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Research article

Combustion and emission characteristics of DI diesel engine fuelled by ethanol injected into the exhaust manifold

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ABSTRACT

Dual fuel diesel engine operation is an important technique used for combustion control in diesel engines. In this study, ethanol is injected into the exhaust manifold of a single cylinder diesel engine. The exhaust valve opens during the intake stroke, enabling vaporized ethanol to enter the cylinder where it is then ignited by diesel fuel injection. The effects of exhaust gas recirculation (EGR) ratios, ethanol injection timing, and ethanol amount are studied. Furthermore, exhaust and intake manifold injection of ethanol compared under the same conditions. These results reveal that ethanol injection into the exhaust manifold increases the apparent heat release rate (AHRR) at the premixed combustion phase. Additionally, the ignition delay increases with ethanol injection by 0.2° crank angle (CA). The indicated mean effective pressure (IMEP) and total heat released per cycle are increased by 8.2% and 14.2%, while the NO_x and soot concentrations are reduced by 88% and 30%, respectively. When compared with exhaust manifold ethanol injection, intake manifold injection results in higher AHRR in the premixed combustion phase, decreased engine performance, an increase in soot production of approximately 35%, and decrease in NO_x of 13%.

1. Introduction

The development of new combustion strategies meant to increase fuel efficiency and reduce the harmful emissions has been driven by multiple factors, including the depletion of conventional fuels, environmental pollution concerns, and tightening exhaust emission standards. Diesel engines, widely used in transportation, electrical power generators, and pumps, play a crucial role in the energy economy. Of additional interest, diesel engines significantly contribute to air pollution and are commonly considered the primary source of NO_x and soot emissions.

Complicating efforts to improve pollution, there is a tradeoff relationship between soot and NO_x formation in diesel engines, which makes the simultaneous reduction of both difficult. Low-temperature combustion (LTC) strategies have been considered efficient for concomitantly reducing NO_x and soot. As the formation reaction of NO_x has high activation energy, low combustion temperatures are able to reduce NO_x emissions [1]. The long ignition delays in LTC additionally provide enough time for fuel and air to mix thoroughly before the start of combustion. This reduces soot formation by diminishing fuel rich regions. The diesel engine dual fuel operation is one example of a practical application of LTC strategies for combustion control and emission reduction [2].

Unfortunately, advanced engine combustion strategies cannot solve all the problems facing our society at present. The use of alternative fuels is imperative for addressing these concepts. Among alternative fuels, ethanol is one of the most widely investigated for use in combination with diesel fuel. Ethanol is an attractive alternative to conventional fuels, as it can be renewably produced from crops such as sugar cane, beetroot, cassava, and sweet sorghum. This increases energy security while reducing reliance on fossil fuels. Globally, ethanol is considered a carbon neutral fuel as the CO_2 produced during combustion is absorbed again during photosynthesis, reducing greenhouse gases. The presence of oxygen in ethanol's chemical composition can potentially reduce soot emissions in diesel engines.

However, use of ethanol fuels in diesel engines suffers from many obstacles. Mixing ethanol with diesel fuel lowers the heat value of combustion. Therefore, higher volumes of ethanol are required to complete the same work as diesel fuel. Ethanol mixes with diesel fuel only in small percentages, and these mixtures are unstable and separate easily in the presence of small amounts of water [3]. Diesel fuel has greater lubricating qualities than ethanol. Furthermore, ethanol has higher latent heat of vaporization then diesel, so mixing ethanol into the fuel leads to charge cooling and combustion quenching [3]. Diesel engines use high cetane number fuels which easy autoignition and short ignition delay,

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Nomenclature

AHRR	Apparent heat release rate
ATDC	After top dead center
B7	Fuel blend of 7% biodiesel and 93% diesel fuel
BDC	Bottom dead center
bsfc	Brake specific fuel consumption
BTE	Brake thermal efficiency
CA	Crank angle
CO	Carbon monoxide
COV	Coefficient of variance
E85	Fuel blend of 85% ethanol and 15% gasoline
EGR	Exhaust gas recirculation
EGR10	EGR ratio of 10%
EGR25	EGR ratio of 25%
EPA	Environmental protection agency
EVC	Exhaust valve close
EVO	Exhaust valve open
FPGA	Field programmable gate array
HC	Hydrocarbon
HCCI	Homogeneous charge compression ignition
IMEP	Indicated mean effective pressure
ITE	Indicated thermal efficiency
IVC	Intake valve close
IVO	Intake valve open
NO_x	Nitric oxides
OH	Hydroxyl group
PFI	Port fuel injection
PM	Particulate matter
ppm	Parts per million
RCCI	Reactivity controlled compression ignition
SOI	Start of injection
TDC	Top dead center
VVA	Variable valve actuating
LTC	Low-temperature combustion

Symbols

N	Number of samples
P	Cylinder pressure [MPa]
Q_{net}	Apparent heat release rate [J/deg]
T	Average gas temperature [K]
V	Cylinder volume [cm^3]
$V_{displace}$	Cylinder displacement volume [cm^3]

Greek symbols

θ	Crank angle degree
κ	Specific heat ratio

whereas ethanol has a low cetane number, low auto-ignition capability, and associated high knock tendency. Using ethanol in diesel engines is not limited to diesel and ethanol blends. Various techniques such as blending, emulsion, fumigation and fuel injection in the intake manifold have been investigated using ethanol in compression ignition engines [4].

The combustion of ethanol and diesel blends in diesel engines has been investigated over the last few decades [3,5–12]. The ethanol to diesel fuel blend ratios in previous studies have not exceeded 20% volumetrically, as higher values may further reduce soot emission but will also affecting engine performance. This is due to the low heating value of ethanol. An ethanol and diesel fuel blending ratio of 15% by volume has been reported as optimum with regards to performance and emissions [3]. These ethanol and diesel fuel blends increase brake thermal efficiency, CO, and HC emissions while reducing soot and NO_x emissions [3,10]. However, higher specific fuel consumption and decreases in torque, power and brake thermal efficiency have been observed as the percent of ethanol in blends increases. This is a result of the low calorific value of ethanol [5–7]. For in-cylinder analysis, adding ethanol to diesel fuel prolongs ignition delay and reduces combustion duration [11,12]. High ethanol percentages in these blends show higher peak cylinder pressures and higher premixed heat release rates compared to unblended diesel [12]. The effect of injection timing on diesel engine performance was investigated by Murcak et al. [9]. They cited that advancing the injection timing 10° CA for ethanol/diesel mixtures gives better engine performance on power, torque, and bsfc compared to standard injection timing of pure diesel fuel [9]. Multiple studies have investigated changes in combustion resulting from the effects of mixing 5, 10 and 15% by mass anhydrous ethanol with a combination of diesel and biodiesel in a diesel engine [11,13,14]. As the ratio of ethanol increased, cylinder pressure and heat release rate were reduced at lower loads and grew at medium or high loads. The 7% biodiesel (B7) containing 15% of ethanol increased fuel consumption by up to 18%, but reduced CO and NO_x emissions by 8% and 10%, respectively [11]. HC emissions were increased at low loads and reduced at high loads [11].

The stability of ethanol and diesel blends is affected mainly by the temperature and water content of the mixture, showing greatest stability at warm ambient temperatures. However, below approximately 10° C, the two fuels separate. This separation can be prevented by adding an emulsifier or co-solvent [15]. Fumigation, an alternative method of introducing alcohols into diesel engines, improves separation prevention at lower temperatures. In dual fuel operation using fumigation, fuels are introduced to air upstream of the manifold at the intake, where premixing with intake air can occur by way of spraying or carbureting [16–20]. Using the fumigation method, it is possible to increase the percentage of injected alcohol over 20% [16].

In-cylinder analysis shows fumigating ethanol results in higher peak pressures, higher heat release rates in the premixed combustion phase, and longer ignition delays at medium or high engine loads [18]. For low engine loads, the heat release rate remains similar between evaluations of ethanol fumigation and pure diesel fuel. The observed longer ignition delay is a result of ethanol's low cetane number and poor autoignition properties. As a consequence of these longer ignition delays, the amount of fuel burned in the premixed phase increases while fuel burned in the diffusion phase decreases. The combustion duration is shortened at medium and high engine loads [18].

When compared to pure diesel combustion, alcohol fumigation also exhibits a higher coefficient of variation of indicated mean effective pressure (COV_{imep}) and reduced maximum in-cylinder temperature [19]. The decrease in cylinder temperature results from ethanol's high latent heat of vaporization [19]. For engine performance parameters, the brake specific fuel consumption increased by 7–12%, a result of ethanol's lower calorific value. Brake thermal efficiency (BTE) decreased at low engine loads by 5–13%, but increased at medium and high engine loads by 2–9% [17]. Emissions were reduced as follows: carbon dioxide by up to 7.2%, nitric oxides by up to 20%, and particulate matter (PM) by up to 57% [16,17]. Additionally, ethanol fumigation increased unburned hydrocarbon (HC) emissions in all load ranges [17].

For several years, great efforts have been devoted to studying ethanol injection using reactivity control compression ignition (RCCI), or port fuel injection (PFI) in the intake manifold. This is a diesel engine dual fuel operation technique [2,21–32] in which two fuels with different autoignition characteristics (one of high reactivity, such as diesel, and the other of low reactivity, such as gasoline or ethanol) are blended inside the combustion chamber [32]. The low reactivity fuel is introduced using port fuel injection, while the high reactivity fuel is directly injected into the cylinder. Combustion phasing is controlled by the relative ratios of these two fuels, and the combustion duration is controlled by spatial stratification between the two fuels [32].

Alternative high reactivity fuels such as biodiesel [31] and gasoline with cetane number improvers have been studied [29,30]. The PFI of alternatives for the lower reactivity conventional fuels, including alcohols such as methanol, ethanol, E85, butanol, and pentanol have been investigated [2,19,30–33]. In some of these, the percentage of fuel injected using PFI was increased by up to 90% for the low reactivity fuel and 10% for the high reactivity fuel [2,30]. For in-cylinder combustion analysis, the introduction of ethanol and E85 increased ignition delay at low and medium loads [22]. Due to the effect of heat of vaporization, increasing injected methanol or E85 is additionally associated with decreases in the peak combustion temperature and final mixture temperature at the end of the compression stroke [27]. For methanol or E85, two characteristic peaks in the heat release rate profile were observed: the first represents combustion of pure diesel fuel, while the second shows burning methanol or E85 [27].

When limited to one diesel injection per cycle, ethanol dual-fuel combustion is limited to ethanol ratios below 50%. Above these levels, blending becomes practically infeasible due to the knocking tendency [25]. Therefore, it is recommended to split diesel injection into two stages: pre-injection and main injection. Early diesel fuel injection raises fuel reactivity in the squish region near the cylinder liner. The main injection, performed closer to the top dead center (TDC), increases reactivity in the central regions [29].

Split diesel injections resulted in higher net indicated efficiency compared to diesel-only operation [28]. For E85 injected as a secondary fuel, NO_x and soot emissions were, respectively, reduced by up to 65% and 29% [28] and reported within the EPA limits of 0.27 g/kWh and 0.013 g/kWh [29]. Decreased nitrogen oxide and soot emissions are a result of the E85 injection, which reduces the excess air factor for the engine. The increased fraction of heat release in the premixed phase of combustion may be an additional contributor for emission reduction [21]. However, NO_x and soot reductions were accompanied by increased CO and HC emissions, particularly at lower loads [21,23]. When the ethanol fraction increased, though, HC and CO emissions were reported to increase [22,25]. Complicating matters, all ethanol and methanol blends reduced soot emissions [22,25].

The effect of intake air temperatures on ethanol combustion in diesel engines was presented by Sarjovaara et al. [22]. A decrease in intake air temperature results in a reduction of the indicated thermal efficiency and exhaust gas temperature, prolonging the ignition delay. This causes later combustion phasing with smaller peak cylinder pressures [22]. Conversely, increasing the intake air temperature reduces the NO_2, HC, CO, formaldehyde and methanol emissions and increases NO, NO_x and soot emissions [26].

It can be concluded from both the previously mentioned articles and results obtained by other investigators [34–36] that the manner in which ethanol is introduced into the engine cylinder influences diesel engine combustion characteristics, impacting both NO_x and soot formation. Previously studied ethanol injection strategies have depended on either injection in the intake manifold or direct injection through blending and emulsion. Although ethanol fumigation and PFI are effective at preventing blend separation and enabling easy shifts of the ethanol-to-diesel ratio in accordance with engine load and speed, the direct injection of ethanol and diesel for blending and emulsion allows ethanol into areas where its presence limits the reduction of emissions. The major drawback, however, with previously developed techniques is that their ethanol injection cools the combustible mixture at the end of the compression stroke. This is a result of ethanol's much higher heat of vaporization (840 kJ/kg) compared to that of diesel (270 kJ/kg). The main consequences of charge cooling and combustion quenching are decreases in the in-cylinder pressure and temperature, NO_x emissions, engine thermal efficiency, and IMEP paired with increased CO and HC emissions.

The present study proposes a new injection strategy to eliminate the charge cooling effect of ethanol. The proposed injection technique depends mainly on ethanol injection into the exhaust manifold, where it is evaporated by the waste heat in exhaust gases. This injection into the exhaust manifold is accompanied by a variable valve actuating system (VVA) which opens the exhaust valve during the intake stroke, allowing evaporated ethanol to enter the engine cylinder. A portion of the remaining exhaust gases are also introduced to the combustion chamber as internal EGR. The vaporized ethanol, EGR, and intake air are mixed during the compression stroke and enhance the next cycle's combustion. This strategy is a kind of retrieval of waste heat energy in which part of exhaust enthalpy is returned to the combustion chamber. This strategy is expected to eliminate the cooling effect of injected ethanol as compared to the previously studied strategies.

Ethanol's heat of vaporization is extracted from the waste heat of hot exhaust gases (300–350 °C) in the exhaust manifold. The main objective of this work was to investigate the effects of injecting ethanol into the exhaust manifold on diesel engine combustion and emissions. A simultaneous evaluation was performed to determine the proposed strategy's effectiveness at avoiding previously reported defects.

