Effects of intake and exhaust manifold water injection on combustion and emission characteristics of a DI diesel engine

Mohamed FARAG*, Hidenori KOSAKA**,
Mahmoud BADY***, **** and Ali K. ABDEL-RAHMAN*, ***

* Energy Resources Engineering Department, Egypt - Japan University of Science and Technology E-JUST, Alexandria 21934, Egypt
E-mail: mohamed.farag@ejust.edu.eg
** Department of Mechanical and Aerospace Engineering, Tokyo Institute of Technology, Tokyo, Japan
*** Mechanical Engineering Department, Assiut University, Assiut, Egypt
**** Mechanical Engineering Department, Islamic University of Medina, Medina, Saudi Arabia

Abstract

Water injection is a technique that has been used for decades to control the combustion and emissions in diesel engines. The effects of water injection at the intake and exhaust manifolds on the combustion and emission characteristics of a direct injection diesel engine are studied in this work. Water injection in the intake manifold increases engine heat losses. Therefore, waste heat in exhaust gases is used to vaporize water before combustion and prevent the water-cooling effect. The injection of 40 mg/cycle of water into the intake and exhaust manifolds were tested with exhaust gas recirculation (EGR) ratios of 10% and 25%. The fuel injection timing, quantity, and engine speed were maintained at constant values to keep the cylinder combustion condition constant. The results show that the exhaust manifold water injection improves engine performance and combustion characteristics and reduces emissions compared to intake manifold water injection. The peak improvement is achieved at exhaust manifold water injection at 25% EGR where the reduction in the brake specific fuel consumption (bsfc) is about 5%, while the increase in the indicated mean effective pressure (IMEP) and the indicated thermal efficiency (ITE) is by 7% and 3%, respectively. On the other hand, the maximum reduction in soot was obtained with exhaust manifold water injection at 10% EGR ratio with reduction ratios of 55%. The intake manifold water injection gives the lowest NO_x emissions with a 88% reduction ratio. The exhaust manifold injection is the recommended technique for water injection to improve engine performance and reduce emissions to avoid the disadvantages of the previously applied techniques.

Key words: Exhaust manifold injection, Water injection, Diesel engine, Waste heat recovery, Heat release rate, Engine performance, Emission characteristics

1. Introduction

Diesel engines play a crucial role in energy economy as they are extensively used in transportation, electrical power generators, and agricultural machines. However, diesel engines contribute to air pollution significantly because of their high level of soot and NO_x emissions. Therefore, development of new combustion strategies is essential to improve fuel combustion efficiency and reduce soot and NO_x emissions. Such improvements can be achieved through preventing emissions formation at the source to reduce the dependence on after-treatment systems. However, soot and NO_x formation in diesel engines have a tradeoff relationship, and the simultaneous reduction for both is difficult. Therefore, water injection into diesel engine is a technique used for in-cylinder combustion control since it is an efficient way for a concomitant reduction in soot and NO_x emissions (Greeves et al., 1977). One effect of water injection on diesel engines is cooling of the intake charge and the flame temperature which is known as the thermal effect. This effect is attributed to the high latent heat of vaporization and specific heat capacity of the injected water, which result in lower flame temperature. Since the formation reaction of NO_x has a high activation energy as indicated

by Turns (1999), the low flame temperature leads to a reduction in NO_x emissions (Armas et al., 2005, Ghojel et al., 2006, Alahmer et al., 2010, Subramanian, 2011, Ma et al., 2014). The other effect of water injection is the cooling of the intake charge. Consequently, the density of in-cylinder air is increased. Then, the total injected mass is increased at the same volume followed by sudden expansion due to water vaporizing within the fuel droplets during the droplet heating period, and this is known as dilution effect (Greeves et al., 1977, Tarlet et al., 2009, Maiboom and Tauzia, 2011). This effect enhances mixing of fuel and air before the start of combustion. Therefore, the fuel rich regions are diminished, and soot formation is reduced (Maiboom and Tauzia, 2011). However, both dilution effect and thermal effect are related as water vapor may terminate the chemical reaction in the gas phase due to the reduced apparent heat release rate (AHRR). The suppression of the chemical reaction may also cause a reduction in flame temperature and consequently a reduction of thermal NO_x emissions (Kadota and Yamasaki, 2002). On the other hand, water injection increases OH radicals that contribute significantly in soot oxidation and reduce the soot emissions, and this is known as the chemical effect (Greeves et al., 1977, Kadota and Yamasaki, 2002).

Water is utilized in the diesel engines either through intake manifold injection or direct water injection into the engine cylinder (Miyamoto et al., 1995, Bedford et al., 2000, Udayakumar et al., 2003, Armas et al., 2005, Tauzia et al., 2010, Subramanian, 2011, Tesfa et al., 2012, Kumar et al., 2013, Ithnin et al., 2014, Mingrui et al., 2016). The direct water injection into the engine cylinder includes injection through separate water injection pump and nozzle, injection through the fuel injector, or water and diesel fuel emulsion. The diesel fuel and water emulsion studies concluded that the emulsion technique reduces flame temperature (Abu-Zaid, 2004, Ithnin et al., 2014). The reduction of the flame temperature results in longer ignition delay and higher engine noise. Additionally, water/diesel emulsion has a disadvantage that the water/fuel blend is constant over a wide range of engine operating conditions and different engine loads may require different blend ratios (Abu-Zaid, 2004). Therefore, water injection in the intake manifold and direct water injection techniques can overcome this problem (Bedford et al., 2000). However, most studies of water injection in the intake manifold confirmed that water exists in areas where it is less efficient in reducing emissions in the combustion chamber. Therefore, intake manifold water injection requires approximately twice the liquid volume for the same reduction in NO_x compared to direct water injection or water/fuel emulsion (Bedford et al., 2000). Tauzia et al. (Tauzia et al., 2010) cited that water to fuel mass fraction of 60-65% is needed to obtain a 50% NO_x reduction. They also mentioned that increase in the injected water quantity leads to a decrease in flame temperature and heat release rate together with a drop in the cylinder pressure and burning rate combined with an increase in the combustion duration. Most studies of water injection into the intake manifold confirmed reduced in NO_x accompanied by a slight growth in PM emissions with an increase in CO and HC emissions (Odaka et al., 1991, Nazha et al., 2001). Therefore, direct in-cylinder water injection through a separate water injection pump and nozzle or injection through the fuel injector has the advantage of changing the water/fuel ratio with engine speed and load. Additionally, the injected water exists in areas where it is more efficient to reduce emissions. However, the main disadvantage of this technique is that an additional water injector should be installed to the engine cylinder and more advanced control system should be used (Bedford et al., 2000, Nishijima et al., 2002).

The previously studied injection techniques have a limitation of reduction of the in-cylinder temperature and heat release rate that occurs due to the charge cooling effect and combustion quenching. The lower in-cylinder temperature leads to a decrease in the thermal NO_x, but a simultaneous increase in CO and HC emissions is reported. Tauzia et al. (Tauzia et al., 2010) cited that the cooling effect of water injection is only responsible for around 30% of NO_x reduction and the other 70% is attributed to other effects of water injection, such as dilution. The evaporation of the injected water before entering the engine cylinder is a proposed way to reduce the NO_x emissions without affecting the combustion temperature and engine efficiency.

The primary objective of this study is to evaluate diesel engine combustion, performance and emissions reduction using water injection into the exhaust manifold. By injecting water into the exhaust manifold, the waste heat of exhaust gases is utilized to vaporize the injected water. Then, the variable valve actuating system controls the opening of exhaust valve during the intake stroke to bring the evaporated water inside engine cylinder with some of the exhaust gases and waste heat. Therefore, the effect of water on charge cooling will be eliminated through this study. Also, the intake and exhaust manifolds injection are compared at the same combustion conditions.

