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Combustion Management of Neat Dimethyl Ether Combustion for Enabling High Efficiency and Low NO_x Production

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Abstract: Modern compression ignition engines heavily rely on exhaust gas recirculation to reduce NO_x emissions. Despite this, complex and expensive after-treatment systems are still necessary to comply with stringent emission regulations. Conventional diesel combustion operates on a robust and readily controllable mode through which the high-pressure fuel injection and combustion processes are intimately coupled. The heterogeneous nature of direct injection systems is liable to the NO_x-soot trade-off inherent to diesel-fueled engines. Dimethyl ether (DME) presents a unique fuel that is reactive, volatile, and oxygenated, offering significant potential to address emission challenges with reduced reliance on aftertreatment systems. In this research, the combustion management of neat DME fuel was investigated using a high-pressure direct injection system. Principally, the suitability of single-shot fuel scheduling as a combustion management technique for DME under low NO_x production was explored. The transient high-pressure injection behaviour of DME was characterized with an offline test bench. A single-cylinder research engine platform was employed to study DME combustion characteristics. A wide range of engine conditions was investigated, including injection pressures from 200 bar up to 880 bar and engine loads from 1 bar up to 17 bar indicated mean effective pressure (IMEP). The combustion management of DME as it relates to fuel injection and operating boundary conditions was emphasized throughout the work. To accomplish this, tests were conducted at direct comparison conditions to diesel operation. Most notably, the DME combustion process finished in a shorter period than diesel, albeit with a significantly longer injection duration. At most operating conditions, the soot emissions were below that of upcoming emission regulations without particulate filter exhaust treatment. Even under high engine load operation—17 bar IMEP—of neat DME, the NO_x emissions could be readily contained via EGR management to 51 ppm engine-out NO_x during which soot reached a maximum of 1.0 FSN. Such operating circumstances of high engine load and low oxygen availability (overall lambda of 1.2) exhibited a deterministic combustion timing control via injection timing while performing with low combustion noise (4.8 bar/°CA) and high burning efficiency (98.5%).

Keywords: dimethyl ether; combustion management; low emissions; clean combustion; alternative fuel

1. Introduction

The heavy-duty transportation sector is largely powered by internal combustion engines (ICEs) owing to their range of power density capabilities, reliability, and convenient handling of liquid fuel as the sole energy source. Compression ignition engines are valuable for such applications because of their inherent high thermal efficiency compared to their spark ignition counterparts. Through the advancements in technologies such as lightweighting, engine downsizing, aftertreatment materials, and combustion management, modern engines have reduced their tailpipe emissions footprint significantly concerning greenhouse gases, e. g., carbon dioxide (CO_2), and harmful combustion byproducts, e.g., nitrogen oxides (NO_x) and soot [1]. Still, it is predicted that the theoretical maximums of efficiency improvement and emission reduction are yet to be achieved, by way of further development in the fueling and combustion management area [2–4].

To ensure standardized progress toward reducing combustion byproducts, a set of tailpipe restrictions have been enacted in different regions throughout the globe, as noted in Table 1. A significant amount of effort has been focused on engine management strategies to meet emission regulation standards. Such regulations set limits on the maximum output of specified species at the tailpipe of on-road vehicles, including CO_2, unburned hydrocarbon (UHC), carbon monoxide (CO), NO_x, and soot. While the CO_2 is predicated by the carbon content of the fuel and thermal efficiency, wherein a higher thermal efficiency will contribute to lowering the CO_2 output, the remaining species are undesired combustion byproducts. The presence of UHC and CO in the exhaust indicates worsened combustion efficiency, whereas NO_x and soot in due course harm the environment and human health. Emission regulations have become stricter over time, with current emission regulations over 90% lower than those introduced 20 years prior [5]. To this end, many improvements have been made successfully by the aftertreatment technology and combustion management, i.e., electronic engine control, electronic sensors, and active fuel and air handling systems [6].

Table 1. The emission standards across various regions for medium- and heavy-duty transportation. Note that the standards are representative of the hot and steady-state test cycles.

Region	NO_x, mg/kWh			Soot, mg/kWh		
	Present (Year)	Planned (year)	↓ %	Present	Planned	↓ %
USA	270 (2007)	47 (2027)	83	13	8	38
EU	400 (2013)	200 (2027)	50	10	8	20
China	400 (2021)	-	-	10	-	-

Present emission limits for NO_x and soot are expected to experience a further reduction up to 83% and 38% in upcoming planned regulations, respectively. The European Union (EU, EURO standards) and the United States of America (USA, EPA standards) advocate distinctive regulation standards, ultimately aimed at similar range of reductions in exhaust emissions. Generally, most countries enact tailpipe emission limits similar to the EU standards in later years for heavy-duty vehicles, albeit with differing testing procedures [7,8].

Modern aftertreatment systems may exceed 90% conversion efficiency with proper operation and maintenance, allowing vehicles to conform to these tight emission regulations [9, 10]. For illustrative purposes, an effective emissions target for engine research remains in the range of 10 to 20 times higher than the regulated tailpipe regulations, as shown in Table 2. In example, at a presumed 90% NO_x conversion efficiency, the tailpipe regulations in the USA of 270 mg/kWh, and planned to become 47 mg/kWh, correspond to engine-out emission limits of 2700 mg/kWh and 470 mg/kWh, respectively. While further improvements in aftertreatment technology are plausible [11], additional emission reductions are expected to be realized through in-cylinder combustion management. Nonetheless, all regulations require a simultaneous reduction of NO_x and soot emissions. This in practice, is an increasingly challenging task for compression ignition combustion systems.

Conventional diesel combustion is a robust and readily controlled combustion mode through which the high-pressure fuel injection and combustion processes are intimately coupled near the top dead centre, as shown in Figure 1. This period of interest is comprised of an extremely rapid sequence of events that is often completed within a few milliseconds. Within this brief time frame, high-pressure fuel is injected, chemical reactions occur, fuel energy is released and emission products are formed, leading to a spatially and temporally complex system [12]. As the fuel injection and combustion are distinct, yet inherently connected processes, it is often the case that the fuel spray duration continues through the combustion event leading to a diffusion flame [13].

Table 2. Sample targets of NO_x and soot reductions via exhaust aftertreatment and in-cylinder combustion. The engine-out emissions are estimated from the aftertreatment efficiencies: NO_x catalytic reduction efficiency of 90%, Soot particulate filter effectiveness of 95%.

USA	NO_x			Soot		
	Tailpipe	Engine-Out		Tailpipe	Engine-Out	
	mg/kWh		ppm	mg/kWh		FSN
Present	270	2700	300~500	13	260	2.4~2.8
Planned	47	470	50~100	8	160	1.8~2.2

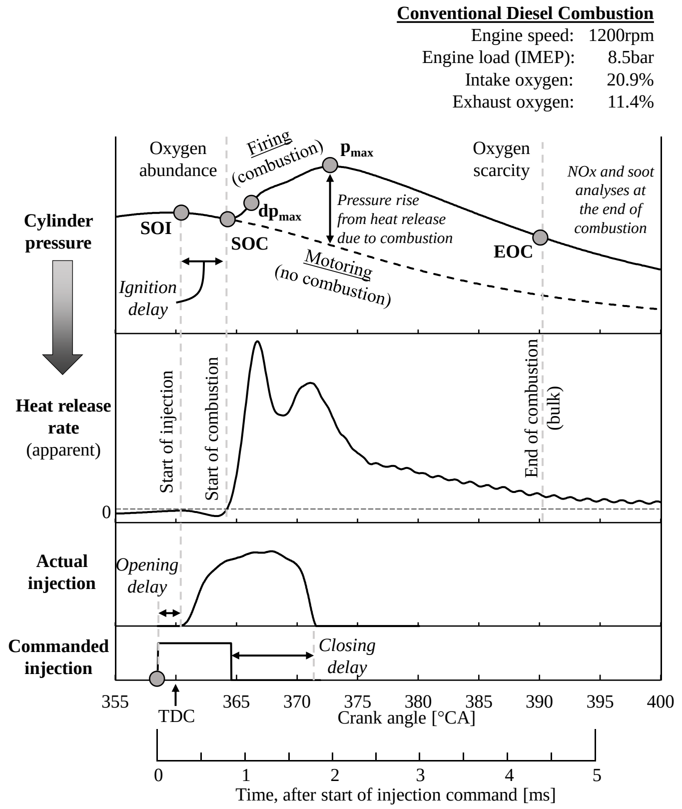


Figure 1. Overview of conventional diesel control and combustion behaviour.