The experiments were performed using a single cylinder direct injection automotive size diesel engine. Ethanol injection into the exhaust manifold was dependent on the timing and displacement of exhaust valve reopening. The movement of the exhaust valve reopening leads to variation in EGR ratios. Therefore, the effects of the EGR ratio, ethanol injection timing, and ethanol injection amount are discussed throughout this study. The same experimental conditions were used to conduct a comparison between injections of ethanol at the intake and exhaust manifolds. The in-cylinder combustion analysis was carried out by evaluating cylinder pressure, apparent heat release rate, ignition delay, combustion phasing, and burn duration in the premixed phase as well as late combustion phase. Engine performance parameters such as indicated mean effective pressure (IMEP), brake specific fuel consumption (bsfc), indicated thermal efficiency (ITE), and total apparent heat released per cycle were also evaluated.

For exhaust emissions, the original intent of adding an oxygenated fuel such as ethanol, as such additions result in more complete combustion, was to reduce carbon monoxide emissions. This is a similar effect to increasing the air available for combustion. However, previous studies have found that increasing the percentage of ethanol may also increase HC and CO [22,25]. While all are important considerations for future investigation, this research focused only on the measurements of NO_x and soot, as these two pollutants are the most common from diesel engines.

2. Experimental setup

The experimental setup was comprised a four stroke, water cooled, single cylinder, direct injection automotive size diesel engine. The engine specifications are presented in Table 1, and schematic diagrams of the experimental setup and measurement system are shown in Fig. 1a and b. The engine has an 89 mm bore and 100 mm stroke with a compression ratio of 15. The combustion chamber is a reentrant type with two intake valves and two exhaust valves. One of the exhaust valves was replaced with an optical access quartz window.

The intake and exhaust valves operated according to a hydraulic variable valve actuating system. The VVA system is operated independently from the engine, and consists of a two hydraulic actuators. One of these is connected to the exhaust valve, while the other is connected to intake valves. The oil pressure was supplied to the two hydraulic actuators at 16 MPa from the same oil pump. A pressure accumulator was placed between the hydraulic actuators and the oil pump to absorb pressure pulsation.

Fig. 1d shows a schematic diagram of the VVA control system. This system consisted of servo valves, a servo amplifier, a laser displacement sensor, a FPGA board (PXI-7854R), and LabVIEW software. The values of the valve timing and displacement were defined using this software. The driving signal was sent from the analog input and output board (ADA16-8/2) to the servo amplifier to amplify the valve event data. The servo valves then directed the position of the hydraulic actuators

Table 1
Engine specifications.

Items	Specifications
Number of cylinders	1
Bore [mm]	89
Stroke [mm]	100
Compression ratio	15:1
Combustion chamber	Reentrant type
Injection system	Common rail injection system
Injection nozzle	8 holes, ϕ = 0.158 mm
Intake system	Supercharged
Valve train	Hydraulic variable valves Two intake valves and one exhaust valve
Ethanol injector	Commercial gasoline direct injector
Intake valve lift	8 mm
Intake valve open	14° CA before top dead center
Intake valve closed	30° CA after bottom dead center
Intake valve open duration	224° CA
Exhaust valve lift	8 mm
Exhaust valve open	39° CA before bottom dead center
Exhaust valve closed	5° CA after top dead center
Exhaust valve open duration	224° CA
Exhaust valve re-open lift (for EGR and ethanol injection)	3 mm (10% EGR) – 4 mm (25% EGR)
Exhaust valve re-open	20° ATDC
Exhaust valve re-close	BDC
Exhaust valve re-open duration	160° CA

and, subsequently, the intake and exhaust valves. The laser displacement sensor was installed on the top of the hydraulic actuators to measure the valves lift profiles in the real time and provide feedback to the servo amplifier controlling valve displacement.

The output timing of the driving signal was synchronized with the engine crank angle (CA). The valve displacement and piston position were detected to prevent valve and piston collision. The valve drive system was tested at different engine speeds and was shown to run stably up to engine speeds of approximately 2000 rpm. Internal EGR was enabled by re-opening the exhaust valve during the intake stroke (by defining the re-opening timing and displacement in the control software) to introduce the exhaust gases back into the engine cylinder. The valve timing and displacement data are provided in Table 1, including exhaust valve re-opening data for the internal EGR and equal EGR ratio for each displacement.

The engine is equipped with an electronically controlled Bosch common rail fuel injection system. This common rail system, capable of exerting 120 MPa rail pressure, is connected to a solenoid fuel injector. The solenoid fuel injector has eight holes, each 0.158 mm in diameter, and was mounted on the center of the cylinder head. The fuel injection pressure, timing, and duration were independently controlled using an electronic injection controller. The injection pressure was held nearly constant throughout the injection period.

Fig. 1d shows a schematic diagram for injection control system. The ethanol injection control system used in this study consisted of an engine control unit, signal source (FPGA PXI-7854R with analog input and output board ADA16-8/2), and LabVIEW software. For each ethanol injection condition, the injection duration (ranging from 2.5 ms to 10 ms) and injection timing were defined using LabVIEW software. The signal was sent from software to the ECU, controlling adjustments of the pulse width according to the desired ethanol quantity and start of injection (SOI) timing. An injection quantity test was performed to determine the ethanol injection quantity as a function of injection duration.

For the actual, physical ethanol injection, an electronically controlled commercial gasoline direct fuel injector was mounted on the exhaust manifold. The ethanol injector was installed facing the exhaust port,

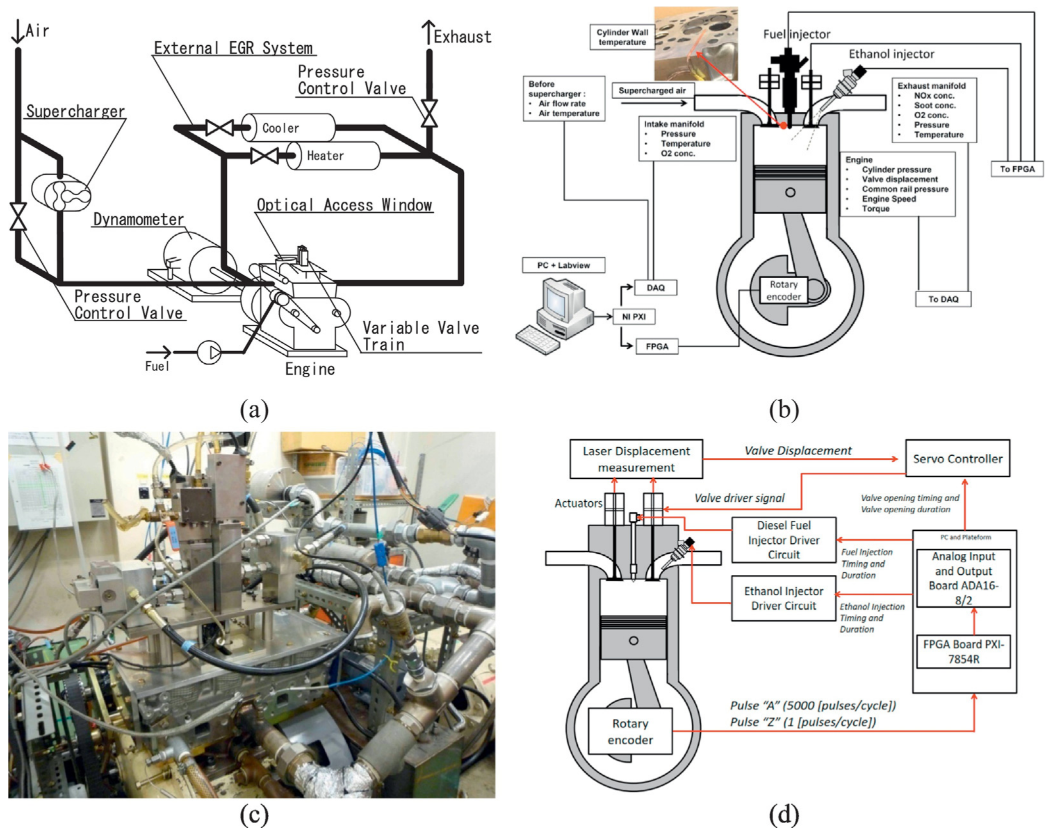


Fig. 1. The diesel engine test system: (a) Experimental system overview. (b) Schematic diagram of the experimental setup and measured parameters. (c) Picture of the experimental setup. (d) Schematic diagram of the variable valve actuating system and injection control systems of diesel and ethanol.

Table 2
Specifications of measuring instruments.

Measured parameter	Instrument	Measuring range	Resolution	Accuracy
NO_x concentration	MEXA – 720 NO_x	0–3000 ppm	1 ppm	±30 ppm or within ±3% of reading scale
Soot concentration	Sokken GSM-3	1–4000 mg/m^3	0.01 mg/m^3	±5% of reading
Intake pressure	Kyowa PHS-B-500Kp	0–500 kPa	–	±0.2% of reading
Exhaust pressure	PE-2KRMT	0–200 kPa	–	±1% of reading
Cylinder pressure	Kistler 6123 with charge amplifier Kistler 5011	0–300 bar	<0.0005 bar	±1%
Air mass flow rate	Laminar air flow LFE-10B	0–15 L/s	–	<±1% FS
Fuel mass flow rate	ONO Sokki gravity flow meter FX-202P with FX 3400	0 to 10 g/s	0.001 g/s	±0.5% of reading ±0.01% FS

and fuel sprays were directed toward the valve top surface. The ethanol supply system consisted of a fuel tank, fuel pump with pressure regulating valve, pressure sensor, port fuel injector, and injector control system. The ethanol was delivered to the port fuel injector at pressures up to 2 MPa using the ethanol fuel pump.

Two methods were used to measure diesel fuel injection quantity. First, the common rail system was operated to allow the injector body temperature to rise to the normal operating conditions, at which point the mass of injected fuel was measured for each of 200 consecutive injections performed. The injector was fixed tightly in the fuel container to prevent evaporated fuel from leaking. For the second technique, the Zeuch method was used to perform an injection rate test for fuel injection quantity as described by Munsin et al. [37].

The single cylinder engine experienced pressure fluctuations in both intake and exhaust pipes. To minimize these fluctuations, a 100-liter capacity surge tank was installed upstream of the intake air. However, supercharged tests showed the surge tank was unable to absorb these pulsations whereas the same surge tank supplied a surge balloon (which attaches to the intake and exhaust pipes) successfully reduced fluctuations. The engine is equipped with a supercharger driven by an independent electrical motor. The engine is coupled with a dynamometer (Yokogawa) used for starting the engine, motoring, absorbing output power, and controlling constant engine load and speed. The maximum absorption power of this system was 15 kW (maximum voltage and current are 220 V and 68.2 A). The dynamometer is equipped with a load cell to measure engine torque.

Fig. 1b shows a schematic diagram of the experimental setup and measurement system. Fig. 1c shows a picture of the experimental system. Specifications of the measurement instrumentation including measuring range, resolution, and accuracy are listed in Table 2. Cylinder pressure was measured using a piezoelectric pressure transducer (Kistler 6123) with charge amplifier (Kistler 5011) at a resolution of 0.144° CA. The intake pressure was measured using a pressure transducer (Kyowa PHS-B-500Kp) installed on the intake manifold. Contrastingly, a water-cooled pressure transducer (Kyowa PE-2KRMT) was used to measure the exhaust pressure at high exhaust temperatures.

Several K-type thermocouples were installed at various points throughout the piping to monitor the intake air, exhaust, lubricant oil, and temperatures of cooling water. The lubricant oil and cooling water circulations were controlled by oil and water pumps operating independently off the engine. Heaters and thermal controllers were used with thermocouples to control temperatures of the intake air, cooling water, and lubricant oil.

Diesel fuel flow rate was measured using a gravity flow meter of Ono Sokki FX-202P with FX 3400 type. The air mass flow rate was measured by a laminar air flow meter (Sokken LFE-10B), installed just before the surge tank. An electronically controlled pressure regulating valve was mounted on the intake manifold to control the desirable boost pressure. Similarly, an exhaust pressure regulating valve was installed so that the exhaust pressure and controlling the internal EGR ratio could be boosted.

A rotary encoder (Nikon RD5000) was connected to the crankcase camshaft, producing 5000 pulses per revolution. Additionally, a photosensor (Omron EE-spx401) was attached to the flange of the crankcase camshaft to produce one pulse per revolution. Pulses from rotary encoder and the photo sensor were necessary for both recording cylinder pressure data and controlling the timing of diesel fuel injection, ethanol injection, and valves. The software and FPGA system, dependent on signals from the rotary encoder and photosensor, were used to monitor the system and acquire data.

The exhaust gases were sampled immediately after the exhaust manifold to measure NO_x, O_2, and soot concentrations. A smoke meter (Sokken GSM-3) was used for soot concentration measurement. The soot sampling line was heated up to 200 °C to prevent soot condensation. The NO_x sensor (Horiba, MEXA-720), installed on the exhaust manifold, provided reliable measurements of the NO_x concentration (range 0–3000 ppm, resolution 1 ppm and accuracy of ±3% of reading scale).

3. Experimental conditions

Table 3 presents the experimental conditions, and Table 4 shows the properties of ethanol and diesel fuel. The current study used absolute ethanol, and the engine speed was controlled at 1000 rpm for all tested conditions.

The combustion process and combustion phasing are influenced by the temperatures of intake air, cooling water, and lubricating oil. Given this, the following controlled temperatures were established: intake air at 65 ± 2 °C, cooling water at 85 ± 2 °C, and lubricating oil at 70 ± 2 °C. The main purpose of temperature control at these values was to achieve conventional diesel combustion conditions with high soot concentration.

Table 3
Experimental conditions.

Engine speed [rpm]	1000
Fuel injection pressure [MPa]	100
Fuel injection amount [mg/cycle]	32
Fuel injection timing [ATDC]	−6
Intake air temperature [°C]	65 ± 2 °C
Cooling water temperature [°C]	85 ± 2 °C
Lubricating oil temperature [°C]	70 ± 2 °C
Intake valve lift [mm]	8
Exhaust valve lift [mm]	8
Exhaust valve reopen lift [mm]	3 (10% EGR), 4 (25% EGR)
Equivalence ratio for the reference diesel fuel case	0.73
IMEP for the reference diesel fuel case [MPa]	0.86

Table 4
Fuel properties [24,38–40].