2. Experimental setup

This study was carried out using a Nissan single cylinder direct injection water-cooled diesel engine. The engine technical specifications are summarized in Table 1. The experimental layout, the measurements and installed instrumentation are shown in Fig. 1(a). The variable valve actuating system, water injection control system and fuel injection control system are shown in Fig. 1(b). The engine was connected to a dynamometer used for engine motoring and absorbing the output power. The dynamometer also controlled the engine load and speed and was equipped with a load cell to measure the engine torque. The engine was also equipped with an electronically-controlled common rail injection system, hydraulic variable valve actuating (VVA) system, and exhaust gas recirculation (EGR) system. The VVA system controls the valve timing and valve lift. Additionally, the diesel fuel and water injection timing and duration were also monitored. The VVA system and the supercharger were operated independent of the engine.
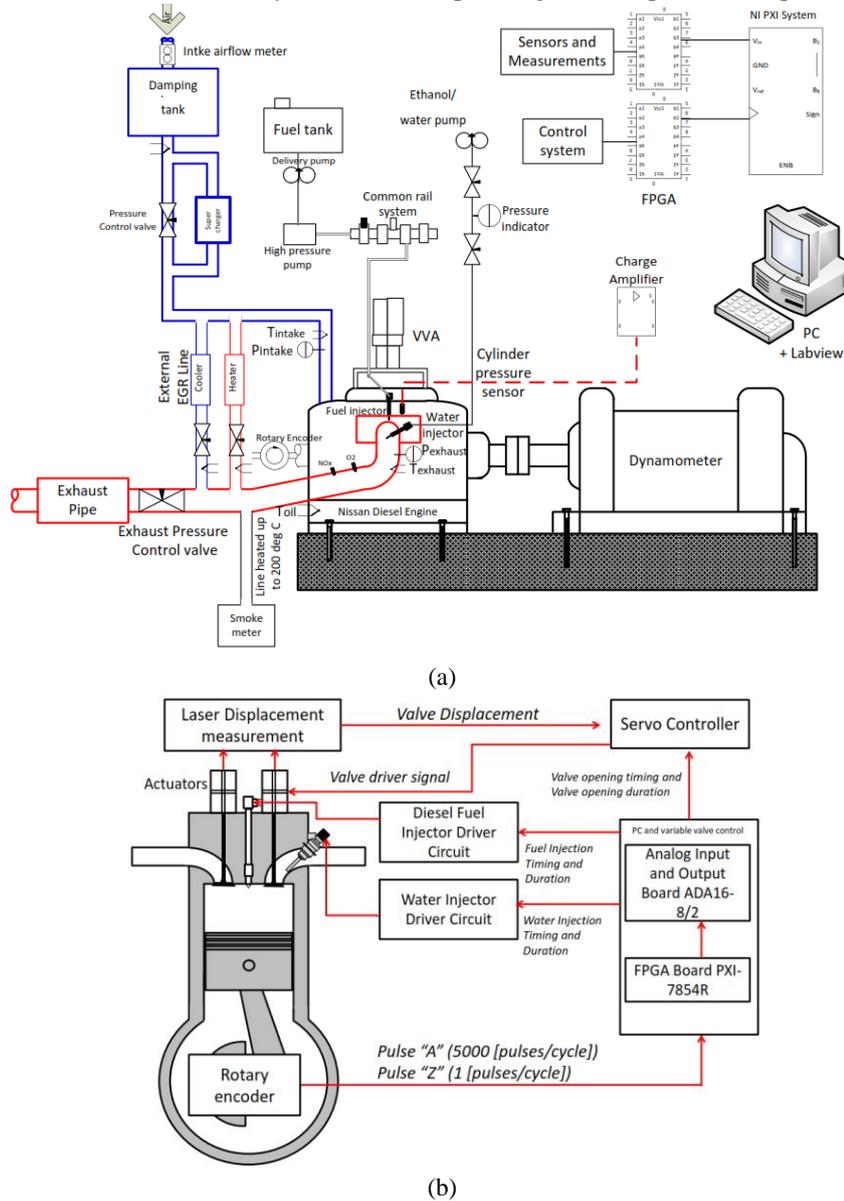

(a)

(b)

Fig. 1 Diesel engine test system (a) Experimental system overview and measurement instrumentation (b) Variable valve actuating, diesel fuel injection and water injection control system

Cylinder pressure measurements were carried out using piezoelectric pressure transducer of Kistler 6123 type. The intake pressure was measured using the strain gauge pressure transducer (Kyowa) that was installed in the intake manifold. However, for the exhaust pressure measurements a water-cooled strain gauge pressure sensor (Kyowa) 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 gases, lubricant oil and cooling water temperatures. Heaters and thermal controllers were used with thermocouples to control the intake air, cooling water,

and lubricant oil temperatures. The air mass flow rate was measured by a laminar air flow meter (Sokken), which was installed just before the surge tank. Electronically-controlled pressure regulating valve was mounted on the intake manifold to control the desirable boost pressure. Similarly, exhaust pressure regulating valve was installed to allow the raising of exhaust pressure and controlling the internal EGR ratio as well. A rotary encoder (Nikon) that produces 5000 pulses per revolution was connected to the crankcase camshaft. A photosensor was also attached to the flange of the crankcase camshaft to produce one pulse per revolution. Pulses from the rotary encoder and the photo sensor were necessary for sampling of cylinder pressure and controlling injection timing and valve timing. Developed program based on LabVIEW software and FPGA system were used to monitor the system and data acquisition depending on the signal from the rotary encoder and photosensor. The exhaust gases were sampled just after the exhaust manifold for the measurement of NO_x, O_2 and soot concentrations. The smoke meter was used for soot concentration measurement. The soot sampling line was heated up to 200° C to prevent soot condensation. A NO_x sensor (Horiba, MEXA-720) was used for NO_x measurements.

A quantitative estimation of the uncertainty in the present measurements was calculated using the procedure by Kline (Kline, 1985). The uncertainties in the measurement of bsfc, IMEP, and engine speed were found to be in the range of 1.8%, 2%, and 0.3% ($\pm$ 3 rpm), respectively.

Table 1 Engine specifications

Engine type	4-Stroke single cylinder water-cooled direct injection diesel engine
Bore [mm]	89
Stroke [mm]	100
Displacement [cm^3]	622
Compression ratio	15:1
Combustion chamber	Reentrant type
Injection system	Common rail injection system
Diesel fuel injector	Solenoid injector type with 8 holes, $\phi = 0.158$ mm
Intake system	Supercharged
Valve train	Hydraulic variable valves two intake valves and one exhaust valve
Water injector	A commercial gasoline direct injector with slit hole size of 0.135 mm x 0.775 mm. The spray is fan type with an angle of 75.6^o achieved at 10 MPa injection pressure, 0.1 MPa surrounding pressure, and 293 K surrounding temperature.

3. Methodology

In this study, the IMEP was used as a measure of the output power. It is considered a measure of engine capacity to do work independent of engine displacement. The IMEP is defined in equation (1) as indicated work divided by the displacement. It is calculated by trapezoidal integration of the cylinder pressure and volume in the expansion stroke for each cycle as follows:

$$IMEP = \frac{\int p dV}{V_{displace}} \tag{1}$$

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

Cylinder pressure data was sampled at every 0.144^o crank angle. For each tested condition, 200 consecutive cycles were sampled and averaged. The average cycle was used to represent the cylinder pressure. Also, the cycle-to-cycle variations were evaluated by calculating the coefficient of variance (COV_{imep}) at each experimental condition according to equation (2). The stability of engine operation was assessed by calculating COV_{imep}, which is defined as the ratio of the standard deviation in IMEP to the mean IMEP over the sampled cycles (Dempsey et al., 2014).

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

where n is the number of samples.

The acceptable level of load variation was somewhat subjective but typically taken to be a COV_{imep} less than 4% (Dempsey et al., 2014). Fig. 2 shows an example of the average cycle used to represent the data obtained from 200 cycles, where the COV_{imep} is equal to 0.8%. For all the tested conditions, the COV_{imep} was not higher than 1%.

For the analysis of in-cylinder pressure, the AHRR was calculated from the measured pressure data and the cylinder volume according to equation (3) (Heywood, 1988).

