Experimental Investigation into the Effects of Two-Stage Injection on Fuel Injection Quantity, Combustion and Emissions in a High-Speed Optical Common Rail Diesel Engine

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Abstract: Diesel combustion and the formation of pollutants are directly influenced by the spatial and temporal distributions of the fuel within the combustion chamber of an internal combustion engine. The requirements for more efficient and responsive diesel engines have led to the introduction and implementation of multiple injection strategies. However, the effects of such injection modes on the hydraulic systems, such as the high pressure pipes and fuel injectors, must be thoroughly examined and compensated for. The objective of this study was to investigate the effects of fuel injection equipment characterisation and optimisation on diesel combustion and emissions with two-stage fuel injection. The fuel injection system was characterised and optimised through the measurement of the fuel injection rate and quantity, in particular, the interaction between the two injection events was quantified and compensated for. The effects of twin and variable split two-stage injection and dwell angle on diesel combustion and emissions were investigated in a high-speed direct injection single-cylinder optical diesel engine using heat release analysis and high-speed fuel spray and combustion visualisation technique. The results indicated that two-stage injection has the potential for simultaneous reduction of NOx and soot emissions. Nevertheless, the studied two-stage strategies resulted in higher soot emission, mainly due to the interaction between two consecutive fuel injection events, whereby the fuel sprays during the second injection were injected into burning regions, generating fuel-rich combustion. In addition, the variable two-stage strategies produced high levels of uHC emission in comparison to single and twin split injection cases. This was mainly attributed to firstly greater fuel quantity injected during the second injection and secondly poor mixing and air utilisation during the second fuel injection event.

Keywords: diesel engine, injection rate, two-stage injection, dwell angle, emissions

1 INTRODUCTION

A significant cost of industrialisation has been the environmental damages inflicted, to a large extent by the use of fossil fuels, within which the significant growth in the use and production of internal combustion engines have since been considered as one of the primary contributing factors. Due to ever increasing concern over the environmental impacts of the exhaust pollutants, emissions legislations have been progressively enforced since the 1960s. In recognition of the need to further reduce vehicle exhaust emissions and the greenhouse effect of CO_2, there has been a lot of interest in developing cleaner and more efficient energy saving vehicle powertrain. In response to social, legislative and environmental pressures, there is a large body of engine research work demonstrating the large energy saving and emissions reducing benefits of two-stage fuel injection. The introduction of common rail (CR) fuel injection system in the 1990s and further advancements thereafter allowed greater control and flexibility on fuel injection pressure, rate, quantity and timing over the entire operating range of diesel engines, facilitating successful application of such fuel injection strategies.

The initial investigations on using alternative injection strategies were primarily focused on the application of pilot and main injections or split injections with equal fuel demand per injection (50%/50%) [**1-3**]. The results demonstrated that shorter ignition delay was achieved due to pilot injection, indicating less premixed combustion, lowering the peak heat release rate (HRR). Therefore, nitrogen oxides (NOx) emission as well as combustion noise was considerably reduced in comparison to the conventional diesel combustion. In addition, the effect of post injection on further reduction of soot emission was examined by Han et al. [**4**] and Farrell et al. [**5**]. Their results showed that soot emission was reduced due to improved soot oxidation which was attributed to higher combustion temperature during mixing controlled combustion phase caused by the combustion of fuel injected during post injection. Furthermore, the potential for further reduction of exhaust emissions using exhaust gas recirculation (EGR) has been extensively investigated. Montgomery and Reitz [**6**] studied the effect of EGR in a heavy-duty diesel engine using 50%/50%, 55%/45% and 70%/30% two-stage injection strategies with EGR levels varying between 10 to 25%. Their investigation revealed the potential for simultaneous reduction of NOx and soot emissions using two-stage injection with EGR. The use of EGR decreased the NOx emission by limiting the peak HRR due to premixed combustion, thus lowering the in-cylinder temperature. The soot emission was reduced due to improved mixing in conjunction with the effect of late fuel injection which resulted in higher in-cylinder temperatures during diffusion combustion, maximising soot oxidation.

In order to better understand the mixing process, Zhang et al. [**7**] carried out a series of investigations involving detailed analysis of fuel-air mixing process in a constant volume vessel through the application of laser absorption scattering. They investigated the mixing process using conventional single injection and compared their results to those obtained through two-stage injection strategies 75%/25%, 50%/50% and 25%/75%. It was reported that the 75%/25% strategy resulted in maximum soot reduction under the tested engine operating conditions. This was mainly attributed to improved mixing due to increased in-cylinder turbulence caused by the combustion of fuel injected during the second injection. Shayler et al. [**8**] compared the combustion and emissions characteristics of single and two-stage injections in a light-duty diesel engine. In this study all the two-stage injection strategies were accompanied by a pilot injection whereby relatively small quantity of fuel was injected in order to improve fuel evaporation. The results indicated that strategies with more fuel quantity during the first injection resulted in less soot emission with no increase in NOx emission.

Koyanagi et al. [**9**] investigated the effects of engine design and operating parameters, in particular injector stability, spray symmetry, nozzle geometry, injection rate, pilot injection and swirl, in a light-duty single-cylinder optical diesel engine with similar production-type combustion chamber geometry. The authors reported that the pilot-main strategy was characterised by a complete premixed combustion of the fuel injected during the pilot injection, 10% of the total injected fuel quantity in this case. Therefore, ignition delay time was reduced due to increased in-cylinder pressure and temperature and the presence of active radicals. Consequently higher soot emission was produced due to deteriorated mixing of the main injected fuel; however, such an effect could be controlled through the use of a suitable dwell angle (DA). Badami et al. [**10**] also studied the effect of pilot injection quantity and DA on diesel combustion and emissions in a light-duty diesel engine. In this study, the effects of pilot injection, less than 1 to 15% of the total injected fuel quantity, were investigated. It was reported that soot and NOx emissions increased as the quantity of the pilot injection increased. The former was attributed to the increase in the in-cylinder pressure and temperature, due to the main combustion advance, while the latter was ascribed to the reduction of the premixed combustion. The same trend was observed as the DA was reduced. The authors also reported on the hydraulic effects of pilot injection on the combustion characteristics and the fluid-dynamic conditions at the start of the main injection.

Schmid et al. [**11**] studied the effect of nozzle hole geometry, rail pressure and pre-injection in a single-cylinder transparent light-duty diesel engine. The authors reported that the pre-injection (i.e. pilot) could lead to shorter ignition delay which in turn could result in the penetration of the main injection into burning regions. Kook and Bae [**12**] examined the effect of two-stage fuel injection in a single-cylinder direct injection (DI) premixed charge compression ignition (PCCI) engine. In this study the majority of the fuel was injected early during the first injection to form the premixed charge while a small quantity of fuel was injected close to top dead centre (TDC), serving as the ignition promoter. The results indicated that simultaneous reduction in emissions as well as improved combustion efficiency can be achieved provided that a combination of optimised fuel quantity, intake air temperature, injection pressure and compression ratio was used. Horibe et al. [**13**] also investigated the effect of two-stage fuel injection on partial PCCI combustion in a constant volume vessel. The first injection quantity was varied from 10 to 40% of the total fuel quantity and the DA was varied between 0.5 to 2ms. They reported on the potential of such an injection mode in reducing NOx emission regardless of the ambient oxygen mole fraction, in particular when the ignition delay was longer. Although longer dwell angel reduced the peak HRR when relatively small quantity of fuel was injected during the first injection, such an effect was not observed with larger first fuel injection quantity.

However, in the case of two-stage strategies with EGR, the results indicated that the first fuel injection quantity had no influence on the peak HRR; it decreased as the DA increased. The results revealed that the final NOx mass per released heat reduced, using two-stage injection, regardless of the ambient oxygen mole fraction.

