

A Study on Combustion Process in Hydrogen/Hydrotreated Vegetable Oil Dual Fuel Operation using Hydroxyl Radical Chemiluminescence

Abstract

This study investigated the combustion processes in hydrogen dual-fuel operation using hydrotreated vegetable oil (HVO) and diesel fuel as pilot fuels. The visualizations of hydrogen dual fuel combustion processes were conducted using hydroxyl radical (OH*) chemiluminescence imaging in an optically accessible rapid compression and expansion machine (RCEM) which can simulate a compression and expansion stroke of a diesel engine. Pilot injection pressures of 40 and 80 MPa and injection quantities of 3, 6 mm³ for diesel fuel and to match the injected energy, 3.14, 6.27 mm³ of HVO were tested. The total excess air ratio was kept constant at 3.0. The RCEM was operated at a constant speed of 900 rpm, with in-cylinder pressure at top dead center (TDC) set to approximately 5.0 MPa. Results demonstrated that using HVO as pilot fuel, compared to diesel fuel, led to shorter ignition delay and combustion duration. OH* chemiluminescence imaging revealed that longer ignition delays observed with diesel fuel resulted in pilot mixture ignition downstream near the piston bowl wall, followed by flame propagation into the hydrogen-air mixture. In contrast, the shorter ignition delays characteristic of HVO caused the pilot mixture to ignite between the injector and the piston bowl wall, with subsequent flame propagation into the hydrogen premixture.

Keywords: Hydrogen dual fuel, HVO, combustion visualization, OH* chemiluminescence, diesel engine

1. Introduction

Internal combustion engine has been a major prime mover in transportation sectors and the main contributor to the greenhouse gas emissions [1]. 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 electricity through electric cars can indirectly emit greenhouse gas, because the electric generation has been reported to mostly emit the greenhouse gas through the combustion of fossil fuel [2]. 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. This is primarily due to the high cost and weight of the batteries, combined with the limited availability of charging infrastructure along long-haul routes [3]. Conventional diesel heavy duty vehicle can travel 1570 to 3138 km per tank (570-1135 liters, based on calculation of approximately 2.3 km/l) [4], whereas battery-electric trucks currently achieve only 100-200 km per charge [5,6]. Extending the range to about 800 km would require a roughly 1000 kWh battery weighing approximately 5500 kg [7], significantly reducing payload and making long-range electrification impractical with current technology [8]. 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, easy access to diesel fuel and the well-established network for its distribution support the ongoing use of diesel engines in heavy duty operations. This presents serious issue as according to the UN Environment Programme (UNEP), although heavy-duty vehicles which are empowered by diesel engines, constitute merely 3.6% of the global automotive trade's total value, they account for 40% of on-road nitrogen oxides (NO_x) emissions, over 60% of on-road particulate matter (PM 2.5), and more than 20% of black carbon emissions [9]. They also account for about 25% of the carbon dioxide (CO₂) emissions in the transportation sector, which corresponds to about 5% of total global CO₂ emissions. [10].

To tackle this issue, replacing the fossil fuel in diesel engine with alternative resources is crucial and prompt way to limit global warming and pollution. This has prompted researchers to explore innovative solutions such as hydrogen, biofuels, and synthetic fuels.

In recent years, hydrogen has garnered considerable attention as a potential future energy source. This interest stems from zero carbon emissions upon hydrogen combustion and its high-speed flame propagation, which can enhance thermal efficiency in internal combustion engines [11,12,13,14,15,16]. Additionally, hydrogen can be derived from various

sources, including renewable resources [17,18]. Spark ignition (SI) hydrogen engines have emerged as the main trend in the industry, largely because they can operate under lean premixed conditions, achieving very low NO_x emissions. However, lean SI combustion is limited in terms of maximum load. While in heavy-duty applications, this limitation can be overcome by increasing engine displacement, this presents a drawback in applications where high power density and compact engines are essential.

For these applications, compression ignition (CI) engines remain more practical. However, in CI engines, utilizing hydrogen as an energy source presents challenges due to its high ignition temperature (858 K [19,20] compared to 483 K for diesel fuel), which is significantly higher than that of diesel fuel [19]. This characteristic renders hydrogen challenging to employ as a sole fuel in the engines, as its limited ignition ability [12,21,22] can result in reduced control over the ignition process, leading to abnormal combustion and restricting the operating conditions of the engine. The abnormal combustion is defined as unintended or uncontrolled ignition, which deviate from normal compression ignition. Pre-ignition and knocking are classified as abnormal combustion. Hydrogen has a very low minimum ignition energy (for reference, approximately 0.02 mJ for hydrogen compared with about 0.25 mJ for gasoline) [19], meaning that ignition can be easily initiated by hot spots in the combustion chamber, particularly at high loads [21,23]. This can lead to pre-ignition when combustion occurs before the intended ignition timing and can also result in knocking due to the rapid pressure rise from uncontrolled autoignition. Such uncontrolled combustion produces severe mechanical stress on the engine, which may eventually damage engine components.

The combustion of hydrogen as sole fuel in CI engines has been conducted by Antunes et al. under homogenous charged compression ignition (HCCI) mode [21]. It was reported that to achieve the ignition of hydrogen, the engine needs to be operated at significant higher compression ratio to about 18 to 25, compared to 10 in SI engine. Another method to achieve ignition is by preheating the intake air. This also allow control of the ignition of hydrogen, by controlling the intake air temperature. In their study, the intake air was preheated to above 85°C. However, increasing the intake air temperature resulted in a decrease in power and brake thermal efficiency. Moreover, hydrogen HCCI operation imposes significant mechanical limitations due to high peak pressures and rapid pressure rise [21], necessitating redesigned and strengthened engine components such as wider crank bearings and thicker piston rings to ensure durability and safe operation.

However, controlling hydrogen ignition is critical for effective application. To address these challenges, the dual-fuel combustion method has shown promise in providing fuel flexibility and improving engine performance. In this approach, hydrogen is combusted using diesel fuel or other liquid fuels as the pilot ignition source. This method 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. Deb et al. [24] 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 demonstrated that higher hydrogen concentrations result in elevated peak in-cylinder pressure and heat release rates, thereby enhancing combustion efficiency.

Hydrogen-diesel dual fuel combustion with higher hydrogen content has been investigated by Tsujimura et al. [25]. 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 exceed approximately 50%, the probability of abnormal combustion, such as pre-ignition, increases, potentially limiting operable conditions.

Moreover, Tomita et al. [26] investigated hydrogen-diesel dual-fuel combustion in a single-cylinder, direct injection diesel engine. The study explored a wide range of injection timings and hydrogen energy fractions (up to 80%) at constant engine speed. It was observed that early diesel pilot fuel injection promoted lean premixed combustion, significantly reducing smoke and NO_x emissions nearly to zero. At mid-range diesel fuel injection timings (2.5-23.7° BTDC), dual peaks are observed, with the first peak is due to heat released from diesel fuel ignition, and second peak attributed to hydrogen combustion initiated by diesel fuel ignition. The study also reported longer ignition delays with higher hydrogen content. The ignition delay was influenced by the mole fraction of oxygen and the temperature of the gas in the cylinder. When the volumetric ratio of hydrogen was increased, the mole fraction of oxygen became smaller. Moreover, the heat loss from the gas to the cylinder wall is greater due to higher thermal conductivity of hydrogen compared to air causing lower gas

temperature, despite having almost same specific heat ratio. CO, CO₂ and smoke was reduced at higher hydrogen content, but NO_x emissions vary depending on the conditions.

Recently, Volvo Group and Westport Fuel Systems launched a joint venture called Cespira to bring hydrogen combustion engines to the market for heavy-duty trucks [27, 28]. This partnership is centered on creating high-pressure direct injection (HPDI) fuel systems for long-distance and off-road uses. Unlike hydrogen fuel cells, these engines burn hydrogen directly within internal combustion systems similar to those used for natural gas, requiring only slight design changes and allowing for the use of lower-purity hydrogen. The key innovation of the system is a patented injector with a dual concentric needle design, which injects a small amount of pilot fuel such as HVO or diesel, before the hydrogen to start combustion. This method results in up to a 97% reduction in CO₂ emissions while keeping NO_x and particulate levels low, in line with current Euro 6 and proposed Euro 7 standards. The technology provides benefits in terms of cost, scalability, and high-load performance, making it a promising option for decarbonizing areas like long-haul transport where battery-electric systems are less feasible [27, 28].