	Diesel	Ethanol
Specific gravity [kg/m^3] at 15.5 °C	0.84	0.785
Viscosity [cP] at 20 °C and 1 atm	3.35	1.2
Molecular weight	170	46.07
Higher heating value [kJ/kg]	46,100	29,700
Lower heating value [kJ/kg]	43,200	26,900
Heat of vaporization [kJ/kg]	270	840
Cetane number	50	8

The diesel injection timing was fixed at −6° ATDC to control combustion near to the TDC. The value −6° ATDC was selected to achieve a fast combustion process, reducing the impact of cyclic variations on engine performance and achieving high IMEP. Additionally, early diesel injection timings such as −13° ATDC or before cause severe knocking, while the advanced injection timings such as 1° ATDC lead to misfires, as reported by Padala et al. [24].

The diesel fuel injection quantity was fixed at 32 mg/cycle, corresponding to 1100 μs at an injection pressure of 100 MPa, to achieve conventional diesel combustion conditions. Lower diesel fuel injection quantities have lower equivalence ratios and form a homogenous mixture with air, at which time the combustion process occurs at premixed phase without late combustion phase. These conditions would result in a combustion process similar to premixed charge compression ignition (PCCI) combustion, which we intended to avoid in favor of more conventional conditions.

The measured parameters in this study were EGR ratio, ethanol injection timing, and ethanol injection amount, as indicated in Table 5. The tested EGR ratios were 10% (EGR10) and 25% (EGR25). Ethanol injection timings of 350°, 400°, and 450° were investigated and adopted with valve timing as shown in Fig. 2. The exhaust valve reopening timing and displacement are important factors affecting ethanol injection into the exhaust manifold.

Fig. 2 shows the ethanol injector reference signals and valve timings at tested ethanol injection timings. The higher peak represents the exhaust valve lift at 8 mm during the exhaust stroke, and the small peak is the exhaust valve reopening at 4 mm during the intake stroke. Ethanol injection timing is represented by the injector reference signal. For ethanol injection timing of 350°, the ethanol SOI is before the exhaust valve reopens, and the end of injection (EOI) is just after the exhaust valve reopens. The 400° ethanol injection timing in Fig. 2b shows that the SOI is after the exhaust valve reopening and the EOI is during the exhaust valve full opening. 450° ethanol injection timing starts as the exhaust valve full opens and ends before the exhaust valve closes. All tested ethanol injection timings were performed during the opening of the intake valve.

Ethanol injection is additionally influenced by the position and angle of the injector on the exhaust manifold. For this study, the injector position was fixed approximately 0.2 m from the exhaust valve at a 45° angle. The tested ethanol injection amounts ranged from 1.2 mg/cycle to 22 mg/cycle, as shown in Table 5. The results of these ethanol injections were compared to EGR without ethanol injection as well as conventional diesel combustion without EGR. An additional comparison between ethanol injection at the same experimental conditions into the intake manifold and exhaust manifold is conducted in the last section of this paper. The ethanol injection amounts and energetic fractions for all tested conditions are provided in Table 5.

4. Methodology

The IMEP was used as a measure of output power. It is considered a valuable measurement of an engine's capacity to do work independent of engine displacement. The IMEP is defined in Eq. (1) as the indicated work divided by the displacement. It is calculated via trapezoidal integration of the cylinder pressure and volume in the expansion stroke for each cycle as follows:

$$\mathrm{IMEP} = \frac{\int pdV}{V_{displace}} \tag{1}$$

where p is the cylinder pressure, V is the cylinder volume, and $V_{displace}$ is the cylinder displacement volume.

Cylinder pressure data was sampled every 0.144° CA (2500 sample from −180 to 180). For each experimental condition, 200 consecutive cycles were sampled and averaged. The average cycle was used to represent the cylinder pressure and apparent heat release rate, as illustrated in Fig. 3a and b. The 200 cycles shown in Fig. 3 represent conventional diesel combustion at 1000 rpm and a load of 0.86 MPa IMEP.

Heat release rate peaks vary from cycle to cycle in the premixed combustion and diffusion phases, as does the burning duration. The magnitude of these variations depends on the speed of burning in the combustion chamber, cyclic cylinder fuel, and air charging variations. The maximum cylinder pressure varies proportionally. When compared, cycles with longer ignition delay have a higher peak of heat release rate in the premixed combustion stage and a lower peak in the diffusion combustion stage than those with shorter ignition delay cycles. In addition, cycles with longer ignition delay have a shorter combustion duration and higher cylinder pressure peaks. The magnitude of changes in maximum cylinder pressure, combustion phasing, and the timing of burning process is dependent on these cyclic variations.

Table 5
Experimental program.

Tested condition	EGR ratio [%]	Ethanol injection amount [mg/cycle]	Diesel injection amount [mg/cycle]	Total Fuel energy per cycle [J/cycle]	ethanol energetic fraction per cycle [%]	Timing of ethanol injection into exhaust manifold [0–720° CA]	Symbol
Effect of internal EGR	10%	–	32	1382.4	–	–	EGR10
	25%						EGR25
Effect of ethanol injection timing	10%	11	32	1678.3	17.6	350°	E11EGR10-350
						400°	E11EGR10-400
						450°	E11EGR10-450
	25%					350°	E11EGR25-350
						400°	E11EGR25-400
						450°	E11EGR25-450
Effect of ethanol injection amount	25%	1.2	32	1414.68	2.3	400°	E1.2
		3.8		1484.62	6.9		E3.8
		4.8		1511.52	8.5		E4.8
		11		1678.3	17.6		E11
		22		1974.2	30.0		E22
Comparison between intake and exhaust manifold ethanol injection	25%	11 (intake injection)	32	1678.3	17.6	400°	E11-In
		11 (exhaust injection)					E11-Ex
		22 (intake injection)		1974.2	30.0		E22-In
		22 (exhaust injection)					E22-Ex

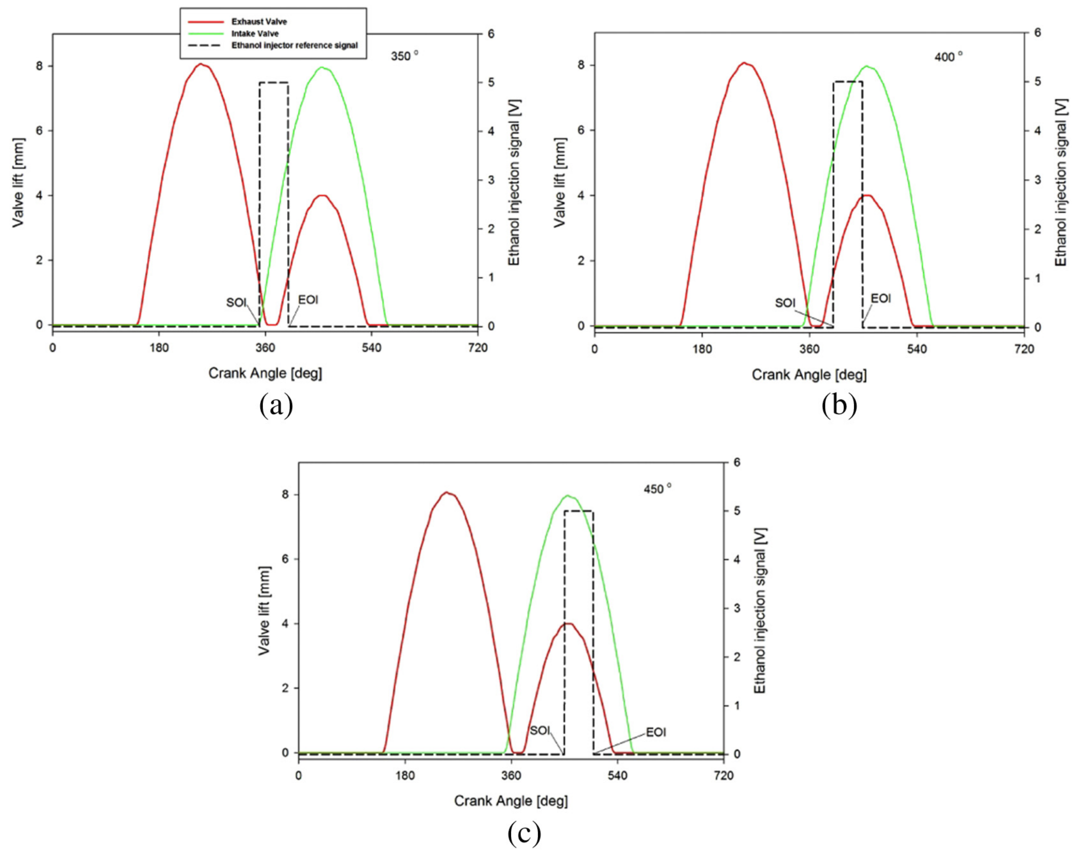


Fig. 2. Ethanol injector reference signal and valve timing at different ethanol injection timing (a) 350° (b) 400° (c) 450°, intake and exhaust valve lift = 8 mm, exhaust valve reopening lift = 4 mm.

The cycle-to-cycle variation was evaluated by calculating the coefficient of variance (COV_{imep}) at each experimental condition according to Eq. (2). Furthermore, the stability of engine operation was assessed by calculating COV_{imep}, which is defined as the ratio of the standard deviation in IMEP and the mean IMEP over the sampled cycles [41].

$$COV = \frac{\frac{\sum |IMEP_{average} - IMEP_i|}{n}}{IMEP_{average}} \times 100 \tag{2}$$

where n is the number of samples.

The acceptable level of load variation was somewhat subjective, but typically taken to be a $COV_{imep} < 4\%$ in accordance with literature [41].

Fig. 3a shows an example of the average cycle used to represent the data obtained from 200 cycles, where the $COV_{imep} = 0.8\%$.

Cylinder pressure was analyzed using the AHRR, calculated from the measured pressure data and the cylinder volume according to Eq. (3) [38]:

$$\frac{dQ_{net}}{d\theta} = \frac{1}{\kappa - 1} V \frac{dP}{d\theta} + \frac{\kappa}{\kappa - 1} P \frac{dV}{d\theta} - \frac{PV}{(\kappa - 1)^2} \frac{d\kappa}{d\theta} \tag{3}$$

where Q_{net} is the apparent rate of heat release, and κ is the specific heat ratio.

The value of the specific heat ratio (κ) is dependent on gas temperature, and was calculated as a function of average gas temperature (T)

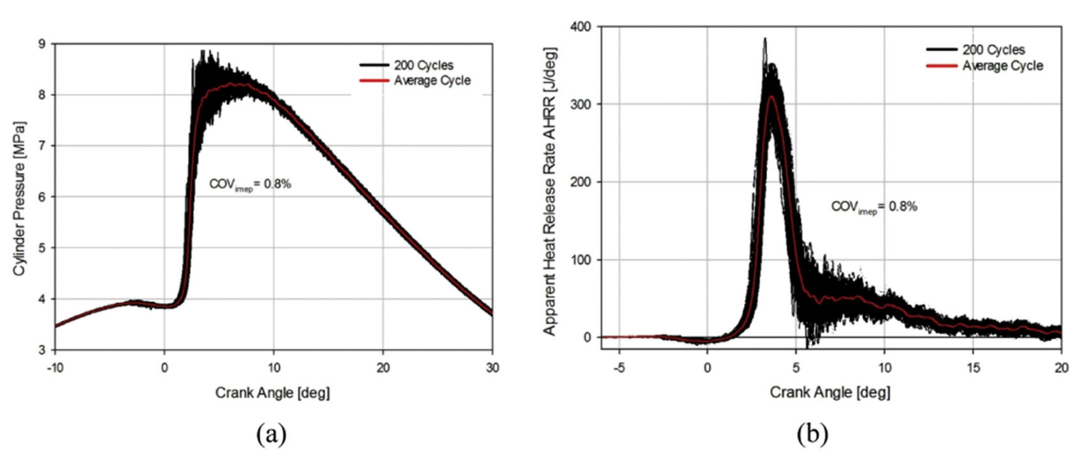


Fig. 3. An average cycle of the conventional diesel combustion that was used to represent the 200 cycles with $COV_{imep} = 0.8\%$ (a) p-θ diagram (b) AHRR.

using Eq. (4):

$$\kappa = 1.386 + 1.776 \times 10^{-4}T - 5.293 \times 10^{-7}T^2 + 4.004 \times 10^{-10}T^3 - 9.932 \times 10^{-14}T^4 \quad (4)$$

The above equation is an approximation formula based on thermophysical property values [1,38] that can be obtained also from:

$$\kappa = cp(T)/cv(T)$$

where cp and cv are the gas specific heat under constant pressure and volume, respectively. The polynomials of cp and cv can be obtained from thermodynamic tables of gases [1,38].

Combustion phasing is determined by CA03, CA50, and CA90. CA03 is the crank angle at which 3% of the total heat is released, as illustrated in Fig. 4a. Similarly, the CA50 and CA90 are defined as the crank angle at which, respectively, 50% and 90% of the total heat has been released. The total heat release amount was calculated according to Eq. (5). The CA03 calculation was used as an indication of the start of the combustion process as well as ignition delay estimation. Ignition delay is defined as the time interval between the start of injection (SOI) and the start of combustion (SOC). The start of injection can be defined as the time that voltage is supplied to the injector (start of energizing). The injector used in the current study has approximately 700 μs injection delay, shown in Fig. 4b to be equivalent to 4.2 CA degrees at 1000 rpm.

The start of combustion is identified differently throughout relevant literature. The SOC was defined as the crank angle when the heat release reaches 0 J after diesel evaporation phase by Sarjovaara et al. [22]. Liu et al. [42] defined the start of combustion as CA05, while it was defined as CA10 by Dempsy et al. [41], Padala et al. [24,43], and Tutak et al. [21]. In previous definitions, SOC can depend on crank angle when the heat release reaches either 0 J or 10% of the total heat released. These different definitions are primarily the result of past attempts to ensure complete evaporation of the diesel fuel and start of the chemical reaction at the selected time. Since the current study uses the waste heat recovery strategy to increase the in-cylinder temperature and enhance fuel evaporation, the start of combustion is defined here as CA03 for a more accurate calculation of ignition delay.

