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Performance Output of Double-Injection Gasoline Compression Ignition (GCI) Combustion in a Common-rail Diesel Engine

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Abstract

The use of low ignition quality fuel in gasoline compression ignition (GCI) engines to form the partially premixed charge during the extended ignition delay period has demonstrated significantly improved engine efficiency and lower smoke/NO_x emissions in many previous investigations. The major advantage of GCI combustion over other advanced regimes achieving similar goals such as kinetics-controlled homogenous-charge compression-ignition (HCCI) engines is a close coupling between the fuel injection event and the combustion phasing, a much needed characteristic for practical engine applications. In the present work, the GCI engine tests were emphasised on a direct comparison between single and double injection strategies at fixed engine operating conditions of 1600 rpm and ~910 kPa net indicated mean effective pressure ($IMEP_n$). The tests were carried out in a single-cylinder light-duty diesel engine equipped with a conventional common-rail fuelling system, which is connected with an Eddy Current (EC) dynamometer. It is found that the double injections implementing the early 170°CA bTDC injection and the late near-TDC injection results in overall smoother pressure traces and lower apparent heat release rates than those obtained from the single injection at 30~10°CA bTDC. While keeping the low 3% coefficient of variations of IMEP, these in-cylinder characteristics of the double injection lead to higher net indicated efficiency by 93% and lower indicated specific fuel consumption (ISFC) by 48%, clearly demonstrating the advantages of independently controlling the mixture premixedness using the early first injection and the combustion phasing with the late second injection.

Introduction

One of the challenging issues in developing next-generation diesel engines is to simultaneously reduce the regulated emissions of nitrogen oxides (NO_x) and particulate matters with no fuel economy penalty. To alleviate this concern, advanced combustion regimes such as low temperature combustion (LTC) diesel [1] and homogenous charge compression ignition (HCCI) [2] have been investigated, all of which aim at increased pre-combustion mixing and reduced flame temperature. LTC utilises a very high rate of exhaust gas recirculation (EGR) to reduce the flame temperature below the soot and NO formation thresholds but the high-load operations are limited due to low power output [3, 4]. The operation range of HCCI engines is also limited because this concept relies on advanced injection timing in the intake stroke for a fully premixed homogeneous charge and therefore a high pressure rise rate and pressure ringing occurs at high loads [4, 5]. More importantly, HCCI is kinetics-controlled [5] and thus the combustion phasing control is not possible. HCCI variants such as stratified charge compression ignition (SCCI) [6] and premixed charge compression ignition (PCCI) [4, 7] have shown to address the high-load limit and combustion control issues, respectively, but they have been only partially successful.

A more practical combustion regime that can overcome aforementioned issues in LTC and HCCI engines is gasoline compression ignition (GCI) in which a low ignition quality fuel (e.g. gasoline) is directly injected into the combustion chamber using a conventional common-rail fuelling system in a diesel engine [8, 9]. The extended ignition delay associated with the use of low ignition quality fuel realises a needed level of pre-mixing for reduced NO_x and particulate emissions [8, 9]. In GCI engines, the direct injection of gasoline or other low cetane number (or high octane number) fuels, which is executed closer to TDC than HCCI or SCCI but earlier than conventional diesel combustion, is believed to cause a partially premixed charge [10]. Therefore, the combustion phasing is closely coupled with the fuel injection event [11] offering a practical advantage over HCCI/SCCI. Compared with LTC regimes, the GCI does not use a high EGR rate and thus overcoming the high-load limit [8, 9]. The moderate EGR is often used to reduce the NO_x emissions of GCI combustion but the particulate emissions are not badly deteriorated [12].

In this paper, the GCI engine tests were performed to compare single and double injection strategies. Of particular interest is the variation of injection timings that is expected to impact the combustion phasing and subsequently engine performance. The in-cylinder pressure traces and net indicated mean effective pressure (IMEP) were measured for various injection timings. The apparent heat release rate (aHRR) and pressure rise rate (PRR) traces were also obtained from the measured in-cylinder pressure traces, which together with the measured injection rate were used to evaluate the net indicated power, net indicated engine efficiency and indicated specific fuel consumption (ISFC).

Experiments

Gasoline has high resistance to auto-ignition which makes it a low-ignition quality fuel (i.e. low cetane number) and thus imparts extended ignition delay [8, 9]. The physical properties of a conventional gasoline fuel is summarised in table 1. The gasoline used in this study has research octane number (RON) of 91 with low cetane number of 10~20 and is easily available in Australian petrol stations.