The hydrogen-diesel dual fuel combustion has demonstrated great benefits through high efficiency and great carbon reduction. However, concurrently, the drive for accelerated carbon neutrality realization in engines has propelled efforts towards biodiesel as well. This alternative fuel offers the potential to emit net zero carbon and decrease dependence on fossil fuels. As the global community continues to seek sustainable energy solutions, the synergy between hydrogen and biofuels could play a crucial role in shaping the future of clean energy technologies. To further improve hydrogen dual fuel combustion using biofuels HVO has been a great candidate as pilot fuel. Its significant properties of high cetane value (calculated cetane index of 95 for HVO, compared to 60 for diesel fuel), exhibits better ignition compared to conventional diesel fuel. Furthermore, unlike ester-type biodiesel fuels, HVO does not exhibit the negative effects particular to biodiesel such as oil aging [29], nor does it lead to increased NO_x emissions or deposit buildup [30]. While ester-type biodiesel typically reduces soot emissions but tends to slightly increase NO_x emission, HVO shows minimal influence on NO_x, thereby avoiding the conventional soot- NO_x trade-off observed with ester-type biodiesel fuels. Moreover, the lack of aromatics of HVO has significantly reduced smoke emissions [31, 32]. Bortel et al. [31] in their study, has clarified the effect of HVO and its blends with diesel fuel combustion onto 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 maximum reduction of

80% with neat HVO, but NO_x emission only demonstrated slight reduction. Smigins et al [32] compared the performance and emissions of a diesel engine using HVO, fossil diesel fuel, 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 fuel. However, HVO slightly increased NO_x emissions by 3%. Moreover, the maximum power, torque and reduction of specific fuel consumption was achieved during HVO5, despite the highest NO_x emissions was observed with an average of 8.8% increase.

Jaworski et al. [33] investigated HVO and its blend with pyrolytic oils in a Euro 3 diesel engine under the Worldwide Harmonized Light Vehicles Test Cycle (WLTC) dynamic test conditions. 3 types of fuels were tested, diesel (B7), pure HVO, and a 90% HVO blend with 5% tire pyrolytic oils (TPO) and 5% polystyrene pyrolytic oils (PSO). B7 refers to European commercial designation for typical diesel fuel with an allowable FAME content of up to 7% by volume. HVO achieved the lowest CO₂, CO, THC, NO_x, and PM emissions. Maximum reduction of CO₂ by up to 9%, NO_x by 22%, and PM by 70% compared to B7, due to its high cetane number and paraffinic nature. The HVO-TPO-PSO mixture increased CO₂ and NO_x higher compared to pure HVO. Fuel consumption for HVO was similar to diesel, while the HVO-TPO-PSO blend showed a slight increase. The study concluded that HVO provided the most favorable trade-off between emissions and efficiency.

Serrano et al. [34] compared HVO-diesel blend (15% HVO, namely HVO15) with diesel (B0) and biodiesel blends (B7, B15, B100) in a Euro V diesel vehicle under New European Driving Cycle (NEDC) and Worldwide harmonized Light vehicles Test Procedures (WLTP) test cycles. The study evaluated fuel consumption, emissions, and engine performance using standardized chassis dynamometer tests. It was observed that HVO15 resulted in almost identical performance to pure diesel, with fuel consumption increasing only slightly (maximum 0.6%) which was significantly lower than the increase observed for B15 (maximum 1.9%). Engine power and torque remained nearly unchanged, indicating that HVO15 could serve as a direct diesel substitute without calibration changes. NO_x emissions were similar or slightly lower than diesel, depending on the test condition, and CO₂ emissions were comparable to diesel.

From these reports, HVO itself has demonstrated great benefits in terms of emission and engine performance and is a great candidate to realize carbon neutrality. However, the combination of HVO with hydrogen or hydrogen-HVO dual fuel engine is still yet not actively studied with only few reports could be found. One example is a dual-fuel operation

conducted by Pinto et al., in which a small amount of hydrogen is added to the intake air as a secondary fuel, while the main fuel is HVO or farnesane biodiesel [35]. They have studied the effect of hydrogen addition of up to 28% of energy fraction to the engine operated with HVO and farnesane biodiesel. The maximum hydrogen energy fraction of 28% represented the highest achievable value that prevented engine knocking and damage while ensuring stable combustion. Emission results showed that neat HVO had lower NO_x , particulate matter (PM), CO, and CO_2 emissions compared to diesel fuel. With hydrogen enrichment, there was a significant reduction in PM, CO, and CO_2 emissions for HVO, although NO_x emissions increased by up to 62%. Hydrogen enrichment slightly reduced brake thermal efficiency (BTE), but both HVO single and dual-fuel modes exhibited higher brake thermal efficiency and lower greenhouse gas emissions compared to diesel-hydrogen dual-fuel operation, suggesting that HVO with hydrogen enrichment is a promising alternative to diesel fuel in terms of both efficiency and emissions reduction [35].

At higher hydrogen concentrations, where hydrogen is fully utilized as the primary fuel and hydrotreated vegetable oil (HVO) serves as the pilot fuel-available knowledge remains limited due to the scarcity of existing studies. To clarify an efficient and low-emission operation strategy, our previous study [36] investigated the injection of a small quantity of HVO to ignite the hydrogen-air mixture. Experiments were conducted in a single-cylinder diesel engine, comparing hydrogen-HVO dual-fuel combustion with hydrogen-diesel dual-fuel combustion. It was found that using HVO as the pilot fuel provides a wider operable range without abnormal combustion at both high and low loads. This is attributed to the higher cetane number of HVO compared to diesel, which reduces the ignition delay. Consequently, a narrower pilot fuel distribution slows down the chemical reactions in the combustion chamber, suppressing hydrogen autoignition during hydrogen-HVO combustion. However, at higher HVO pilot quantities, thermal efficiency decreased, primarily due to increased heat transfer to the cylinder walls (cooling loss). The cause of this increased cooling loss, calculated via the energy balance equation, remains unclear. Combustion visualization can help clarify this behavior by revealing the combustion location and flame propagation speed; for example, combustion occurring near the piston wall and squish areas may explain the increased energy losses.

Regarding hydrogen dual fuel utilizing diesel as pilot fuel, until recently, quite a few visualization reports have been conducted. Rajasegar et al. [37] explored the effects of hydrogen addition on the ignition and combustion characteristics of a pilot diesel-spray in lean-premixed hydrogen-internal combustion engines operating in dual-fuel mode. Their

investigation utilized sophisticated imaging methods, such as high-speed cool-flame and OH* chemiluminescence techniques, to examine the ignition behavior of hydrogen-diesel dual-fuel systems. The researchers also performed a comparative evaluation with natural gas dual-fuel combustion. Results from the flame visualization experiments demonstrated that the presence of hydrogen significantly hindered the transition between first stage and second-stage autoignition reactions, resulting in smaller ignition kernels.

Moreover, in our previous study [38], an optically accessible rapid compression and expansion machine (RCEM) was utilized to visualize hydrogen-diesel dual-fuel combustion, examining factors such as hydrogen concentration, pilot injection pressure, and pilot fuel quantity. Through OH* chemiluminescence imaging and direct photography, the results demonstrated that varying pilot injection pressures and quantity influenced the pilot spray development and combustion process. Lower pilot injection pressures were found to cause a rapid, conical expansion of the pilot flame near the injector. Conversely, higher injection pressures resulted in an expanded area of intense OH* chemiluminescence near the piston bowl wall, indicating increased cooling losses. Meanwhile, with reduced pilot injection quantity, ignition began near the injection nozzle, and the high-temperature region gradually spread throughout the piston bowl. This resulted in an extended combustion duration.

However, since HVO differs from diesel fuel 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 fuel is used as the pilot fuel. Our previous study [36] also demonstrated this when examining engine performance results using HVO as pilot fuel in hydrogen dual fuel mode. It was found that HVO achieved slightly higher thermal efficiency than diesel fuel at high hydrogen concentrations, attributed to reduced cooling loss. However, increasing the HVO injection quantity results in greater cooling loss, which subsequently deteriorates thermal efficiency. Therefore, to clarify the cause of the performance differences between HVO and diesel fuel, and to ensure a reliable operation strategy for HVO as pilot fuel in hydrogen dual fuel mode, the visualization of hydrogen-HVO dual fuel mode is conducted in this study. By comparing these visualizations with the hydrogen-diesel dual fuel mode, the combustion location and flame formation process are investigated.

2. Experimental setup and conditions

2.1 Pilot fuels

HVO and diesel fuel JIS No. 2 served as the pilot fuels in this study. The characteristics of these pilot fuels and hydrogen are presented in Table 1. The density of HVO is lower than that of diesel fuel because HVO is mostly composed of linear hydrocarbons and contains no aromatic content. Although the cetane index is applicable only to petroleum-derived diesel fuel because it is derived from correlations between density and distillation characteristics specific to petroleum hydrocarbons and beyond the scope for paraffinic fuels such as HVO, it was applied here to HVO. It must be stated here that while a cetane index cannot be directly applied to HVO, an estimated cetane index was calculated as an indicator of relative ignitability. Diesel fuel has a cetane index of 60, whereas the calculated value of the cetane index of HVO is 95, suggesting that HVO is expected to have higher ignitability than diesel fuel. 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 under various conditions.

Table 1. Fuel properties. Data on HVO taken from Ref. [36].