The duration between CA03 and CA50 shows the burn duration for the premixed combustion stage [21,24]. The CA50 determines the end of premixed combustion phase and the start of the late combustion stage. The CA90 indicates the end of the combustion process [41]. The late combustion phase duration is determined to be the duration between CA50 and CA90 [21,24].

$$Q_{net} = \int_{\theta_{IVC}}^{\theta} \left(\frac{dQ_{net}}{d\theta}\right) d\theta \quad (5)$$

The final result, a description of engine performance, was calculated using initial measurements of variables such as cylinder pressure and air flow rate. The levels of uncertainty in engine performance parameters such as bsfc and IMEP are due to the acceptable error levels in the initial measurements. A quantitative estimation of the uncertainty levels in engine performance parameters was calculated following the method proposed by Kline and McClintock [44] and Holman [45], using Eq.(6).

$$R = R(x_1, x_2, x_3 \ldots, x_n)$$

$$w_R = \left[\left(\frac{\partial R}{\partial x_1} w_1\right)^2 + \left(\frac{\partial R}{\partial x_2} w_2\right)^2 + \ldots + \left(\frac{\partial R}{\partial x_n} w_n\right)^2\right]^{1/2} \quad (6)$$

where R, the calculated engine performance parameter under consideration, is a function of the measured independent variables $x_1, x_2, \ldots x_n$. The uncertainty in the final result (w_R) is calculated using Eq. (6) with $w_1, w_2, \ldots w_n$ defined as the uncertainties in the measured independent variables. The uncertainty in the measurements of IMEP, bsfc, ITE and engine speed were found to be 1%, 1.6%, 0.7% and 0.28% (±3 rpm), respectively. For each experimental condition, 200 samples were taken for 200 consecutive cycles. The average value of these obtained results is used to describe variation in engine performance, presented in the figures shown in the results and discussion section. Error bars were added to the figures to show maximum and minimum values of the 200 samples.

5. Results and discussion

5.1. Effect of internal EGR ratio

The main aim of injecting ethanol into the exhaust manifold was to eliminate the ethanol's endothermic effect. The introduction of ethanol to the engine cylinder following initial injection into exhaust manifold involves some of the exhaust gases. This system provides a measure of waste heat recovery, in which the part of exhaust enthalpy is used to evaporate ethanol and returned to the combustion chamber, enhancing combustion in the next cycle. This section primarily investigates the effect resulting from EGR without ethanol injection to distinguish between the EGR effect and ethanol addition effect in next sections.

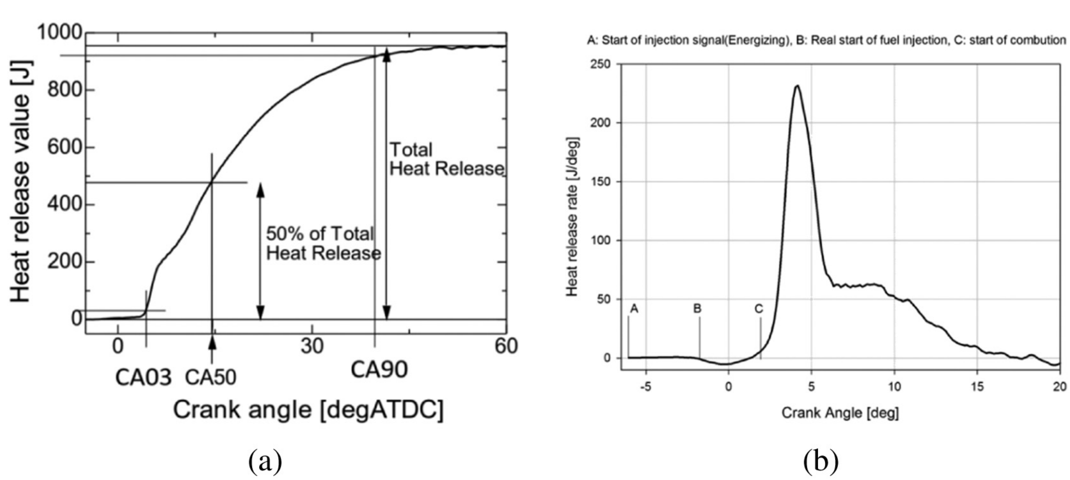
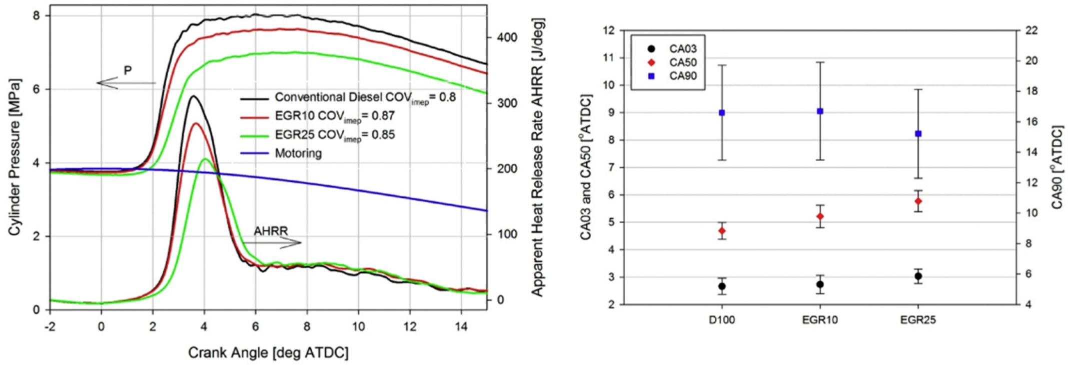


Fig. 4. Combustion phasing and start of injection signal (a) definition of CA03, CA50, and CA90 (b) start of energizing and start of injection.



(a) (b)

Fig. 5. Effect of different EGR ratios on combustion: (a) Cylinder pressure and AHRR; (b) Combustion phasing CA03, CA50, and CA90.

The exhaust valve can lift 3 mm or 4 mm to reopen during the intake stroke, which respectively correspond to 10% EGR (EGR10) and 25% EGR (EGR25) ratios. Fig. 5 shows the measured cylinder pressure, AHRR, and combustion phasing at different EGR ratios. The cylinder pressure decreases with increasing EGR ratios, as shown in Fig. 5a. Compared to conventional diesel combustion, the peak cylinder pressure declined by 6.5% for EGR10 and by 13.6% for EGR25. This can be attributed to the fact that increasing EGR ratio decreases the incoming concentration of O_2, which in turn decelerates O_2 and fuel mixing and resulting in an extension of the flame region. Thus, the quantity of CO_2 and H_2O gases that absorb released heat increases as a result of lower flame temperature and AHRR, as illustrated in Fig. 5a. The relation can be drawn that increased amounts of H_2O and CO_2 will raise the inlet heat capacity, concurrently decreasing gas temperature during combustion.

Further comparison showed that the AHRR during the premixed combustion phase decreased with EGR compared to conventional diesel combustion. However, increased AHRR was seen in the mixing controlled combustion phase as the EGR ratio increased. This is because, as the EGR ratio increases, the amount of fuel burned in the premixed phase becomes lower than that burned in the mixing controlled combustion phase.

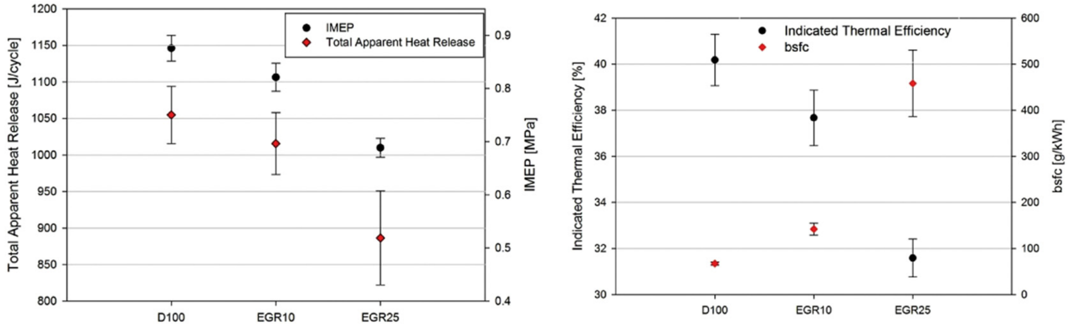
The analysis of the AHRR in Fig. 5a and combustion phasing in Fig. 5b reveals that the ignition delay period increases as the percentage of EGR increased. The CA03 (an indication of ignition delay) increased with increasing EGR ratios, reaching 2.72° for EGR10 and 3.00° ATDC for EGR25 compared to 2.69° ATDC for conventional diesel. Similarly, CA50 was 5.2° for EGR10 and 5.8° ATDC for EGR25 compared to 4.7° ATDC for conventional diesel. This is a result of the causal relationship between increased EGR ratios, lower O_2 amounts, and longer ignition delays. It can also be shown that combustion ends earlier with increasing EGR ratios: the CA90 for EGR25 is 14.9° ATDC compared to 16.4° ATDC for conventional diesel combustion. The earlier completion of CA90 is due to a lack of O_2, resulting in incomplete combustion. This reduces both the total heat released per cycle and IMEP, as shown in Fig. 6a.

The total heat release decreased by 3% for EGR10 and 16% for EGR25 compared to conventional diesel fuel. Furthermore, the IMEP declined while EGR ratios increased: IMEP drops by 6% for EGR10 and 21% for EGR25. Consequently, the bsfc increases dramatically for EGR10 and EGR25, and the indicated thermal efficiency is also reduced relative to conventional diesel by 6% and 21% for EGR10 and EGR25, respectively. These results are shown in Fig. 6b.

The reduction of flame temperature is due to the higher heat capacity of H_2O and CO_2 as well as the decrease in O_2 concentration at the inlet. Consequently, the decreased fuel mixing efficiency led to reductions in NO_x emissions but increased the soot produced, as shown in Fig.7. NO_x emissions were reduced, respectively, by 52% and 85% for EGR10 and EGR25. Conversely, soot concentration increased three times for EGR10 and five times with EGR25 compared to conventional diesel combustion.

5.2. Effect of ethanol injection timing

The injected amount of ethanol was fixed to 11 mg/cycle, representative of 17% of the total added heat per cycle as indicated in Table 5. Ethanol injection timings of 350°, 400°, and 450° were investigated. With the proper ethanol injection timing and EGR ratios, excellent combustion characteristics and lower soot and NO_x emissions can be obtained. Fig. 8a and b show the cylinder pressure and AHRR for ethanol



(a) (b)

Fig. 6. Effect of different EGR ratios on engine performance parameters: (a) Total heat released and IMEP (b) indicated thermal efficiency and bsfc.

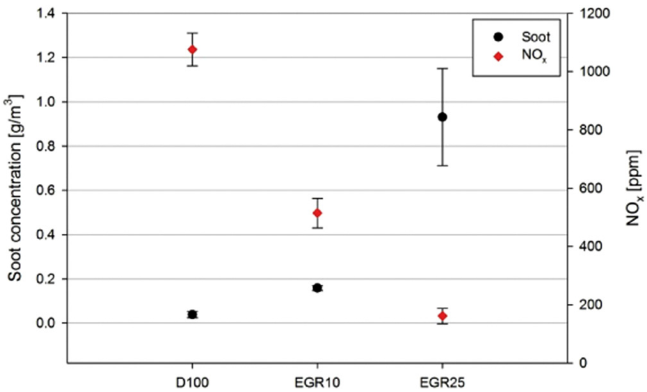


Fig. 7. Effect of different EGR ratios on soot and NO_x concentrations.

injection at EGR10 and EGR25, respectively. By injecting ethanol, the peak cylinder pressure was increased by 1%, 2% and 1.5% for injection timings of 350°, 400°, and 450°, respectively, compared to EGR10 without ethanol injection. The increase in cylinder pressure was attributed to the increase in input energy given by injecting ethanol. On the other hand, the cylinder pressure decreased by 1.5% on average for all tested ethanol injection timings (350°, 400°, and 450°) at EGR25 as compared to the same EGR condition without ethanol injection.

EGR25 occurred at 4 mm exhaust valve lift, an opening measurement that should have delivered more ethanol into the engine cylinder. However, when the delivered amount of ethanol is combined with a high EGR ratio, the effect of ethanol on the cylinder peak pressure is reduced. Ethanol injection experiments showed that maximum cylinder pressure was achieved with an injection timing of 400° CA at EGR10. Fig. 8a shows that premixed combustion phase AHRR is higher for ethanol injection than EGR10 without ethanol injection. This means the tested ethanol injections increase the amount of fuel burned under premixed conditions when compared with EGR.

The low cetane number of ethanol contributes to this effect, as ethanol injection prolongs ignition delay and gives a longer mixing time between fuel and air, resulting in a higher amount of fuel burned under premixed conditions. Additionally, ethanol injection was shown to increase the total heat added per cycle and, consequently, the amount of fuel burned under premixed conditions in comparison to EGR without ethanol injection.

For EGR10, the value of CA03 was not significantly changed between EGR with and without ethanol injection. However, at EGR25, ethanol injection increased the ignition delay increases by 0.2° CA. As reiterated in Table 3, ethanol has a low cetane number (8), and displays both overall poor ignitability and low autoignition quality relative to diesel fuel. Previous studies [11,12,18] concluded that ethanol injection increases the ignition delay. The current study was able to reduce this effect by vaporizing ethanol before it entered the engine cylinder, reducing total ethanol evaporation time. Using this technique, ethanol injection in exhaust manifold gives a similar ignition delay to cases without ethanol injection.

While combustion duration, shown in Fig. 8c was reduced when ethanol was injected at EGR25 without ethanol injection, EGR10 without ethanol injection did not have the same effect. The shorter combustion duration at EGR25 is due to the high EGR ratio, resulting in incomplete combustion. Consequently, soot emission increases at both of EGR10 and EGR25 as shown in Fig. 9. However, under the combined conditions of injection timing at 350° CA and EGR10, ethanol injection decreased soot emissions by 68%. At EGR25, the soot emission was decreased by 29% at 400° CA ethanol injection timing. Soot is reduced in these conditions because the addition of oxygenated ethanol fuel enhances soot oxidation.