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

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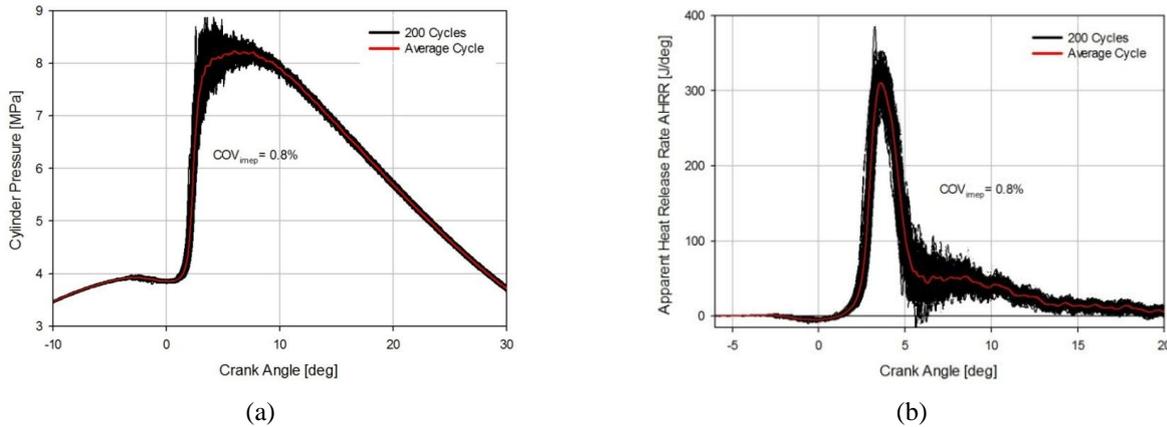

Fig. 2 The average cycle of the conventional diesel combustion that was used to represent the 200 cycles with COV_{imep} = 0.8% (a) p-θ diagram (b) AHRR.

The value of specific heat ratio (κ) depends on the gas temperature, and it was calculated from equation (4) as a function of average gas temperature (T).

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

The combustion phasing was determined at CA03, CA50, and CA90. CA03 here refers to the crank angle at which 3% of the total heat is released as illustrated in Fig. 3. The heat release amount was calculated according to equation (5). The CA03 calculation was used as an indication of the start of the combustion process, and it is used for ignition delay estimation. Similarly, CA50 and CA90 are defined as the crank angle at which 50% and 90% of the total heat is released respectively. The CA50 determines the end of premixed combustion phase and the start of the late combustion stage. The duration between CA03 and CA50 shows the burn duration for the premixed combustion stage. CA90 indicates the end of the combustion process. The late combustion phase duration was determined as the duration between CA50 and CA90. Therefore, the control of CA50 timing leads to control of combustion phasing (premixed combustion phase and late combustion phase).

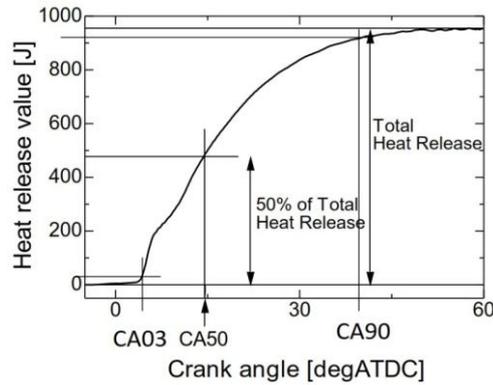$$Q_{net} = \int_{\theta_{IVC}}^{\theta} (\frac{dQ_{net}}{d\theta}) d\theta \tag{5}$$

Fig. 3 Definition of CA03, CA50, and CA90.

4. Experimental conditions and procedure

The experimental investigation was carried out at conditions shown in Table 2. The cooling water, lubrication oil, and intake air temperature were controlled at the listed values to maintain a constant heat loss for all conditions. Also, the fuel injection timing, quantity, and engine speed were controlled at the desired values to keep the cylinder combustion condition constant. Additionally, the used experimental conditions targeted a high load conventional combustion for high soot concentration. The intake and exhaust manifold pressures were controlled at 0.12 MPa to attain a balance between the intake and exhaust manifolds pressures. The balance between the intake and exhaust manifolds pressures allows the water injected into exhaust manifold to re-enter the combustion chamber when opening the exhaust valve during the intake stroke. The primary variable in this investigation is the water injection location either in the intake or exhaust manifolds.

Water injection timings and durations for both locations are presented in Table 2. The water injection quantity was fixed at 40 mg/cycle, corresponding to two injections of 8 ms duration separated by 1 ms duration (8 ms – 1ms – 8 ms) at an injection pressure of 2 MPa. The injector signal and the valve timings for both cases are shown in Fig. 4. The diesel fuel injection quantity was fixed at 32 mg/cycle, corresponding to 1100 μs at an injection pressure of 100 MPa.

The comparison of combustion process between intake and exhaust manifold injection was done with respect to cylinder pressure, AHRR, ignition delay, and combustion phasing. While, the IMEP, bsfc, ITE and total heat generated per cycle were considered to evaluate engine performance. Also, NO_x and soot emissions were measured for exhaust emissions analysis.

Table 2 Experimental conditions

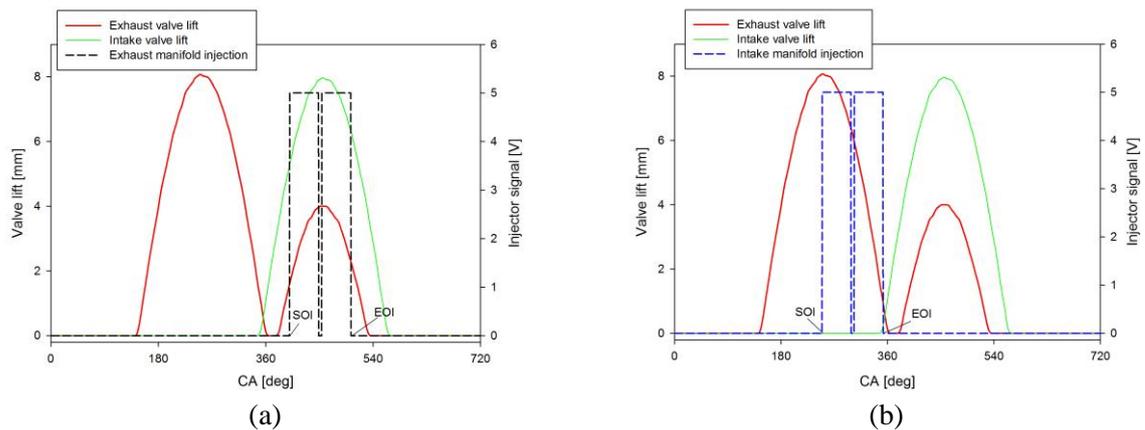|	
Engine speed [rpm]	1000
Fuel injection quantity [mg/cycle]	32 (1100 μs)
Fuel injection timing [ATDC]	-6
Fuel injection pressure [MPa]	100
Exhaust manifold injection timing [0 -720 deg] SOI	400° (8ms-1ms-8ms)
Intake manifold injection timing [0 -720 deg] SOI	250° (8ms-1ms-8ms)
Water injection amount [mg/cycle]	40
Water injection pressure [MPa]	2
Intake air temperature [^{o}C]	65
Cooling water temperature [^{o}C]	85
Lubrication oil temperature [^{o}C]	70
Intake pressure [MPa]	0.12
Exhaust pressure [MPa]	0.12
Intake valve lift, IVO, IVC	8 mm, 14 ^{o}BTDC, 30 ^{o}ABDC
Exhaust valve lift, EVO, EVC	8 mm, 39 ^{o}BBDC, 5 ^{o}ATDC
Exhaust valve reopen lift [mm] and EGR ratio [%]	3mm (10% EGR), 4mm (25% EGR)

Fig. 4 Water injection timing and duration with valve timing for (a) Exhaust injection (b) Intake injection.

The experimental test procedure discussed in the present study starts by switching on the cooling water and lubricant oil systems and then enabling their heating systems to raise the cooling water and lubrication oil temperatures to 85° C and 70° C, respectively. Then, the engine motoring is started by operating the supercharger, intake air heating system, and the VVA system. After that, the fuel injection pressure was raised up to 100 MPa, and the fuel injection timing and duration were set at -6 ATDC and 1100 μs and the fuel injection started. Next, the rack arm was used to control the engine speed at 1000 rpm. After operating the engine for more than 1500 cycles using diesel fuel, the steady state condition was achieved, and data sampling started. For exhaust gas recirculation, the exhaust valve was opened during the intake stroke using the VVA control system according to the timing shown in Table 2 and Fig. 4. For water injection, the water pump was switched on, and the water line pressure increased to 2 MPa and the water injection timing and duration specified to start injection into the intake or exhaust manifold as shown in Table 2 and Fig. 4.

