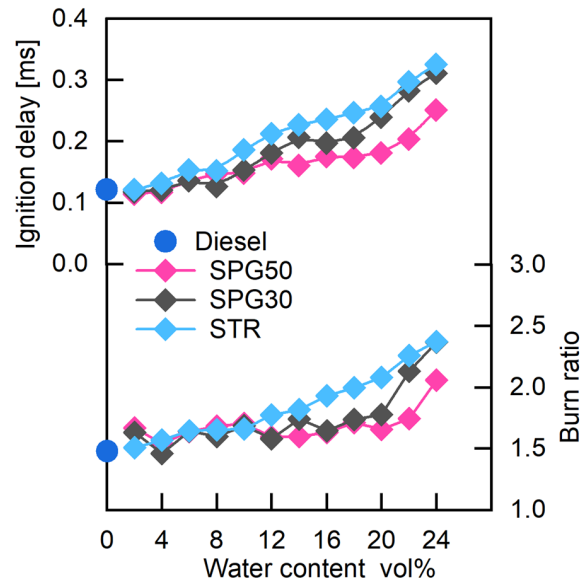


Figure 4.8: Variations of water content versus ignition delay and burn ratio of water-emulsified fuels.

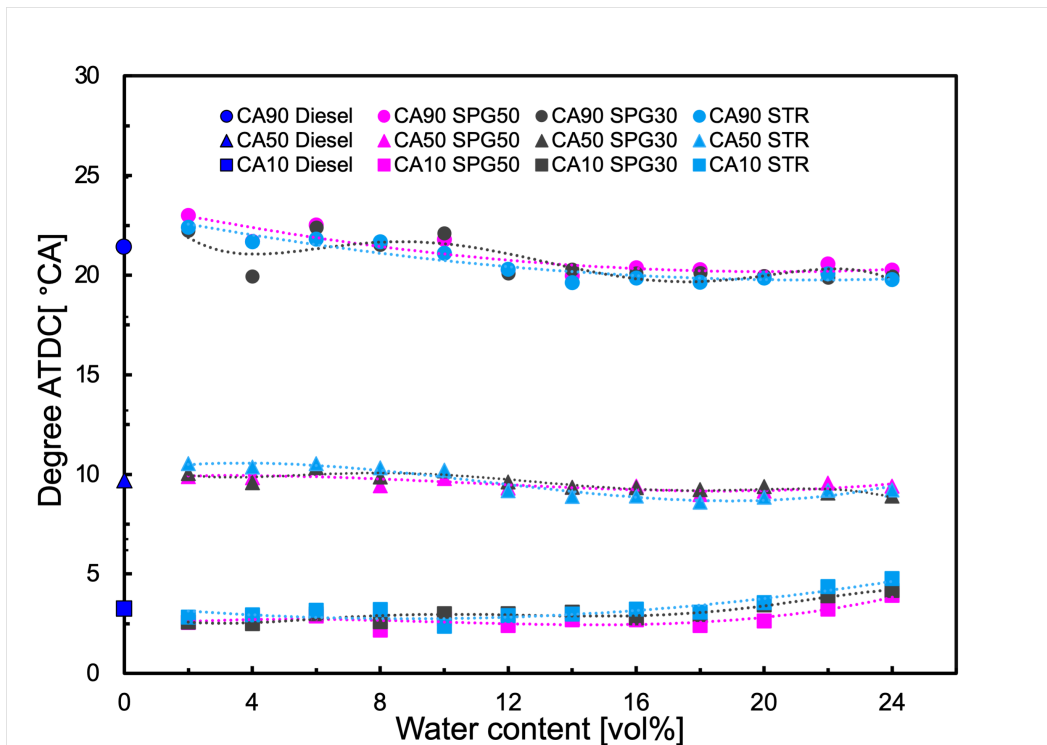


Figure 4.9: Combustion phases of test fuels.

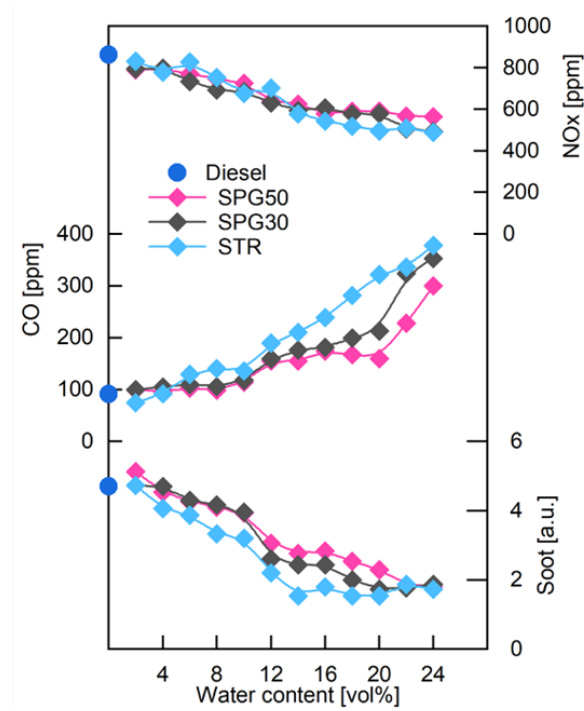


Figure 4.10: Water content of emulsified fuels versus NO_x, CO and soot emission.