Su et al. [**14**] studied the effect of two-stage fuel injection on homogeneous charge compression ignition (HCCI) diesel combustion both numerically and experimentally. The results revealed the potential of two-stage injection strategy in triggering the charge ignition due to the presence of locally rich fuel parcels under certain operating conditions, thus extending the HCCI operating range. Furthermore, the results indicated that there exists an optimal second injection timing with which maximum engine output is produced for a given fuel split ratio. Mobasheri et al. [**15**] also examined the effect of two-stage injection with different EGR rate on diesel combustion and emissions using CFD modelling. The results indicated that such an injection mode is capable of simultaneous reduction of soot and NOx emissions provided that optimal DA, fuel split ratio and EGR rate is used.

Bruneaux and Maligne [**16**] investigated the mixing and combustion processes of two-stage injection with short and long DA in a high pressure, high temperature vessel with similar thermodynamic conditions to those present in the combustion chamber of a diesel engine during injection. Several optical diagnostic techniques including laser-induced exciplex fluorescence, laser-induced fluorescence and particle image velocimetry were adopted for detailed analysis of fuel evaporation, mixture formation, in-cylinder flow field and combustion characteristics. The results indicated that in the case of short DA, increase in the mixing rate at the head of the second injection was detected due to the interaction between the first and the second injections. This interaction and the resulting effects on mixing were reduced with longer DA. In addition, in the case of two-stage injection, the ignition of the second injection was promoted by the entrainment of high temperature gases originating from the combustion of premixed fuel and air from the first injection, therefore the combustion of the second injection was more fuel-rich and consequently more soot and poly-aromatic hydrocarbons were formed. This effect was more pronounced with longer DA, this was attributed to the presence of low temperature intermediates closer to the injector nozzle at the timing of the second injection which resulted in faster entrainment of higher temperature gases into the fuel sprays during the second injection, provoking faster ignition and consequently more fuel-rich combustion.

Abdullah et al. [**17**] investigated the effect of higher pressure two-stage injection on engine performance and emissions in a V6 2.8 litre turbocharged DI diesel engine. The results showed that high injection pressure with EGR has a great potential in controlling NOx emission and engine performance simultaneously. Swor et al. [**18**] investigated the effect of adaptive injection strategy on two-stage injection with variable injection pressure in a single-cylinder premixed compression ignition engine. The objective of this investigation was to reduce spray-wall impingement, control combustion phasing and limit pressure rise rate. The fuel injection system was capable of producing low and high injection pressures, 300 and 1200 bar respectively. The former prevented spray-wall impingement of early injections caused by long liquid penetration lengths while the latter improved fuel atomisation and mixing processes during the second injection. Two combustion modes were investigated; single stage heat release (SHR) combustion and two-stage combustion (TSC). The SHR proved to have better fuel economy and lower emissions in comparison to TSC. The results indicated that the adaptive two-stage injection has the potential to achieve similar engine performance to that of highly diluted low temperature combustion but at a lower EGR rate, however, small increase in emissions was reported.

Although, hydraulic effects of two-stage fuel injection has been previously reported [**10**, **19-25**], the effects of advanced fuel injection modes on the fuel injection system are yet to be fully explored and few detailed in-cylinder studies have been carried out on the interaction of two-stage fuel injection and its effect on fuel injection quantity, combustion characteristics and exhaust emissions. The objective of this study was to investigate the effects of fuel injection equipment (FIE) characterisation and optimisation on diesel combustion and emissions with two-stage fuel injection. The twin and variable split two-stage injection strategies with variable DA were investigated in a single-cylinder DI high-speed optical diesel engine by means of conventional heat release analysis and high-speed fuel spray and combustion visualisation technique. In order to quantify and compensate for the interactions between the two consecutive fuel injection events in the solenoid CR fuel injector, a fuel injection characterisation rig and fuel bulk modulus measurement device were utilised. Therefore, in the first part of the paper, the principle and application of the fuel injection rate measurement technique is presented. Then the experimental setup and in-cylinder measurement techniques are described. Effects of two-stage fuel injection and the interactions of two consecutive injection events on diesel combustion and emissions are then presented and discussed.

2 FIE CHARACTERISATION

In this study, a fuel injection rate characterisation rig was utilised in order to characterise and optimise the CR fuel injection system for the application of two-stage and multiple injection strategies. The evaluation method employed in this study was based on Zeuch's method presented by Ishikawa et al. [**26**]. The principle of this technique is based on the injection of fuel into a constant volume chamber filled with the selected fuel for the measurements, in this case commercially available diesel fuel. Consequently, the pressure inside the chamber increases, this augmentation is proportional to the quantity of fuel injected. Therefore, the rise in pressure (ΔP) can be determined by the following expression as a function of the change in volume (ΔV),

$$\Delta P = k \frac{\Delta V}{V} \qquad (1)$$

Where k bulk modulus of fuel

V total volume of the chamber

The fuel injection rate can be determined by differentiating equation (1) with respect to time (t) as depicted below,

$$\frac{dV}{dt} = \frac{V}{k} \times \frac{dP}{dt} \qquad (2)$$

In the case of a single injection, fuel quantity and injection duration as a function of injection pulse width could be measured through the application of this technique. In addition, interference between injections could also be identified for two-stage or multiple injections, whereby substantial variations in the quantity and rate of fuel injection could be experienced. Detailed explanation of the experimental setup/procedure can be found in [**24**, **25**].

3 ENGINE AND MEASUREMENT TECHNIQUES

All experimental testing in this study was carried out in a single-cylinder high-speed optical engine equipped with a production cylinder head, designed to be representative of a typical modern high-speed DI diesel engine. The engine specifications are depicted in Table 1. The engine consisted of a Ricardo Hydra engine crankcase, extended cylinder block and piston, and a standard production Ford 2.0 litre ZSD 420 Duratorq cylinder head.

Table 1. Single-cylinder optical engine specifications

Ricardo Hydra Single-Cylinder Engine	
Bore	86 mm
Stroke	86 mm
Swept Volume	499 cm^3
Compression Ratio	16:1
Piston Bowl	43.4/11.6 mm
	Re-entrant bowl with flat bottom
Swirl Ratio (Ricardo)	1.4

The FIE consisted of a fuel filter, a 12V low pressure pump which drew the filtered fuel from the fuel tank, a first generation Bosch high pressure pump and a Delphi CR fuel injector. The injector utilised in this study was a Delphi multi-hole valve covered orifice (VCO) injector capable of injecting up to 1600 bar. The specifications of the FIE are listed in Table 2.

Table 2. Fuel injection system specifications

Injection System	
1st Generation CR System	
Maximum Injection Pressure	1350 bar
Delphi Standard VCO Injector	
Number of Holes	6
Hole Size	0.154 mm
Cone Angle	154°
Flow Rate	0.697 l/min

In this study, a Kistler 6125 piezoelectric pressure transducer was installed in place of the glow plug for in-cylinder pressure measurement and heat release analysis. Optical access was provided by the Bowditch piston design which allowed for the visualisation of the combustion chamber through the axis of the cylinder via a glass window, made from fused silica, mounted in the crown of the piston. An extended piston and cylinder block were required in order to accommodate such an optical configuration which consisted of lower and upper parts with a 45° angled mirror, made of glass with aluminised front surface, between the sections. Therefore, the combustion chamber and cylinder walls could be fully visualised through such an optical setup, Figure 1.