The CA50 control was enabled by setting the required crank angle for controlling CA50 in the program. The CA50 control was based on a calculation of CA50 in the previous cycle using cylinder pressure data and equations (3)-(5). The duration from the start of injection to CA50 was also determined. Then, the start of injection in the following cycle was adjusted to a proper timing to control CA50 at the desired value. Therefore, CA50 control is based on cycle-to-cycle control strategy and based on the variation of the start of injection timing from cycle-to-cycle to maintain the CA50 at the desired crank angle as shown in Fig. 5. The previously described control strategy is illustrated in Fig. 5(a) for 20 consecutive cycles for conventional diesel combustion using normal operating conditions and CA50 control. For normal operating conditions the start of injection (SOI) was fixed at -6° ATDC, and the CA50 varied accordingly. For CA50 control, the SOI was varied depending on the value of CA50 in the previous cycle to maintain the CA50 in the next cycle according to the control condition. Fig. 5(b) shows the difference between normal operating conditions and CA50 control for various water injection conditions. The plotted points in CA50 control value is the average value for 200 consecutive cycles, and the error bar is added to show the maximum and minimum CA50 value in the sampled 200 points. For normal operating condition, the SOI was fixed at -6° ATDC, and the CA50 varied accordingly while for CA50 control, the SOI was varied to control CA50 at 4.8^{o}ATDC.

Finally, the different readings from the measuring devices for a particular test were recorded at steady state condition of the engine operation. This step was repeated to cover the EGR conditions and water injection conditions according to the test program summarized in Table 3.

5. Results and discussion

The combustion analysis, mechanical performance and the exhaust emissions of a diesel engine are discussed when water was injected into intake and exhaust manifolds with and without CA50 control. The experimental work was conducted as stated in the test program shown in Table 3. For combustion analysis, the cylinder pressure, AHRR, ignition delay and combustion phasing are illustrated versus the crank angle. The performance parameters such as bsfc, ITE, IMEP, and total heat generation are discussed at different water injection conditions. Similarly, the emission concentrations of NO$_x$ and soot are introduced for the various tested conditions.

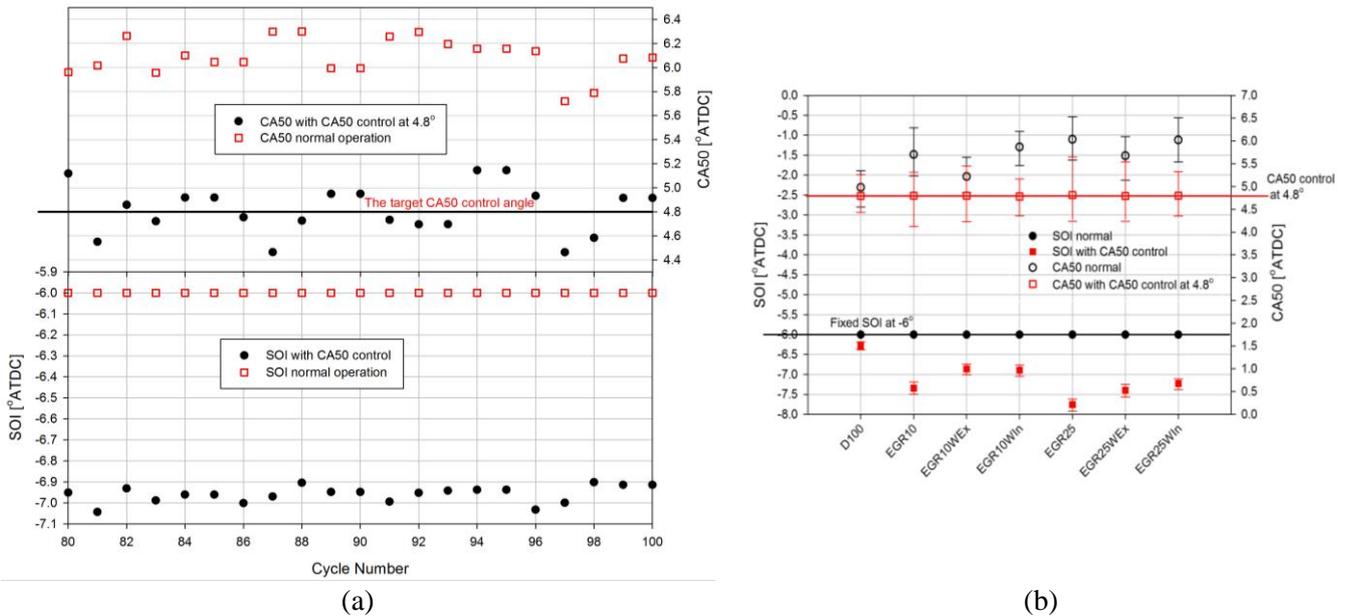(a)   (b)

Fig. 5 Definition of CA50 control (a) Conventional diesel cycle-to-cycle variation of SOI for CA50 control at 4.8^o for 20 consecutive cycles (b) SOI and CA50 for the different condition with and without CA50 control.

Table 3 Experimental program

Condition	Equivalence ratio	EGR ratio	Water injection location	Speed, rpm
D100		--	--	
EGR10		10%	--	
EGR10WEx		10%	Exhaust manifold	
EGR10WIn	0.72	10%	Intake manifold	1000
EGR25		25%	--	
EGR25WEx		25%	Exhaust manifold	
EGR25WIn		25%	Intake manifold	

5.1 Combustion characteristics

The investigation of the EGR effect on the combustion and emission of the diesel engine was conducted first as a preliminary experiment to differentiate between the effect of EGR and the effect of water injection. The experimental conditions and adapted program of this experiment are shown in Table 2 and Table 3. The values of exhaust gases recirculation ratio were 10% (EGR10) and 25% EGR (EGR25). Fig. 6 and Fig. 7 show the cylinder pressure and AHRR for EGR10 and EGR25, respectively. Fig. 8 shows the ignition delay and combustion phasing for EGR10 and EGR25 with and without water injection. The cylinder pressure decreased by 6.5% and 13.6% for EGR10 and EGR25 respectively compared to conventional diesel combustion. This reduction can be attributed to the fact that increasing EGR ratio decreases the inlet O_2 concentration which decelerates mixing between O_2 and fuel resulting in the extension of the flame region. Also, the quantity of CO_2 gas and H_2O vapor that absorb the released heat increased. The higher the amount of H_2O and CO_2, the higher the inlet heat capacity which decreases the flame temperature and AHRR as illustrated in Fig. 6(b) and Fig. 7(b). Additionally, the EGR leads to poor mixing process between the fuel and oxygen resulting in longer ignition delay and lower AHRR at the premixed combustion zone. However, the AHRR at the mixing controlled combustion phase was higher with an increase in the EGR ratio. The CA03 (which gives an indication of ignition delay) increased with increasing rate of EGR where it become 2.72^o and 3.00^o ATDC for EGR10 and EGR25 compared to 2.69^o ATDC for conventional diesel. Similarly, CA50 was 5.2^o and 5.8^o ATDC for EGR10 and EGR25 compared to 4.7^o ATDC for regular diesel. The higher the EGR ratio, the shorter the combustion duration (duration from CA03 to CA90).

With injection of water into the exhaust manifold, the start of combustion was similar to that of regular diesel fuel. However, the AHRR for EGR25WEx and EGR10WEx was increased at the premixed combustion phase compared to EGR25. This increase is attributed to the dilution effect of water that enhances the mixing process between fuel and air

and results in a higher premixed combustion while the diffusion combustion remains the same. Also, the charge cooling effect eliminated and no change was noted in the ignition delay due to water evaporation in exhaust manifold before combustion. On the other hand, the injection of water into intake manifold led to cooling of the intake charge. The reduction of the intake air temperature increases the in-cylinder air density which means that a given volume of gas entrained by fuel spray contains a bigger mass of air. Thus, the fuel and air mixing process was enhanced and resulted in a higher combustion pressure and a higher AHRR at the premixed phase, and this was evident with 10% EGR ratio as shown in Fig. 6(a). Also, the influence of water injection into intake manifold on ignition delay is given in Fig. 8. The intake manifold water injection achieves longer ignition delay compared to exhaust manifold injection. The longer ignition delay is also responsible for increasing the time at which the fuel and air are mixed. Consequently, diesel fuel is mostly burned under a premixed condition with intake manifold water injection as shown in Fig. 6(b) and Fig. 7(b). Also, Fig. 8(a) indicates that intake manifold water injection has a longer combustion duration compared to exhaust manifold injection and conventional diesel combustion.