High-temperature diesel combustion as shown here is known for high efficiency but is prone to generate high levels of NO_x emissions and, under high load conditions, soot. Substantial reductions in engine-out NO_x emissions for compression ignition engines can be readily achieved through exhaust gas recirculation (EGR) [14]. However, a simultaneous increase in soot because of lower oxygen availability and lower combustion temperatures is often an unintended consequence of NO_x mitigation efforts through EGR [15]. In general, lower oxygen availability at the start of combustion predictably lowers NO_x emissions, whereas higher oxygen availability towards the end of combustion lowers soot emissions. Many operating boundary conditions act as suitable means of manipulating the physical and chemical influences to manage combustion, such as injection pressure, fuel scheduling, intake boost pressure, combustion phasing, and more. Nonetheless, the simultaneous balance of NO_x and soot continues to prove challenging as the inevitable NO_x-soot trade-off inherent to diesel-fueled engines persists [16,17].

The diffusion flame is a prominent phenomenon in compression ignition combustion as it facilitates a large portion of heat release under elevated engine loads [13]. It is often characterized by its mixing-controlled combustion features [18]. The close coupling of fuel injection and combustion amplifies the influence of the fuel properties on the subsequent combustion behaviour. A diffusion flame is generally undesired for enabling high NO_x and soot emissions. In stark contrast, the diffusion flame provides a high-efficiency burning mode that is readily controlled in timing and intensity. As it is the undesired rise in soot emissions that limits the amount of EGR, the elimination of soot formation via fuel properties allocates a direct in-cylinder NO_x management via EGR, reducing fuel scheduling and combustion management complexities.

Fuel properties such as reactivity, fuel-borne oxygen content, and volatility have been proven advantageous for direct injection combustion applications for high efficiency and low engine-out soot emissions. A list of conventional and alternative fuels has been organized into three characteristics in Figure 2. A chemically reactive fuel, i. e., recognized by its adequate cetane number, enables high controllability and combustion completeness, tending to produce lower combustion noise [19]. On the other hand, fuels with high cetane numbers promote a greater degree of diffusion combustion and, in turn, may augment soot emissions [20]. An oxygenated fuel alleviates the dependency on oxygen availability from the cylinder charge during fuel spray and improves combustion completeness. Kitamura et al. [21] found that soot formation within the high-pressure fuel spray flame could be significantly reduced at >30% oxygen content. Lastly, volatility primarily enhances the atomization rate of fuel and air. Even though a volatile fuel may be subject to difficulties in liquid-phase high-pressure fuel injection, its application significantly influences the fuel spray and subsequent combustion event [22]. The period from fuel spray to autoignition involves both physical and chemical delays, wherein the physical delay dominates [23]. In high-pressure fuel sprays, Konno et al. [24] showed that the mixing quality of dimethyl ether (DME) at 600 bar injection pressure was equivalent to that of diesel at 2000 bar at half the nozzle orifice diameter. Teng and McCandless [25] explained that the rapid fuel evaporation was most notable around the perimeter of spray plume, effectively showing a turbulent-jet-like nature. Such behaviour may be most valuable during the diffusion flame when the rate of reaction is governed by the mixing rate. A faster diffusion combustion rate correspondingly improves controllability and enables a high efficiency, ultra-low emissions mode of combustion [26]. Through this comparison, the unique value of DME is apparent; a fuel possesses unique qualities of reactivity, volatility, and oxygen content.

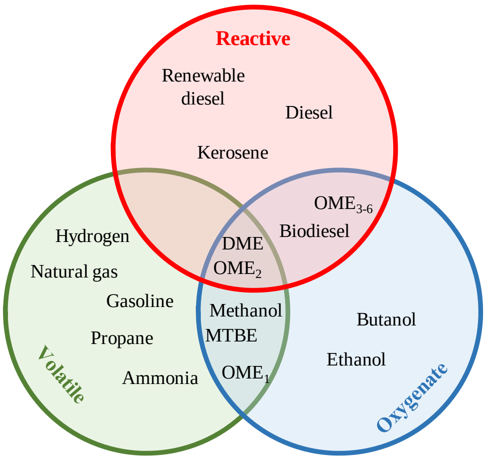


Figure 2. Qualities of conventional and alternative fuel.

The potential of DME as a fuel for compression ignition engines has been widely investigated in academia and industry. The advantageous near-zero soot tendency of DME is clear, alongside the availability to be produced from a multitude of renewable feedstocks, presenting DME as an ideal fuel and sustainable energy source for future heavy-duty powertrains [27,28]. Apart from the engineering challenges associated with handling onboard high-pressure liquified DME, the engine performance is largely similar to that of diesel but with little inheritance NO_x-soot trade-off [29–32]. In this respect, it is valuable to understand the combustion behaviour, akin to the research and development of diesel engines [33]. While engine performance appears similar, the unique fuel properties of DME necessitate adjustments in combustion management strategies. The fuel scheduling necessitates adjustments as the lower energy density raises the injection quantity ~55% by mass. With conventional single-shot fueling strategies, the injection duration will be extended significantly, especially as the physical properties of DME permit lower injection pressures. On the other hand, the extreme mitigation of soot emissions with DME can be leveraged for better control through prioritizing the diffusion combustion mode.

In this work, the injection and combustion characteristics of neat DME fuel are investigated in a high-pressure direct injection combustion application. This research aims to compile and extend the understanding of DME combustion in a conventional direct injection combustion application The transient fuel injection and combustion processes are investigated with an offline test bench and online research engine platform, covering injection pressures from 200 bar up to 880 bar and engine loads from 1 bar up to 17 bar indicated mean effective pressure (IMEP). The combustion management of DME as it relates to fuel injection and operating boundary conditions is emphasized throughout the work. To accomplish this, tests were conducted at direct comparison conditions to diesel operation. For diesel engine tests, a higher fuel pressure of up to 1500 bar was applied to constructively match the lower volatility of diesel. The discussion primarily emphasizes comparative results to diesel fuel which acts as a reference for modern-day combustion management strategies. This comparison provides insight into the suitability of single-shot fuel scheduling as a combustion management technique for DME under low NO_x emissions.

2. Methodology

The heterogeneous nature of direct injection combustion involves an intimate coupling of fuel delivering process and the combustion process. In this study, the management of DME fuel injection and combustion was investigated in a single-shot strategy. Diesel fuel was also investigated at matching conditions for relevance. An offline rate of injection measurement bench was used to study the injection characteristics of the fuels and the combustion characteristics were investigated on a single cylinder research engine. The fueling system was configured to adapt on-board and off-board high-pressure fueling systems suitable for diesel and DME fuels, respectively. This setup has enabled high-pressure direct injection and combustion studies with alternative fuels including those of low viscosity, high vapour pressure, and inadequate elastomer compatibility.

2.1. Injection Characterization

Due to the predominance of diffusion combustion control, effective combustion control requires fast and precise instrumentation that can be readily adjusted. The unique physical fuel properties of DME compared with diesel fuel raise concern about the fuel injection behaviour regarding the real transient discharge process and the subsequent plume formation. In this section, the system specifications and testing procedure will be discussed.

The Bosch long-tube method was utilized to analyze the injection rate profile for DME fuel. The results were then compared to those of diesel injection under the same conditions. The rate of injection measurement platform included a high-pressure injection pump, an injector installed in a reservoir connected to a 50-meter-long tube, and a host computer for monitoring, control, and data acquisition. An overview of the rate of injection platform is illustrated in Figure 3. The signals transmitted and received during testing were managed by an RT-FPGA system interfaced with the host computer, all orchestrated through the National Instruments LabVIEW software environment.

The fuel properties of DME prevent its direct substitution for diesel fuel using traditional diesel injection systems due to DME's higher vapour pressure, lower viscosity and lubricity, and its corrosive effects on elastomers. A list of selected fuel properties is given in Table 3. To address these challenges, a pneumatically driven double-air drive piston pump (Maximator LSF100-2) was used to supply high-pressure DME. This system accommodates a variety of alternative fuels regardless of physical properties, as the plunger was pneumatically driven and the sealing materials are chemically resistant. Liquid DME was fed from the bottom of a 1-gallon stainless steel tank, with the DME pressure maintained at approximately 20 bar absolute using a nitrogen supply at the top of the tank. Throughout the tests, a back-pressure range of 30~50 bar yielded the best results without noticeable differences.