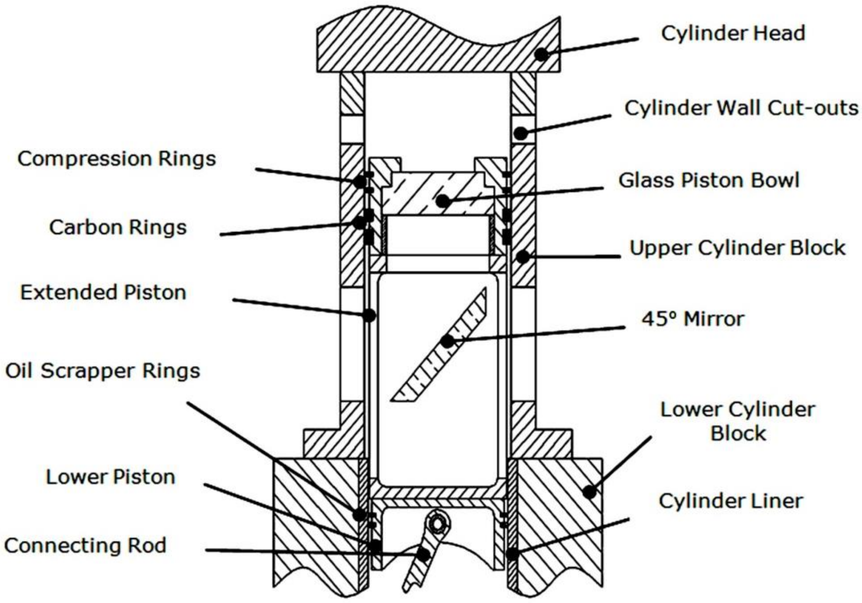

Figure 1. Sectional schematic view of the single-cylinder optical engine

The observation field through the piston crown window is illustrated in Figure 2. In addition, the upper cylinder block had three rectangular wall cut-outs which can be fitted with glass windows, made of fused silica, for side optical access. Two of these windows were in the same plane allowing laser sheet imaging, Figure 1, while the third window was positioned at 90°, premeditated for imaging and detection purposes.

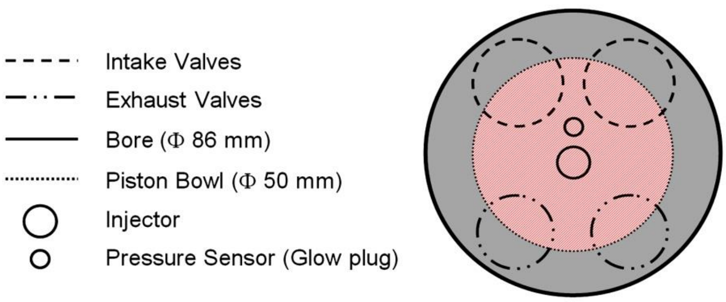

Figure 2. Observation field through piston crown window

3.1 Conventional measurements

In-cylinder pressure data from 20 consecutive engine cycles was recorded for the studied cases, from which the ensemble-averaged net HRR was calculated through the application of the first law of thermodynamics [**27**], equation (3).

$$HRR = \frac{\gamma}{\gamma - 1} P \frac{dV}{dt} + \frac{1}{\gamma - 1} V \frac{dP}{dt} \qquad (3)$$

In addition, the soot and gaseous exhaust emissions were measured to assess the effects of two-stage fuel injection and DA on engine performance and emissions. The gaseous exhaust emissions of CO, CO_2, O_2, uHC and NOx were measured by means of a Horiba MEXA-7170DEGR analyser system while soot emission was measured using an AVL 415 smoke meter.

3.2 High-speed fuel spray and combustion visualisation

High-speed video imaging was employed to record colour images of fuel sprays during the fuel injection period and subsequent combustion process, Figure 3. In order to visualise the fuel sprays, a high repetition copper vapour laser was utilised. The output of the laser was delivered to the engine via an optical fibre so that the whole combustion chamber could be illuminated by the divergent laser beam leaving the optical fibre. The short laser pulse of 30ns effectively defined the exposure time of the spray images, which was critical to obtain sharp images of sprays travelling at very high speeds. A NAC Memrecam FX6000 high-speed colour video camera was utilised which was equipped with a high-speed complementary metal–oxide semiconductor (CMOS) sensor. The high-speed camera was synchronised to the pulsed laser outputs to capture videos at 10,000 frames per second with an image resolution of 512 x 248 pixels. A Nikon 50mm f.1.4 lens was used.

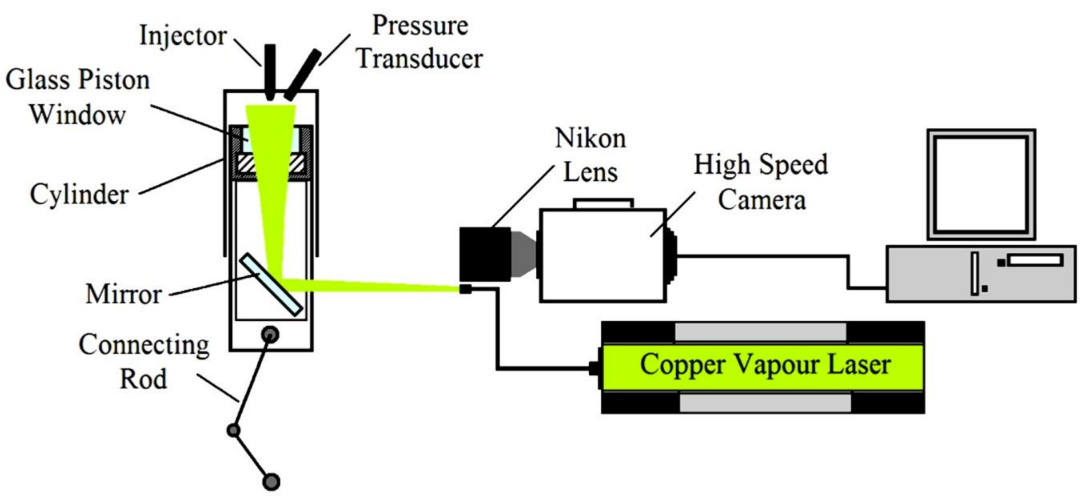

Figure 3. Schematic diagram of direct combustion visualisation setup

A light-emitting diode installed in the field of view of the camera provided the reference signal at TDC in the high-speed video images. It is important to note that although the first frame with visible fuel spray is referred to as the start of injection, the actual start of injection in some of the presented frames may not correspond to this point due to inter frame rate of the high-speed camera utilised.

4 ENGINE OPERATING CONDITIONS

The mechanical and thermal stresses the optical components are exposed to during firing cycles are substantial, thus the operating time of optical engines is limited. As a result, both the coolant and the lubricating oil were heated prior to the experiment to ensure reasonable engine temperature was achieved, this also minimised excessive window fouling. In this study injection pulse widths were set to achieve 50%/50% two-stage fuel injection with variable DA under part load operating conditions at the engine speed of 1500rpm, Table 3.

Table 3. Test conditions

Intake Air	100°C Naturally Aspirated (NA)
Engine Speed	1500 rpm
Fuel	Commercially Available 49.1 CN Diesel
Fuelling Demand	20 mm^3/cycle
Load	≈ 72% of NA Full Load, 27.7:1 AFR
Injection	1200 bar
Piston Bowl	Glass – Pressure, Optical Techniques Metal – Soot, Emissions

The two-stage fuel Injection strategies were selected such that the injection timing of the second injection remained constant at TDC while the injection timing of the first injection was varied. A typical profile of the current that energises the solenoid of the electronically controlled fuel injector is illustrated in Figure 4.