Combustion phasing control was applied by controlling the CA50 at 4.8° ATDC. The main purpose of CA50 control is to suppress the change in combustion phase due to water injection. The water addition changes the combustion phase, and the thermal efficiency will be changed as well. This change should be eliminated from the discussions on the advantages of water addition on the engine performance. Applying the CA50 control for all the test conditions makes the SOI earlier so that the CA50 for all conditions occur at the same crank angle. Consequently, combustion process and AHRR with applying CA50 control occurs earlier than that without CA50 control. Also, the early start of injection elongates the time at which the fuel and air mix and results in a higher in-cylinder pressure for all the tested conditions, and consequently a higher power as shown in Fig. 6 and Fig. 7. The AHRR curve indicates that the amount of fuel burned under premixed condition decreased and the diffusion combustion improved with CA50 control. This change in AHRR curve is shown for EGR10CA50, EGR10CA50WEx, EGR25CA50 and EGR25CA50WEx cases in Fig. 6(b) and Fig. 7(b). Also, Fig. 8(b) illustrates that with CA50 control, the combustion duration increased compared to Fig. 8(a), without CA50 control.

The effect of water injection on the intake air, cylinder wall, and exhaust gases temperatures is shown in Fig. 9(a). The intake manifold temperature was controlled at $65 \pm 2°$ C. The intake manifold temperature was approximately the same for conventional diesel combustion, EGR without water injection and exhaust manifold water injection. However, with water injection in the intake manifold, the intake air temperature reduced by 1.1% compared to other conditions due to water injection. The CA50 control condition did not affect the intake air temperature variation when compared to conditions without CA50 control. The cylinder wall temperature attained the same trend of intake air temperature. The cylinder wall temperature for water injection into intake air is lower than that of other conditions by 4.9% and 7% for EGR10WIn and EGR25WIn, respectively. For exhaust gases temperature, EGR10 and EGR25 cases show an increase in the exhaust manifold temperature compared to D100 due to partially closed pressure regulating valve, which is installed in the exhaust manifold, to allow the raising of exhaust pressure and controlling the EGR ratio. With water injection into the exhaust manifold, the exhaust gases temperature reduced by 6.4% and 5.2% for EGR10WEx and EGR25WEx compared to corresponding EGR values without water injection. This reduction is due to the water evaporation in the exhaust pipe. For water injection in the intake manifold and conventional diesel conditions, the exhaust manifold temperature is almost similar.

The average gas temperature was calculated for the various tested conditions as shown in Fig. 9(b) and Fig. 9(c). For EGR10 and EGR25, the average gas temperature reduced compared to D100 due to the increase in CO_2 and H_2O amount which enhance the in-cylinder heat capacity and absorb the released heat leading to a reduction in flame temperature and average gas temperature. For water injection cases, the H_2O quantity increased and in-cylinder heat capacity was enhanced compared to EGR without water injection cases, which results in a further reduction in flame temperature as shown for the EGR25WIn case in Fig. 9(c). Additionally, water injection into the intake manifold leads to charge cooling because of the heat of vaporization of water. However, EGR10WIn shows a higher average gas temperature, and this may be due to the enhancement of fuel and air mixing process as a result of water dilution effect. Also, water injection into exhaust manifold cases (EGR10WEx and EGR25WEx) show a higher average gas temperature compared to EGR without water injection, and this is attributed to the fact that water vaporized in the exhaust pipe and not in the combustion chamber. Therefore, water injection into exhaust manifold eliminates the cooling effect of water. The CA50 control for the tested conditions enables early SOI and longer mixing time between air and fuel leading to a rapid increase in in-cylinder temperature in premixed combustion phase, while for the late

combustion phase, the average gas temperature is reduced for CA50 control compared to normal operating conditions.
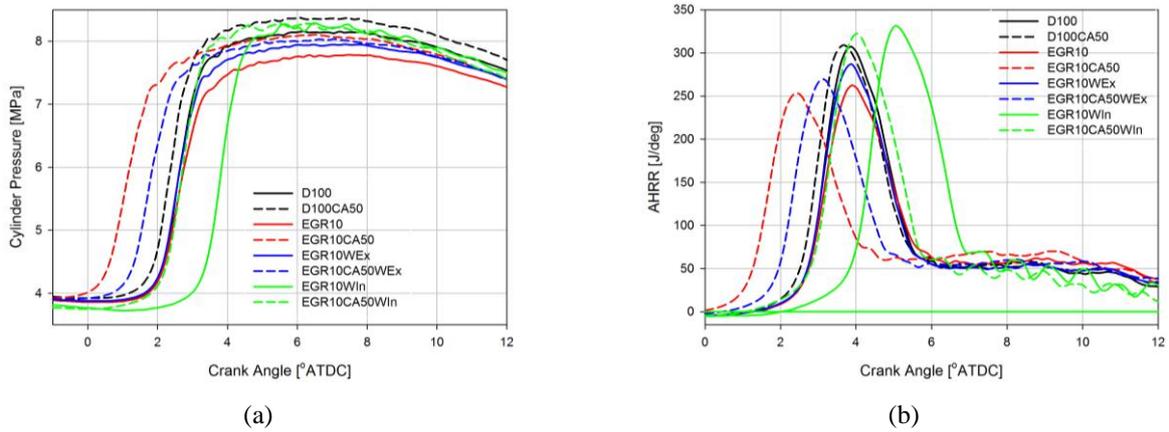

(a)

(b)

Fig. 6 The variation of cylinder pressure and heat release rate with crank angle for EGR of 10% with water injection of 40 mg/cycle into intake and exhaust manifolds (a) p-θ (b) AHRR
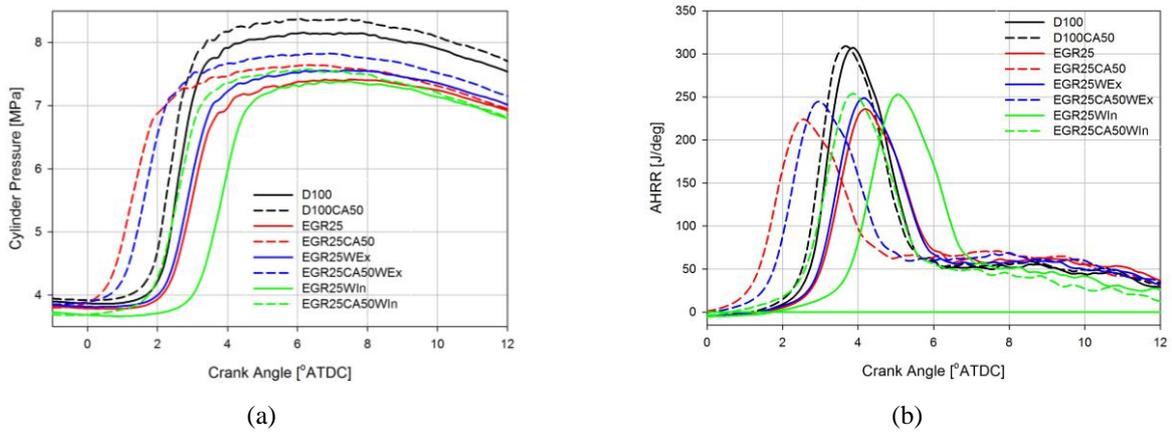

(a)

(b)