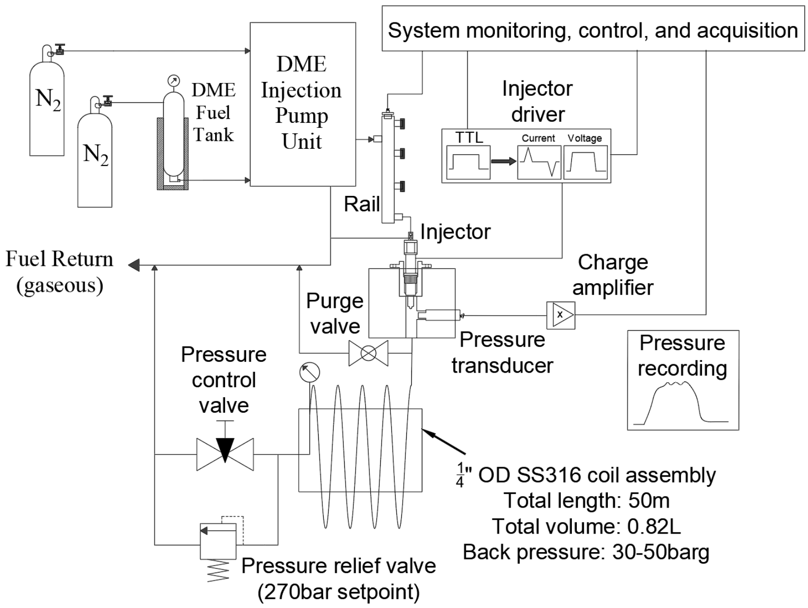


Figure 3. Rate of injection research platform.

Table 3. Comparison of fuel properties between the standard ultra-low sulfur diesel (ULSD) and DME.

Fuel Property	ULSD	DME
Chemical formula	$C_nH_{1.77n}$	$CH_3O\ CH_3$
Cetane number	40–50	55–68
Oxygen content [%]	0	34.8
Lower heating value [MJ/kg]	42.9	27.6
Density [kg/m^3]	831	667 [1]
Vapor pressure (20 °C) [bar]	<0.10	5.30
Kinematic viscosity [mm^2/s]	1.9–4.1	<0.15 [1]

[1] DME in a saturated liquid state.

A stainless-steel chamber was designed and fabricated to hold a conventional diesel injector (8 × 140 µm). The pressure transducer was flush-mounted on the side of the chamber. Signals were sent to a piezoelectric injector driver to send the tuned opening and closing power signals to the injector. The pressure inside the chamber increased as the fuel was injected into the chamber. The corresponding rate of injection shape is proportional to the magnitude of pressure change recorded by the pressure transducer. The theoretical calculation from the measured chamber pressure mass flow rate is proportional to the total nozzle hole area and inversely proportional to the local speed of sound for the respective fluid [34].

Hydraulic delays can be categorized into opening and closing delays. More detailed information about the offline testing methodology can be found in [35]. The start of fuel delivery is considered reliable and has been validated with high-speed video recordings. The end of fuel delivery required assumptions in data processing. In this work, the raw signal exhibits an exponential delay as the ROI decreases at the end of fuel delivery. It is assumed that the needle effectively cuts off the fuel upon closing. Consequently, the end of the fuel delivery slope was extended using a linear fit line.

The system was validated by recording an extended period and observing the pressure wave reflections. The reflected frequencies for diesel and DME resulted in estimated speeds of sound of 1389 m/s and 980 m/s, respectively. Similar speeds of sound for DME have been confirmed by Kapus and Ofner [13], as well as other studies [14 – 16]. The agreement between the speed of sound measurements in this study and those reported in the literature instills confidence in the results obtained from the long tube system.

2.2. Single-Cylinder Research Engine

The engine is configured for direct injection CI research with an independent fuel injection system. A summary of engine geometry specifications is given in Table 4. An overview of the complete engine system is shown in Figure 4. All data acquisition and engine control programs are designed within the National Instruments LabVIEW software environment. The raw pressure data and fuel scheduling programs were programmed into the real-time operating systems on embedded NI controllers. A piezoelectric pressure transducer was flush-mounted to the cylinder head. The pressure measurements are synced with the optical encoder at a frequency of 7200 points per cycle, or every 0.1 °CA. Each operating condition logged a continuous 200 cycles of pressure recordings. Thermocouples and pressure sensors are fitted throughout the system for feedback during operation and safety. A pair of independent conditioning units maintained the temperature and pressure of the lubricant (80 °C at 2.4 bar) and coolant (80 °C). The intake air was supplied from a dry and oil-free air compressor. The air is guided through an electro-pneumatic regulator to control the operating intake boost pressure. Two large tanks were positioned at the intake and exhaust to minimize airflow variations and pressure wave actions into either manifold. To add, the intake volumetric flow rate was measured with a roots-type flowmeter fitted between the regulator and intake surge tank.

Table 4. Research engine specifications.

Engine Configuration	Single-Cylinder
Swept volume	744 cm^3
Bore × Stroke	95 mm × 105 mm
Piston bowl	Omega
Compression ratio	16.5:1
Injector specifications	7 × 152 μm, 156° umbrella angle
Engine speed	1200 rpm
Intake pressure	150~250 kPa absolute

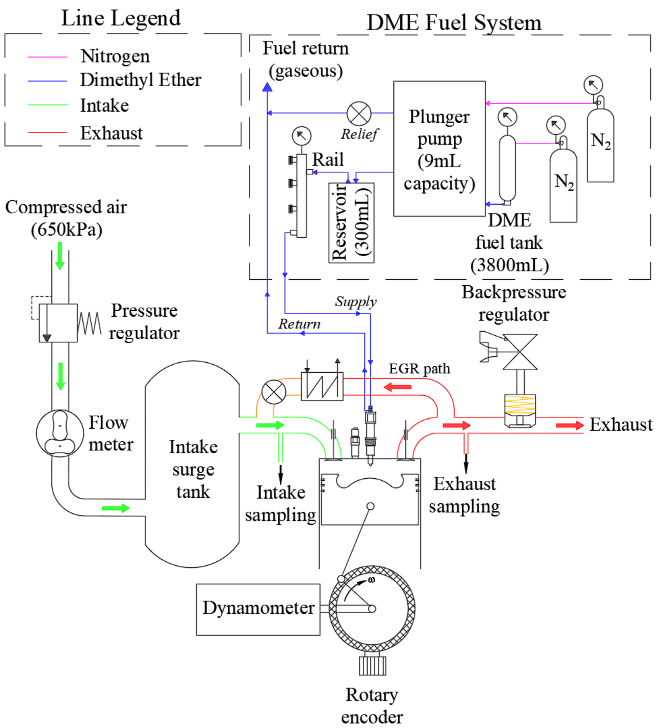


Figure 4. Single-cylinder engine configured for high-pressure direct injection DME combustion research.

The apparent heat release rate (HRR) was calculated from the cylinder pressure, based on the first law of thermodynamic fundamentals:

$$\frac{dQ_{app}}{d\theta} = \frac{1}{\gamma - 1}\left[\gamma \cdot p \cdot \frac{dV}{d\theta} + V \cdot \frac{dp}{d\theta}\right] \tag{1}$$

where, $\frac{dQ_{app}}{d\theta}$ is the apparent HRR (J/°CA), γ is the specific heat ratio, p is the cylinder pressure (Pa), V is the volume (m^3), and θ is the change in crank angle (°CA).

A set of emission analyzers including California Analytical Instruments was used to measure the standard regulated emissions, including nitrogen oxides (NO_x), total hydrocarbon (THC), carbon monoxide (CO), carbon dioxide (CO_2), and oxygen (O_2). An AVL-415S variable sampling smoke meter was used to measure the smoke content in the exhaust.

The same type of pneumatic-driven pump (Maximator LSF100-2) used for investigating injection characteristics was also employed on the engine platform. The cyclic characteristic of the pneumatic pump leads to frequent pressure fluctuations during pneumatic piston recycling and further deteriorates under high-frequency applications. A large high-pressure accumulator was fitted between the pump and injector, in series to the fuel rail, to mitigate the pressure drop during the single-plunger pump resetting event, detailed information about the setup could be found in [36]. The fuel return was directed to the atmosphere. As a result of the open-loop fueling system, the fuel flow rate was not directly measured but rather estimated via the carbon balance of the exhaust gas contents.

3. Results and Discussion

The compression ignition combustion event in a reciprocating engine is a fast and complex process often involving an overlap of physical interactions (fuel injection) and chemical reactions (combustion) simultaneously. The combustion timing and process are strongly dependent upon the fuel injection timing, injection pressure, and injection duration. Effective combustion management strategies are directly linked to the injection process, notably the actual fuel spray timing related to the electronic control signals and accurate calibration of the total fuel supplied. Therefore, this work first investigates the transient fuel injection process using an offline test bench followed by combustion studies using a single-cylinder research engine. All DME tests are matched with separate diesel tests for relevance to modern diesel engine management strategies.