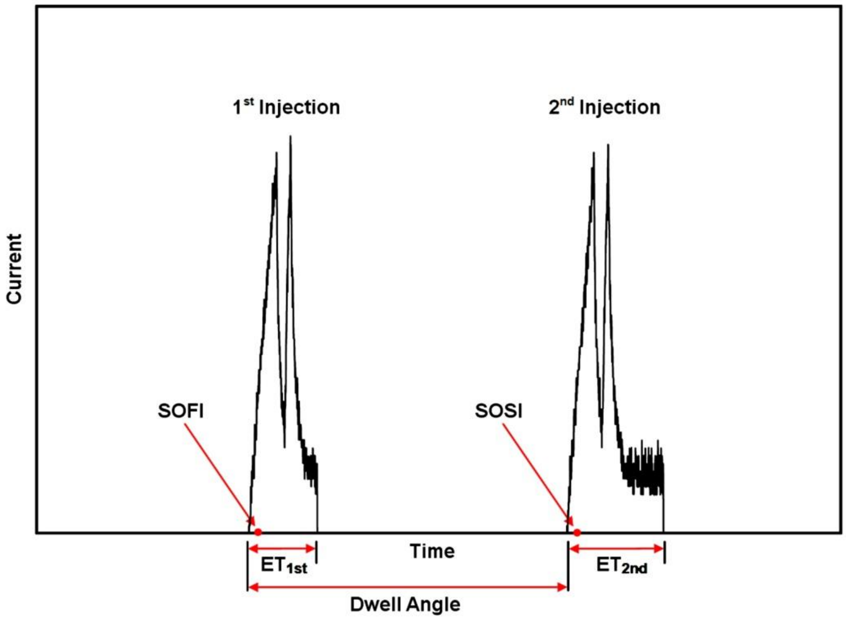

Figure 4. Schematic representation of injector current waveform

The current waveform supplied to the injector can be characterised by the following definitions,

- SOFI: Start of the first injection
- ET_{1st}: Energising time of the first injection
- SOSI: Start of the second injection
- ET_{2nd}: Energising time of the second injection
- Dwell Angle: The time interval between the start of the first injection and the start of the second injection

It is important to note that there exists a time lag between the start of the injector current waveform and the start of fuel injection due to the inertia of the system as illustrated in Figure 4. The authors have previously reported on the hydraulic effects of two-stage injection on fuel injection quantity, combustion characteristics and emissions [**24**, **25**]. It was identified that the fuel injection quantity during the second injection was substantially influenced by the first injection when they were closely spaced in time. Consequently, the FIE was thoroughly calibrated in order to achieve equal fuel injection quantities during the first and the second fuel injection events through the adjustment of the second injection duration. The strategies with identical first and second injection durations are referred to as 50%/50% twin split while the strategies with variable second injection duration are referred to as variable split, Table 4. In addition single injection strategies were performed, serving as the baseline for comparative analysis of the results.

Table 4. Injection strategies

Strategy	Test No	SOI (CAD ATDC)		ET (μs)		DA (CAD)
Single Injection	**A1**	-15		596		N/A
	A2	-10		596		
	A3	-5		596		
	A4	TDC		596		
		1st	**2nd**	**ET1st**	**ET2nd**	
50%/50% Twin Split	**T1**	-25	TDC	422	422	25
	T2	-20	TDC	422	422	20
	T3	-15	TDC	422	422	15
	T4	-10	TDC	422	422	10
50%/50% Variable Split	**V1**	-25	TDC	422	390	25
	V2	-20	TDC	422	478	20
	V3	-15	TDC	422	390	15
	V4	-10	TDC	422	520	10

The fuel injection rate profiles of the twin and variable split fuel injection strategies are presented in Figure 5.

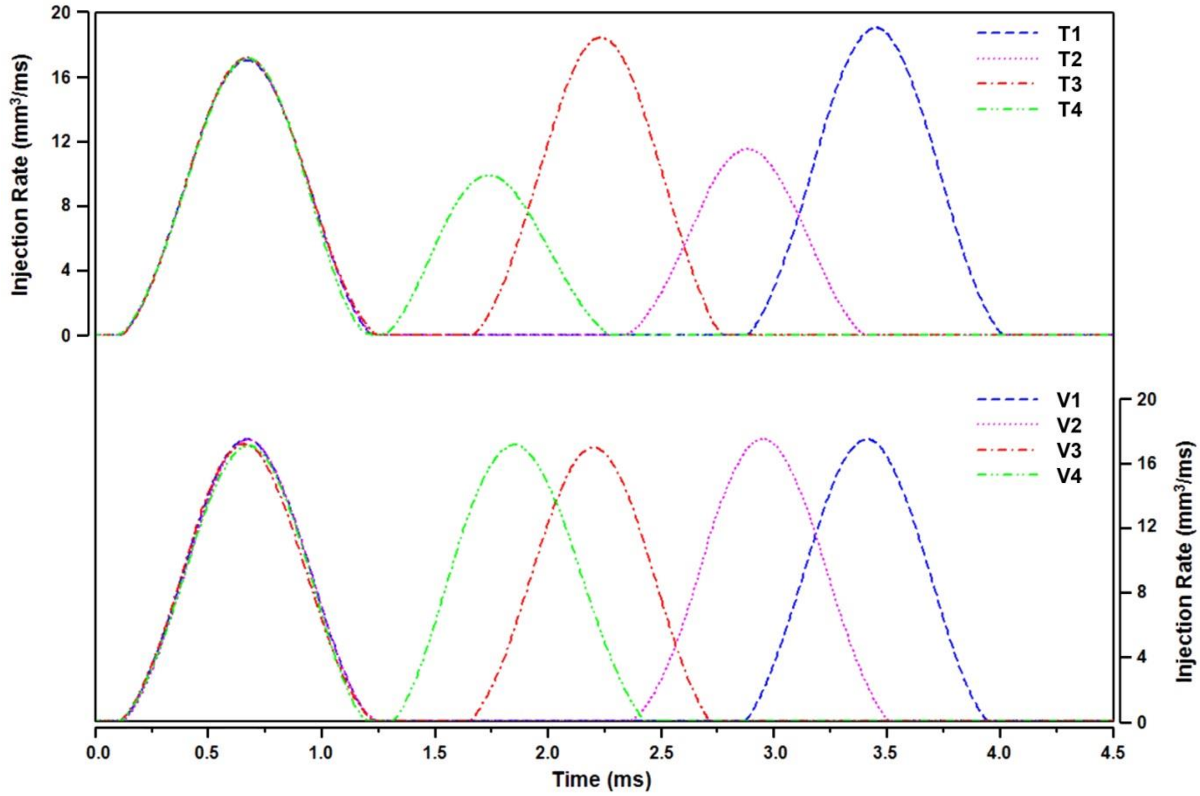

Figure 5. Injection rate profiles of variable and twin split injection strategies

It should be noted that the twin split injections were obtained with the same energising time for the first and second injections and the difference in the fuel injection quantity in the second injection was mainly attributed to the design limitations of the injector utilised as well as the adverse effects of pressure waves in the high pressure fuel line, caused by the first injection [**24**, **25**]. Consequently, the FIE was thoroughly calibrated and the second fuel injection pulse width was optimised such that the quantity of fuel injected during the second injection was identical to that of the first injection, Figure 5 and Table 5. Thus, true effects of 50%/50% two-stage fuel injection with variable DA on diesel combustion and emissions can be identified.

Table 5. Fuel injection quantity

Test Number	Injection Pressure (bar)	1st Fuel Injection Quantity (mm^3)		2nd Fuel Injection Quantity (mm^3)		Total Fuel Injection Quantity (mm^3)		Total Fuel Quantity Percentage Error (%)
		Desired	Actual	Desired	Actual	Desired	Actual	
T1	1200	10.0	10.0	10.0	11.3	20.0	21.3	+6.5
T2	1200	10.0	10.0	10.0	8.7	20.0	18.7	-6.5
T3	1200	10.0	10.0	10.0	11.6	20.0	21.6	+8.0
T4	1200	10.0	10.0	10.0	7.3	20.0	17.3	-13.5
V1	1200	10.0	10.0	10.0	10.0	20.0	20.0	0.0
V2	1200	10.0	10.0	10.0	10.0	20.0	20.0	0.0
V3	1200	10.0	10.0	10.0	10.0	20.0	20.0	0.0
V4	1200	10.0	10.0	10.0	10.0	20.0	20.0	0.0

5 RESULTS AND DISCUSSION

The calculated and measured data for the injection and combustion characteristics are listed in Table 6. The Injection duration for each injection strategy was determined based on the injection rate profiles obtained through Zeuch's method while the IMEP values and the start of combustion timings were calculated from the in-cylinder pressure and HRR data respectively. The Ignition delay was also determined from the HRR data as the time between the start of the first injection and the onset of the main HRR.