Fig. 7 The variation of cylinder pressure and heat release rate with crank angle for EGR of 25% with water injection of 40 mg/cycle into intake and exhaust manifolds (a) p-θ (b) AHRR
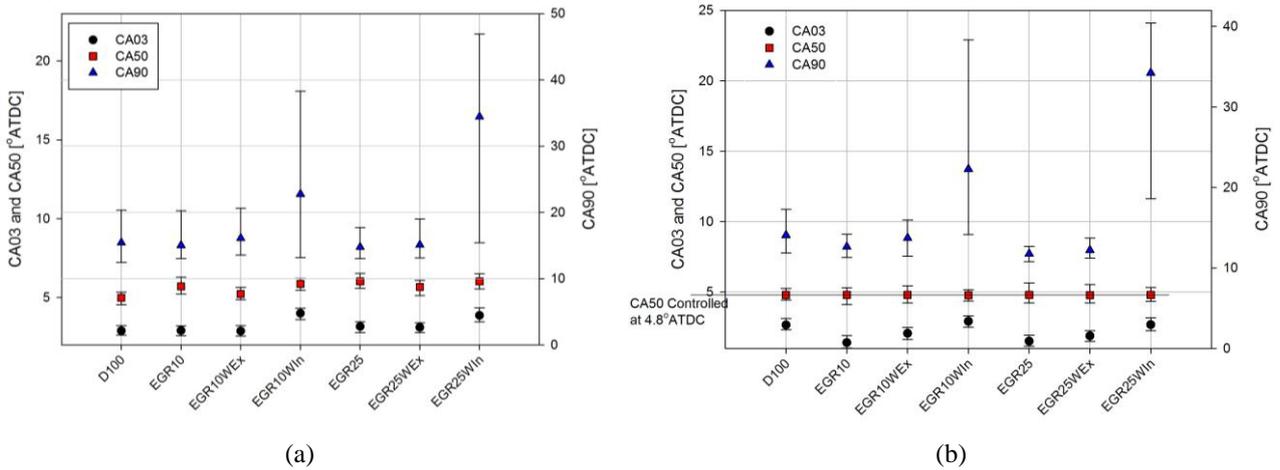

(a)

(b)

Fig. 8 Ignition delay and combustion phasing for 10% and 25% EGR and water injection into intake and exhaust manifolds (a) Without CA50 control (b) With CA50 control

(a)   (b)
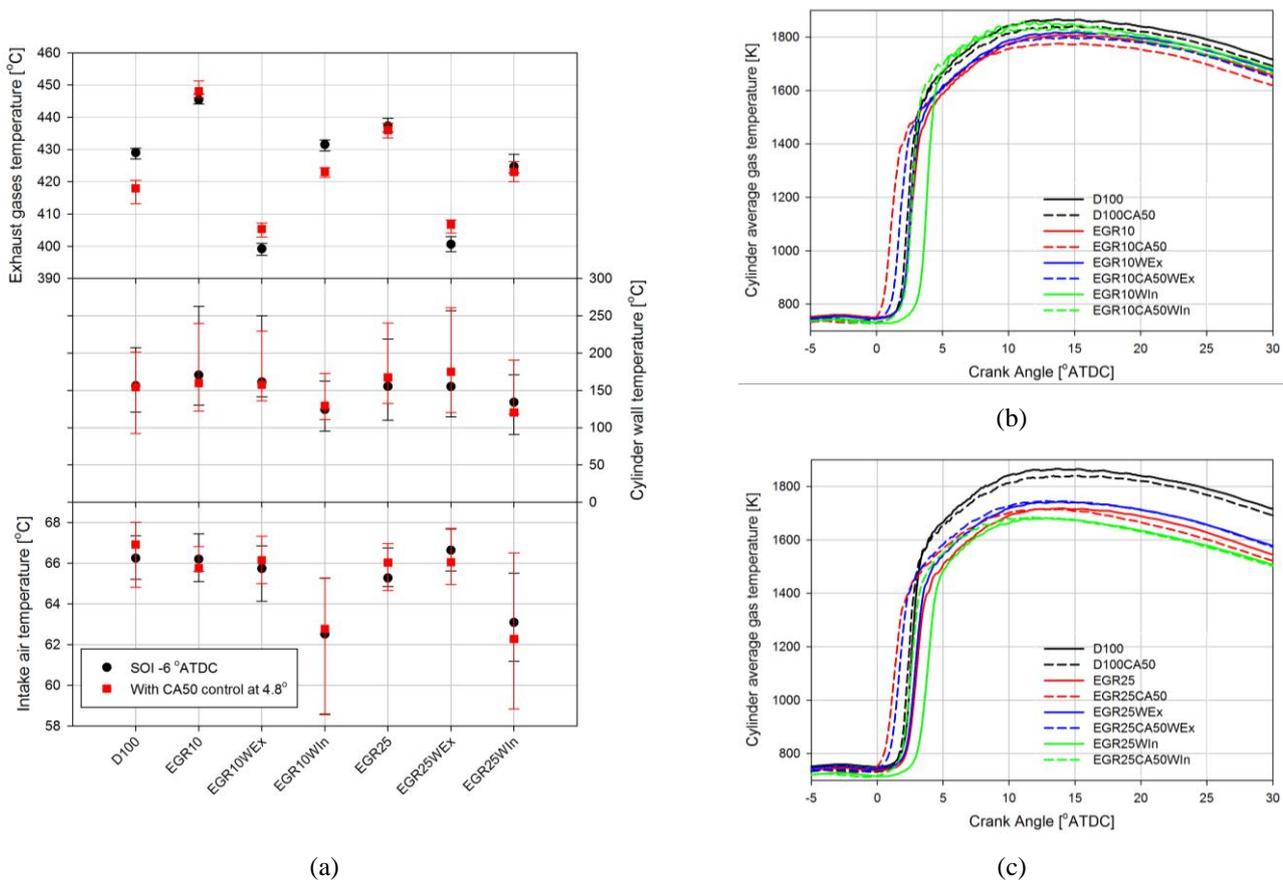
(c)

Fig. 9 Temperature measurement for 10% and 25% EGR and water injection into intake and exhaust manifolds (a) Intake air, cylinder wall and exhaust gases temperatures (b) Average gas temperature for 10% EGR (c) Average gas temperature for 25% EGR with and without CA50

5.2 Engine performance

The indicated mean effective pressure and total heat generated per cycle are shown in Fig. 10. The plotted points are the average of 200 cycles data for each condition. The error bars are added to the figure to show the maximum and minimum value of the 200 cycles. The IMEP declined by 6% and 21% for EGR10 and EGR25 respectively. This is due to the reduction of the cylinder pressure with the increase of the EGR ratio. By applying water injection, the IMEP slightly increased for both of water injection into intake and exhaust manifolds compared to EGR10. Also, the IMEP increased for EGR25WEx compared to EGR25 by 3% while the intake water injection gives the same IMEP value as EGR25. With CA50 control, which is applied to improve the IMEP, the conventional diesel combustion and EGR25WEx are the only conditions at which the IMEP improved by 4% and 7% respectively. However, the other condition does not show significant difference between with or without CA50 control. As IMEP provides a measure of the engine output power, water injection into exhaust manifold for EGR25 improves the engine power compared to the intake manifold injection. This improvement is because the exhaust manifold water injection eliminates the water cooling effect and enhances the combustion process by reuse of the waste heat from exhaust gases. However, the intake manifold injection decreases the combustion pressure and temperature and consequently the engine output power.

The total heat generated which is calculated from the integration of the AHRR decreased by 3% and 16% for EGR10 and EGR25 compared to that of conventional diesel. However, there was no effect of water injection on the heat generation in the case of EGR10. At EGR25, the total heat generation remains the same for EGR25 and EGR25WEx, while, it decreased for EGR25WIn case by 10% compared to EGR25. This behavior confirms that the cooling effect is eliminated by exhaust manifold injection but exists for intake manifold injection although the energy balance for both cases is the same. Also, water injection into intake manifold increases the heat losses in the engine cylinder. The CA50 control achieves a noticeable improvement in the total heat generated except for the intake manifold water injection. The heat generated in the case of EGR25WEx with CA50 control is higher by 5% compared

to without CA50 control.