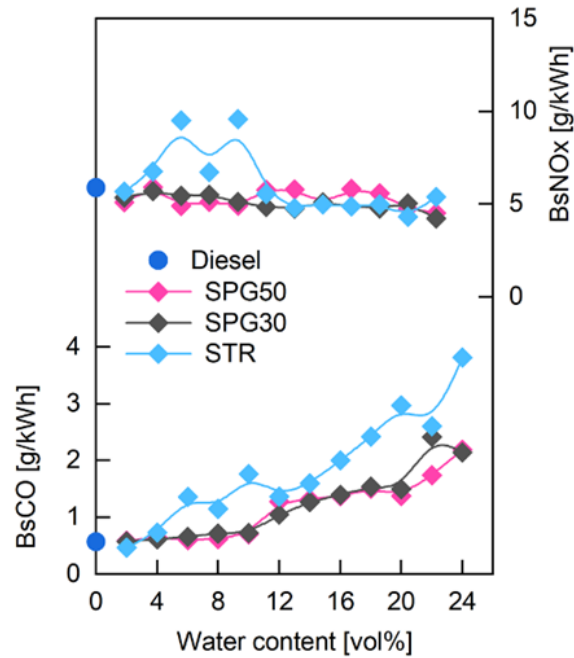


Figure 4.11: Water content of emulsified fuels versus BsNO_x and BsCO.

Tables

Table 4.1: Properties of surfactants.

Product name	Type	Composition	HLB Value
Emasol O-10(F)	Nonionic	Sorbitan monooleate	4.3
Emanon 4110	Nonionic	Polyethylene glycol monooleate	11.6

Table 4.2: Experimental conditions for emulsified fuel.

Fuel	STR	SPG30	SPG50
Diesel [vol%]	96, 94, 92, 90, 88, 86, 84, 82, 80, 78, 76, 74		
Emanon 4110 [vol%]	0.46		
Emasol O-10(F) [vol%]	1.54		
Water [vol%]	2, 4, 6, 8, 10, 12, 14, 16, 18, 20, 22, 24		

Table 4.3: Specifications of test engine [18].

Model	YANMAR L70V6-MEJ2
Fuel injection system	Jerk pump driven direct injection
Displacement [cc]	320
Compression ratio	21.1
Operating speed [rpm]	2300

Chapter 5 Dissertation summary and conclusion

Diesel engines are valued for their efficiency but face challenges from emissions such as CO₂, NO_x and PM. Conventional methods for mitigating these emissions, including exhaust after-treatment systems and advanced combustion control, frequently result in increased system complexity and cost. Consequently, fuel modification has emerged as a promising alternative, aiming to reduce emissions by altering fuel properties rather than engine hardware. Hydrogen is a key candidate for carbon neutrality due to its carbon-free combustion and diverse production sources; however, its high ignition temperature and propensity for knocking limit its direct use in diesel engines. Utilizing high-cetane pilot fuels such as HVO could address these challenges, offering improved ignitability and reduced carbon emissions, thereby making it a promising option for hydrogen-diesel dual-fuel systems. However, to ensure optimal safe operation with low emissions and high efficiency at various loads, knowledge regarding the engine emissions and performance across various parameters is required. Furthermore, to determine the optimal pilot fuel injection strategies in hydrogen dual fuel operation, understanding the location of pilot ignition kernels and the progression of the pilot flame towards the hydrogen premixed fuel is crucial, in which the visualization of combustion is essential. However, when carbon is reduced in internal combustion engine, NO_x emission problem emerges. Carbon emission often reported to related with NO_x emission, in which it show trade off relationship. Emulsifying fuel with water can both reduce NO_x and carbon emission, but the knowledge on optimum water content and water droplet diameter that can both improve engine emission, without worsening the engine performance is essential. Therefore, following studies has been conducted and documented.