Conversely, NO_x emission was decreased by 52% for EGR10 and 85% for EGR25 without ethanol injection. Ethanol injection, increased NO_x emissions by 16% compared to the cases without injection. The increased NO_x emissions result from the increased cylinder pressure and volume of burned fuel in the premixed phase associated with ethanol injection.

Total heat release is shown in conjunction with IMEP values in Fig. 10a. It can be seen that the highest IMEP and heat release values for EGR10 and EGR25 were attained at ethanol injection timing of 400° CA. The increase in added heat value is responsible for these effects. This effect becomes unclear at high EGR ratios, as can be seen with EGR25. The bsfc and indicated thermal efficiency are shown in Fig. 10b for tested conditions. Ethanol injection at 400° CA gives a higher indicated thermal efficiency and lower bsfc relative to 350° and 450° injection timings for both of EGR10 and EGR25.

As the combustion stability, indicated by COV_{imep}, remained below 1% for all tested conditions it can be determined that ethanol injection into exhaust manifold has no influence on combustion stability. This contrasts with literature accounts of intake manifold injection [19].

After evaluating the obtained results, the ratio EGR25 and 400° CA injection timing were selected to conduct further experiments investigating the effect of different ethanol injection amounts on combustion characteristics and emissions.

5.3. Effect of ethanol injection amounts

This section discusses the effect of the ethanol injection amount. Tested ethanol injection amounts were 1.2 mg, 3.8 mg, 4.8 mg, 11 mg and 22 mg. This varied between 2.5% to 30% of the total added energy per cycle, as indicated in Table 5. The equivalence ratios of different ethanol and diesel fuel conditions are shown in Fig. 11, in which the values of ϕ range from 0.73–1.25. As stated above, the ethanol injection timing was fixed at the 400° CA, and EGR ratio was kept constant at 25%. Fig. 12a shows the cylinder pressure and AHRR. The results indicate that increasing the amount of ethanol injected corresponds with higher the cylinder peak pressure.

The peak cylinder pressure increased, respectively, by 0.8%, 1%, 1.5%, 2% and 2.5% for the volumes listed above relative to EGR25 without ethanol injection. The AHRR in the premixed combustion phase increased with as the volume of ethanol injected increased. For the mixing controlled combustion phase and late combustion phase, the AHRR was higher for ethanol injection than EGR25 without ethanol injection. The combustion phasing for ethanol injection is shown in Fig. 12b. Ignition delay was similar with and without ethanol injection under the fixed test conditions. Although ethanol injection prolonged the ignition delay due to the higher latent heat of vaporization and lower cetane number [11,12], exhaust manifold injection reduced these effects. This is a result of the vaporization of injected ethanol prior to entering the combustion chamber. Therefore, exhaust manifold injection of ethanol has the advantage of short ignition delay combined with a higher combustion rate in the premixed phase.

It was also seen that combustion ends earlier when ethanol is injected. The combustion duration is 13.6° CA for conventional diesel combustion, but decreases to 11.9° CA for EGR25. The combustion duration decreases with ethanol injection to the following timespans: 9.8° for 1.2 mg, 10° for 3.8 mg, 9.7° for 4.8 mg, 10.2° for 11 mg, and 10.3° for 22 mg.

The IMEP and total heat release per cycle are shown in Fig. 13a. Increasing injected ethanol leads to an increase in the total heat added per cycle and consequently increases the heat released and IMEP. When compared to EGR25 without ethanol injection, the IMEP increased by 3%, 5%, 4.5%, 6.7% and 8.2% respective to injected ethanol volumes listed above. Similarly, the total heat released increased by 10%, 13.5%, 11.5%, 14.2% and 14% for these ethanol injection volumes. Fig. 13b shows that the indicated thermal efficiency is reduced as ethanol injection timing increases. Similarly, the bsfc is increased with increasing the ethanol injection volumes.

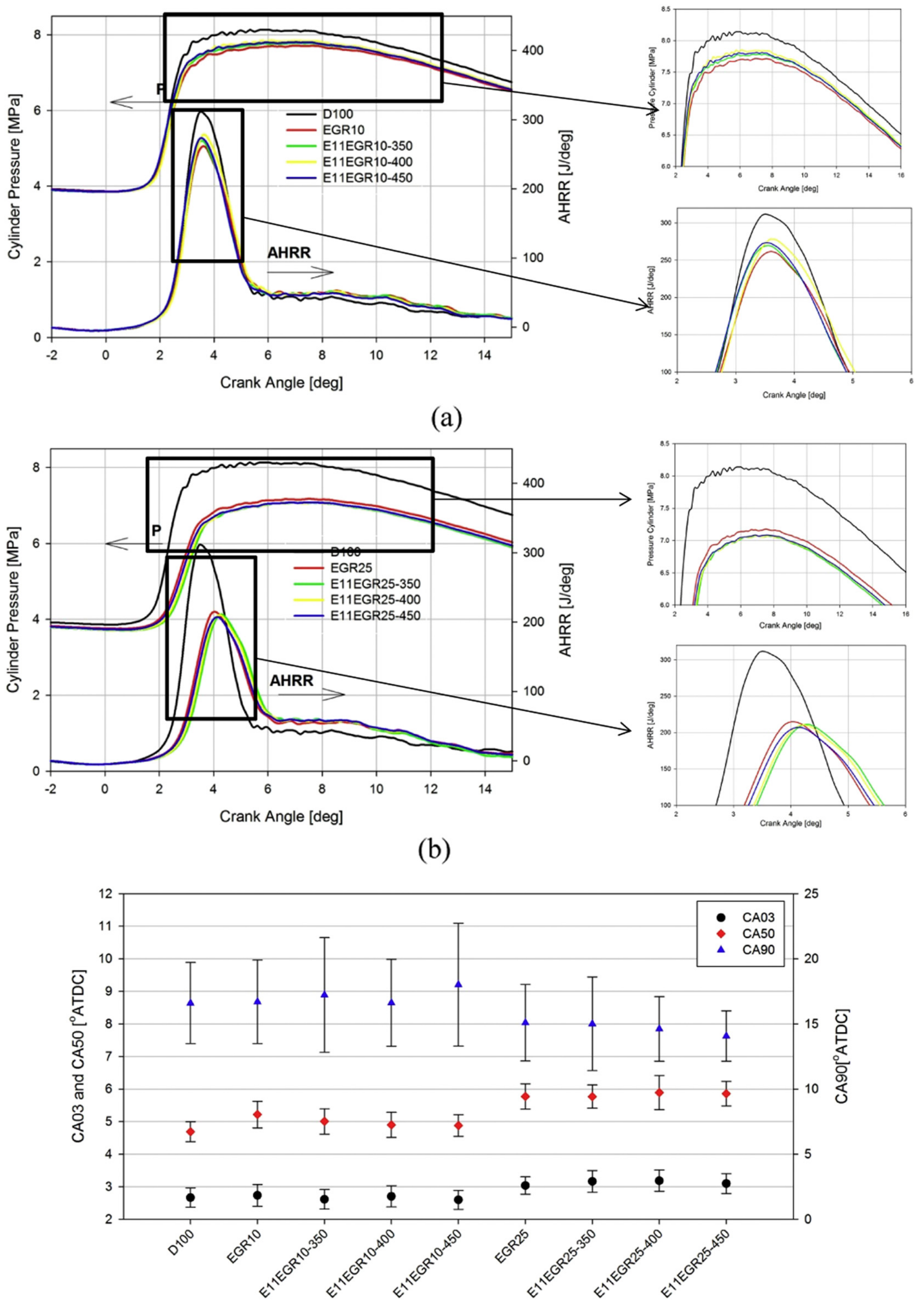


Fig. 8. Effect of various ethanol injection timing and EGR ratios on: (a) Cylinder pressure and AHRR at 10% EGR. (b) Cylinder pressure and AHRR at 25% EGR. (c) Combustion phasing CA03, CA50, and CA90.

The soot concentration increased dramatically for EGR25 without ethanol injection, as shown in Fig.14. Experimental values showed that soot concentration decreased as the volume of ethanol injected increased. Soot decreased by 5% for 1.2 mg injected ethanol, 14% for 3.8 mg, 13% for 4.8 mg, 29% for 11 mg, and 27% for 22 mg when compared to EGR25 without ethanol injection. Increasing of ethanol injection volumes also leads to an increase of the OH radicals, promoting soot oxidation [46].

The generated OH radicals in the flame region react immediately with soot particles, and burning progresses at the soot surface;

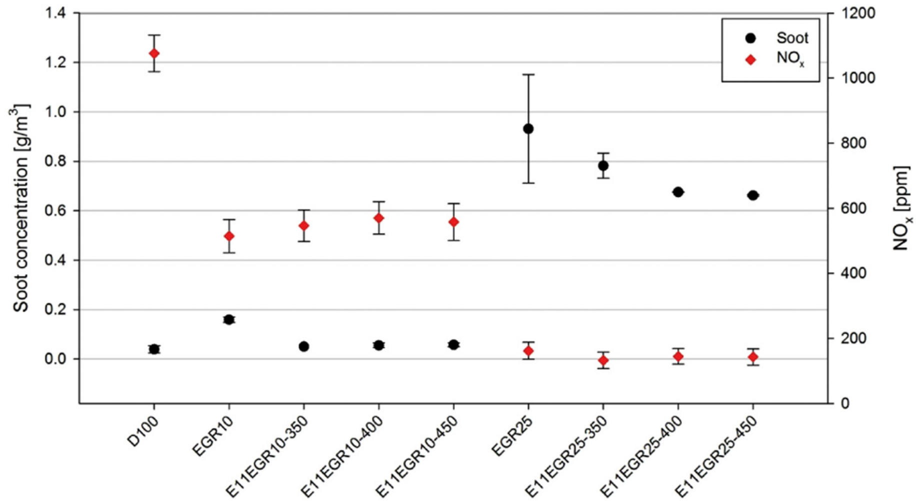
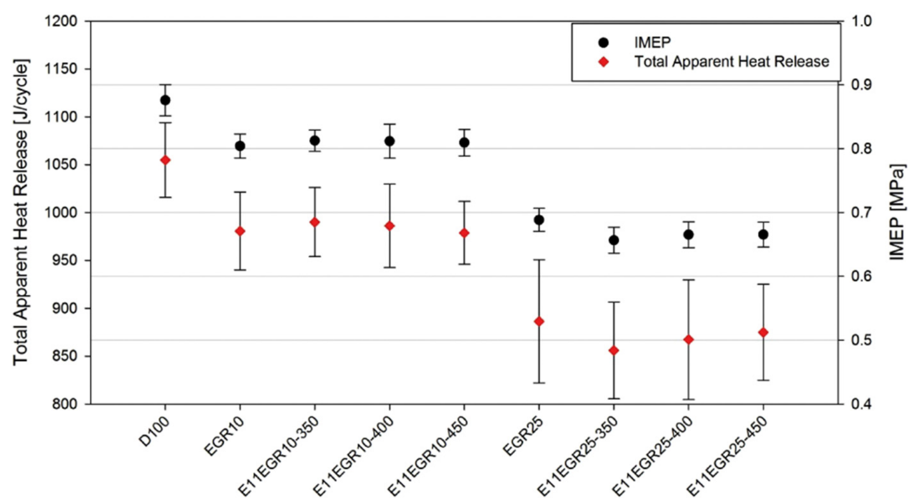
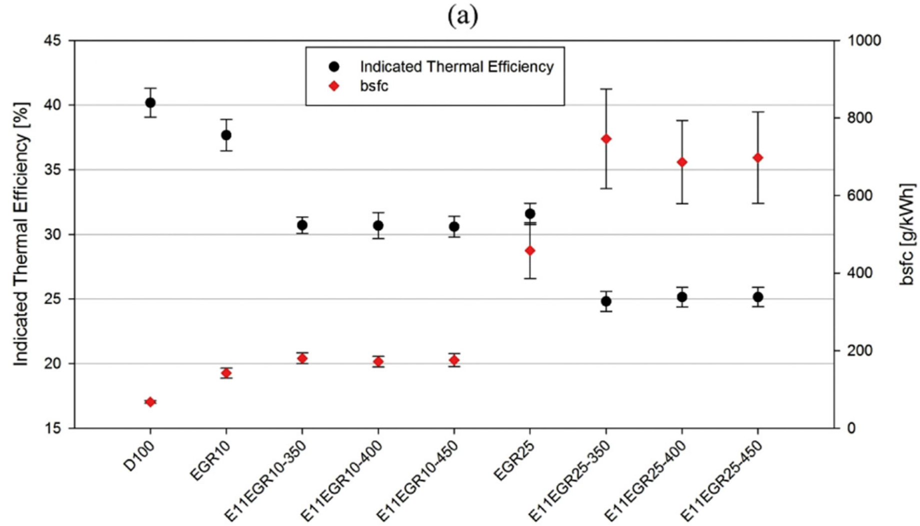


Fig. 9. Effect of various ethanol injection timings and EGR ratios on NO_x and soot emissions.

consequently, soot particles and OH radicals cannot coexist [46]. Conversely, NO_x emissions increased with ethanol injection relative to EGR25 without ethanol injection. This is due to a higher AHRR during the premixed combustion phase that results in higher combustion temperatures. NO_x emissions increased slightly as injection volumes increased, with values 7%, 10%, 7.5%, 12% and 14% higher for the



(a)



(b)

Fig. 10. Effects of ethanol injection timing and EGR ratio on: (a) IMEP and total heat release (b) indicated thermal efficiency and bsfc.