3.1. Injection Characteristics

The high-pressure injection process often takes place in less than a couple of milliseconds. Modern fueling systems incorporating a common-rail high-pressure accumulator and electronically controlled, hydraulically actuated injector can achieve extreme precision in both fuel spray timing and quantity. The precise control of fuel allows precise control of the combustion process and, in turn, stimulates the development of fuel techniques for optimizing engine performance and emission output.

The behaviour of the brief injection process is affected by the physical properties of the fluid under high pressure. It is important to realize such effects that develop from replacing diesel with an alternative fuel, especially in the case of volatile, liquified DME fuel. The transient injection process of DME demonstrates a higher exit velocity spray to diesel under matching injection pressure (600 bar) and commanded duration (1.0 ms), as shown by the solid lines in Figure 5. A single diesel baseline case is compared to three DME cases, of which one case matches operating conditions and two cases match fuel energy supplied via extended injection duration or injection pressure. The commanded square pulse signal of 1.0 ms and the injector driver power signal (current) is shown at the top. The delay between the start of the electronic signal command to the start of fuel delivery is evident. Although the target injection duration was 1.0 ms, the actual fuel supply event persisted up to 1.7 ms and was offset from the desired timing.

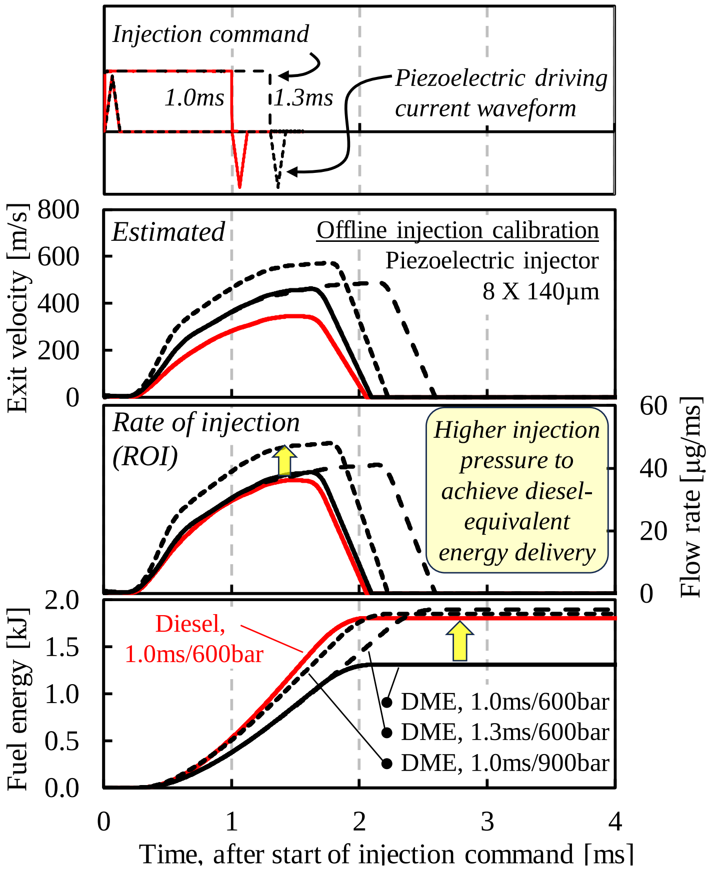


Figure 5. Injection control signals and the transient hydraulic response of a piezoelectric injector operating on diesel and DME fuels.

The fuel spray of DME reached greater exit velocity than that of diesel. It is expected that the lower viscosity of neat DME contributes significantly to this effect. A balance of higher exit velocity yet the lower liquid density of DME presents a very similar mass flow rate behaviour to diesel. Coincidentally, this result implies that for the same injection pressure and injection duration command of DME and diesel, a similar injection quantity by mass, rather than by volume, is expected. The overlap in fuel supply by mass with DME was observed by the work of Youn et al. [37]. To match fuel supply quantity on an energy basis, ~55% higher fuel by mass must be supplied. The additional fuel may be supplied via extended injection duration or increased injection pressure as demonstrated by the long and short dashed lines in Figure 5, respectively. It was possible to achieve similar energy supply quantities at 1.3 ms commanded duration under 600 bar or 1.0 ms commanded duration under 900 bar fuel pressure.

The offline test bench platform is valuable for a deep understanding of the transient behaviour of the injector. However, a higher frequency operation with additional variables such as warm injector body temperature and rail pressure fluctuations during actual engine testing can lead to variations. A steady-state firing operation of high-pressure DME was conducted at 10 Hz (1200 rpm) with 80 ° C engine block temperature, as shown in Figure 6. The injection quantity by mass (upper) and energy (lower) are depicted. The diesel fuel flow rate (FFR) was measured directly via a fuel flow meter. However, due to the lack of closed-loop fueling for this system, the DME fuel flow rate was estimated via exhaust carbon balancing.

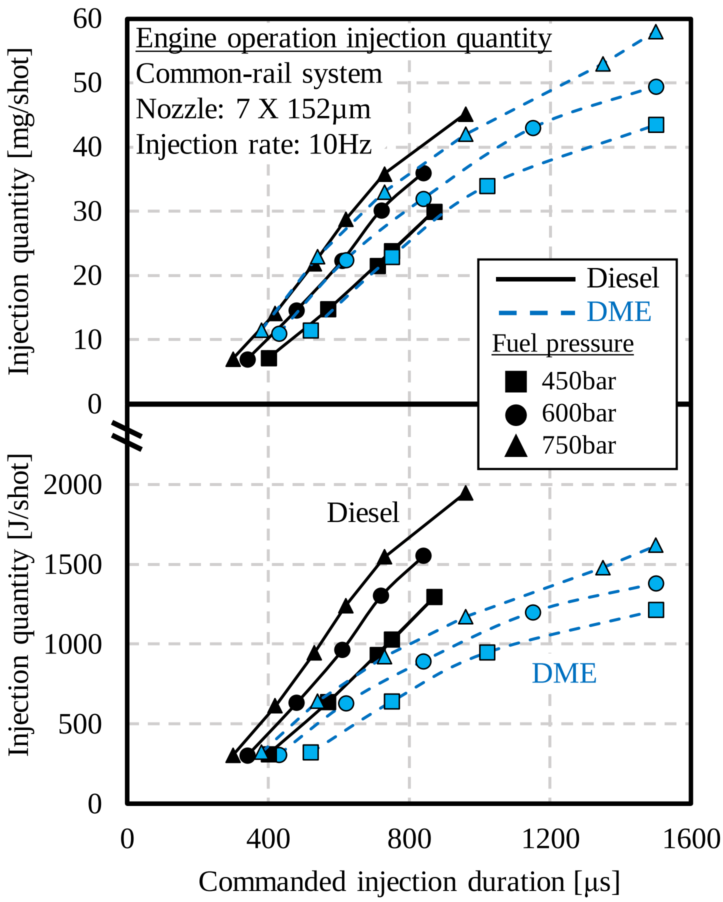


Figure 6. The estimated fuel injection quantity of diesel and DME by mass (upper) and by energy (lower) under engine operation at various fuel pressures.

It is apparent from Figure 6 (upper) that a short range of injection durations may realize similar injection quantities, wherein the commanded injection pulses are shorter than 700 μs. In such a range, the injection supply energy is ~35% less, as shown in Figure 6 (lower). However, further extending the injection duration demonstrated a progressive deviation in the injection quantity of DME as compared to diesel. This deviation may be explained by the higher compressibility and its role during frequent and long transient injection events. To confirm the estimation method for DME injection quantity, the evaluation of cycle-by-cycle cumulative apparent heat release normalized to the fuel supply energy was conducted. Due to the nature of high-temperature direct injection combustion at matching combustion phasing, it was expected that the fuel-to-heat conversion of diesel would be similar to DME. Such evidence was realized in Figure 7a and provides validation to the injection quantity deviation at extended injection durations.

To attain matching engine load at the same operating conditions, the cumulative apparent heat release during a DME combustion cycle should be the same as diesel, as shown in Figure 7b. In brief, for engine management of DME fuel, if the fuel supply quantity by energy is equivalent then the corresponding engine load will be realized. By correlating the equivalent injection quantity by energy from Figure 6 (lower), the equivalent engine load was realized. For example, under 750 bar injection pressure, a commanded injection duration of 1500 μs with DME fuel was equivalent to 730 μs with diesel.