Chapter 1 is an introductory chapter which sets the context for a research study on alternative fuels in internal combustion engines, primarily focusing on hydrogen dual fuel and water-emulsified fuels. To address carbon emission problems, this chapter begins by highlighting the urgency of addressing global warming, emphasizing the significant contribution of greenhouse gas emissions from various sectors, particularly electricity production and transportation. The chapter then introduces hydrogen as a promising alternative energy source in this sectors, discussing its potential applications and the

strategies of different countries, especially Japan, in promoting hydrogen usage. Then, the challenges and potential benefits of using hydrogen in internal combustion engines is discussed, particularly in diesel engine using dual-fuel mode with pilot fuels like diesel or HVO, which motivate the study in chapter 2 and chapter 3. Then, to address NO_x emission issues, emulsified fuel is introduced to be studied in chapter 4.

Chapter 2 presented an experimental investigation on hydrogendiesel dual-fuel and hydrogen-HVO dual-fuel operations in a single-cylinder diesel engine. The engine specifications included an 85.0 mm bore, 96.9 mm stroke, and 14.3 compression ratio, with a common rail fuel injection system and supercharger. Hydrogen was adjusted by modifying the hydrogen excess air ratio from 2.5 to 4.0 in 0.5 increments. A gas injector introduced hydrogen into the intake pipe, while diesel fuel and HVO were injected as pilot fuels at a constant 80 MPa pressure. Pilot fuel quantities were established at 3, 6, and 13 mm^3/cycle . Intake and exhaust pressures ranged from 100 to 220 kPa in 20 kPa intervals. The engine maintained a constant 1,800 rpm speed throughout all experimental conditions. Pilot injection timing was adjusted to maintain a consistent ignition timing at the TDC for all conditions. The findings revealed that HVO fuel reduces smoke and provides combustion and performance characteristics comparable to diesel. Under constant intake pressure, NO_x and smoke increase with a lower hydrogen excess air ratio. Decreasing pilot injection quantity reduces both NO_x and smoke, while increasing intake pressure boosts thermal efficiency by lowering cooling losses. At low hydrogen excess air ratios, abnormal combustion like knocking or pre-ignition is more likely, though HVO can operate at lower hydrogen excess air ratios (higher hydrogen concentration) compared to diesel fuel. When used as the pilot fuel, HVO enables operation at lower loads than diesel. Contour plots suggest that the distribution of pilot fuel spray, influenced by factors like ignition delay and hydrogen concentrations, primarily drives differences in combustion behavior. It is assumed that the shorter ignition delay of HVO allows for a narrower spray distribution, reducing autoignition tendencies and enabling operation at higher hydrogen concentrations and low loads. To achieve high efficiency and low emission when operating using HVO and diesel as pilot fuel, the engine is recommended to be operated at around $\phi_{\text{H}_2}=3.5$ and $\phi_{\text{H}_2}=4.0$, respectively.

Chapter 3 discusses the visualization of hydrogen dual-fuel combustion processes

using an optically accessible rapid compression and expansion machine (RCEM). The effect of hydrogen concentration, pilot injection pressure and pilot injection quantity was studied while utilizing diesel as pilot fuel. A small amount of diesel fuel was injected as a pilot fuel with injection pressures of 40, 80, and 120 MPa using a common rail injection system. The injection amount of diesel fuel was varied as 3, 6, and 13 mm³. The amount of hydrogen was manipulated by varying the total excess air ratio (λ_{total}) at 3 and 4. Then, as HVO exhibit shorter ignition delay which could affect the combustion behavior, the visualization of hydrogen-HVO combustion process was also conducted, while hydrogen-diesel dual fuel operations were studied as reference. Employing the same RCEM during the study with diesel as pilot fuel, with the exception that the RCEM is oriented vertically to prevent oil from ascending onto the piston. The study tested pilot injection pressures of 40 and 80 MPa and HVO pilot injection quantities of 3.14 and 6.27 mm³ to match the injected energy of 3 and 6 mm³ diesel fuel. The total excess air ratio was maintained at 3.0. The hydrogen-air mixture was introduced into the combustion chamber and the pilot injection of diesel fuel was used as the ignition source. The RCEM was operated at a constant speed of 900 rpm, and the in-cylinder pressure were set at 5 MPa. The combustion processes were visualized by direct photography and OH* chemiluminescence photography using a high-speed camera and an image intensifier. Based on the results, it can be concluded that a decrease in the total excess air ratio with constant pilot injection reduces combustion duration, shifting the heat release rate from a dual to a single peak, and shorter combustion duration. Comparing different pilot fuel types, diesel fuel pilot spray ignites after impacting the piston bowl due to a longer ignition delay, while HVO, with a shorter ignition delay, causes ignition before wall impact, with the flame formed in the spray cone shape. Smaller pilot injection quantity leads to low peak of heat release rate and ignition of small pilot flame near the nozzle with longer combustion duration, while increasing pilot quantity spreads combustion with momentum to the squish area. Higher pilot injection pressure leads to lower peak of heat release rate and longer combustion duration. Visualization of combustion process demonstrated ignition after injection, spreading the high-temperature region along the bowl wall. At lower pilot injection pressure, HRR exhibit higher peak and short combustion duration. Visualization demonstrated a dense spray jet

shaped high-temperature region in the middle of combustion chamber quickly formed after ignition. With HVO, the heat release rate exhibit slight decrease in ignition delay, resulting in the flame becomes slightly broader. Visualization studies of combustion location and flame formation revealed a strategy to ensure high efficiency through combustion location in the center of combustion chamber, and thick and dense flame formation.