The schematic diagram of a naturally-aspirated single-cylinder common-rail diesel engine for GCI combustion applications is shown in Fig. 1. The direct-injection light-duty diesel engine shares the production engine head but three of the four cylinders were deactivated. As listed in table 2, the engine has a single-cylinder displacement volume of 497.8 cm^3 with bore and stroke of 83 mm and 92 mm, respectively. The geometric compression ratio is 17.7 and the swirl ratio is 1.4 according to manufacturer's specification. The shape of the piston crown is cylindrical bowl as depicted in the cross-sectional view of the combustion chamber in Fig. 1. In order to reduce the pressure fluctuations (one of the prominent issues of single-cylinder engines) in the intake and exhaust manifold, two 60-litre surge tanks are positioned on the intake and exhaust sides. The intake air temperature was fixed at 80°C. The temperature of coolant that was circulated through the cylinder head and liner was maintained at 90°C using a water heater/circulator (ThermalCare Aquatherm RA Series). A conventional common-rail fuel injection system (Bosch CP3) was used to provide 50 MPa of injection pressure for the direct injection of gasoline via a solenoid-type injector. The injector nozzle has seven holes with nominal hole diameter of 134 µm, the included angle of 150°, and the Bosch K-factor of 1.5. The nozzle is mini-sac type with a convergent, hydro-eroded orifice and a discharge coefficient of 0.86 (i.e. KS1.5/86 according to Bosch specifications). The hydraulic flow rate (HFR) was 400 cm^3 and is measured for time duration of 30 s. The common-rail pressure, injection timings, the number of injections and injection duration were accurately controlled by electronic injection control unit (Zenobalti ZB-9013P). The engine was connected to an Eddy Current (EC) dynamometer (FroudeHoffmann AG-30HS) for constant speed tests [13].

The GCI engine operating conditions are also summarised in table 2. For the single injection strategy, the injection timing was varied from 30 to 10°CA bTDC. The range was limited due to knocking and misfiring issues at the respective extremes which caused unstable engine operations. For the double injection strategy, the early first-injection timing was fixed at 170°CA bTDC to provide partially pre-mixed charge condition similar to Ref. [13] while the late near-TDC second injection executed at between 12°CA bTDC and 3°CA aTDC was for triggering the combustion [10]. While the injection conditions were varied, the net IMEP was measured at about 910 kPa. For the same level of net IMEP, however, the single injection required higher injected mass as noted in table 2. The injected fuel mass was measured using a Bosch tube-type injection rate meter [13]. During GCI combustion tests, the in-cylinder pressure traces were recorded using a piezo-electric pressure transducer (Kistler 6056A1).

Results and Discussion

Figure 2 shows the in-cylinder pressure, pressure rise rate (PRR) and apparent heat release rate (aHRR) for the single injection (left) and double injection strategies (right). The results are plotted for various injection timings as discussed previously in table 2. For both injection strategies, a clear trend is found such that the advanced injection leads to the increased in-cylinder pressure and PRR. At the same time, the combustion occurs at earlier crank angles (i.e. more advanced combustion phasing). The observed trends clearly demonstrate that the combustion phasing of GCI combustion is closely coupled with the injection timing as in conventional diesel combustion. As mentioned previously, the control of combustion phasing through the variations in injection timing is a critical factor for practical applications, which indicates the advantages of GCI over other advanced combustion regimes. Between the two injection strategies, the double injection shows overall lower in-cylinder pressure and PRR, likely due to the increased charge premixing and combustion occurring in locally leaner mixtures.