Beyond merely engine load management, the precise timing of fuel delivery is important for combustion management. The delays between the command pulse and actual fuel spray were quantified and presented in Figure 8. The opening delay represents the period between the start of the injection command pulse and to actual injection. Likewise, the closing delay represents the period between the end of the injection command pulse and to actual injection. Both delay periods involve a multitude of effects including electrical (power signal delivery), mechanical (needle motion), and hydraulic (fluid dynamics). Nonetheless, each influence is expected to be present in a similar manner for injection research with the offline injection test bench and on the engine test.

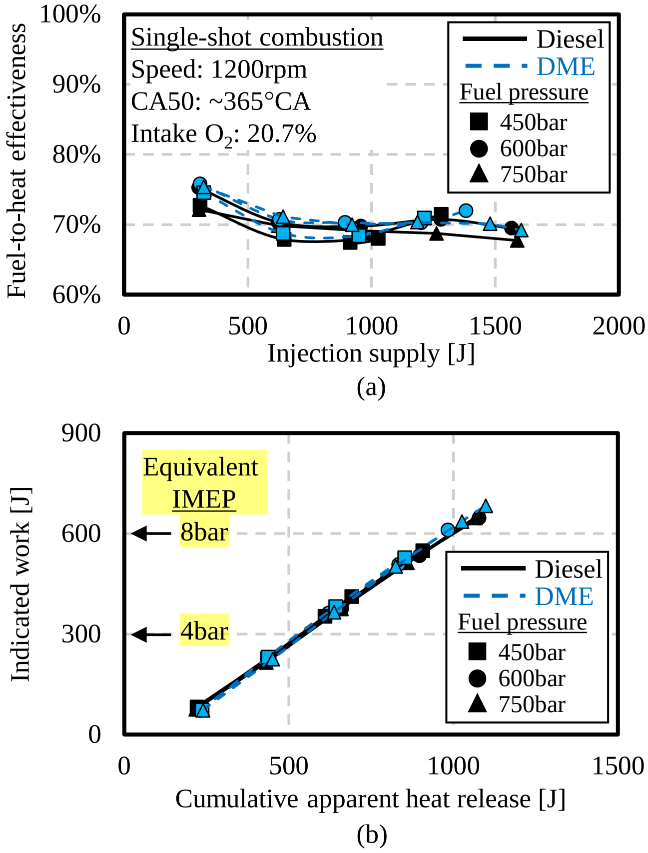


Figure 7. (**a**) The fuel-to-heat release conversion effectiveness of diesel and DME. (**b**) The indicated work produced for the respective cumulative cycle heat released.

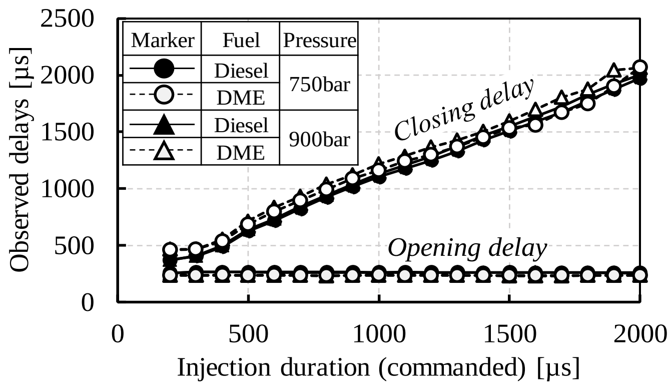


Figure 8. A summary of observed injector opening delays and closing delays operating with diesel and DME.

The injection delays were relatively similar between DME and diesel. The opening delay was unaffected by the injection duration and injection pressure. A nearly steady opening delay of ~240 µs and ~260 µs was observed for DME and diesel, respectively. The marginal reduction in actual fuel injection timing was confirmed using high-speed video recordings of the injection process [35]. In stark contrast, the closing delay steadily extended with injection duration. The extension in closing delay was comparable between DME and diesel.

The spray penetration of the fuel can extensively alter the performance and emissions of the combustion cycle. With insufficient penetration, the fuel-air interactions will deteriorate whereas too much fuel penetration will promote piston- and wall-wetting. In terms of fuel property impact on spray penetration, prevalent spray models [38,39] estimate that only liquid density plays a leading role. Moreover, this influence is only for a brief period (0.3~1.0 ms after the start of injection), whereafter the fluid properties have minimal impact. However, the boiling point of DME is significantly lower than diesel and has demonstrated a flash boiling phenomenon under elevated ambient temperature [40]. Fortunately, under elevated ambient pressures up to engine-like conditions, the boiling point of a fluid rises significantly and is not expected to experience flash boiling. Previous work studied DME spray characteristics and showed a fuel penetration pattern similar to diesel under elevated temperature and pressure conditions [22].

The injection characteristics of DME proved very similar to diesel under engine-relevant conditions. Most notably, the injection quantity by mass and the injection delays were proved to be similar. The fuel injection behaviour of high-pressure DME injection is expected to be similar to diesel. Such results suggest that the combustion management understanding, developments, and practices for modern diesel engines may be applied for DME as necessary.

3.2. Combustion Management

The characteristic overlap of injection and combustion events in a compression ignition combustion system promotes the formation of a diffusion flame. This inevitable phenomenon is that localized pockets of fuel-air at a stoichiometric ratio burn with a maximum flame temperature. Moreover, the cylinder pressures during combustion are further elevated by the use of a high compression ratio and intake pressure. Both factors of high temperature and high pressure promote the formation of NO_x [41]. As such, the NO_x formation is often higher than its spark ignition counterpart.

To mitigate NO_x formation, it is valuable to reduce the oxygen availability in the cylinder thereby lowering the peak flame temperatures within the diffusion flame [42]. In application, this is realized through engine exhaust gas recirculation (EGR), i.e., a lower oxygen concentration, into the fresh air intake stream. By doing so, a significant reduction in engine-out NO_x emissions can be achieved, as shown in Figure 9. A multitude of engine loads were tested under a full-range EGR sweep wherein the EGR alone could lower NO_x emissions by up to 99%. With higher engine load, the concentration of NO_x emissions (top) increased. In stark contrast, the NO_x emissions nearly overlap in the conversion to an indicated power basis, the standard for heavy-duty emission regulations. EGR dilution provides a direct and predictable pathway toward low NO_x formation.

As the NO_x regulations are planned to be reduced further (recall Tables 1 and 2), in-cylinder strategies will be necessary. Following the common trend depicted in Figure 9 (bottom), the intake oxygen conditions for planned regulations (47 mg/kWh) would incur a drop from 17~18% to 14~15%. An intake oxygen range of 14~15% provided a repeatable NO_x emission below 100 ppm.

Apart from intake dilution to limit flame temperatures, combustion timing retard can be an effective method to lower NO_x emissions. In doing so, the peak in-cylinder pressure is lowered and the heat release shape distribution favours premixed combustion as the time for mixing is enhanced, as shown in Figure 10. Three cases of varying combustion timings were controlled by delaying the injection timing. A delay in CA50 of 12 °CA realized a NO_x emissions reduction of 55%, from 93 ppm to 42 ppm. It is interesting to note that while a lower NO_x emission was achieved, a higher exhaust temperature may be expected due to the shift in later combustion. For the same reason, the shift away from the top dead center yielded a loss in thermal efficiency while combustion efficiency remained high.

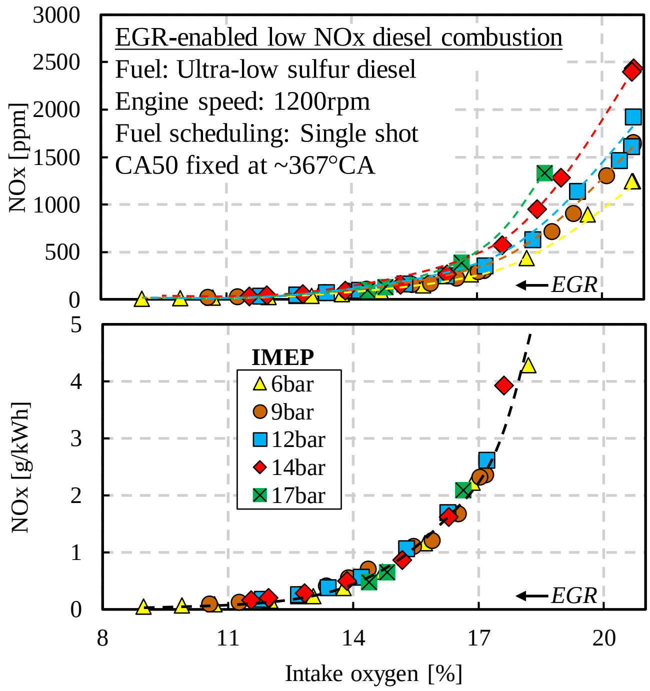


Figure 9. Overview of the EGR-enabled NO_x control strategy.