The bsfc and ITE are shown in Fig. 11 for the different tested conditions. The bsfc increased with increasing EGR ratio. By applying exhaust manifold water injection, the EGR10WEx case enhanced the bsfc by 19% compared to EGR10 case, while the bsfc decreased by 5% for EGR25WEx compared to the EGR25 without water injection. However, for intake manifold water injection, the bsfc increased drastically by 127% and 70% for EGR10WIn and EGR25WIn respectively compared to EGR without water injection. This increase is due to the reduction of the engine output power that results from the cooling effect of water injection and the increase of the heat losses due to intake manifold water injection. Applying the CA50 control shows a reduction in bsfc with EGR10WIn and EGR25WEx cases by 6% and 7%, respectively. The other cases show no change even with CA50 control. The indicated thermal efficiency reduced by 6% and 21% for EGR10 and EGR25 respectively compared to conventional diesel as shown in Fig. 11. For exhaust manifold water injection, the indicated thermal efficiency improved by 3% for both of EGR10WEx and EGR25WEx compared to EGR without water injection. For intake manifold water injection, the ITE increased by 4% for EGR10WIn and decreased by 2% for EGR25WIn compared to EGR without water injection. By applying CA50 control, the ITE reduced for most of the tested cases except for the conventional diesel and EGR25WEx cases.
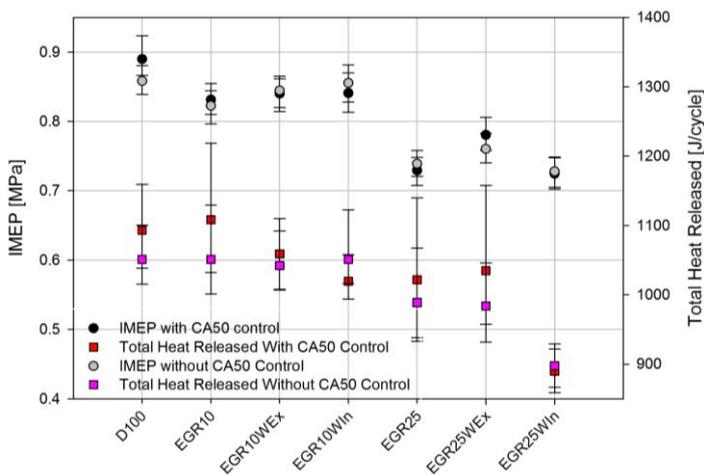

Fig. 10 IMEP and total heat released for10% and 25% EGR and water injection into intake and exhaust manifold with and without CA50 control
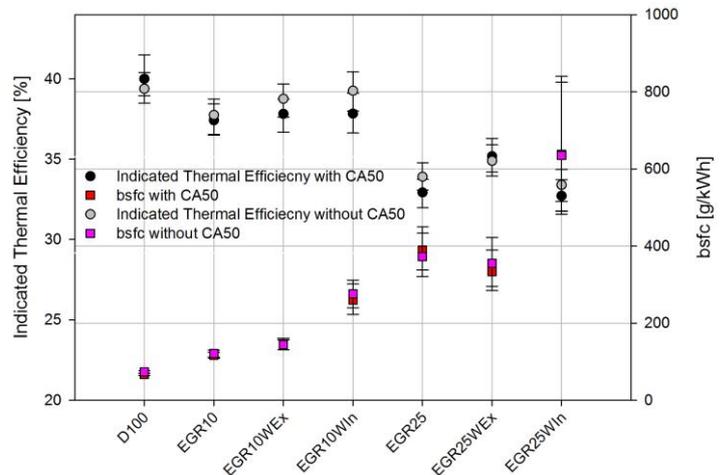

Fig. 11 Indicated thermal efficiency and bsfc for10% and 25% EGR and water injection into intake and exhaust manifold with and without CA50 control

5.3 Soot and NOₓ emissions

The EGR led to a reduction in flame temperature due to the higher heat capacity of H_2O and CO_2. EGR also decreases the inlet O_2 concentration and accordingly the mixing efficiency with the fuel. Consequently, the NO_x emissions decreased due to the low flame temperature and soot emissions increased due to the poor mixing between the fuel and the air as shown in Fig. 12. The NO_x emissions were reduced by 52% and 85% for EGR10 and EGR25, respectively. The NO_x emissions reduced further with water injection by 16% and 56% for EGR10WEx and EGR10WIn respectively compared to EGR10's case. For the case of EGR25, EGR25WIn reduced NO_x emissions by 41% while EGR25WEx increased NO_x by 5%. Hence, it is clear that the reduction in NO_x emissions with intake manifold water injection is more than that of exhaust manifold water injection. This is due to a combination of dilution and thermal effect and the higher heat losses. However, the reduction noted with exhaust manifold water injection is mostly due to the dilution effect of water injection as cited by Tauzia et al. (Tauzia et al., 2010). Applying CA50 control leads to increase in NO_x emissions for all the tested conditions. As there is a tradeoff between NO_x and soot, soot emissions increased with EGR10 by 44%. However, EGR10WEx reduced soot emissions by 55% compared to the case of EGR without water injection. Also, the soot emissions were reduced by 10% for the case of EGR10WIn. For 25% EGR, soot emission increased significantly. However, it is worth noting that water injection into exhaust manifold reduces the soot emissions by 35% compared to EGR25 without water injection. Also, with EGR25WIn, soot emissions reduced by 7% compared to EGR25. The reduction in soot emissions using water injection in exhaust manifold is higher than that of intake manifold. The decrease in soot emissions is as a result of the enhancement of fuel

and air mixing due to the dilution effect by exhaust manifold water injection. However, the lowering of flame temperature by intake manifold water injection affects soot oxidation and consequently increases the soot formation. Applying CA50 control reduced soot emissions by 10%, 23%, 7% and 5% for D100, EGR10WIn, EGR25WEx, and EGR25WIn respectively. Soot emissions on the other hand increased by 4%, 68%, and 13% for EGR10, EGR10WEx, and EGR25.
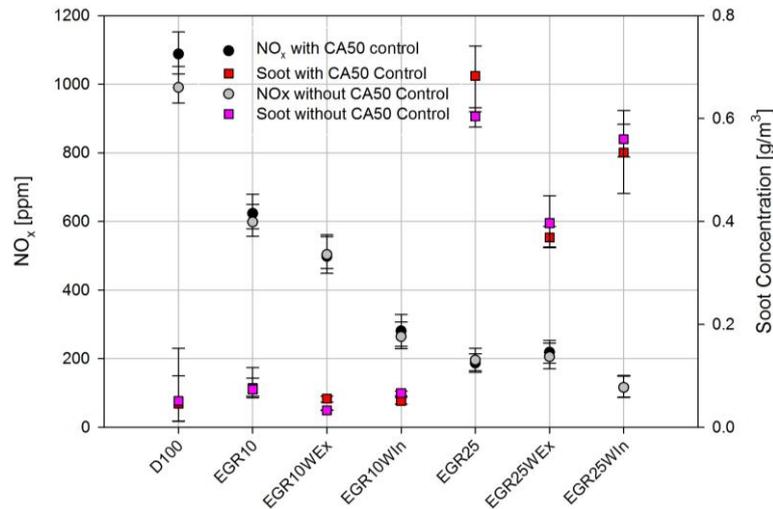

Fig. 12 Soot and NO_x emissions for 10% and 25% EGR with water injection into intake and exhaust manifold with and without CA50 control

Several practical questions arise when dealing with water injection into the exhaust manifold. The technique used throughout this research work shows that the water injected to exhaust manifold evaporates and enters to the cylinder during valve overlapping period at the intake stroke. The experimental results of this study indicate that the combustion, engine performance and exhaust emissions are influenced by water injection into exhaust manifold, which confirms that the water injected into exhaust manifold is sucked inside the cylinder. However, it is speculated that entire quantity of injected water at exhaust manifold will not enter to the cylinder as part of evaporated water will flow to the exhaust pipe. Therefore, further investigation is required to accurately quantify the amount of water sucked into the combustion chamber and the amount that escapes in the exhaust system. Additionally, more research work should be conducted to study the variation of initial H_2O concentration of intake air while using water injection in the exhaust manifold. The effect of exhaust manifold temperature on injector performance is also an important factor that effect on the exhaust manifold injection strategy and needs further investigations including spray characteristics. Further studies are required to investigate the dual fuel technique using exhaust manifold injection and the investigation should be extended to include several fuels and various engine operating conditions.