	Test method	Unit	HVO	Diesel fuel	Hydrogen [*]
Density (15°C)	JIS K2249-1	kg/m ³	780.4	822.5	0.084
Kinematic viscosity (30°C)	JIS K 2283	mm ² /s	3.848	3.564	105 (20°C) ^{**}
Lower heating value	JPI-5S-65-11	MJ/kg	43.67	42.99	120
95% distillation point	JIS K 2254	°C	295	358	-
Cetane number	JIS K 2280-5	-	-	54.5	-
Cetane index	JIS K 2280-5	-	95 (reference value)	60	-
Autoignition temperature	-	K	477 ^{***}	483	858 [19, 20]

^{*}The data for hydrogen were obtained from Refs. [23, 39], independent of the test method. The density and kinematic viscosity values are reported at 1 atm.

^{**}Kinematic viscosity of hydrogen is reported at 20 °C, while HVO and diesel fuel values are at 30 °C.

^{***}The data on autoignition temperature of HVO was obtained from Ref. [40].

2.2 Visualization of combustion process using RCEM

An RCEM was employed to visualize a single combustion process in hydrogen-dual fuel mode. Adapted from a single-cylinder, naturally aspirated four-stroke diesel engine (Yanmar NFD170), the RCEM is capable of performing a single compression and expansion stroke under controlled conditions. It features a custom-designed cylinder head equipped with a quartz window and an independent intake/exhaust valve to facilitate combustion visualization. The specifications of the RCEM used in this experiment are summarized in Table 2. The experimental setup of this study is illustrated in Figure 1.

Table 2. Technical specifications of the RCEM. Data taken from Ref. [38].

Bore × Stroke	mm	102 × 105
Displacement	cc	857
Compression ratio	-	15.4
Pilot fuel injector hole diameter × number of holes	mm	φ0.123 × 7 holes
Spray cone angle	deg	156

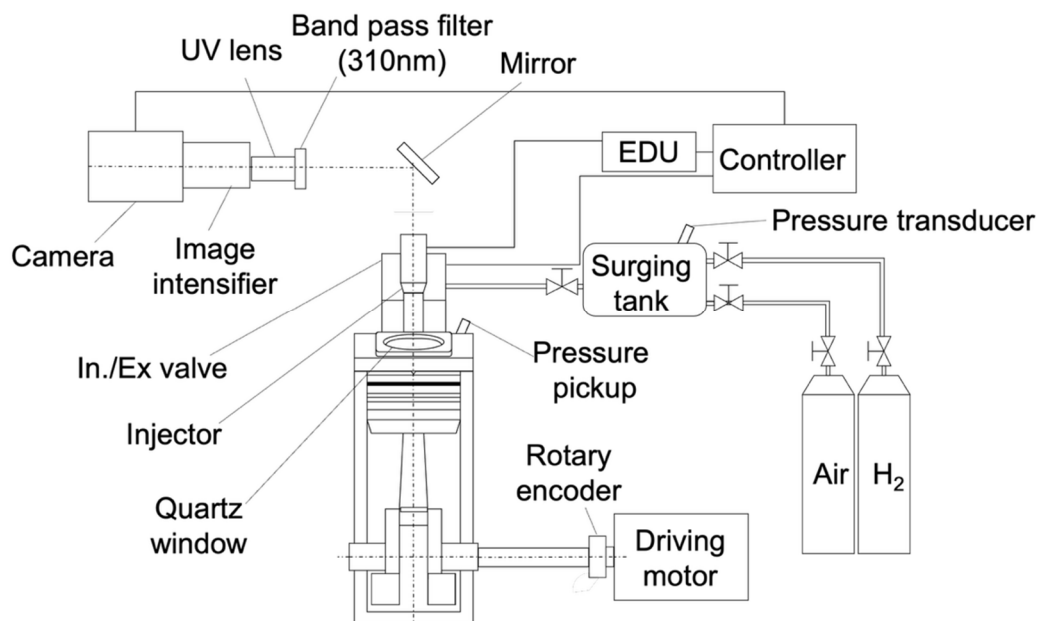


Figure 1. Schematic of experimental setup. Reprinted with permission from Ref. [38]; copyright SAE International.

Hydrogen (lower heating value: 120 MJ/kg) and air are mixed in the mixing tank outside the cylinder before being inducted into the cylinder through the independently controlled intake/exhaust valve. To ensure correct mixture concentration, the amount of the hydrogen and air supplied to the tank was monitored by measuring the absolute pressure in the tank using an absolute pressure transducer (KYOWA, PAB-A-500KP). The pilot fuel is delivered through a common rail injection system and a fuel injector (Denso, G3P), similar to the setup used in experiments during continuous engine operation in our previous study [36]. The intake and exhaust timing, as well as fuel injection timing, were synchronized with the crank angle positions, detected by a rotary encoder connected to the shaft. To precisely control the intake valve opening and closing as well as the fuel injection timing, a timing controller (LabSmith LC880) was employed.

During operation, the engine was externally motor-driven at 900 rpm using a driving motor (Mitsubishi Electric SF-PR18.5KW 4P 200V) and general-purpose inverter (Mitsubishi Electric, FR-A820-18.5K-E1). During cranking, the intake/exhaust valve is opened, allowing the mixture of hydrogen and air to be compressed and as intake temperature increases until reaching the prescribed temperature, a trigger signal was sent from the timing controller to close the intake/exhaust valve and initiate fuel injection at a specified crank angle. This setup enables a flexible and accurate testing conditions as in single compression and expansion stroke of a diesel engine.

Regarding combustion visualization, OH* chemiluminescence imaging has been employed as visualization method in this study. The OH* chemiluminescence is emitted from high temperature regions during combustion, hence the flame formation from both pilot fuels and hydrogen combustion could be observed. It was captured through a mirror reflecting the quartz window installed in the cylinder head, as shown in Figure 1. The combustion was captured using a high-speed camera (Photron Fastcam SA-Z). The chemiluminescence wavelength for OH* was 309 nm, hence, a bandpass filter (Asahi Spectra UZ0310; center wavelength 310 nm, half-width 10 nm) and a UV lens (Nikon UV-Nikkor; focal length 105 mm, F value 4.5) was employed. Due to the low intensity of OH* chemiluminescence, an image intensifier (Hamamatsu C10880-03F) was included in the setup to amplify the signal. The imaging conditions are detailed in Table 3.

Table 3. Imaging conditions for the combustion visualizations. Data taken from Ref. [38].

Frame rate	fps	50400
Wavelength	nm	310 (FWHM 10)
Range	pixel $\times$ pixel	512 $\times$ 512
Resolution	mm/pixel	0.0947
Exposure (camera)	μ s	18.23
I.I. gain	-	880/900
I.I. gate	-	open

The outline of the visualization range used to capture combustion and spray behavior in the cylinder is shown in Figure 2. The quartz 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 left half of the field of view covers the combustion chamber cavity, while the right half corresponds to the squish area.

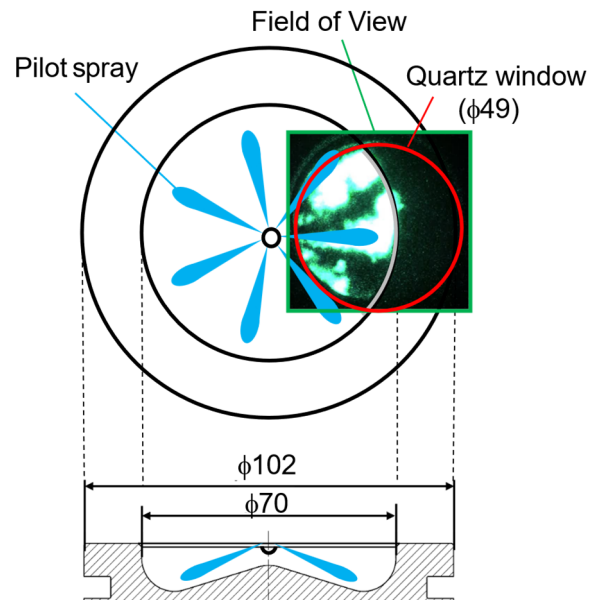


Figure 2. Schematic of visualization area.

The experimental conditions for this study are shown in Table 4. The in-cylinder pressure at TDC was set to 5 MPa, the temperature to 700 K, and the overall excess air ratio, λ_{total} was fixed at 3.0. These in-cylinder pressure and temperature conditions are chosen to

represent those in an engine with a compression ratio of 14.3 operating at a frequently used gross IMEP of 0.8 MPa and an intake pressure of 140 kPa. Moreover, $\lambda_{total} = 3$ corresponds to the highest hydrogen concentration that avoid preignition. Attempts to increase the hydrogen concentration by reducing the λ_{total} to 2.5 resulted in frequent preignition, reducing feasibility of the experiment. λ_{total} is defined as the overall excess air ratio considering both the pilot fuel and hydrogen, and is calculated using Eq. (1). 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, q_j was set to 6.27 mm³ to match the injected energy of the diesel fuel. Additionally, to investigate the effect of injection pressure on combustion characteristics, experiments were conducted at a reduced injection pressure of 40 MPa. A lower injection pressure is expected to reduce spray penetration and prolong the injection duration, which may enhance combustion uniformity. Furthermore, to analyze the combustion process at higher hydrogen energy fractions, experiments were also conducted with half the HVO injection quantity, $q_j = 3.14$ mm³. The energy ratio replaced by hydrogen is approximately 93% under this condition, while at 6.27 mm³, the hydrogen energy ratio is approximately 86%. These high substitution ratios demonstrated that dual fuel operation enabled a significant replacement of the pilot fuel with hydrogen. For all conditions, the pilot injection timing was adjusted to maintain a constant ignition timing of 2°ATDC ($\pm 1^\circ$ 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 (the point where the derivative of the heat release rate reaches its maximum value) intersects the crank angle axis. The illustration of how ignition timing is determined will be explained later in the discussion of Figure 3.

$$\lambda_{total} = \frac{m_{air,actual}}{m_{air\ stoich,pilot} + m_{air,stoich,H2}} \quad (1)$$

where $m_{air,actual}$ is the actual mass of air supplied, and $m_{air\ stoich, pilot}$ and $m_{air\ stoich, H2}$ represent the stoichiometric air masses required to completely burn the supplied pilot fuel and hydrogen quantities, respectively.