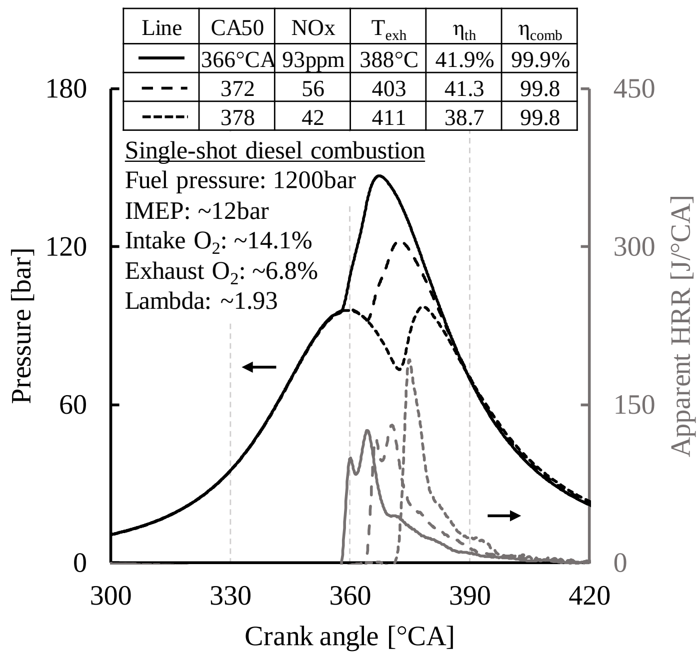


Figure 10. The combustion behaviour from retarding injection timing and combustion phasing (CA50) to mitigate NO_x emissions.

The engine-out soot emissions from combustion equal the net sum of soot formation minus soot oxidation [13]. By limiting oxygen availability inside the combustion chamber, the oxidation rates are reduced. The excess availability of oxygen after the bulk of combustion effectively lowers engine-out smoke emissions via soot oxidation processes. By limiting oxygen availability, net soot emissions rise exponentially, as shown in Figure 11. Therefore, common NO_x mitigation strategies via dilution management often succumb to excessive soot increase that requires further combustion management techniques. Some strategies manipulate the fuel scheduling via early timing or split into multiple shots to decouple the injection and combustion processes [33]. Zheng et al. [17] demonstrated the feasibility of using 8 shots of close-coupled injections to achieve near-homogeneous charged compression ignition with combustion timing control. Fuel scheduling manipulation is a feasible strategy to further minimize soot emissions, however, challenges in engine calibration may arise due to complexity.

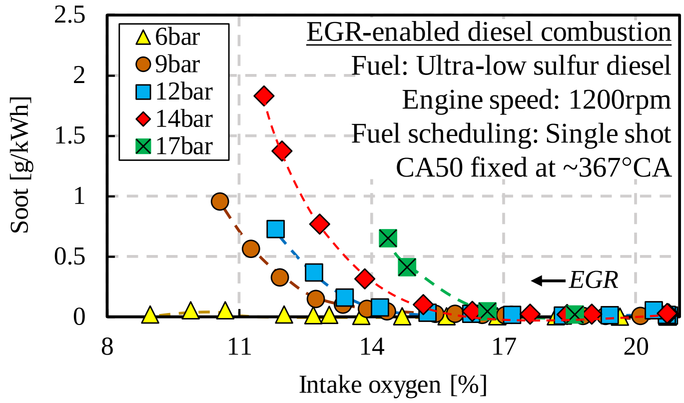


Figure 11. Overview of the effect of EGR on soot emissions.

To mitigate soot formation, the process of soot formation and soot oxidation is centred. Unlike NO_x emissions that are formed from atmospheric air (nitrogen- and oxygen-containing), soot is generally formed from the fuel carbon as a byproduct of combustion. As such, fuel properties play a more significant role in soot formation and oxidation behaviour. Moreover, physical and chemical factors can influence engine-out soot emissions.

A common method to minimize soot formation is through higher injection pressure, as shown in Figure 12. By increasing the fuel pressure, finer droplets are spread throughout the fuel spray and enhance the vaporization processes significantly. In this brief dataset, up to 99% reductions could be achieved. The importance of injection pressure becomes more prevalent at higher engine loads. It is noted that the reductions in smoke emissions via higher injection pressure are subject to a lesser brake efficiency as the pump power draw increases. However, the improvement in soot emissions is essential to meet emission regulations and its adoption has shown huge success with onboard diesel fuel systems extending over 2700 bar.

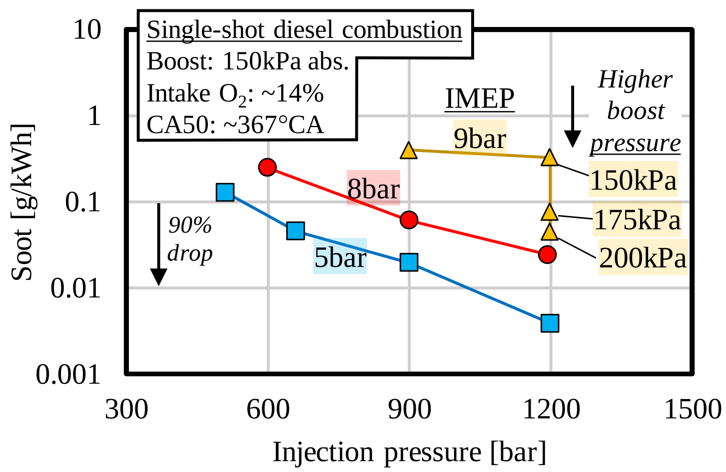


Figure 12. An overview of soot mitigation strategies via increased injection pressure and intake pressure.

The suppression of oxygen is essential for NO_x reduction but is detrimental to the soot oxidation process. Considering NO_x reduction via EGR to be essential to meet emission regulations, oxygen availability may be increased via intake boost pressure, as demonstrated with the 9 bar IMEP case in Figure 12. While the intake oxygen concentration remained fixed to manage NO_x emissions, it was possible to minimize engine-out soot emissions via increased charge density and therefore higher exhaust oxygen for oxidation processes.

The management of NO_x emissions is challenged with a rise in soot emissions. Proper combustion management for handling both NO_x and soot emissions simultaneously require a tight balance of EGR, intake boost, and injection pressure operating conditions. This is especially important with increasing engine load wherein high boost and high fuel pressure are essential. As load increases, the regulation of intake oxygen allows for a lesser EGR rate as the oxygen in the exhaust becomes scarcer. Aside from combustion and emissions, mechanical challenges become more challenging with higher conditions:

· EGR rates introduce unstable flow control, oil degradation through blow-by, and carbonaceous residuals buildup on valves and other surfaces;

· Boost pressures increase wearing on head gaskets and the turbocharge sizing, in turn, fabrication tolerances and sliding friction;

· Fuel pressures increase the parasitic power draw from the crankshaft.

A summary of the NOx-soot trade-off challenge for diesel combustion is shown in Figure 13a. A wide-range EGR sweep was conducted for each engine load setpoint. Furthermore, the fuel pressure and intake boost were increased with engine load to appropriately manage the soot emissions below excessive levels. Overlaid on the dataset are yellow areas of present and planned emission limits with aftertreatment technology for the USA heavy-duty sector. A small red area can be observed as the engine-out limits without aftertreatment. It is apparent in conventional diesel combustion conditions that elevated engine loads significantly challenge soot emissions. It is expected that even higher fuel and intake boost pressures would further limit soot emissions. However, the challenge remains as NO_x and soot emission limits are reduced together.