The aHRR traces shown at the bottom of Fig. 2 provides more detailed information about GCI combustion and its dependency on the single and double injection strategies. For the single injection, the peak of aHRR increases when the injection timing is advanced from 10 to 20°CA bTDC. The advanced combustion phasing and hence the main combustion event occurring closer to TDC explain this increase. However, as the injection timing is further advanced to 25 and 30°CA bTDC, the peak of aHRR decreases. This is because the significantly increased premixing

Property / Fuel		Gasoline
Density (@15°C), kg/m^3		730
Lower heating value, MJ/kg		44.4
Viscosity (@40°C), mm^2/s		0.75
Research octane number (RON)		91
Cetane number		10-20
CHO wt.%	C	84.21
	H	15.79
	O	0

Table 1. Fuel properties

Engine Specifications			
Displacement		497.8 cm^3	
Bore		83 mm	
Stroke		92 mm	
Compression ratio		17.7	
Swirl ratio		1.4	
Injection system		7-hole Bosch common-rail Nominal hole diameter: 134 µm Included angle: 150° K-factor: 1.5 Discharge coefficient: 0.86 HFR: 400 cm^3 for 30s	
Operating Conditions			
Coolant temperature [°C]		90	
Intake air temperature [°C]		80	
Engine speed [rpm]		1600	
Injection pressure [MPa]		50	
Fuel injection strategy		Single	Double
Injection timing [°CA bTDC]	1^{st} injection	30, 25, 20, 15, 10	170
	2^{nd} injection	-	12, 9, 6, 3, 0, -3
Total injection mass [mg]		36	20.2
Overall equivalence ratio [ϕ]		0.96	0.53
Net IMEP [kPa]		~910	

Table 2. Engine specifications and operating conditions

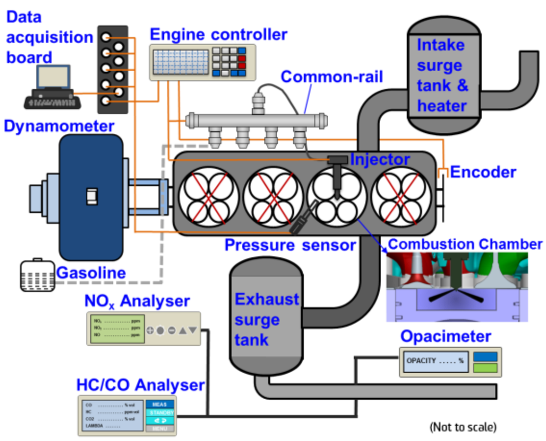

Figure 1. Schematic diagram of a single-cylinder diesel engine and diagnostic tools for GCI combustion experiments

caused locally leaner mixtures, which reduced the rate of heat release. The in-cylinder pressure and PRR traces for the single injection do not show this decreasing trend because the combustion occurs very close to TDC. For the double injection strategy, the aHRR shows a lower value than that of the single injection, similar to the in-cylinder pressure and PRR. As mentioned previously, the increased premixing due to the early first injection at 170°CA bTDC resulted in this lower aHRR. A noticeable trend from Fig. 2 for the double injection is the second peak in the aHRR traces, which is lower than the first peak occurring near TDC. It was thought that the first peak of aHRR is due to combustion of the premixed charge of the first injection whereas the second peak is due to combustion of the remaining mixtures of the first injection as well as the second injection.

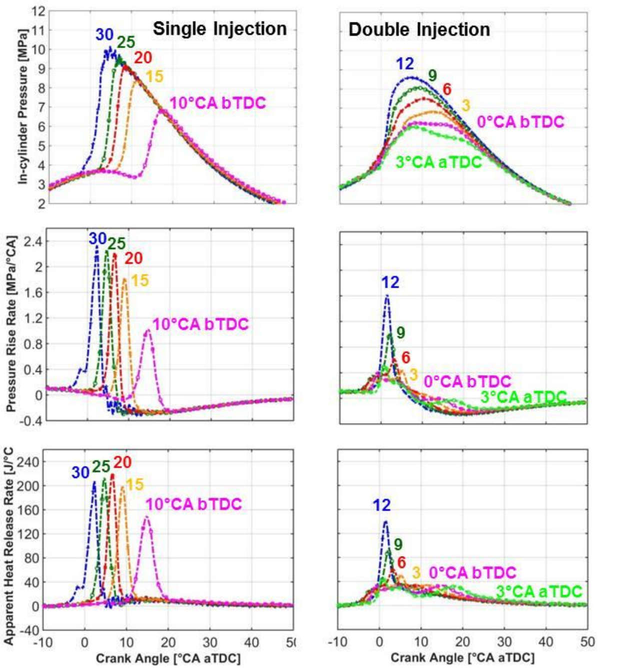

Figure 2. Effect of injection timings on the in-cylinder pressure, PRR and aHRR on GCI engine combustion for the single and double injection strategies. The first injection timing was fixed at 170°CA bTDC for the double injection.