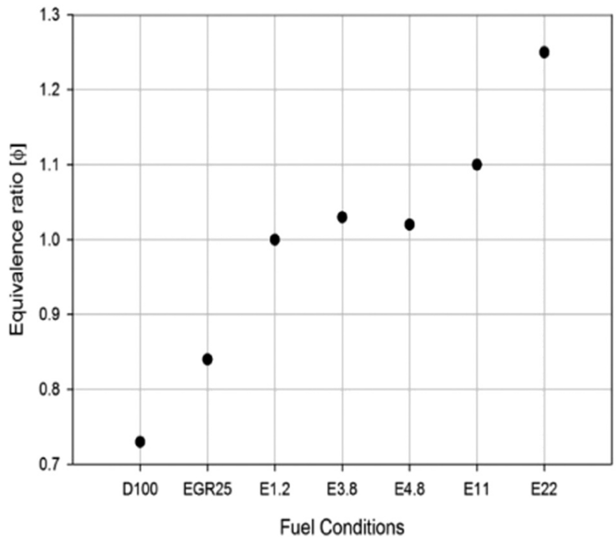


Fig. 11. Equivalence ratio for the different fuel conditions.

injection volumes that have been listed relative to the baseline without injection.

It can thus be stated that ethanol injection improves soot emissions compared to EGR without ethanol injection, but does not affect the NO_x emission significantly. As a tradeoff, injecting large amounts of ethanol reduces the indicated thermal efficiency of the engine and increases brake specific fuel consumption, as shown in Fig. 13b. Therefore, small volumes of ethanol can be recommended for injection to reduce soot without affecting the NO_x emissions, bsfc or indicated thermal efficiency.

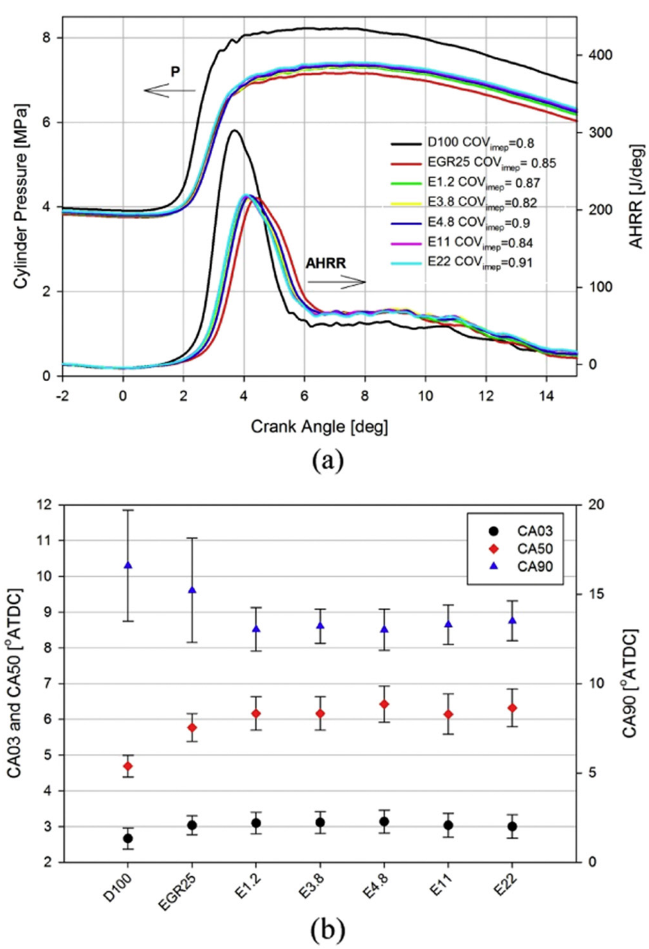


Fig. 12. Effect of different ethanol injection amounts on combustion: (a) Cylinder pressure and AHRR (b) combustion phasing CA03, CA50 and CA90.

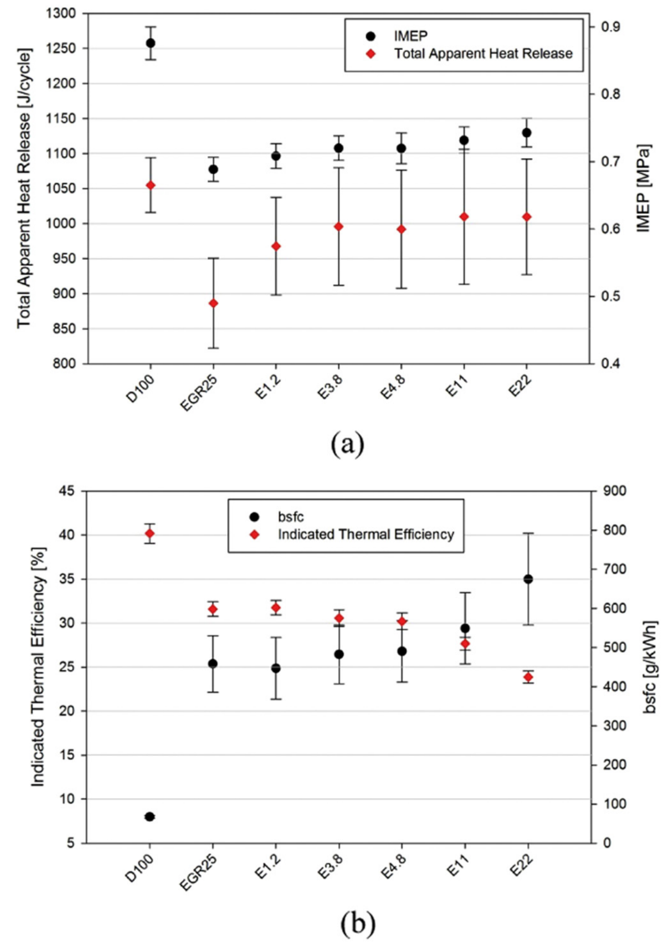


Fig. 13. Effect of different ethanol injection amounts on engine performance: (a) IMEP and total heat release (b) indicated thermal efficiency and bsfc.

Further experiments were conducted under the same fixed conditions to compare ethanol injection into exhaust and intake manifolds.

5.4. Comparison between intake and exhaust manifold ethanol injection

Ethanol injection into the intake manifold was compared with ethanol injection into the exhaust manifold at a fixed EGR ratio of 25%. The

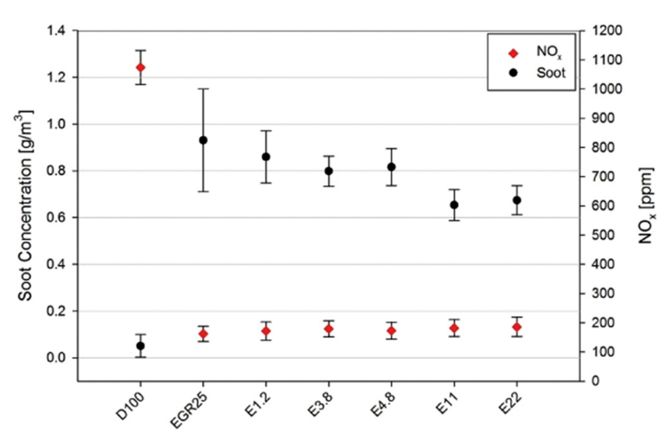


Fig. 14. Effect of various ethanol injection amounts on NO_x and soot emissions.

exhaust port was re-opened with the same value during the intake stroke in both cases. This means the enthalpy of exhaust gases was controlled in both cases, exhaust and intake port injection, to gives approximately the same energy balance in both cases.

The formulation of fuel mixtures is known to play a crucial role in combustion and formation of emissions. Consequently, knowledge of the distribution of fuel, air, and exhaust gases can increase knowledge of the combustion process. The main differences between intake and exhaust manifold injection were the ethanol vaporization and mixture formation.

In the case of exhaust manifold injection, the temperature of exhaust gases was 350 °C, allowing complete vaporization of ethanol (the reported ethanol boiling point is 79 °C) [47]. The mixture of the vaporized ethanol and exhaust gases was introduced inside the engine cylinder with a lake of oxygen, creating a rich fuel condition that gave longer ignition delays and burned in the late combustion phase.

Contrastingly, the upstream injection of ethanol into the intake manifold at 65 °C leaves ethanol in the liquid phase when it is introduced into the cylinder, where mixing with air that increases the tendency of the fuel blend to burn under premixed conditions with a shorter ignition delay. Following these observations, combustion analysis showed that ethanol injection in the intake manifold achieves a higher peak cylinder pressure compared to exhaust manifold injection, shown in Fig. 15a and b.

The AHRR in the premixed combustion phase for intake manifold injection is also higher than that of exhaust manifold injection, illustrated in Fig. 16a and b, which agrees with results available in the literature [12]. Exhaust manifold injection, however, achieves a higher AHRR in the mixing control combustion phase compared to intake manifold injection. Additionally, shorter ignition delay for the intake manifold injection relative to exhaust manifold injection indicates rapid burning of the ethanol-air mixture in the former case, as reported in Fig. 16c. The method of mixture formation between ethanol, exhaust gases, and air is the main factor affecting the start (CA03) and duration of combustion.

In the case of ethanol injection into the exhaust manifold, the exhaust manifold temperature ranges from 300 to 350 °C, and injected ethanol was evaporated and mixed with exhaust gases. During the intake stroke, both intake and exhaust valves open by 8 mm and 4 mm, respectively. The well-mixed charge of evaporated ethanol and exhaust gases is introduced from the exhaust port while supercharged air is introduced from the intake port. The two start to mix in the cylinder. Mixing between air and ethanol, in this case, is insufficient and may take longer to start the combustion process. It is possible this is the cause of longer ignition delays (higher CA03) seen in exhaust manifold injection.

For intake manifold injection of ethanol, the intake manifold temperature was controlled at 65 ± 2 °C. The ethanol was injected prior to intake valve opening, giving enough mixing time for ethanol and supercharged air to form a homogenous mixture in the intake manifold. During the intake stroke, this mixture is introduced from the intake port while hot exhaust gases are introduced from the exhaust port. Thus, the mixing between ethanol and air in intake manifold injection is sufficient for the final mixture to be prepared to ignite upon valve opening, and consequently, a lower CA03 is observed.

The duration from CA03 to CA50 is longer for exhaust compared to intake manifold injection, as shown in Fig. 16c. This reflects the lower burning rate for exhaust manifold injection in the premixed combustion phase. The combustion duration from CA03 to CA90 was 35.1° for 11 mg/cycle and 26.8° for 22 mg/cycle for intake manifold injection, whereas exhaust manifold injection showed durations of 10.1° for 11 mg/cycle and 10.3° for 22 mg/cycle. The main reason for the shorter combustion duration for exhaust manifold injection is the higher AHRR in the combustion control region.

The IMEP and total heat released per cycle are shown in Fig. 17a. Ethanol injection into the exhaust manifold gives a slightly higher IMEP as compared to intake manifold injection. The total heat released for exhaust manifold ethanol injection was greater than that of intake manifold injection by 11% for 11 mg/cycle and 8% for 22 mg/cycle ethanol injection. This is because ethanol injected into the intake manifold is vaporized by the generated heat in-cylinder, leading to a lower measured heat release per cycle. The bsfc for ethanol injection into exhaust manifold was lower than that of intake manifold injection, as shown in Fig. 17b. At the same time, the indicated thermal efficiency was higher with ethanol injection into the exhaust manifold relative to intake injection.

As for exhaust emissions, Fig. 18 shows the soot and NO_x concentrations associated with ethanol injection into the intake and exhaust manifolds. The emission characteristics are seen to improve with exhaust manifold ethanol injection. Comparing exhaust manifold lowered soot emissions by, respectively, 24% and 53% with injection amounts of 11 mg/cycle and 22 mg/cycle relative to intake manifold injection under the same conditions. The longer ignition delays allow more premixing which results in lower soot concentrations [47].

However, mixture formation also plays a crucial role in soot formation [47]. Burning ethanol in the late combustion phase with exhaust manifold injection enhanced the soot oxidation and decreased soot concentrations compared to ethanol burning in the premixed combustion phase, where it occurs in intake manifold injection. As previously discussed, ethanol burning in the late combustion phase results in increased OH radical concentrations, leading to the decrease in soot [46].

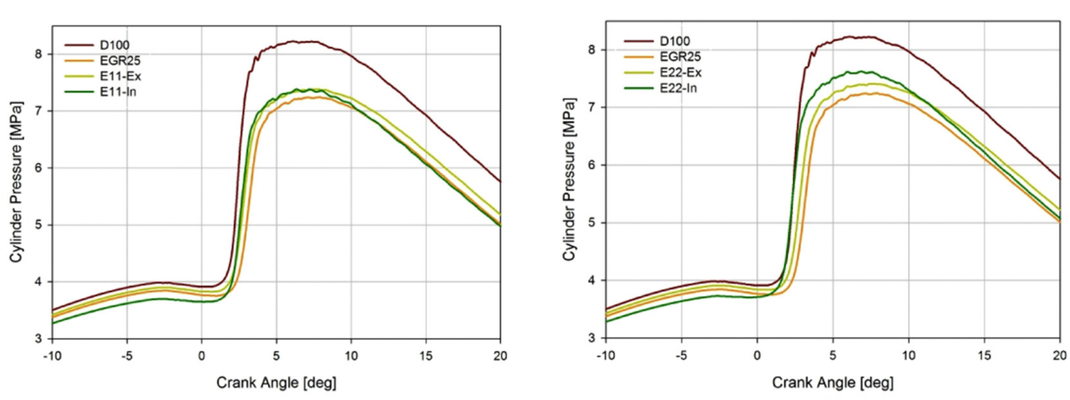


Fig. 15. Comparison of measured cylinder pressure between ethanol injection into intake and exhaust manifolds at different ethanol injection amounts: (a) 11 mg injection (b) 22 mg injection.