Under these conditions, the ambient density at TDC is expected to remain constant for a given injection quantity. Furthermore, since the oxygen demand of HVO and diesel fuels is nearly identical, variations in fuel type are not expected to significantly affect the ambient density. The difference in injection quantity between the two cases is only 3.13 mm³ (equivalent to approximately 2.4 mg of fuel), resulting in a negligible change in oxygen demand and, consequently, a minor variation in ambient density of approximately 0.2 kg/m³.

Therefore, it can be concluded that the ambient density at TDC is effectively uniform across all test conditions in this study.

Table 4. Operation conditions utilizing HVO and diesel fuel as pilot fuel. Data taken from Ref. [38].

	Unit	HVO		Diesel fuel
Cranking speed	rpm	900		
In-cylinder pressure at TDC	MPa	5.0±0.1 MPa		
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	6.00
H ₂ excess air ratio	-	3.23	3.49	3.63
H ₂ energy ratio	%	93.1	86.1	85.3
Ignition timing	°ATDC	2.0 ± 1.0 (by pilot fuel injection timing adjustment)		
Injection pressure p_j	MPa	40, 80		

3. Results and discussions

3.1 Combustion process and heat release rate of diesel-H₂ combustion

To study 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 $q_j = 6\text{ mm}^3$. Figure 3 shows the plots of in-cylinder pressure and heat release rate versus the crank angle. In the results, the injection duration of the pilot fuel is indicated using a straight horizontal line and marked as 'Injection' in the graphs. Under this condition, the injection started at approximately -10°CA and finished at approximately -7°CA . As previously mentioned, the ignition timing is determined by identifying the crank angle at which the tangent to the rapid rise in the heat release rate (the point where the derivative of the heat release rate reaches its maximum value) intersects the crank angle axis. This method is illustrated in Figure 3, where the red dashed lines represent the tangent, and the ignition timing is indicated as 'SOC'. In the graph, black colored dashed lines were added at certain crank angles to facilitate reference to the heat release rate behavior and combustion images at

specific crank angles later. When observing the heat release rate, a small heat release increase, likely caused by low-temperature oxidation, was observed approximately -1°CA , followed by a rapid increase in the heat release rate obtained from approximately 2°CA . This continues until 7°CA and declines after 10°CA . Under this condition, where diesel served as the pilot fuel, the ignition delay was notably long. This delay is defined as the interval from the start of injection (SOI) to the ignition timing, measuring approximately 12°CA under this condition. As a result of the extended ignition delay, the heat release rate increases slowly, with combustion persisting until around 20°CA . The combustion duration under this condition is approximately 18°CA . To study further this heat release rate behavior on combustion process and flame location, OH^* chemiluminescence visualization images corresponding to the heat release rates shown in Figure 3 are presented and discussed in Figure 4.

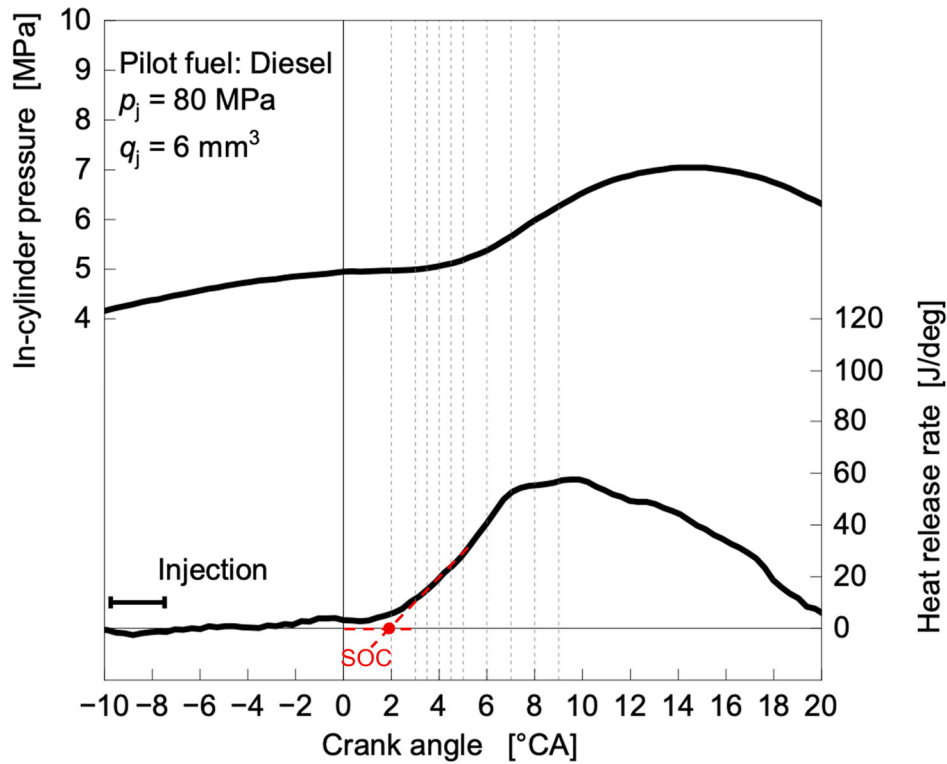


Figure 3. In-cylinder pressure and heat release rate (diesel fuel, $p_j = 80 \text{ MPa}$, $q_j = 6 \text{ mm}^3$).

Figure 4 provides insight into the combustion process by showing the visualization images of combustion in hydrogen-diesel dual fuel mode within the visualization range using OH^* chemiluminescence at different crank angles. In the images, the red circle marks the visualization window, the white curve represents the edge of piston bowl, and the arrow indicates the spray direction. Weak luminescence was first seen at 3°CA until 4°CA , along the spray axis, at a location close to piston bowl wall. This is mainly attributed to the

autoignition of diesel fuel. The region then spread from the points along the piston bowl wall, matching the rise of heat release rate in Figure 3. At the same time, a weaker luminescence region spread slowly toward the center of the visualization area, which is believed to be a propagating flame in the hydrogen-diesel mixture triggered by the autoignition of the pilot fuel. During this combustion, the ignition delay was long. Under this operating condition, the pilot injection quantity is small, and ignition occurs after a sufficiently long delay, well after the end of injection. This long ignition delay allows extensive mixing of the pilot fuel with the hydrogen-air charge, leading to more homogeneous and locally leaner mixture at the time of ignition, which corresponds to a lower local equivalence ratio. Consequently, weak luminescence was observed, and this correlates with the slow heat release rate in Figure 3.

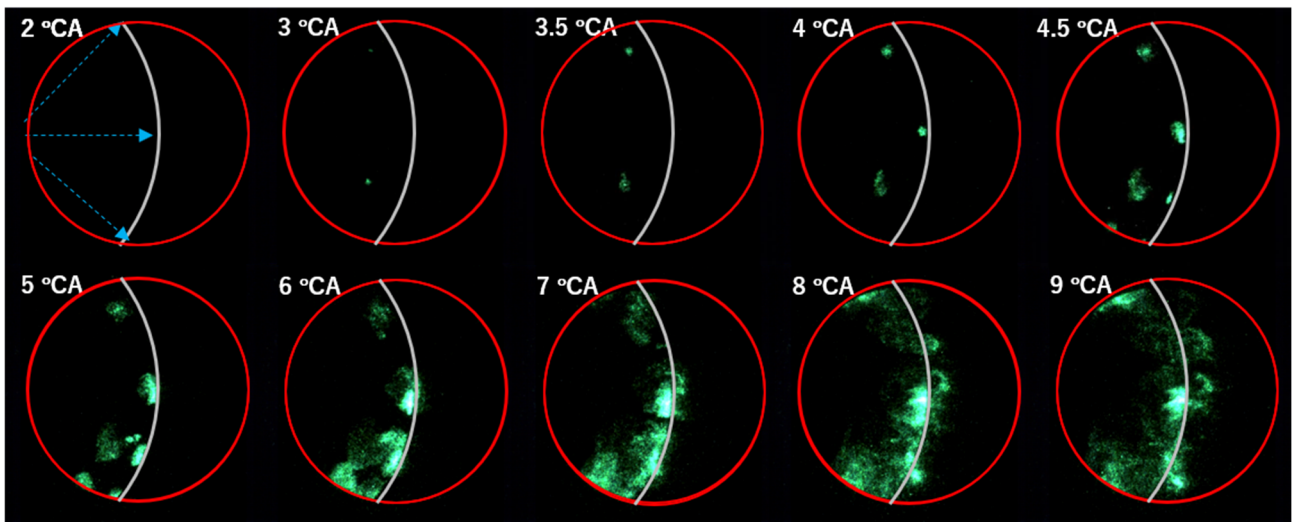


Figure 4. Images of OH* chemiluminescence (pilot fuel: diesel fuel, $p_j = 80$ MPa, $q_j = 6$ mm³).