6. Conclusions

In this study, the effects of water injection into the intake manifold and exhaust manifold on the combustion and exhaust emissions of the direct injection diesel engine have been investigated experimentally. Major conclusions can be revealed as follows:

- The combustion analysis showed a reduction in cylinder peak pressure by 6.5% and 13.6% for EGR10 and EGR25 respectively, compared to pure diesel fuel combustion without EGR.
- The water injection into intake and exhaust manifold led to an increase in the cylinder peak pressure compared to EGR without water injection. The increase in cylinder peak pressure for intake manifold water injection was higher than that of exhaust manifold water injection. This is due to the reduction of the intake air temperature increasing the in-cylinder air density. Thus the volume of gas entrained by the fuel spray contains a bigger mass of air that improves the mixing with fuel and results in a higher combustion pressure.
- The intake manifold water injection results in longer ignition delay compared to exhaust manifold injection. The longer ignition delay is also responsible for the increase in time at which the fuel and air are mixed. Consequently, diesel fuel is mostly burned under a premixed condition with intake manifold water injection.
- Water injection into exhaust manifold achieves a higher average gas temperature compared to EGR without

water injection, and this is owing to the latent heat of vaporization of water being extracted from the exhaust pipe and not from in-cylinder heat. However, the charge cooling and combustion quenching are observed with water injection in the intake manifold.

- The total heat generated per cycle was not affected by exhaust manifold water injection while it decreased by 10% in the case of intake manifold water injection owing to the fact that water injection into intake manifold increasing the heat losses in the engine cylinder.

- The exhaust manifold water injection increased the bsfc by 19% compared to EGR10's case, while the bsfc decreased by 5% for EGR25WEx compared to the EGR25 without water injection. With intake manifold water injection, the bsfc is increased drastically by 127% and 70% for EGR10WIn and EGR25WIn respectively compared to EGR without water injection.

- The indicated thermal efficiency improved with exhaust manifold water injection compared to water injection into intake manifold due to the high heat losses in intake manifold water injection.

- The reduction in NO_x emissions with intake manifold water injection is more than that of exhaust manifold water injection due to the combination of dilution and thermal effect and the higher heat losses. The NO_x emissions reduced with water injection by 16% and 56% for EGR10WEx and EGR10WIn respectively compared to EGR10 case. For EGR25 case, EGR25WIn reduce NO_x emissions by 41% while EGR25WEx increase NO_x by 5%. The reduction with exhaust manifold water injection is attributed mostly to the dilution effect of water injection.

- The decrease in soot emissions with water exhaust manifold injection is higher than that of intake manifold water injection. The reduction in soot emissions is due to the enhancement of fuel and air mixing as a result of the dilution effect of exhaust manifold water injection. However, the lowering of flame temperature due to the intake manifold water injection affects soot oxidation and consequently increases the soot formation.

This study recommends the use of exhaust manifold water injection as it improves the engine emissions without increasing the in-cylinder heat losses. The waste heat recovery that is associated with exhaust manifold injection improves the engine efficiency and heat balance. However, further investigations are required to accurately quantify the amount of water sucked into the combustion chamber and the amount that escapes into the exhaust system. Also, the effect of exhaust manifold temperature on injector performance is also an important factor that affects the exhaust manifold injection strategy and needs further investigation including spray characteristics. Further studies are required to investigate the dual fuel technique using exhaust manifold injection and extending the investigations to include several fuels and various engine operating conditions.

Acknowledgements

References

Abu-Zaid, M., Performance of single cylinder, direct injection diesel engine using water fuel emulsions. Energy Conversion and Management, Vol.45, No.5 (2004), pp.697–705.

Alahmer, A., Yamin, J., Sakhrieh, A., and Hamdan, M.A., Engine performance using emulsified diesel fuel. Energy Conversion and Management, Vol.51, No.8 (2010) , pp.1708–1713.

Armas, O., Ballesteros, R., Martos, F.J., and Agudelo, J.R., Characterization of light duty diesel engine pollutant emissions using water-emulsified fuel. Fuel, Vol.84, No.7 (2005), pp.1011–1018.

Bedford, F., Rutland, C., Dittrich, P., Raab, A. and Wirbeleit, F., Effects of direct water injection on DI diesel engine combustion. SAE Technical Paper No.2000-01-2938, (2000).

Dempsey, A.B., Walker, N.R., Gingrich, E., and Reitz, R.D., Comparison of low temperature combustion strategies for advanced compression ignition engines with a focus on controllability. Combustion Science and Technology, Vol.186, No.2 (2014), pp.210–241.

Ghojel, J., Honnery, D., and Al-Khaleefi, K., Performance, emissions and heat release characteristics of direct injection diesel engine operating on diesel oil emulsion. Applied Thermal Engineering, Vol.26, No.17 (2006), pp.2132–2141.

Greeves, G., Khan, I.M., and Onion, G., Effects of water introduction on diesel engine combustion and emissions.

International Symposium on Combustion, Vol.16, No.1 (1977), pp.321–336.

Heywood, J.B., Internal combustion engine fundamentals (1988), p.510, McGraw-Hill.

Ithnin, A.M., Noge, H., Kadir, H.A., and Jazair, W., An overview of utilizing water-in-diesel emulsion fuel in diesel engine and its potential research study. Journal of the Energy Institute, Vol.87, No.4 (2014), pp.273–288.

Kadota, T. and Yamasaki, H., Recent advances in the combustion of water fuel emulsion. Progress in Energy and Combustion Science, Vol.28, No.5 (2002), pp.385–404.

Kline, S.J., The purposes of uncertainty analysis. Journal of Fluids Engineering, Vol.107, No.2 (1985), pp.153–160.

Kumar, N., Sharma, A., and Vibhanshu, V., Performance analyses of diesel engine at different injection angles using water diesel emulsion. SAE Technical Papers No.2013-01-2170, (2013).

Ma, X., Zhang, F., Han, K., Zhu, Z., and Liu, Y., Effects of intake manifold water injection on combustion and emissions of diesel engine. Energy Procedia, Vol.61 (2014), pp.777–781.

Maiboom, A. and Tauzia, X., NO_x and PM emissions reduction on an automotive HSDI diesel engine with water-in-diesel emulsion and EGR: An experimental study. Fuel, Vol.90, No.11 (2011), pp.3179–3192.

Mingrui, W., Thanh Sa, N., Turkson, R.F., Jinping, L., and Guanlun, G., Water injection for higher engine performance and lower emissions. Journal of the Energy Institute, Vol.90, No.2 (2016), pp.285-299.

Miyamoto, N., Ogawa, H., Wang, J. and Ohashi, H., Significant NO_x reductions with direct water injection in the sub-chamber of an IDI diesel engine. SAE Technical Paper No.950609, (1995).

Nazha, M.A.A., Rajakaruna, H., and Wagstaff, S.A., 2001. The use of emulsion, water induction and EGR for controlling diesel engine emissions. SAE Technical Paper No.2001-01-1941, (2001).

Nishijima, Y., Asaumi, Y., and Aoyagi, Y., Impingement spray system with direct water injection for premixed lean diesel combustion control. SAE Technical Paper No.2002-01-0109, (2002).

Odaka, M., Koike, N., Tsukamoto, Y., Narusawa, K., and Yoshida, K., Effects of EGR with a supplemental manifold water injection to control exhaust emissions from heavy-duty diesel powered vehicles. SAE Technical Paper No.910739, (1991).

Subramanian, K.A., A comparison of water-diesel emulsion and timed injection of water into the intake manifold of a diesel engine for simultaneous control of NO and smoke emissions. Energy Conversion and Management, Vol.52, No.2 (2011), pp.849–857.

Tarlet, D., Bellettre, J., Tazerout, M., and Rahmouni, C., Prediction of micro-explosion delay of emulsified fuel droplets. International Journal of Thermal Sciences, Vol.48, No.2 (2009), pp.449–460.

Tauzia, X., Maiboom, A., and Shah, S.R., Experimental study of inlet manifold water injection on combustion and emissions of an automotive direct injection diesel engine. Energy, Vol.35, No.9 (2010), pp.3628–3639.

Tesfa, B., Mishra, R., Gu, F., and Ball, A.D., Water injection effects on the performance and emission characteristics of a CI engine operating with biodiesel. Renewable Energy, Vol.37, No.1 (2012), pp.333–344.

Turns, S.R., An introduction to combustion: concepts and applications (1999), p.171, McGraw-Hill.

Udayakumar, R., Sundaram, S., and S.C. Johnson, Reduction of NO_x emissions by water injection in to the inlet manifold of a DI diesel engine. SAE Technical Paper No.2003-01-0264, (2003).