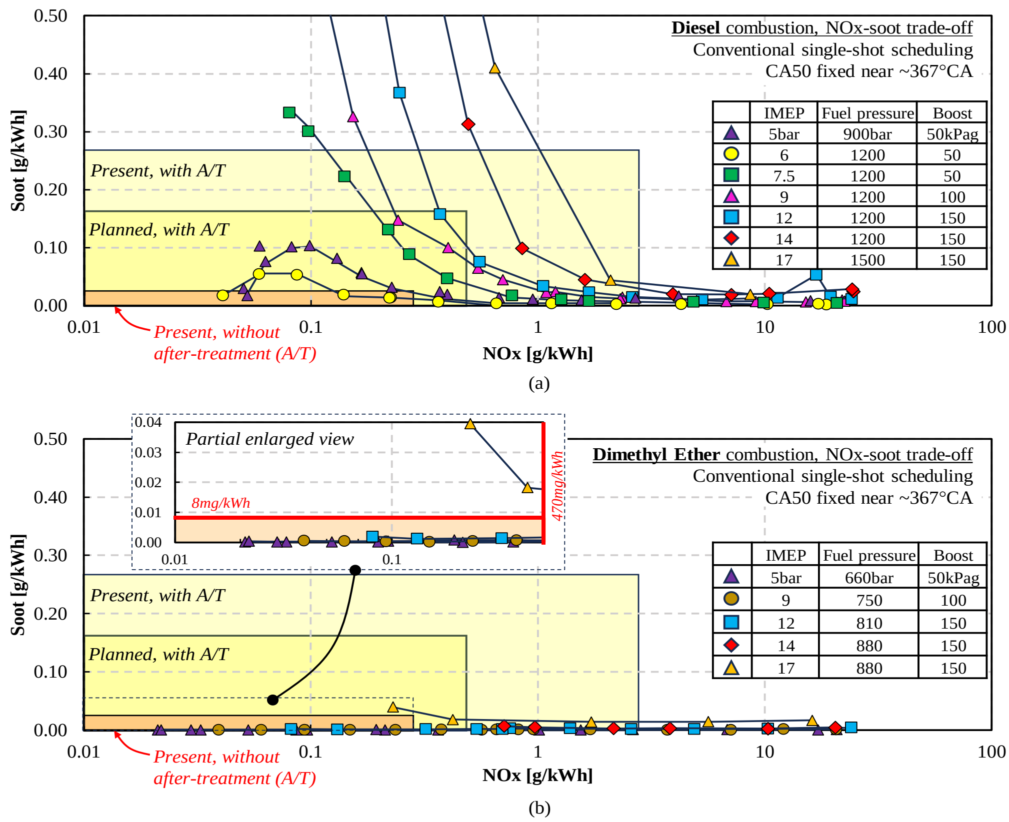


Figure 13. A summary of conventional NO_x-soot trade-off for (**a**) diesel and (**b**) neat DME combustion. Single-shot fuel scheduling with injection timing adjusted to fix CA50 near 367 °CA. Overlaid maps demonstrate the present and planned emission regulations in the USA with and without after-treatment technology.

A similar engine management approach was conducted with neat DME fuel to observe its respective NOx-soot trade-off, as shown in Figure 13b. The primary advantage of DME combustion for compression ignition combustion is clear, the significantly lower soot emissions among all dilution levels. The influence of intake oxygen dilution of NO_x emissions follows similar to that of diesel, in a predictable manner across engine loads. However, extremely low soot emissions were observed.

Under high-temperature combustion, soot emissions are not of concern for diesel or DME combustion. Interestingly, though the longer injection duration of DME extended the diffusion combustion mode, the combustion phasing (CA50) remained similar to diesel under matching operating conditions, as shown in Figure 14a. While the start of injection was matched, at low engine load (short injection pulse) the CA50 of DME was shorter, owing to an earlier ignition delay period and shorter overall combustion duration that is primarily premixed combustion, as demonstrated in Figure 14b. On the other hand, increasing the engine load (longer injection pulses) led to an overlap of CA50 between DME and diesel as the combustion became diffusion-dominant. Correspondingly, the shorter combustion duration of DME was especially noticeable. A similar combustion behaviour is observed from injection to CA50, the deviation of diesel occurs beyond CA50 to the end of combustion. It appears that the second half of the combustion rate for diesel is hindered after the end of injection. In stark contrast, DME had a sharper end of combustion behaviour and was more closely coupled to the end of injection timing.

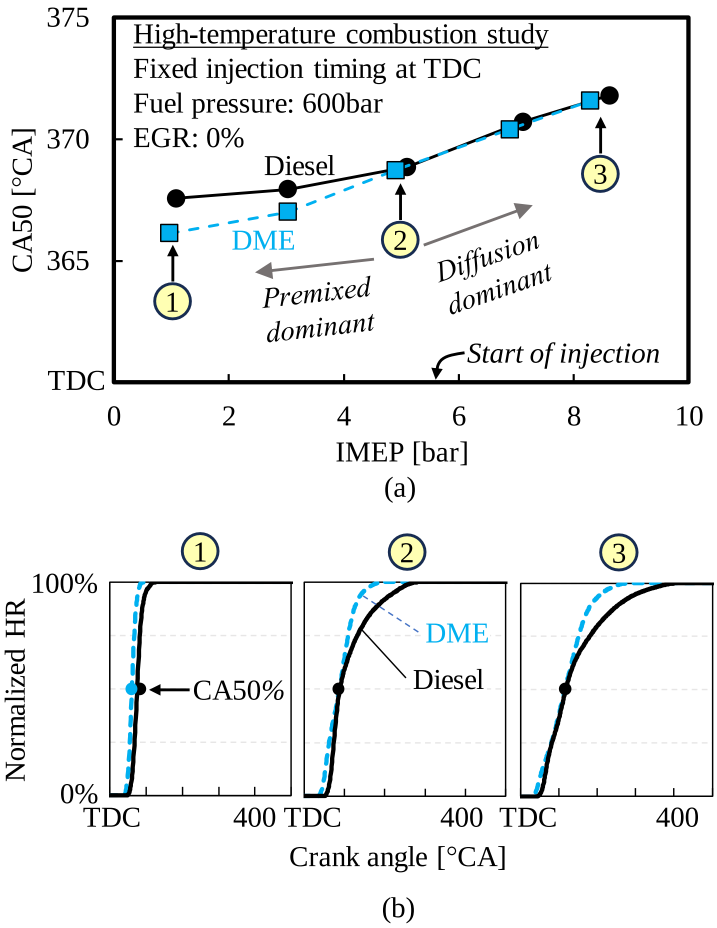
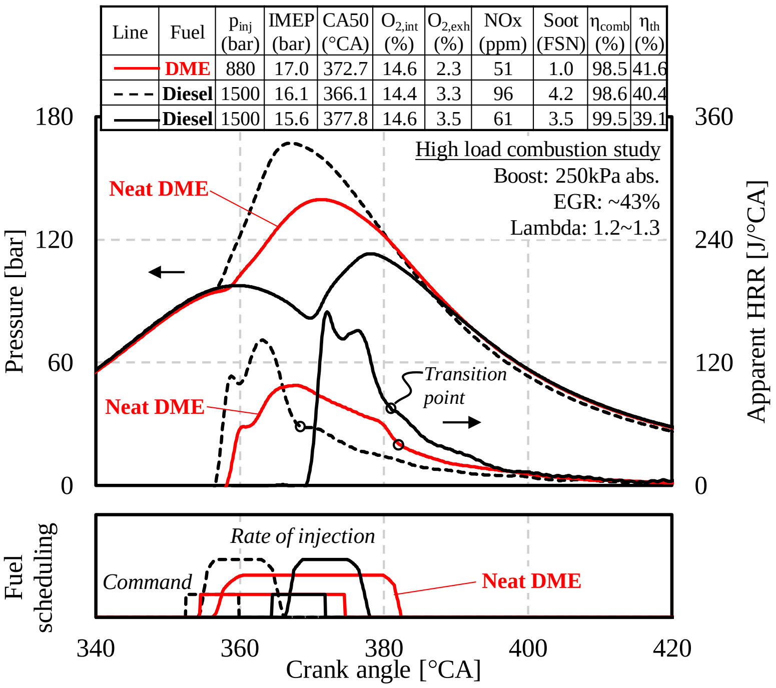


Figure 14. (**a**) The combustion phasing (CA50) timing of DME compared to diesel under matching injection timing and engine load (IMEP). (**b**) Corresponding normalized heat release (HR) patterns.

It is postulated that the fuel properties of DME enable a clean diffusion flame for internal combustion engines. The higher volatility, higher reactivity, fuel-borne oxygen, and lower fuel supply rates enhance the diffusion combustion process. More specifically, the volatility shortens the time required for atomization, the reactivity enhances combustion completeness, the oxygen content minimizes the dependency on surrounding air entrainment, and the lower rate of injection permits the processes to progress without accumulation. This accumulation behaviour may be perceived as the final combustion period after the injection is complete wherein a significant portion of fuel energy remains to burn and the oxygen availability is most scarce. This difference in combustion processes is valuable to the management of DME combustion as an extended overlap of injection and combustion does not realize the same challenges as diesel since the soot emissions are very low.