3.2 Effects of pilot fuel on combustion process

To investigate differences in the combustion process based on the pilot fuel, HVO was then used as a pilot fuel. The 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 fuel. Figure 5 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 at 2°CA, HVO is injected at approximately -5°CA, approximately 5°CA later than when diesel fuel was injected as pilot fuel. This is due to higher cetane properties of HVO compared to diesel fuel. However, the injection duration of HVO was almost identical to that of diesel fuel, owing to the very small difference in injection quantity. In this study, the intake (mixing tank) pressure was not fixed; instead, the pressure at top dead center (TDC) was

maintained at 5.0 MPa. Since the exact TDC pressure resulting from a given mixing tank pressure was not known beforehand, several trial-and-error adjustments were performed to achieve the target TDC pressure. For practicality, a tolerance of ± 0.1 MPa was allowed. Under these conditions, the measured TDC pressure was 4.95 MPa for diesel and 5.06 MPa for HVO. Therefore, the slight difference in intake pressure between the two cases is likely responsible for the observed variation in the in-cylinder pressure before TDC. When observing the heat release rate, similar to diesel fuel condition, 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 fuel, the decline was more gradual, extending to 20°CA . The shorter ignition delay of HVO causes ignition to occur before significant fuel dilution can take place. As a result, the fuel-surrounding gas mixture at the ignition location remains relatively concentrated, producing a locally concentrated mixture. Moreover, the ignition delay was enough that the fuel is premixed well with air and hydrogen, but shorter compared to diesel fuel condition, that resulted in less excessive mixing of the pilot fuel, compared to diesel fuel. This leads to a higher heat release rate, and shorter combustion duration (approximately 12°CA). Furthermore, HVO exhibits dual peak heat release rate compared to single peak observed from diesel fuel. As reported by Zhou et al. [41] and Akcay et al. [42], 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 marked as '①' and the second peak is attributed to the combustion of premixed air-hydrogen, illustrated as '②' in Figure 5. The excessive mixing during diesel fuel as pilot fuel combustion resulted in the combustion of pilot fuel and hydrogen occurs at almost simultaneously, hence the single peak observed and this is marked as '①+②' in Figure 5. To investigate further regarding this matter, the combustion process is studied in the next figure. However, the third peak attributed to autoignition of premixed hydrogen at end gas region i.e. PREMIER combustion [43] or knocking could not be observed under this study, due to lack of residual heat effect and low combustion temperature under this RCEM operation.

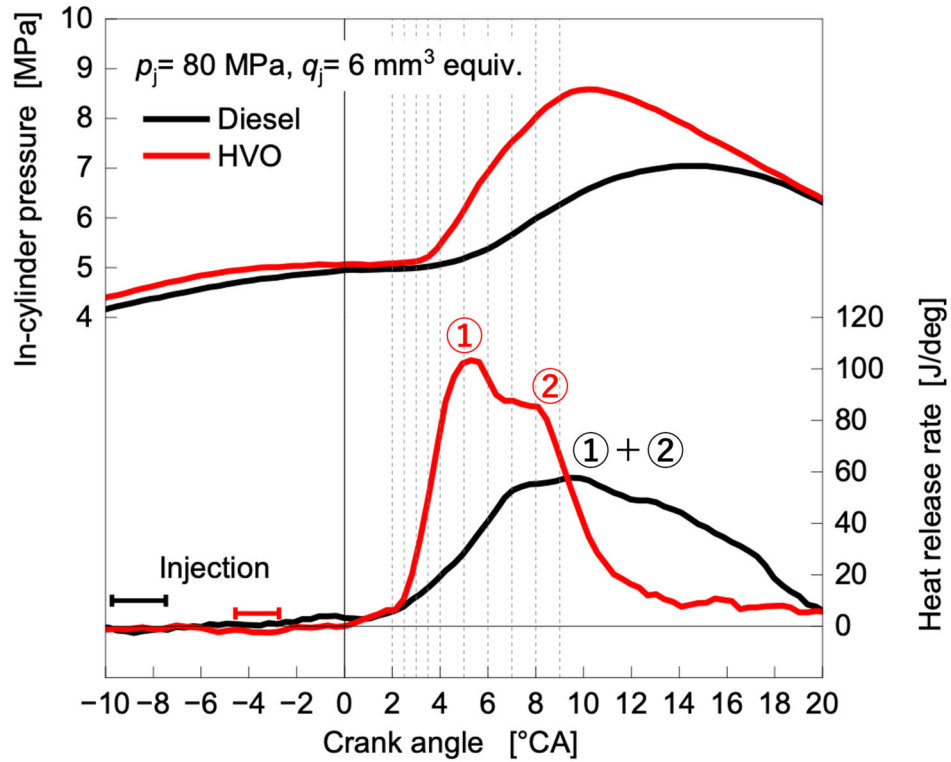


Figure 5. In-cylinder pressure and heat release rate (comparison of diesel fuel and HVO).

Figure 6 exhibits the visualization images of combustion in hydrogen-HVO dual fuel mode within the visualization range using OH^* chemiluminescence at different crank angles. These images correspond to heat release rate data using HVO as pilot fuel in Figure 5. 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 pilot fuel spray. This corresponds with the appearance of the second peak of heat release rate from 6°CA in Figure 5.

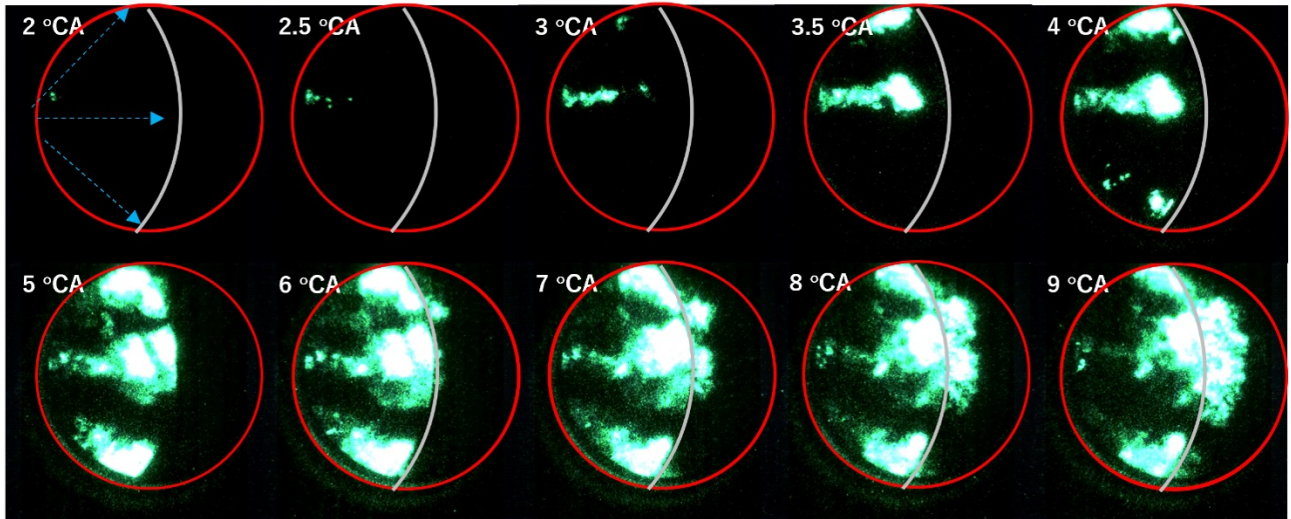


Figure 6. Images of OH* chemiluminescence (pilot fuel: HVO, $p_j = 80$ MPa, $q_j = 6.27$ mm³).

To facilitate a direct comparison between diesel fuel and HVO as pilot fuel in hydroegen dual fuel combustion characteristics, Figure 7 presents the OH* chemiluminescence images at selected crank angles together with the corresponding in cylinder pressure and heat release rate profiles. The upper row shows the images for diesel fuel (previously shown in Figure 4), and the lower row shows those for HVO (previously shown in Figure 6) at 4°CA, 6°CA and 8°CA. The heat relase rate traces on the right (from Figure 5) allow correlation between the visual flame development and the combustion progress for both pilot fuels. It was observed that the luminescence region attributed to pilot fuel combustion was more intense with HVO and formed broader ignition kernels along spray axis. This contrast is attributed to the difference in ignition delay and the resulting mixture states. Diesel fuel, having a longer ignition delay, experienced ‘over-mixing,’ where the fuel dispersed excessively before ignition, HVO’s higher cetane number leads to a significantly shorter ignition delay. This suppressed over-mixing, allowing ignition to occur while the pilot vapor was more concentrated. Consequently, HVO shows the stronger and more widespread OH* chemiluminescence, which corresponds to the higher heat release rate.