Table 6. Injection and combustion characteristics

Test Number	Injection Pressure (bar)	Total Injection Duration (CAD)	IMEP (bar)	Start of Main Combustion (CAD ATDC)	Ignition Delay (CAD)
A1	1200	5.4	3.06	-6.6	8.4
A2	1200	5.4	3.17	-2.6	7.4
A3	1200	5.4	3.46	2.6	7.6
A4	1200	5.4	4.43	12.6	12.6
T1	1200	7.6	3.62	-13.2	11.8
T2	1200	7.6	2.65	-12.2	7.8
T3	1200	7.6	4.24	-8.6	6.4
T4	1200	7.6	1.98	-5.2	4.8
V1	1200	7.3	4.04	-14.6	10.4
V2	1200	8.1	5.16	-11.6	8.4
V3	1200	7.3	4.81	-8.2	6.8
V4	1200	8.5	5.47	-3.6	6.4

The in-cylinder pressure and HRR data for the single injection strategies are illustrated in Figure 6. The peak in-cylinder pressure decreased as the injection timing was retarded. This was mainly attributed to late initiation of the combustion process during the expansion stroke where the piston was descending after TDC due to retarded fuel injection timing.

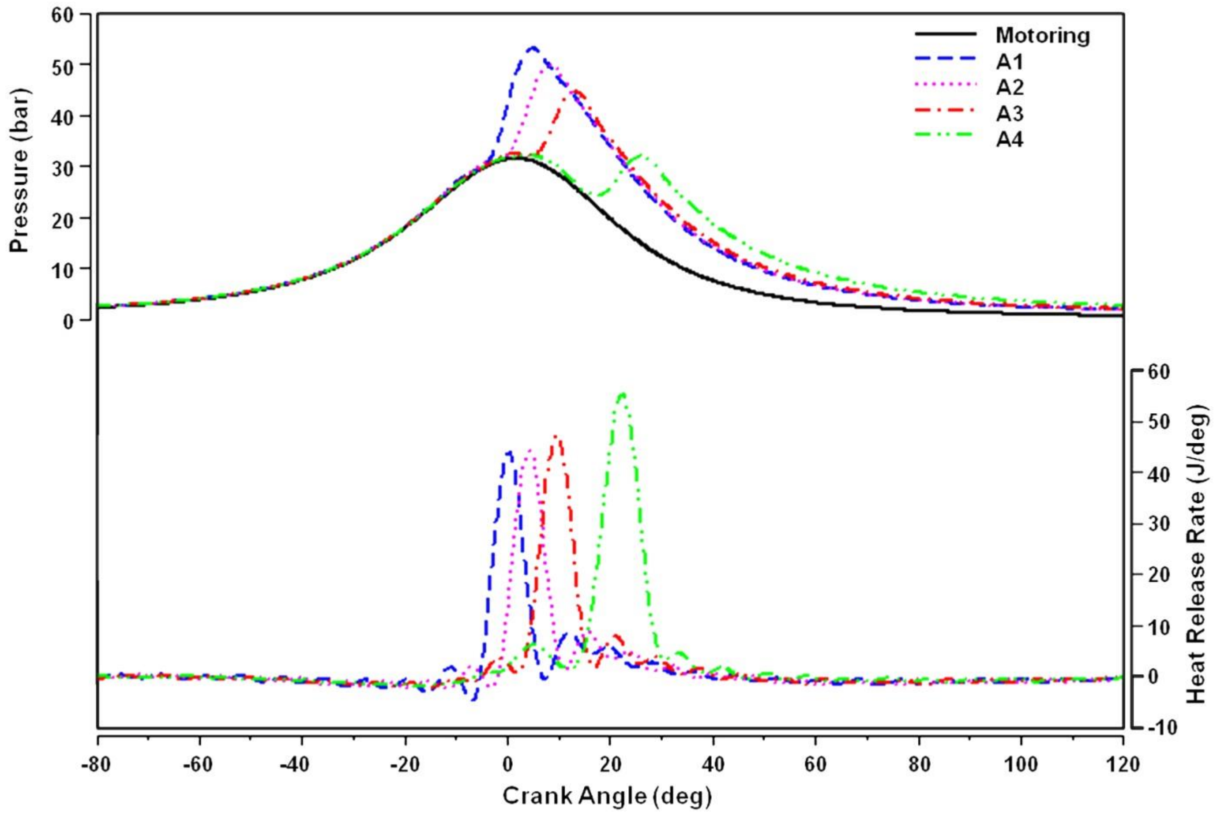

Figure 6. In-cylinder pressure and HRR for single injection strategies

The HRR curves exhibited a drop due to charge cooling effect shortly after the onset of the fuel injection. Subsequently, the rate of heat release rapidly increased due to the premixed combustion. As given in Table 6, the ignition delay period decreased and the percentage of premixed combustion lessened as the injection timing was retarded towards TDC, A1-3. The reduction in the amount of heat release from the fast premixed combustion with late injection was compensated for by the faster heat release of combustion in a smaller volume near TDC. As a result, the peak HRR remained almost constant for A1, A2 and A3 strategies. However, this trend was not observed for A4 strategy. In this case the ignition delay was much longer, thus more heat release took place due to premixed combustion which resulted in greater HRR. This was mainly due to initiation of the combustion during the expansion stroke when the in-cylinder pressure and temperature were relatively lower. However, the peak in-cylinder pressure was lower as the combustion occurred in a larger cylinder volume.

The in-cylinder pressure and HRR data for both variable and twin split strategies are depicted in Figures 7 and 8 respectively. As shown in these figures, when the two-stage injection was employed, in-cylinder pressure and HRR curves exhibited different trends to those of single injections. In the case of twin split, the peak in-cylinder pressure first increased as the first injection was retarded from 25 to 15 crank angle degree (CAD) BTDC; T1-T3; and then dropped slightly in the cases of 10 CAD DA, T4. The former was mainly due to improved fuel evaporation and mixing process as the first fuel injection took place closer to TDC when the in-cylinder pressure and temperature were higher while the latter was mainly due to late initiation of combustion process during the expansion stroke where the piston was descending after TDC due to retarded fuel injection timing. Nevertheless, the trend observed in the peak in-cylinder pressure was not solely due to the differences in the in-cylinder thermodynamic conditions, variations in the actual total fuel injection quantity, Figure 5, played an important part.

The effect of inconsistencies in the total fuel injection quantity is evident in Figure 7, whereby the in-cylinder pressure tracers for the variable split strategies exhibited a different trend. The peak in-cylinder pressure first increased as the first injection was retarded from 25 to 20 CAD BTDC, V1-V2, and then progressively decreased as the first injection timing was further retarded, V3 and V4. The former was due improved fuel evaporation and mixing process as the first fuel injection took place closer to TDC when the in-cylinder pressure and temperature were higher while the latter was due to late initiation of the combustion process during the expansion stroke where the piston was descending after TDC due to retarded fuel injection timing. The change in the in-cylinder pressure was much more consistent since identical fuel quantities were injected during the first and the second injections.