Water emulsified fuels serve as a complementary approach to reduce NO_x emission by lowering combustion temperatures, thus reducing NO_x and particulate emissions, and possibility to enhancing engine performance. Chapter 4 elucidates the combustion and emission characteristics of diesel fuel emulsified with water, focusing on the effects of varying water concentrations and droplet diameter. Emulsified fuels containing 2 to 24vol% water, with droplet sizes of $< 3 \mu\text{m}$, $26.5 \mu\text{m}$, and $39.8 \mu\text{m}$, were prepared by utilizing strong stirring and membrane emulsification techniques. These fuels were evaluated in a single-cylinder, air-cooled direct injection diesel engine operating at a constant 2300 rpm. It was concluded that increasing water content improves pollutant emissions, significantly reducing NO_x and soot due to lower combustion temperatures and reduced carbon input, though CO emissions show a trade-off. Higher water content results in lower heat input, reducing output power for all fuels, but this can be slightly improved by increasing water droplet size. A water content of 20vol% is favorable for improving thermal efficiency with larger water droplet emulsified fuel, as the increased ignition delay boosts the peak heat release rate. Efficiency declines beyond these water content levels due to increased burn ratio and higher energy losses. Both water content and droplet size increase burn ratio by extending ignition delay, shifting combustion to the diffusion-controlled phase, although an optimal range exists for both factors to maintain a balance between premixed and diffusion combustion. Larger water droplets reduce water droplet number density and lead to combustion behaviors similar to neat diesel fuel.

List of publications and conference presentations

Chapter 2

Published paper: G. A. Mukhtar, K. Tange, S. Nakatani, N. Horibe, H. Kawanabe, G. Morita, K. Hiraoka and K. Koda, Performance and Emissions of a Hydrogen Dual-Fuel Engine Using Diesel and HVO as Pilot Fuels, SAE Technical Paper 2024-014286, 2024. DOI:10.4271/2024-01-4286.

Conference presentation: G. A. Mukhtar, K. Tange, S. Nakatani, N. Horibe, H. Kawanabe, G. Morita, K. Hiraoka and K. Koda, Performance and Emissions of a Hydrogen Dual-Fuel Engine Using Diesel and HVO as Pilot Fuels, SAE Energy & Propulsion Conference & Exhibition, November 12-14, 2024, Columbus, Ohio, USA.

Chapter 3

Published paper: G. A. Mukhtar, K. Shimogawa, N. Horibe, J. Hayashi, H. Kawanabe, G. Morita and K. Hiraoka, Combustion Analysis of Hydrogen-DDF Mode Based on OH* Chemiluminescence Images, SAE Technical Paper 2024-012367, 2024. DOI: 10.4271/2024-01-2367.

Conference presentation: G. A. Mukhtar, K. Shimogawa, N. Horibe, J. Hayashi, H. Kawanabe, G. Morita and K. Hiraoka, Combustion Analysis of Hydrogen-DDF Mode Based on OH* Chemiluminescence Images, SAE WCX 2024, April 16-18, 2024, Detroit, Michigan, USA.

In preparation for submission: G. A. Mukhtar, N. Une, N. Horibe, J. Hayashi, H. Kawanabe, K. Hiraoka and K. Koda, A Study on Combustion Process in Hydrogen-HVO Dual Fuel Operation using OH* Chemiluminescence.

Chapter 4

Published paper: G. A. B. Mukhtar, T. Nagata, J. Hayashi, N. Horibe, H. Kawanabe, and M. Fuchihata, Analysis of emission and combustion behaviors of water-emulsified diesel with various water contents and droplet diameters, JSME Journal of Thermal Science and Technology, vol. 20, no. 1, 2025. DOI: 10.1299/jtst.24-00385.

Conference presentation: G. A. Mukhtar, M. Fuchihata, Effect of water droplet diameter and water content in emulsified fuel on the operation characteristics of diesel engine, International Conference on Power Engineering (ICOPE-2021), October 17-21, 2021, Virtual Conference, online.

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