In contrast, the longer ignition delay of diesel fuel leads to increased dilution rate of the pilot fuel mixture, resulting in a more dilute fuel-vapor mixture. Therefore, the combustion progressed slowly, and the temperature was low; therefore, a lower luminescence intensity was observed when diesel fuel was used as the pilot fuel.

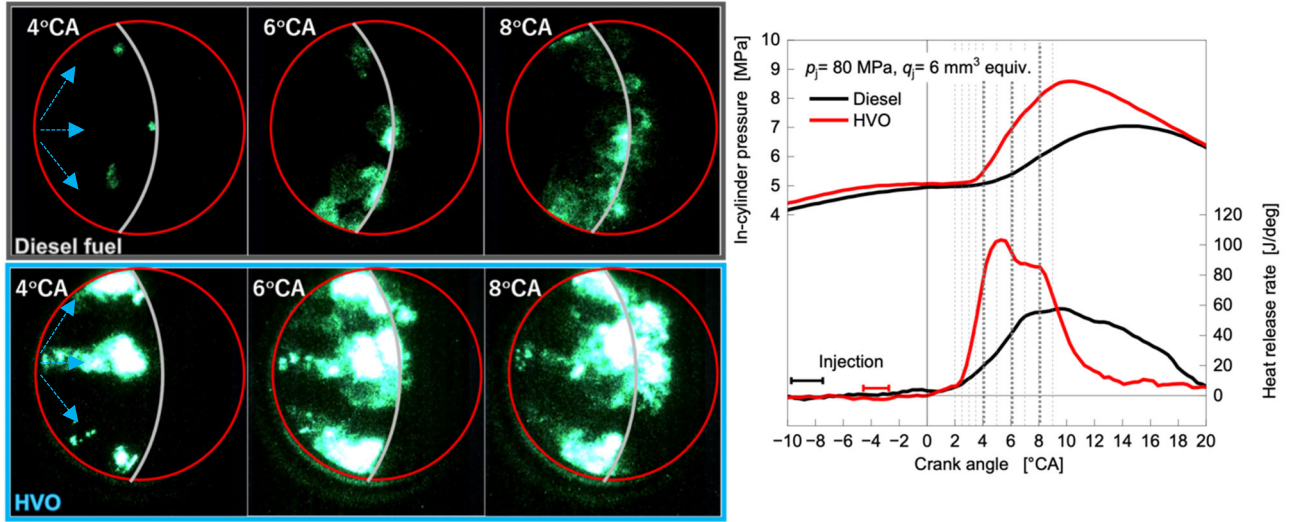


Figure 7. Comparison of OH* chemiluminescence images at selected crank angles for diesel fuel (top row) and HVO (bottom row) as pilot fuel combustion, together with the corresponding in-cylinder pressure and heat release rate profiles.

Although the residual heat under continuous engine operation could not be simulated in this study, these combustion images can possibly serve as a reference when comparing the engine performance of diesel fuel and HVO combustion in hydrogen dual fuel operation. During HVO combustion, as seen in the visualization images, ignition occurs very close to the injector, forming a spray shape. This results in combustion taking place far from the piston bowl wall, leading to reduced heat loss to the wall. In contrast, when diesel fuel was used as pilot fuel, the ignition delay was relatively long, resulting in combustion occurring well after the injection completes, and thus taking place around the piston bowl wall. This could result in higher cooling loss.

It could be concluded from HVO combustion behavior that a shorter ignition delay suppresses flame development from occurring close to the wall. To further suppress the ignition from occurring close to the wall, a lower pilot injection pressure is hypothesized to promote ignition further from the wall due to a longer injection duration and reduced spray penetration. This condition may lead to the formation of locally rich regions around the pilot spray prior to ignition, producing a high-temperature flame with greater momentum that spreads more uniformly throughout the combustion chamber. Therefore, the study was extended to investigate the effects of reduced pilot injection pressure.

3.3 Effects of pilot injection pressure on combustion process

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 injection pressure was decreased to 40 MPa for both HVO and diesel fuel. Figure 8 illustrates the in-cylinder pressure and heat release rate under both conditions. When using diesel fuel as pilot fuel, reducing the injection pressure resulted in delayed injection timing and longer injection duration, as can be observed from the green injection duration line starting at -8°CA . This resulted in shorter ignition delay leading to steeper increase in heat release rate and a shorter combustion duration (roughly 14°CA) than 80 MPa condition (18°CA). However, the ignition delay is long enough that the pilot fuel is premixed with the surrounding air and hydrogen, causing the combustion of hydrogen and pilot fuel occurs at almost same time, resulting in a single stage combustion as can be observed from the heat release rate peak. When HVO was used as pilot fuel, no delay in injection timing and similar combustion duration is noticeable with reduced injection pressure, although the injection duration becomes marginally longer. When observing the heat release rate, the reduced pilot injection pressure exhibits dual peak of heat release rate which indicates dual stage of combustion. The slightly delayed injection duration resulted in

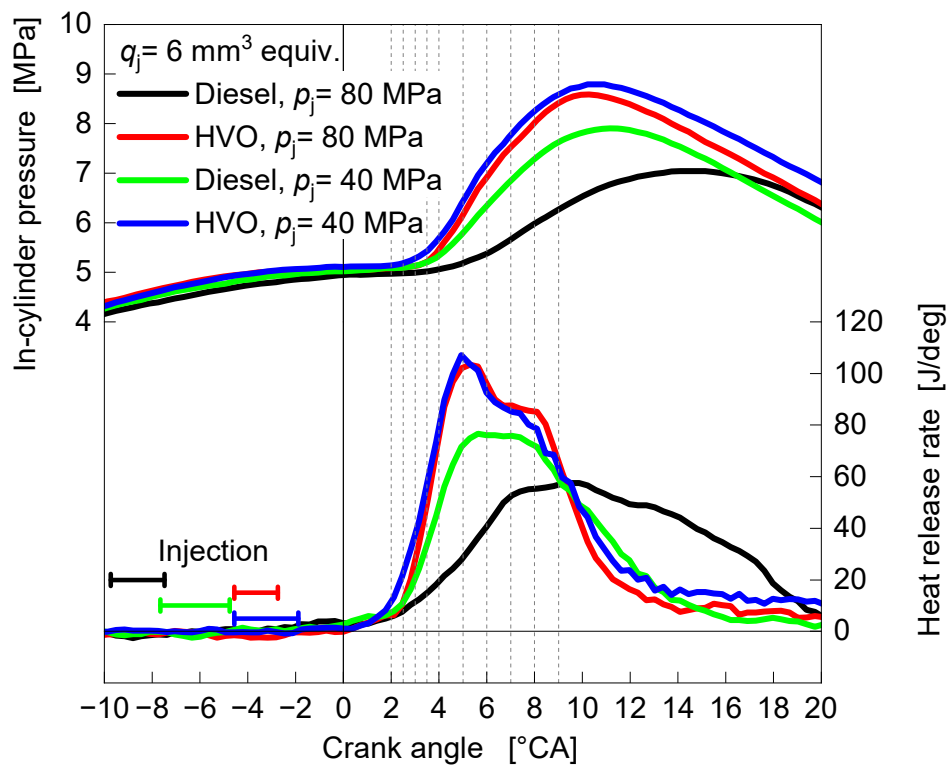


Figure 8. Effect of pilot injection pressure on in-cylinder pressure and heat-release rate (pilot fuel: diesel fuel and HVO).

slightly higher peak of heat release rate, particularly during the first stage of combustion in which pilot fuel was mainly combusted. The second stage of combustion exhibits slightly reduced maximum peak when the injection pressure was reduced.

Figure 9 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 Figure 6. The high intensity suggests that a lower injection pressure prevented significant dilution 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.