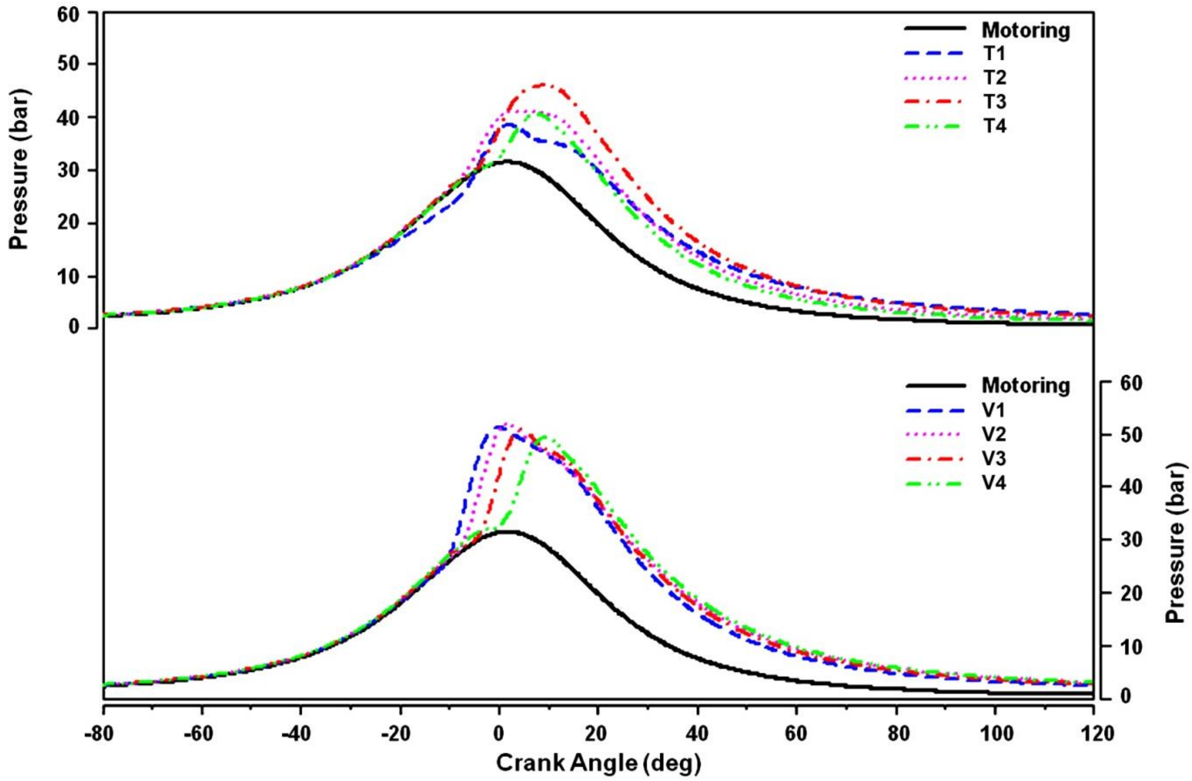

Figure 7. In-cylinder pressure for 50%/50% two-stage injection strategies

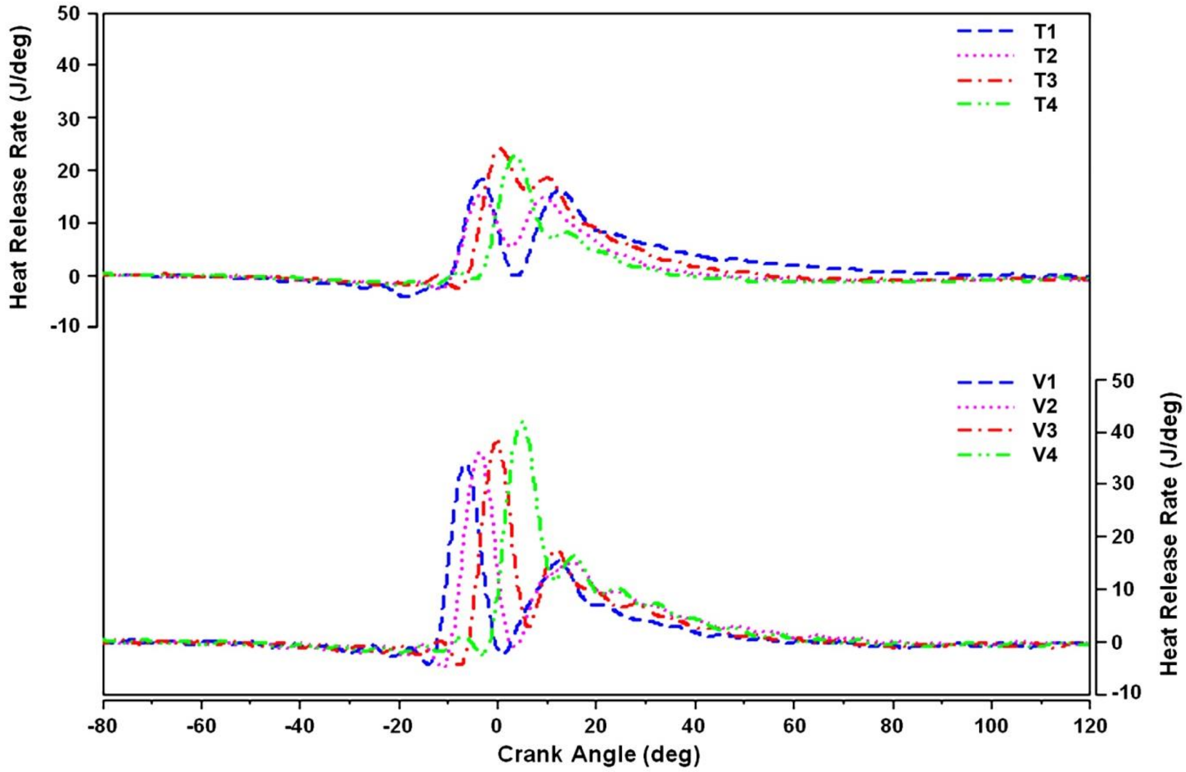

Figure 8. In-cylinder HRR for 50%/50% two-stage injection strategies

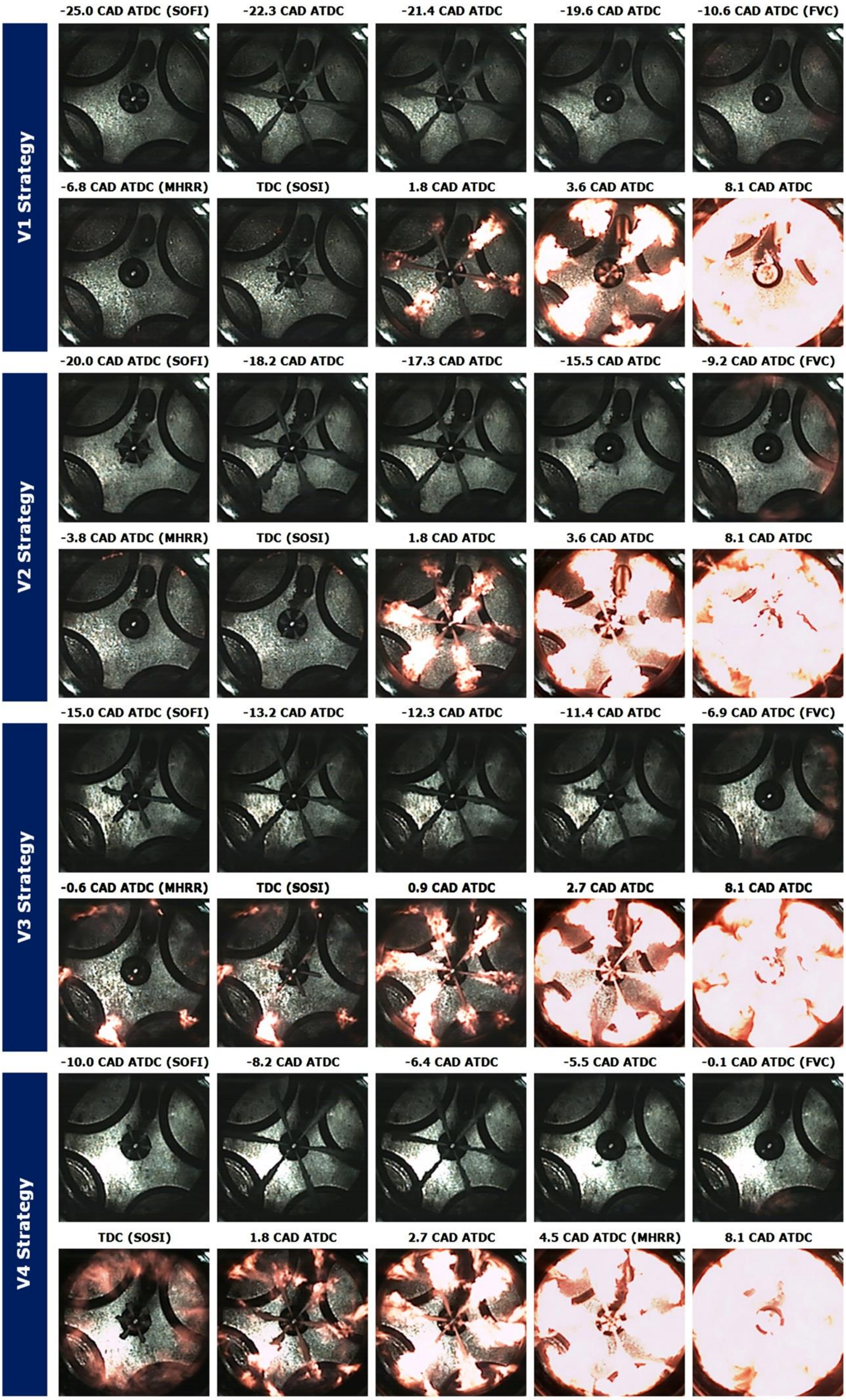

Figure 9. Combustion image sequence of variable split strategies

The HRR curves in Figure 8 exhibited a drop due to charge cooling effect shortly after the onset of the first fuel injection. Subsequently, the rate of heat release rapidly increased due to the premixed combustion. As shown in Table 6, the ignition delay period decreased, due to improved fuel evaporation and mixing effects closer to TDC, as the injection timing was retarded. However, unlike the single injection strategies the percentage of combustion during the peak heat release period increased with shorter ignition delay. In order to further examine this phenomenon and to establish the underlying causes, high speed fuel spray and combustion visualisation technique was applied, Figure 9, in which SOFI, first visible combustion (FVC), SOSI and the maximum heat release rate (MHRR) for V1, V2, V3 and V4 strategies were studied and compared.

The asymmetry of the fuel sprays during the initial stages of fuel injection is evident in Figure 9. At low needle lift condition the VCO injectors suffer from uneven distribution of pressure field inside the nozzle holes due to eccentric or radial motion of the needle tip, caused by the close proximity of the needle seat to the nozzle holes. The high speed images in Figure 9 revealed that the increase in the percentage of combustion near and shortly after TDC during the peak HRR period was due to the interaction between the first and the second injections, whereby the fuel sprays during the second injection were injected into burning regions caused by the combustion of premixed fuel and air from the first injection, generating fuel-rich combustion. As a result, the premixed combustion of fuel from the first injection was intensified by the participation of the second injection. The effect of such an interaction became more pronounced as the first injection timing was moved closer