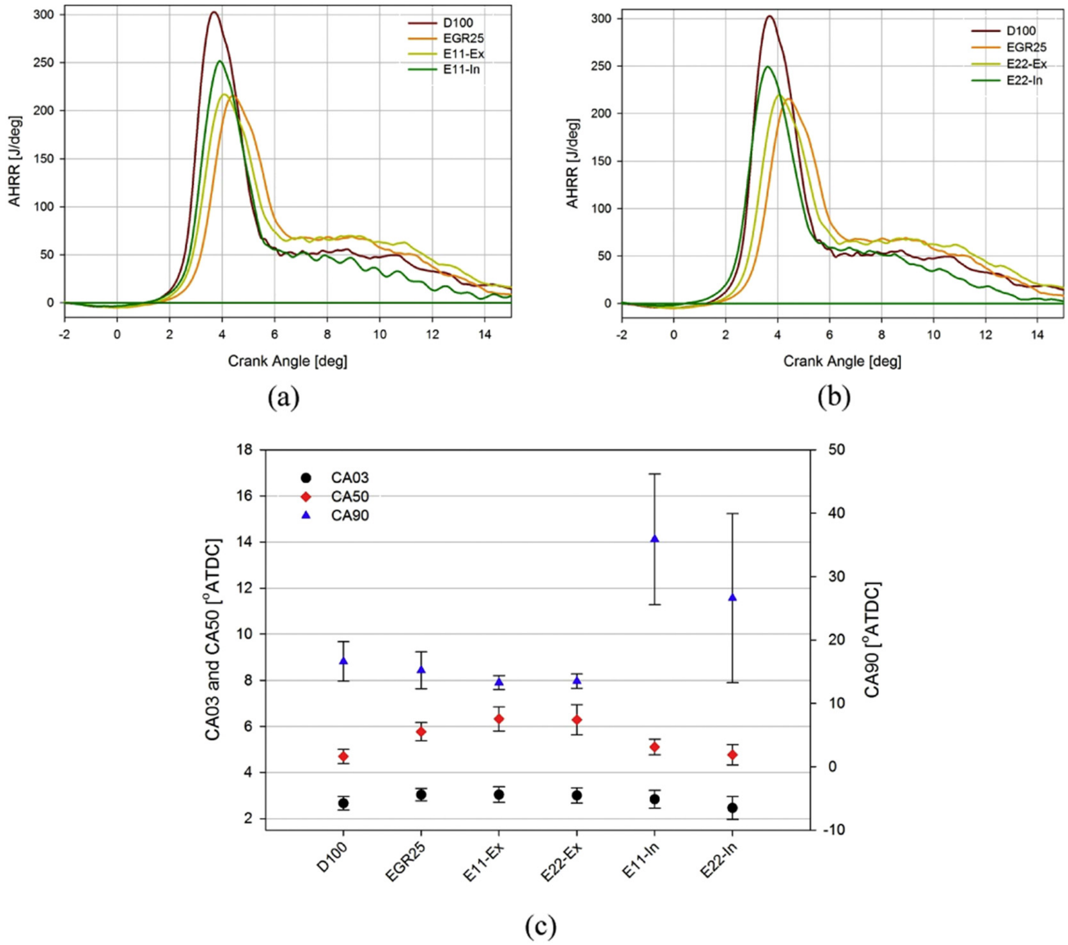


Fig. 16. Comparison of AHRR and combustion phasing between ethanol injection into intake and exhaust manifold at different ethanol injection amounts: (a) 11 mg, (b) 22 mg, and (c) combustion phasing CA03, CA50, and CA90.

The NO_x emissions for intake manifold injection were slightly lower than that of exhaust manifold injection by 7.5% for 11 mg/cycle and 13% for 22 mg/cycle. This decrease results from the lower combustion temperature of intake manifold injection due to in-cylinder heat release that vaporizes the injected ethanol.

As an overall evaluation for the ethanol injection into the intake and exhaust manifolds, it can be said that the engine combustion characteristics, performance, and exhaust emissions were improved significantly with ethanol injection into exhaust manifold relative to intake manifold injection.

From the aforementioned discussion on combustion characteristics of both intake and exhaust manifold injection, it can be summarized that exhaust manifold injection eliminates the ethanol endothermic effect and decreases the engine heat losses. Furthermore, exhaust manifold injection improves combustion timing while shortening the combustion duration. For engine performance and emissions, the comparison between ethanol injection into the intake and exhaust manifolds is summarized in Fig. 19. Here, the percentage of reduction or increase is shown in comparison to EGR25 without ethanol injection. The highest reduction in soot emissions was achieved using exhaust manifold injection. Intake manifold injection was seen to increase soot emissions in the case of E22In. The increase in NO_x emissions was higher for exhaust manifold injection relative to intake manifold injection. However, the NO_x emission level remained below 200 ppm, which is lower than conventional diesel combustion by 80%. Both intake and exhaust manifold injection techniques are indicated to have the same thermal efficiency reduction percentage. The total apparent heat release is higher in exhaust manifold injection than intake manifold injection due to the elimination of the ethanol cooling effect. Additionally, the bsfc is improved in exhaust manifold injection.

Several practical questions arise when dealing with ethanol injection into the exhaust manifold. These include the effects of the hot valve on early ignition of ethanol and the high temperature of the exhaust manifold on the injector operation, as well as the possibility of ethanol evaporation inside the ethanol injector. The autoignition temperature of ethanol is equal to 366 °C, as reported by Hansen et al. [15]. The exhaust manifold temperatures monitored during experiment ranged from 300 to 350 °C. Additionally, the oxygen concentration in the exhaust manifold was very low, unlikely to form a mixture with ethanol that can easily autoignite. Given these conditions, early ignition of ethanol resulting from the heated exhaust valve was not observed. Additionally, the exhaust valve profile was monitored during all performed experiments, and operated smoothly and without variation. The potential effects of the exhaust manifold temperature, including valves, on the early ignition of ethanol requires further examination and study. The additional possibility of ethanol evaporation inside the injector requires further investigation as well.

Aleiferis et al. [48] studied the effect of fuel temperatures (ranging between 20 and 90 °C) and pressure (0.5 and 1 bar) on the in-nozzle cavitation and spray formation of ethanol by using a real-size optical injector in direct-injection spark-ignition engines. They concluded that increasing the fuel temperature from 20 °C to 50 °C did not cause a notable difference in spray formation or in-nozzle cavitation. At the peak ethanol temperature and pressure, 90 °C and 1.0 bar, they cited that deformation in the nozzle occurred due to the reaction of ethanol with the optical material, resulting in a reduction of the cone angle such that

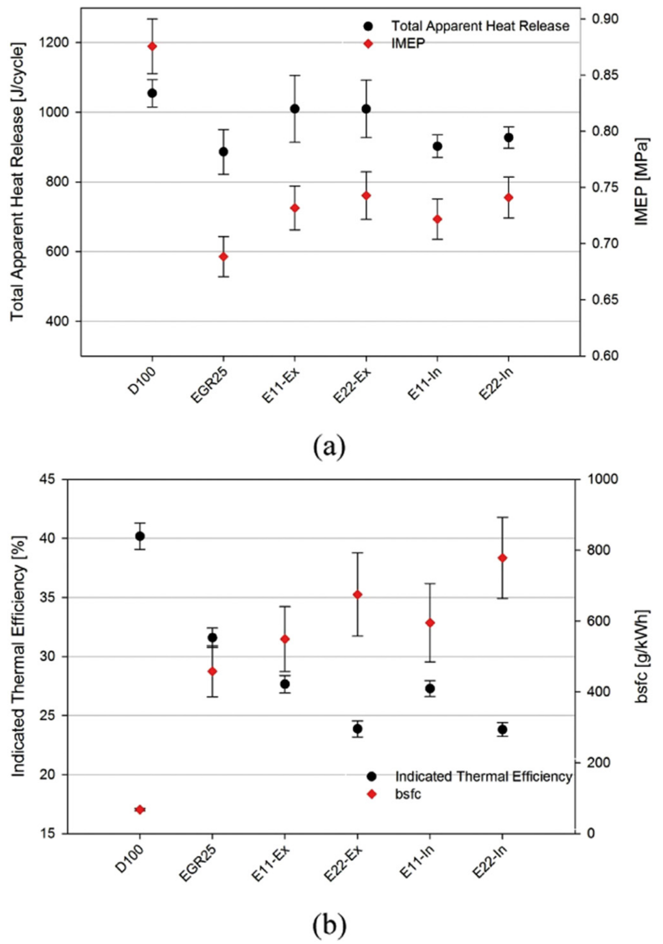


Fig. 17. Comparisons between ethanol injections into the intake and exhaust manifolds at different ethanol injection volumes regarding: (a) IMEP and total heat release (b) indicated thermal efficiency and bsfc.

ethanol exhibited a more compact spray with increased cavitation. They added that at 90 °C, ethanol atomization was greatly improved for pressures of 0.5 and 1.0 bar.

The second practical question that has arisen is whether it can be confirmed that 100% of ethanol amounts injected into the exhaust are sucked into the engine cylinder. The experimental results of this study were instead focused on showing the effects of different ethanol injection amounts on combustion. Variations in engine performance and combustion were detected with the change in the quantity of ethanol

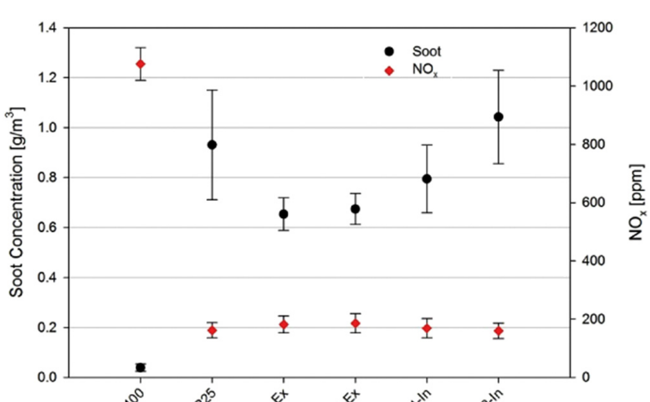


Fig. 18. Comparison between ethanol injection into intake and exhaust manifold regarding the emitted NO_x and soot concentrations.

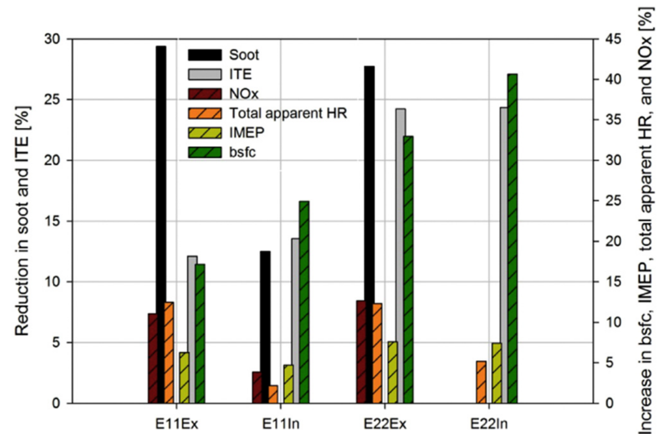


Fig. 19. Comparison between intake and exhaust manifold injection at 11 mg and 22 mg ethanol injection showing the reduction in soot and ITE, and the increase in bsfc, IMEP, total heat release, and NO_x.

injected, thereby confirming that ethanol injected into exhaust manifold is at least in some part sucked inside the cylinder. However, further investigation will be required to accurately identify the amount that is introduced in-cylinder comparative to any amounts that escape in the exhaust system. Additionally, further studies are needed to investigate the process of exhaust manifold injection and extend the investigation to include several fuels in various engine operating loads and speed.

6. Conclusions

In the present study, ethanol injection into the exhaust manifold was investigated in dual fuel combustion using a single cylinder diesel engine. Ethanol was injected into exhaust manifold while diesel fuel was directly injected in-cylinder. The purpose of this injection strategy was to utilize the enthalpy of exhaust gases to vaporize ethanol prior to combustion, reducing soot and NO_x emissions without decreasing the combustion temperature.

Experiments were carried out at varying ethanol injection timings, volumes, and EGR ratios. Furthermore, ethanol injection into intake manifold was compared to ethanol injection into the exhaust manifold under otherwise constant experimental conditions. Engine operating conditions such as engine speed, intake temperature, coolant temperature and lubricating oil temperature were controlled at fixed values for all tested conditions. The following conclusions can be drawn from the results:

- Ethanol injection into the exhaust manifold was tested at a fixed ethanol volume, 11 mg/cycle, and selected ethanol injection timings of 350°, 400°, and 450°. These results were compared to EGR without ethanol injection. The peak cylinder pressure increased by up to 2% with ethanol injection, and the highest cylinder pressure was attained at 400°. Ethanol injection also increased the AHRR in the premixed combustion phase, implying the amount of fuel burned under premixed conditions was greater with ethanol injection. The ignition delay increased by 0.2° CA for ethanol injection, a result of ethanol's low cetane number and poor autoignition properties. The emitted NO_x concentrations were reduced by up to 16% at 350° injection timing and 25% EGR. The soot concentration was decreased by 68% for ethanol injection at 10% EGR and 29% at ethanol injection timing of 400° and 25% EGR.
- The effect of ethanol injection volume was tested with multiple injection amounts ranging between 1.2 mg/cycle and 22 mg/cycle. It was found that the more ethanol that was injected, the longer the ignition delay and the higher the AHRR at premixed combustion phase became. Increasing the amount of injected ethanol also increased the total heat release by 14% and IMEP by 8% relative to EGR without

ethanol injection. Soot concentrations were reduced by 29% at an ethanol injection volume of 22 mg/cycle. However, NO_x emissions were increased by 14% under these conditions. It was also found that increasing the amount of the injected ethanol reduces the combustion efficiency. Therefore, only a small quantity of ethanol is recommended for soot reduction without affecting combustion efficiency.

- To compare between the injection of ethanol at the intake and exhaust manifolds, experimental conditions were maintained as equally as possible. The same exhaust valve was opened with the same timing (during intake stroke) for both cases, which means it is reasonable to assume EGR ratio and energy balance were equal for both cases. For exhaust manifold injection, the mixture of the vaporized ethanol and exhaust gases was introduced with a lake of oxygen inside the engine cylinder, leading to a rich fuel condition that gave longer ignition delays and burned during the late combustion phase. Conversely, the upstream injection of ethanol into the intake manifold at 65 °C leaves ethanol in the liquid phase, and sufficiently mixed with air, when it is introduced into the cylinder that the tendency of the mixture to burned under premixed conditions increases with a shorter ignition delay. Ethanol injection into the exhaust manifold gives a higher IMEP compared to intake manifold injection. Additionally, the total heat released for ethanol injection into exhaust manifold was greater than that of intake manifold injection by 11% given the same injection quantity. Exhaust manifold injection lowered soot emissions by up to 53% compared to injection into the intake manifold, but the NO_x emissions for intake manifold injection were lower by up to 13%. Although the waste heat of exhaust gases was used for both cases, only intake manifold injection experienced the standard ethanol cooling effect.