Under dilution conditions, the majority of conditions for DME fuel realize ultra-low soot emissions, below that of planned emission regulations without aftertreatment particulate filter technology. However, when isolating for extremely challenging conditions of low oxygen availability and high fuel supply quantity, it was observed that DME did produce notable soot emissions (1.0 FSN), that is at 17 bar IMEP and 14.6% intake oxygen. The in-cylinder combustion behaviour was further explored, as shown in Figure 15. Two cases for diesel were compared under similar engine loads and retarded timing for realizing lower NO_x emissions. The target of these conditions was low NO_x emissions in the range of 50~100 ppm target.



Line	Fuel	p_{inj} (bar)	IMEP (bar)	CA50 (°CA)	$O_{2,int}$ (%)	$O_{2,exh}$ (%)	NOx (ppm)	Soot (FSN)	η_{comb} (%)	η_{th} (%)
—— (red)	DME	880	17.0	372.7	14.6	2.3	51	1.0	98.5	41.6
- - -	Diesel	1500	16.1	366.1	14.4	3.3	96	4.2	98.6	40.4
——	Diesel	1500	15.6	377.8	14.6	3.5	61	3.5	99.5	39.1

Figure 15. The high load operation of DME and diesel targeting low NO_x (<100 ppm) production.

The heat release rate magnitude of DME combustion is notably lower than diesel throughout. The combustion proceeds similarly from premixed combustion followed by a period of diffusion combustion. However, beyond the diffusion combustion, near the end of injection timing shown at the bottom of Figure 14, a transition point in the heat release can be observed as noted with a black dot. For the 366.1 °CA phasing diesel case, a significant shift is observed whereafter a slowly reducing yet sustained heat release is observed. The later phasing condition had a less significant transition, albeit still apparent. In the case of DME, however, the transition point is followed by a lower-intensity end-burn behaviour. It is interesting to note that with higher engine load, the cylinder pressure and its maximum pressure rise rate were reduced. Such trends in engine performance parameters were consistent across a range of engine loads, as detailed in Table 5.

A comparison among the lowest achievable engine-out NO_x emissions at matching combustion phasing without a significant drop in combustion efficiency was considered. At the lowest engine load, the highest dilution levels were capable of reaching low-temperature combustion regions and, as such, reaching ultra-low engine out NO_x emissions of <10 ppm. Throughout, DME experienced less combustion noise and comparable indicated thermal and combustion efficiencies all the while operating at lower fuel injection pressure and producing much significantly less soot emissions.

Table 5. A summary of operating conditions, combustion characteristics and engine performance metrics under low, medium, and high load for diesel and DME combustion. The operating conditions were adjusted separately to achieve similar engine load, combustion phasing, and low NO_x targets.

Combustion Mode	Low Load		Medium Load		High Load	
Operating conditions	Diesel	DME	Diesel	DME	Diesel	DME
IMEP (bar)	4.7	4.2	8.7	8.9	15.8	17.0
Fuel pressure (bar)	900	320	1200	750	1500	880
Injection duration (μs)	480	1200	600	1500	1030	2800
Intake pressure (kPa abs.)	150	150	200	200	250	250
CA50 (°CA)	365.3	364.5	364.8	366.0	372.0	372.7
Intake O_2 (%)	9.2	9.8	10.6	10.8	14.6	14.6
EGR (% wt.)	68	77	64	62	42	44
Combustion characteristics						
Ignition delay (°CA)	11.1	11.3	8.1	7.2	5.9	6.5
Combustion duration (°CA)	22.6	23.7	27.9	29.1	35.1	39.5
Maximum pressure (bar)	78.1	76.4	115.2	115.1	140	139.6
Peak pressure rise rate (bar/°CA)	4.8	4.0	10.5	6.1	6.7	4.8
Engine performance						
NO_x (ppm)	8	8	16	13	78	50
NO_x (mg/kWh)	43	44	82	60	393	230
Smoke (FSN)	0.24	<0.01	4.79	0.02	4.9	0.95
Smoke (mg/kWh)	9.4	<0.5	955	0.8	1020	40.5
$\eta_{thermal,\ indicated}$ (%)	36.9	41.8	40.2	39.9	39.7	41.6
$\eta_{combustion}$ (%)	97.2	97.6	98.9	99.0	99.0	98.5

4. Conclusion

The application of DME fuel as an ideal fuel for compression ignition engines would enable a sustainable power source that is not bound to the same NO_x-soot limitation inherent to diesel-fueled combustion. This work made an effort to realize the suitability of a single-shot fueling strategy as a combustion management strategy to enable high-efficiency DME combustion with low NO_x production. The EGR-enabled NO_x management technique for DME was investigated. The expected transient injection response of DME as compared to diesel was revealed. The results were then correlated to the combustion analysis of high-pressure DME engine tests under a variety of operating conditions with a focus on meeting planned NO_x emissions targets in the heavy-duty sector. The presented work communicated the following.

The high-pressure fuel spray response of DME showed similar features as diesel. Offline injection tests proved the opening and closing delay to be marginally affected by DME fuel. The fuel spray of DME reached ~25% greater exit velocity than that of diesel. However, lower liquid density led to an estimated similar injection flow rate by mass. Engine tests confirmed this behaviour under 700 μs wherein the estimated injection quantities were equivalent. Extending the injection duration demonstrated a progressive deviation in the injection quantity of DME as compared to diesel. Nevertheless, higher injection quantities (~55% by mass) realized via injection pressure or injection duration are required to match the energy supply quantity.

The high-temperature combustion of DME was faster than that of diesel. The heat release rate magnitude of DME combustion was lower than diesel. At a fixed injection timing at the top dead center, medium load operation showed a similar combustion phasing (CA50) timing, albeit with a significantly longer injection duration for DME. The deviations in combustion were apparent from CA50 until the end of combustion where the combustion of DME fuel showed a sharper end of combustion that was more closely coupled to the end of injection timing.

A wider range of EGR rates are applicable with DME combustion. At most operating conditions, the soot emissions are below that of upcoming emission regulations without particulate filter exhaust treatment. Under very low oxygen availability and high fuel supply quantity, i.e., lambda 1.2 operating at 17 bar IMEP and 51 ppm NOx, it was observed that DME produced engine-out soot at 1.0 FSN, suggesting a potential necessity of a particulate filter. Nonetheless, the single-shot management of EGR-enabled low NO_x combustion with DME proved effective. It is postulated that lower engine-out NO_x and soot are attainable under all engine loads, especially higher engine loads.

The involved combustion management strategies of diesel systems were expected to affect DME combustion with similar outcomes. EGR dilution is effective in both cases for NO_x mitigation. Mixing enhancement via injection pressure is suggested to improve DME combustion, however to a lower degree than diesel. Although the low sooting tendency of DME incurs a small benefit from injection pressure, operating at considerable dilution and high engine load could find value in extending the injection pressure over 1000 bar. Mixing enhancement via multi-pulse injection scheduling may provide little advantage as the nature of DME mixing quality is high. The high reactivity lessens the applicability of the early-shot multiple injection strategy but favourably produces lower combustion noise with single-shot scheduling.

As a neat fuel, DME is readily applicable for compression ignition systems. Its unique physical properties present engineering challenges in its low-pressure liquified handling but establish combustion and emission tendencies which are valuable to direct injection fueling. This work proved the validity of simply extending the injection duration to compensate the loss of energy density without challenge in combustion efficiency or soot rise. As such, a primarily single-shot approach to fuel scheduling for DME is expectedly sufficient for effective combustion management and in-cylinder emission mitigation. In cases of high fuel supply and dilution (e.g., >17 bar IMEP at <100 ppm NO_x) may require special consideration to avoid engine-out soot and fully eliminate the need of particulate filter in the exhaust treatment. In stark contrast, the suitability of a dual-fuel DME/diesel remains attractive but further complicates the low-pressure handling system. On-board blending of DME/diesel has its inherent challenges whereas a dedicated dual-fuel direct injector may be more appropriate. In such application, it would be essential to avoid diesel injection during combustion whereafter DME injection could continue throughout combustion to take advantage of the clean diffusion characteristics. Empirical testing is necessary to validate such application and evaluate the influence of soot precursors from premixed diesel spray combustion towards forming unsolicited soot emissions.

Future work will focus on the high load and dilution operation of DME and effective combustion management strategies, that is when soot was observed at engine load over 16 bar IMEP and intake oxygen in the range of 14 to 15%. Here, higher injection pressure and fuel scheduling strategies alone may eliminate the particulate filter requirement for DME engines to meet planned emission regulations.

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