to TDC, where higher percentage of premixed combustion took place prior to the onset of the second fuel injection, Figure 8. Furthermore, this phenomenon was more accentuated for the variable split cases since more fuel was injected during the second injection, Figure 9. Unlike T1-4 cases, the maximum HRR for the variable split cases steadily increased as the first injection timing was retarded. Thus, it is evident that the design limitations of the FIE as well as the possibility of interactions between the fuel injections during the combustion are amongst the complications that may be encountered in the application of two-stage or multiple injection strategies. Therefore, thorough characterisation of the FIE and careful selection of injection strategies are essential for satisfactory implementation of such advanced injection modes.

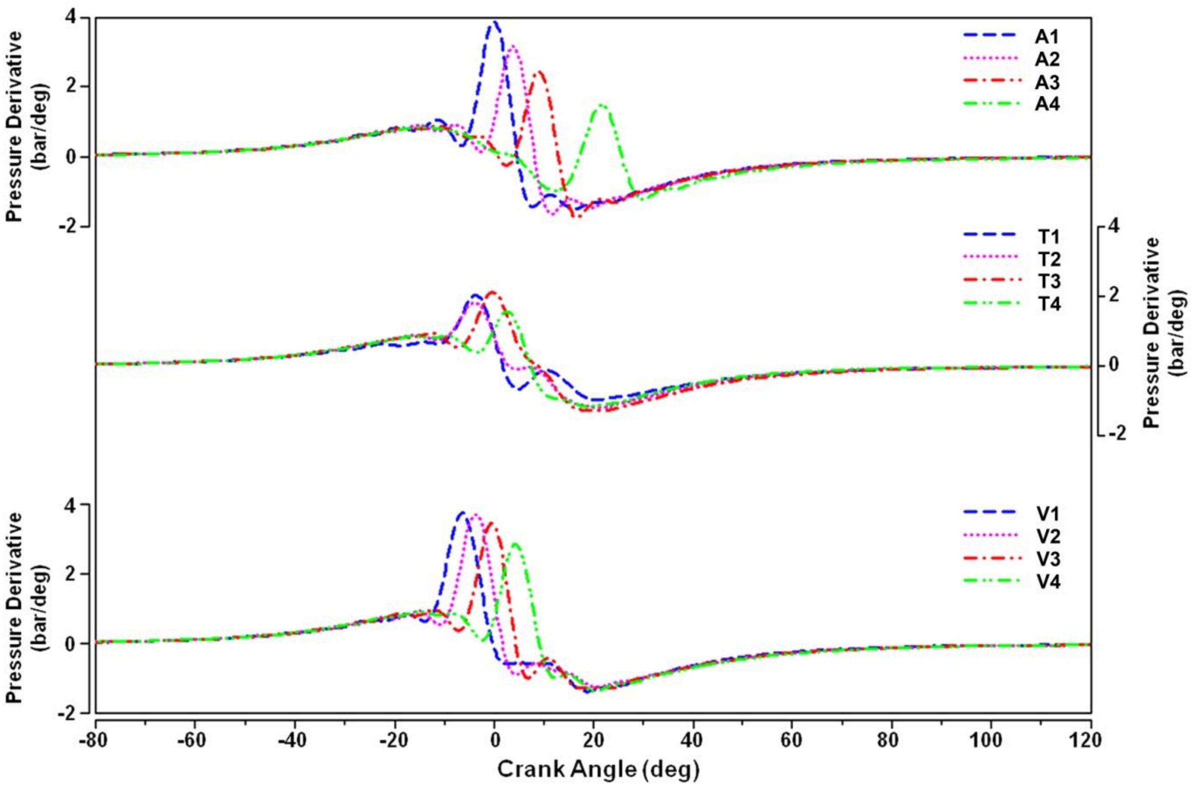

Figure 10. In-cylinder pressure derivative for single and two-stage injection strategies

The in-cylinder pressure derivative for the studied single and two-stage injection strategies is presented in Figure 10. The in-cylinder pressure derivative is an indication of the combustion noise and the mechanical stresses the engine parts are exposed to. The rate of in-cylinder pressure rise steadily decreased as the fuel injection timing was retarded, mainly due to late initiation of the combustion process. As evident in Figure 10, the quantity of fuel injected during the second injection had a significant impact on the rate of in-cylinder pressure rise. Unlike T1-4 strategies, V1-4 strategies exhibited higher rate of pressure rise in comparison to those of single injection strategies, mainly due to higher degree of interaction between the first and the second injections. The results suggest that the combustion noise and also the mechanical stresses the engine parts are exposed to can be notably decreased with two-stage fuel injection. However, these aspects depend on the fuel injection strategy, fuel injection quantity and the engine speed, thus thorough assessment of such a phenomenon under various engine operating conditions is required in order to draw an explicit conclusion.