Comparing the ethanol injection strategies used in previous studies, such as intake port or direct injection, exhaust manifold injection is an effective method for enhancing ethanol vaporization prior to entering the combustion chamber and reduces the long ignition delay. This new proposed method also enhances ethanol burning in the late combustion phase, improving the soot oxidation. Additionally, a combination of ethanol injection, high EGR rate, and late fuel injection timing enabled low-temperature combustion in diesel engines. Such techniques can simultaneously reduce soot and NO_x emissions while also improving the engine efficiency.

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References

[1] S.R. Turns, An Introduction to Combustion: Concepts and Applications, second ed. WCB/McGraw-Hill, Boston, Massachusetts, 1999.
[2] R.D. Reitz, G. Duraisamy, Review of high efficiency and clean reactivity controlled compression ignition (RCCI) combustion in internal combustion engines, Prog. Energy Combust. Sci. 46 (2015) 12–71, http://dx.doi.org/10.1016/j.pecs.2014.05.003.
[3] M. Abu-Qudais, O. Haddad, M. Qudaisat, The effect of alcohol fumigation on diesel engine performance and emissions, Energy Convers. Manag. 41 (2000) 389–399 doi:http://dx.doi.org/10.1016/S0196-8904(99)00099-0.
[4] E.E. Ecklund, R.L. Bechtold, T.J. Timbario, P.W. McCallum, State-of-the-art report on the use of alcohols in diesel engines, SAE Technical Paper (No. 840118), 1984http://dx.doi.org/10.4271/840118.
[5] Ö. Can, I. Çelikten, N. Usta, Effects of ethanol addition on performance and emissions of a turbocharged indirect injection diesel engine running at different injection pressures, Energy Convers. Manag. 45 (2004) 2429–2440, http://dx.doi.org/10.1016/j.enconman.2003.11.024.
[6] C. Sayin, M. Canakci, Effects of injection timing on the engine performance and exhaust emissions of a dual-fuel diesel engine, Energy Convers. Manag. 50 (2009) 203–213, http://dx.doi.org/10.1016/j.enconman.2008.06.007.
[7] D.C. Rakopoulos, C.D. Rakopoulos, E.C. Kakaras, E.G. Giakoumis, Effects of ethanol-diesel fuel blends on the performance and exhaust emissions of heavy duty DI diesel engine, Energy Convers. Manag. 49 (2008) 3155–3162, http://dx.doi.org/10.1016/j.enconman.2008.05.023.
[8] J. Huang, Y. Wang, S. Li, A.P. Roskilly, H. Yu, H. Li, Experimental investigation on the performance and emissions of a diesel engine fuelled with ethanol-diesel blends, Appl. Therm. Eng. 29 (2009) 2484–2490, http://dx.doi.org/10.1016/j.applthermaleng.2008.12.016.
[9] A. Murcak, C. Hasimoglu, I. Cevik, H. Kahraman, Effect of injection timing to performance of a diesel engine fuelled with different diesel–ethanol mixtures, Fuel 153 (2015) 569–577, http://dx.doi.org/10.1016/j.fuel.2015.03.028.
[10] I. Ahmed, Oxygenated diesel: emissions and performance characteristics of ethanol-diesel blends in CI engines, SAE Pap. (2001) 1–2475, http://dx.doi.org/10.4271/2001-01-2475.
[11] A. De Oliveira, A.M. De Morais, O.S. Valente, J.R. Sodré, Combustion characteristics, performance and emissions from a diesel power generator fuelled by B7-ethanol blends, Fuel Process. Technol. 139 (2015) 67–72, http://dx.doi.org/10.1016/j.fuproc.2015.08.010.
[12] G.K. Prashant, D.B. Lata, P.C. Joshi, Investigations on the effect of methanol blend on the combustion parameters of dual fuel diesel engine, Appl. Therm. Eng. 103 (2016) 187–194, http://dx.doi.org/10.1016/j.applthermaleng.2016.04.061.
[13] G. Labeckas, S. Slavinskas, Comparative performance of direct injection diesel engine operating on ethanol, petrol and rapeseed oil blends, Energy Convers. Manag. 50 (2009) 792–801, http://dx.doi.org/10.1016/j.enconman.2008.09.029.
[14] G. Labeckas, S. Slavinskas, M. Mažeika, The effect of ethanol-diesel-biodiesel blends on combustion, performance and emissions of a direct injection diesel engine, Energy Convers. Manag. 79 (2014) 698–720, http://dx.doi.org/10.1016/j.enconman.2013.12.064.
[15] A.C. Hansen, Q. Zhang, P.W.L. Lyne, Ethanol-diesel fuel blends - a review, Bioresour. Technol. 96 (2005) 277–285, http://dx.doi.org/10.1016/j.biortech.2004.04.007.
[16] B.S. Chauhan, N. Kumar, S.S. Pal, Y. Du Jun, Experimental studies on fumigation of ethanol in a small capacity diesel engine, Energy 36 (2011) 1030–1038, http://dx.doi.org/10.1016/j.energy.2010.12.005.
[17] A. Imran, M. Varman, H.H. Masjuki, M.A. Kalam, Review on alcohol fumigation on diesel engine: a viable alternative dual fuel technology for satisfactory engine performance and reduction of environment concerning emission, Renew. Sust. Energ. Rev. 26 (2013) 739–751, http://dx.doi.org/10.1016/j.rser.2013.05.070.
[18] K.S. Tsang, Z.H. Zhang, C.S. Cheung, T.L. Chan, Reducing emissions of a diesel engine using fumigation ethanol and a diesel oxidation catalyst, Energy Fuel 24 (2010) 6156–6165, http://dx.doi.org/10.1021/ef100899z.
[19] A.F. López, M. Cadrazco, A.F. Agudelo, L.A. Corredor, J.A. Vélez, J.R. Agudelo, Impact of n-butanol and hydrous ethanol fumigation on the performance and pollutant emissions of an automotive diesel engine, Fuel 153 (2015) 483–491, http://dx.doi.org/10.1016/j.fuel.2015.03.022.
[20] Z. Şahin, O. Durgun, M. Kurt, Experimental investigation of improving diesel combustion and engine performance by ethanol fumigation-heat release and flammability analysis, Energy Convers. Manag. 89 (2015) 175–187, http://dx.doi.org/10.1016/j.enconman.2014.09.053.
[21] W. Tutak, Bioethanol E85 as a fuel for dual fuel diesel engine, Energy Convers. Manag. 86 (2014) 39–48, http://dx.doi.org/10.1016/j.enconman.2014.05.016.
[22] T. Sarjovaara, M. Larmi, V. Vuorinen, Effect of charge air temperature on E85 dual-fuel diesel combustion, Fuel 153 (2015) 6–12, http://dx.doi.org/10.1016/j.fuel.2015.02.096.
[23] R.F. Britto, C.A. Martins, Emission analysis of a diesel engine operating in diesel-ethanol dual-fuel mode, Fuel 148 (2015) 191–201, http://dx.doi.org/10.1016/j.fuel.2015.01.008.
[24] S. Padala, C. Woo, S. Kook, E.R. Hawkes, Ethanol utilisation in a diesel engine using dual-fuelling technology, Fuel 109 (2013) 597–607, http://dx.doi.org/10.1016/j.fuel.2013.03.049.
[25] T. Sarjovaara, J. Alantie, M. Larmi, Ethanol dual-fuel combustion concept on heavy duty engine, Energy 63 (2013) 76–85, http://dx.doi.org/10.1016/j.energy.2013.10.053.
[26] W. Pan, C. Yao, G. Han, H. Wei, Q. Wang, The impact of intake air temperature on performance and exhaust emissions of a diesel methanol dual fuel engine, Fuel 162 (2015) 101–110, http://dx.doi.org/10.1016/j.fuel.2015.08.073.
[27] W. Tutak, K. Lukács, S. Szwaja, Á. Bereczky, Alcohol-diesel fuel combustion in the compression ignition engine, Fuel 154 (2015) 196–206, http://dx.doi.org/10.1016/j.fuel.2015.03.071.
[28] V.B. Pedrozo, I. May, M. Dalla Nora, A. Cairns, H. Zhao, Experimental analysis of ethanol dual-fuel combustion in a heavy-duty diesel engine: an optimisation at low load, Appl. Energy 165 (2016) 166–182, http://dx.doi.org/10.1016/j.apenergy.2015.12.052.
[29] R. Hanson, S. Kokjohn, D. Splitter, R.D. Reitz, Fuel effects on reactivity controlled compression ignition (RCCI) combustion at low load, SAE Int. J. Engines 4 (2011) 394–411, http://dx.doi.org/10.4271/2011-01-0361.
[30] D. Splitter, R. Hanson, S. Kokjohn, R.D. Reitz, Reactivity controlled compression ignition (RCCI) heavy-duty engine operation at mid-and high-loads with conventional and alternative fuels. No. 2011-01-0363, SAE Technical Paper, 2011http://dx.doi.org/10.4271/2011-01-0363.
[31] D.Z. Zhou, W.M. Yang, H. An, J. Li, C. Shu, A numerical study on RCCI engine fueled by biodiesel/methanol, Energy Convers. Manag. 89 (2015) 798–807, http://dx.doi.org/10.1016/j.enconman.2014.10.054.
[32] R.D. Reitz, R.M. Hanson, D.A. Splitter, S.L. Kokjohn, Engine combustion control via fuel reactivity stratification. U.S. Patent 8616177 B2, issued December 31, 2013.
[33] B.R. Kumar, S. Saravanan, Use of higher alcohol biofuels in diesel engines: a review, Renew. Sust. Energ. Rev. 60 (2016) 84–115, http://dx.doi.org/10.1016/j.rser.2016.01.085.

[34] S.H. Park, I.M. Youn, C.S. Lee, Influence of two-stage injection and exhaust gas recirculation on the emissions reduction in an ethanol-blended diesel-fueled four-cylinder diesel engine, Fuel Process. Technol. 91 (2010) 1753–1760, http://dx.doi.org/10.1016/j.fuproc.2010.07.016.
[35] W. Tutak, A. Jamrozik, M. Pyrc, M. Sobiepanski, Investigation on combustion process and emissions characteristic in direct injection diesel engine powered by wet ethanol using blend mode, Fuel Process. Technol. 149 (2016) 86–95, http://dx.doi.org/10.1016/j.fuproc.2016.04.009.
[36] O. Armas, R. García-Contreras, Á. Ramos, Pollutant emissions from New European Driving Cycle with ethanol and butanol diesel blends, Fuel Process. Technol. 122 (2014) 64–71, http://dx.doi.org/10.1016/j.fuproc.2014.01.023.
[37] R. Munsin, Y. Laoonual, S. Jugjai, M. Matsuki, H. Kosaka, Effect of glycerol ethoxylate as an ignition improver on injection and combustion characteristics of hydrous ethanol under CI engine condition, Energy Convers. Manag. 98 (2015) 282–289, http://dx.doi.org/10.1016/j.enconman.2015.03.116.
[38] J.B. Heywood, Internal Combustion Engine Fundamentals, Mcgraw-Hill, New York, 1988.
[39] B.M. Masum, H.H. Masjuki, M.A. Kalam, I.M. Rizwanul Fattah, S.M. Palash, M.J. Abedin, Effect of ethanol-gasoline blend on NOx emission in SI engine, Renew. Sust. Energ. Rev. 24 (2013) 209–222, http://dx.doi.org/10.1016/j.rser.2013.03.046.
[40] D.K. Jamuwa, D. Sharma, S.L. Soni, Experimental investigation of performance, exhaust emission and combustion parameters of stationary compression ignition engine using ethanol fumigation in dual fuel mode, Energy Convers. Manag. 115 (2016) 221–231, http://dx.doi.org/10.1016/j.enconman.2016.02.055.
[41] A.B. Dempsey, N.R. Walker, E. Gingrich, R.D. Reitz, Comparison of low temperature combustion strategies for advanced compression ignition engines with a focus on controllability, Combust. Sci. Technol. 186 (2014) 210–241, http://dx.doi.org/10.1080/00102202.2013.858137.
[42] J. Liu, A. Yao, C. Yao, Effects of diesel injection pressure on the performance and emissions of a HD common-rail diesel engine fueled with diesel/methanol dual fuel, Fuel 140 (2015) 192–200, http://dx.doi.org/10.1016/j.fuel.2014.09.109.
[43] S. Padala, S. Kook, E.R. Hawkes, Effect of ethanol port-fuel-injector position on dual-fuel combustion in an automotive-size diesel engine, Energy Fuel 28 (2014) 340–348, http://dx.doi.org/10.1021/ef401479s.
[44] S.J. Kline, F.A. McClintock, Describing uncertainties in single-sample experiments, Mech. Eng. 75 (1953) 3–8.
[45] J.P. Holman, W.J. Gajda, Experimental Methods for Engineers, McGraw-Hill, New York, 1994.
[46] H. Kosaka, T. Nishigaki, T. Kamimoto, T. Sano, A. Matsutani, S. Harada, Simultaneous 2-D imaging of OH radicals and soot in a diesel flame by laser sheet techniques. No. 960834, SAE Technical Paper, 1996http://dx.doi.org/10.4271/960834.
[47] M.K. Bobba, C.L. Genzale, M.P.B. Musculus, Effect of ignition delay on in-cylinder soot characteristics of a heavy duty diesel engine operating at low temperature conditions, SAE Int. J. Engines 2 (2009)http://dx.doi.org/10.4271/2009-01-0946.
[48] P.G. Aleiferis, J. Serras-Pereira, A. Augoye, T.J. Davies, R.F. Cracknell, D. Richardson, Effect of fuel temperature on in-nozzle cavitation and spray formation of liquid hydrocarbons and alcohols from a real-size optical injector for direct-injection spark-ignition engines, Int. J. Heat Mass Transf. 53 (2010) 4588–4606, http://dx.doi.org/10.1016/j.ijheatmasstransfer.2010.06.033.