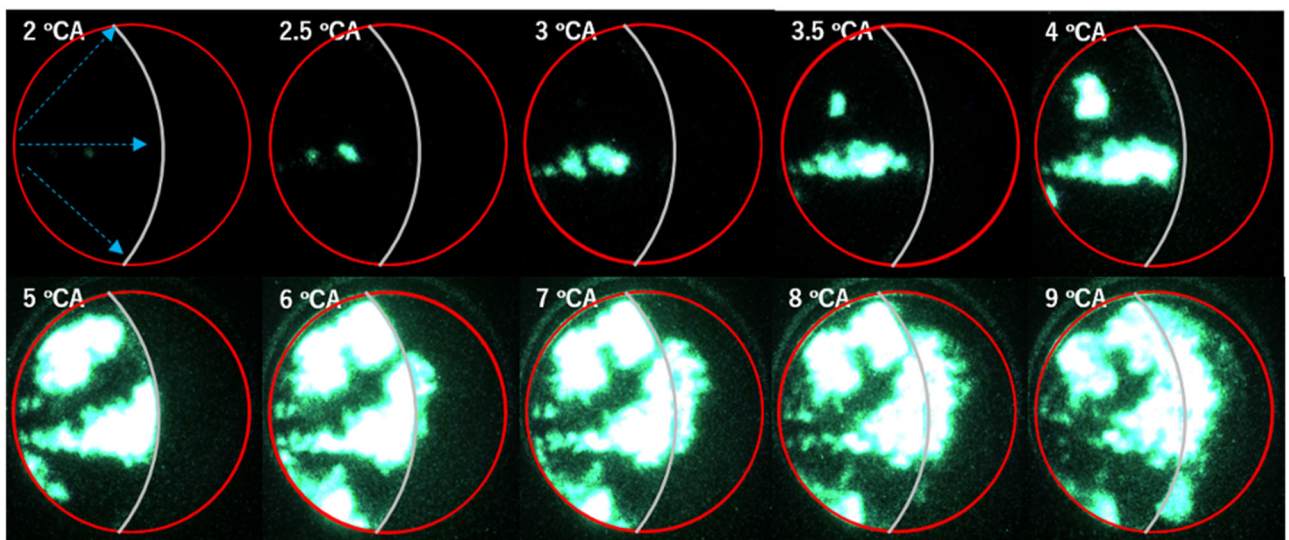


Figure 9. Images of OH* chemiluminescence (pilot fuel: HVO, $p_j = 40$ MPa, $q_j = 6.27 \text{ mm}^3$).

Figure 10 shows the combustion images obtained using diesel fuel as pilot fuel, injected with 40 MPa pressure. When diesel fuel was injected, luminescence was initiated upstream near the spray base at 2.5°CA and progressed downstream to the squish area. This followed by propagation of weak luminescence region at the same time. Compared to the higher injection pressure visualization result in Figure 4, the luminescence intensity is relatively higher, indicating that the temperature is higher. This contributed to a shorter combustion duration. It could be concluded that lower pilot injection pressure resulted in lower dilution rate of the pilot fuel, providing favorable ignition condition for the pilot fuel. The observation that lower pilot injection pressure leads to higher OH* intensity, while seemingly counterintuitive, can be explained by the specific combustion regime in our study. Under these operating conditions, the pilot injection quantity is small, and ignition occurs with a sufficient delay, well after the end of injection. This behavior is analogous to premixed controlled compression ignition (PCCI) combustion. In this PCCI-like regime, the duration between the end of injection and the start of combustion provides ample time for mixing. Consequently, a higher injection pressure does not just "improve" mixing; it promotes "over-mixing". This excessive mixing leads to a more dispersed, homogeneous, and thus locally leaner mixture at the ignition timing. This leaner mixture results in a lower combustion temperature and, consequently, a weaker OH* intensity. Conversely, lowering the pilot pressure suppresses this over-mixing. This maintains a more stratified and locally richer mixture, leading to a higher combustion temperature and the stronger OH* intensity observed in our results.

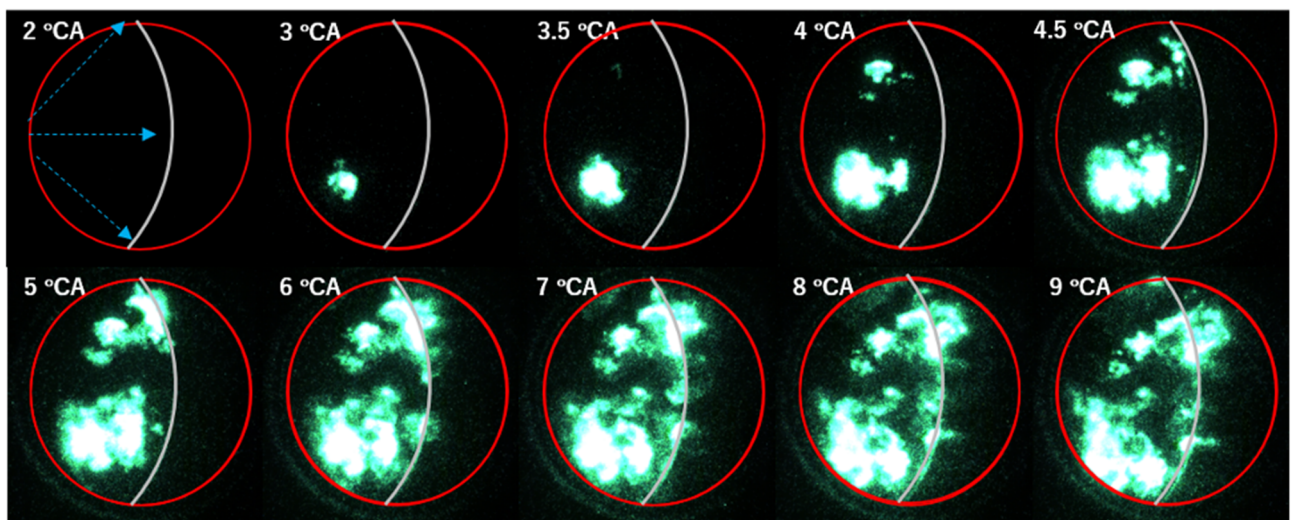


Figure 10. Images of OH* chemiluminescence (pilot fuel: Diesel fuel, $p_j = 40$ MPa, $q_j = 6 \text{ mm}^3$).

As the amount of surrounding air and hydrogen could also affect the mixing rate of the pilot fuel, this visualization was then further continued with studying different hydrogen mixing ratio.

3.4 Effects of hydrogen mixing ratio on combustion process

To investigate the effect of increasing the hydrogen mixing ratio while maintaining a constant total excess air ratio, the pilot injection quantity of HVO was reduced to 3.14 mm³ at an injection pressure of 80 MPa. Figure 11 illustrates the in-cylinder pressure and heat release rate, with the results for 6.27 mm³ injection provided for comparison. The hydrogen mixing ratio increased roughly from 86% to 93% when the injection quantity was reduced. This information can be obtained from Table 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. Despite a slight decrease in the excess air ratio for hydrogen, the combustion duration is also extended until approximately 18°CA. Due to the differences of amount of energy substituted by hydrogen, the cumulative heat release rate

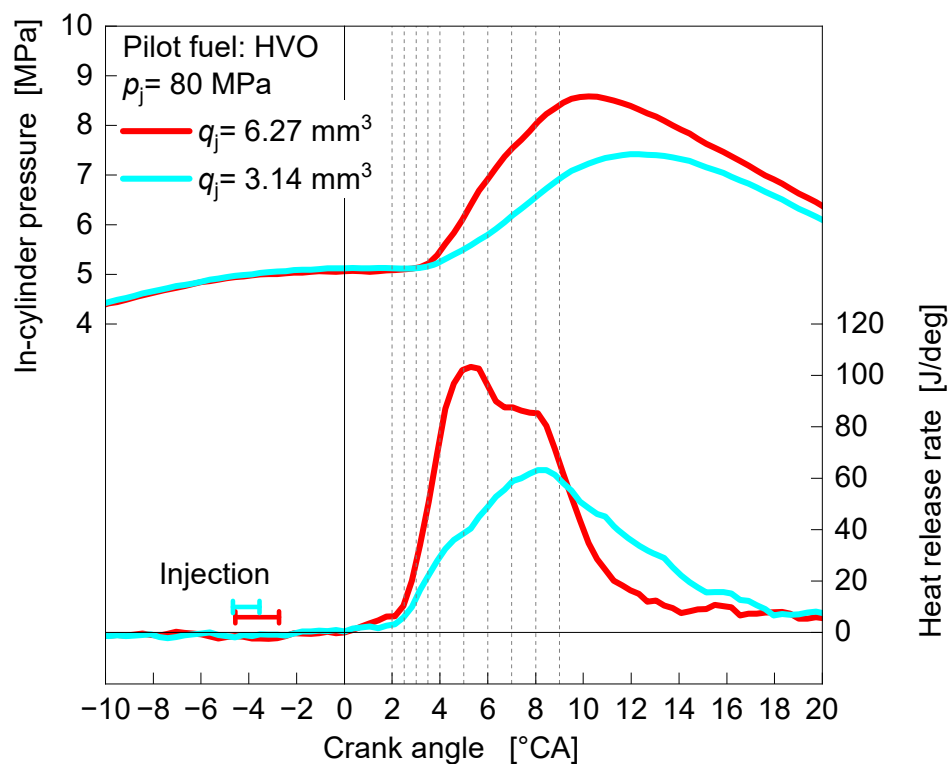


Figure 11. Effect of pilot injection quantity, q_j under fixed total excess air ratio condition on in-cylinder pressure and heat-release rate (pilot fuel: HVO, $p_j = 80$ MPa).

is shown to evaluate the overall heat release development and energy conversion efficiency by hydrogen.