The soot, NOx and uHC emissions results are depicted in Figure 11. In the case of the twin split strategies the substantial variations in the engine output and exhaust emissions were primarily caused by the inconsistency in the total fuel injection quantity, due to the effect of tow-stage injection and the DA, as well as the interaction between the first and the second injections. Though, in the case of the variable split strategies simultaneous reduction in soot and NOx emissions was observed as the injection timing of the first injection was retarded from 25 to 15 CAD BTDC, V1-3, mainly due to improved mixing and air utilisation. However, in the case of V4 strategy both NOx and soot emissions increased, the former was attributed to poor soot oxidation due to late initiation of the combustion while the latter was due to high level of interaction between the first and the second injections whereby at the start of the second injection intense flame was present due to the combustion of premixed fuel and air caused by the first injection, Figure 9. It is important to note that although NOx emission was lower than single injection strategies, due to greater control on the combustion process, soot emission was higher. This was mainly due to high degree of interaction between the first and the second injections which was even more pronounced in the case of variable split strategies since more fuel was injected during the second injection.

In the case of V1 strategy, the uHC emission was significantly higher due to poor fuel evaporation and mixing since the first fuel injection took place at 25 CAD BTDC when the in-cylinder pressure and temperature were relatively lower. However, as the injection timing was retarded from 20 to 15 CAD BTDC, V2 and V3 strategies, uHC emission progressively decreased, mainly due to improved fuel evaporation and mixing processes. Though, in the case of V4 strategy uHC emission increased due to late initiation of the combustion during the expansion stroke which firstly deteriorated the fuel evaporation and mixing processes and secondly shortened the time available for the combustion to complete.

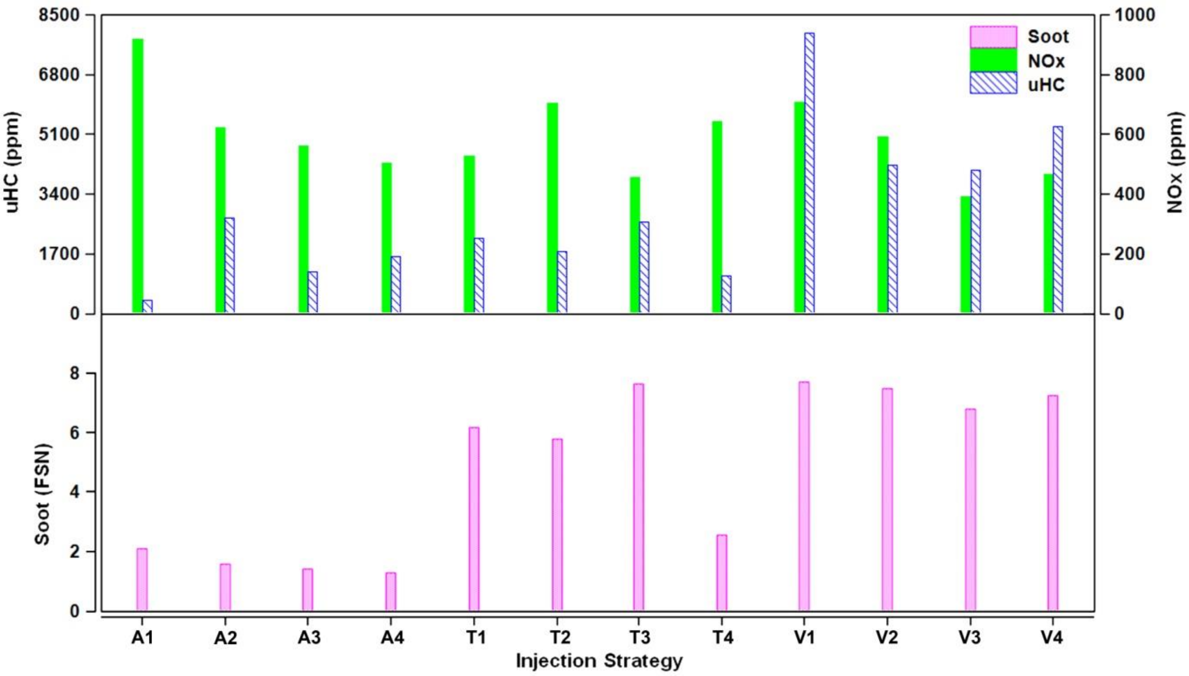

Figure 11. Exhaust emissions for single and two-stage injection strategies

6 CONCLUSIONS

In this paper, the effects of FIE characterisation and optimisation on fuel injection system performance and in-cylinder mixture formation, combustion and emissions, with variable and twin split fuel injection, were investigated in a single-cylinder CR fuel injection optical diesel engine. The fuel injection rate and interaction between two consecutive fuel injection events were quantified and compensated for through the adjustment of the second injection duration. The in-cylinder process was studied by means of detailed heat release analysis and high-speed fuel spray and combustion visualisation technique. The main findings can be summarised as follows: regarding the studied two-stage injection strategies with variable DA at 1200 bar injection pressure and the engine speed of 1500 rpm.

- Significant interactions were found between two closely spaced two-stage injections in a solenoid CR fuel injector, mainly due to the limitations of the injector utilised and the presence of pressure waves in the fuel line. The FIE needed to be properly calibrated such that the fuel injection quantity during the first and the second injections were identical.

- The in-cylinder high-speed fuel spray and combustion visualisation technique provided direct evidence of the interaction between combustion and fuel injection. The direct fuel injection into the premixed combustion from the first injection led to more pronounced HRR seen amongst the studied strategies. This phenomenon was even more pronounced for the variable split cases since more fuel was injected during the second injection.

- The results from single and two-stage strategies were compared, indicating the possibility of simultaneous reduction of NOx and soot emissions with two-stage injection. Nevertheless, the studied two-stage strategies resulted in higher soot emission, mainly due to the interaction between two consecutive fuel injection events, whereby the fuel sprays during the second injection were injected into burning regions as well as reduced soot oxidation which was further exacerbated in the case of variable split strategies since more fuel was injected during the second injection. In addition, the variable split cases produced high levels of uHC emission in comparison to single and twin split cases. This was mainly attributed to firstly greater fuel quantity injected during the second injection and secondly poor mixing and air utilisation during the second fuel injection event.

Although the engine operating conditions and the injection parameters investigated in this study were different to those previously reported in the literature, the findings are in good agreement with the findings previously reported by other research groups [**3-5**, **10**, **11**, **16**]. Nevertheless, an explicit conclusion on the advantages of two-stage fuel injection in comparison to that of conventional single injection can only be drawn provided that the effects of such an injection mode under various engine operating conditions are thoroughly assessed.

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APPENDIX

Notation

ΔP	change in pressure
ΔV	change in volume
γ	ratio of specific heats
k	bulk modulus
P	pressure
t	time
V	volume

Abbreviations

AFR	air-fuel ratio
ATDC	after top dead centre
BTDC	before top dead centre
CAD	crank angle degree
CMOS	complementary metal–oxide semiconductor
CR	common rail
DA	dwell angle
DI	direct injection
EGR	exhaust gas recirculation
ET	energising time
FIE	fuel injection equipment
FSN	filter smoke number
FVC	first visible combustion
HCCI	homogenous charge compression ignition
HRR	heat release rate
MHRR	maximum heat release rate
NA	naturally aspirated
NOx	nitrogen oxides
PCCI	premixed charged compression ignition
SHR	single stage heat release
SOFI	start of first injection
SOSI	start of second injection
TDC	top dead centre
TSC	two-stage combustion
uHC	unburned hydrocarbon
VCO	valve covered orifice