Figure 12 shows the cumulative heat release under both pilot quantity conditions. As expected from the heat release rate behavior in Figure 11, higher pilot quantity (lower hydrogen mixing ratio) results in an earlier attainment of the maximum cumulative heat release and the maximum value at point close to TDC is higher.

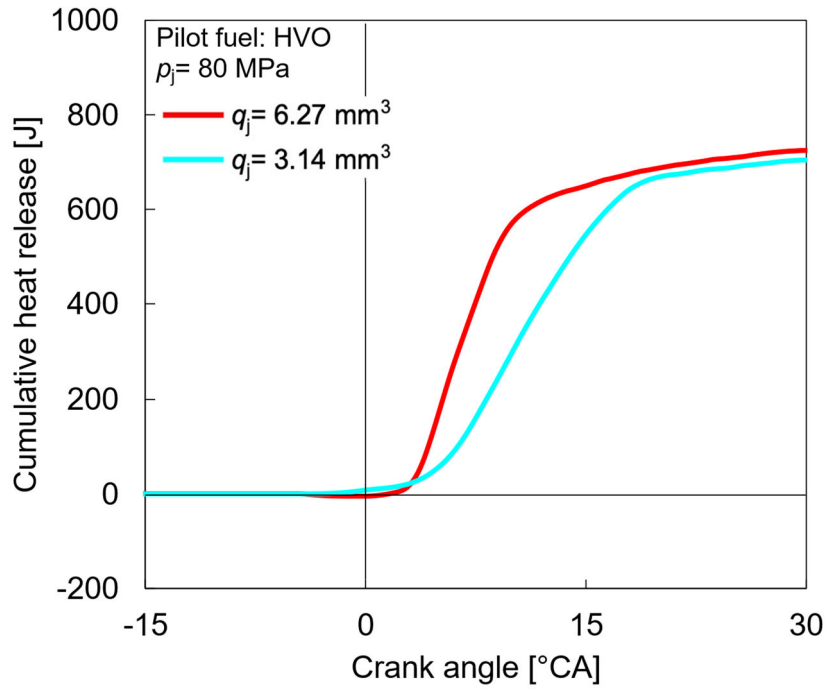


Figure 12. Effect of pilot injection quantity, q_j on cumulative heat release (pilot fuel: HVO, $p_j = 80 \text{ MPa}$).

Figure 13 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 Figure 6. The reduced pilot quantity led to a smaller spray region, resulting in a higher spray dilution rate. This resulted in a less concentrated local pilot flame, which consequently led to a lower flame temperature. This explains the low heat release rate and extended combustion duration.

After 5°CA , weak luminescence, possibly attributed to hydrogen combustion, propagated around the pilot flame. When comparing visualization images at 6.27 mm^3 pilot quantity in Figure 6, the luminescence from hydrogen premixed luminescence intensity was

smaller than when a larger quantity of pilot fuel was utilized. The smaller autoignition flame in the pilot fuel resulted in slower flame propagation through the hydrogen-air mixture, despite the lower excess air ratio as could be referred from Table 4. Consequently, hydrogen combustion required additional time to propagate throughout the combustion chamber.

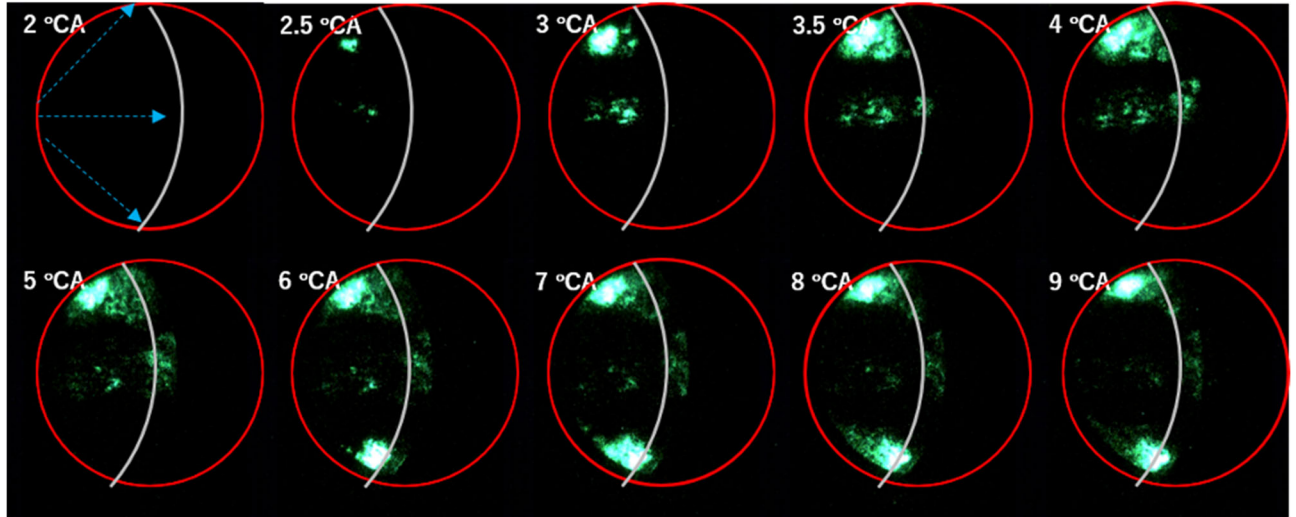


Figure 13. Images of OH* chemiluminescence (pilot fuel: HVO, $p_j = 80$ MPa, $q_j = 3.14$ mm³).

4. Conclusion

To clarify the distinctions in the combustion process between diesel fuel and HVO as pilot fuels in hydrogen dual-fuel combustion, the combustion location and flame formation were visualized using RCEM and OH* chemiluminescence imaging. The results revealed the following:

- (1) When diesel fuel is employed as the pilot fuel, the pilot sprays ignite after impacting the piston bowl. In contrast, 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. Consequently, the combustion period is shortened when HVO is utilized.
- (2) When using diesel fuel and HVO as pilot fuels, reducing the pilot injection pressure from 80 MPa to 40 MPa inhibits the dilution of the pilot spray mixture. This results in shortened combustion duration under both conditions. Under the condition diesel fuel as pilot fuel, reducing the pilot injection pressure leads to ignition of the pilot spray vapor before impact with the piston bowl wall. This improves the ignition of diesel fuel.

(3) When the hydrogen mixing ratio is increased by decreasing the pilot fuel injection quantity at a constant total excess air ratio, the pilot spray flame diminishes in size, and the peak heat release rate decreases. Therefore, despite the low excess air in the hydrogen premix, flame propagation throughout the combustion chamber requires an extended duration.

(4) Compared to diesel fuel, HVO tends to ignite further from the piston bowl wall under most conditions due to its higher ignitability. This suggests that using HVO as a pilot fuel could result in lower cooling losses.

While observations from OH* visualization images in the optically accessible RCEM experiments offer valuable insights into fundamental combustion processes, in this study, it was challenging to clearly ascertain whether the high-temperature region results from the combustion of the pilot fuel or the hydrogen, and the transition of flame propagation from the pilot fuel to the hydrogen remains indistinct. Therefore, further confirmation through complementary diagnostics, such as simultaneous high-speed imaging of multiple chemiluminescent species is necessary. These approaches would enable a more precise differentiation of the respective contributions of pilot fuel and hydrogen combustion, and a clearer characterization of the flame propagation mechanism between the two fuels, to realize a better injection strategy in hydrogen dual fuel operation.

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Appendix

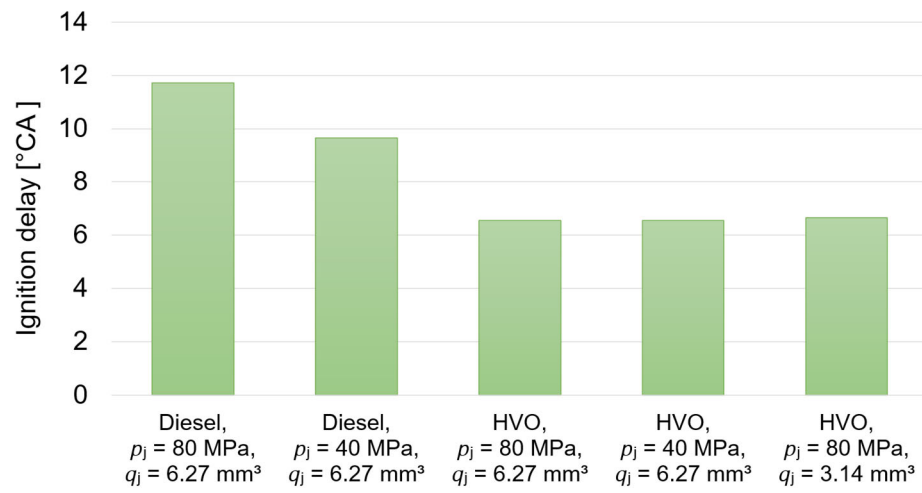


Figure A1. Summary plot on the ignition delay of all cases in this study.