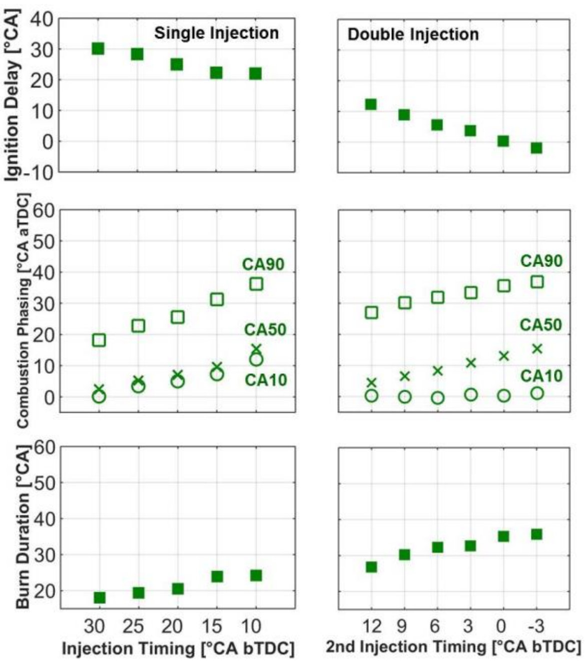

Figure 3. Effect of injection timings on ignition delay, combustion phasing and burn duration of GCI engine combustion for the single and double injection strategies

The aHRR of Fig. 2 was further analysed to derive the ignition delay, combustion phasing, and burn duration, as shown in Fig. 3. The combustion phasing was characterised by measuring the crank angle corresponding to 10%, 50%, and 90% heat release (i.e. CA10, CA50, and CA90). The ignition delay is shown for the crank angles between the start of injection and CA10 while the burn duration is calculated for the crank angles between CA10 and CA90. The estimated ignition delay suggests that the single injection had longer premixing time than the double injection. However, the ignition delay was shorter for the double injection simply because the time between the start of the second injection and CA10 was calculated. If the fact that a half of the fuel was delivered at early 170°CA bTDC, it was thought that the charge premixedness would be higher for the double injection. Figure 3 indeed shows that the CA50 and CA90 are very similar between the two injection strategies but the burn duration is longer for the double injection strategy, which indicates a higher level of premixing and locally leaner mixtures. With the advancement of injection timings, both injection strategies show increased ignition delay, earlier combustion phasing, and shorter burn duration (or higher burning rate).

The calculated net IMEP and coefficient of variation of IMEP using the measured in-cylinder pressure is shown in Fig. 4. The response of GCI combustion to the variations of injection timings is very different between the single and double injection strategies. For example, the advanced injection timing of the single injection leads to the decreased IMEP, which also results in the increased CoV of IMEP. As it was shown in the in-cylinder pressure traces (Fig. 2), the main combustion occurring earlier than TDC caused negative work, which decreased the IMEP. When the injection timing was very late at 10°CA bTDC, the over-retarded combustion phasing and the heat release occurring late in the expansion stroke also decreased IMEP, resulting in the highest IMEP measured for 20 and 15°CA bTDC injection timings. By contrast, the net IMEP shows a monotonically increasing trend with advanced injection timings, which is attributed to the increased in-cylinder pressure and the main combustion occurring after TDC. The CoV of IMEP for the double injection is remained low at about 4%, which indicates stable engine operations.

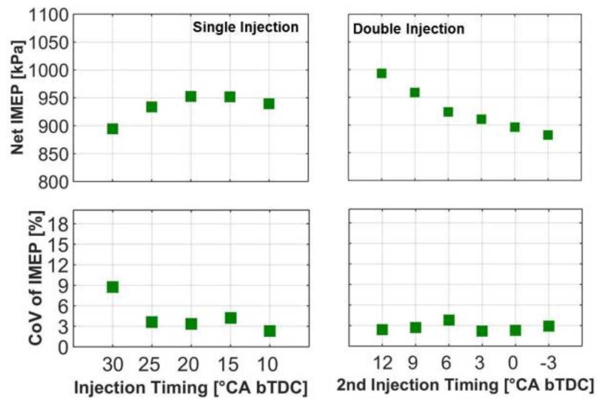

Figure 4. Effect of injection timings on the net IMEP and CoV of IMEP of GCI engine combustion for the single and double injection strategies

As mentioned previously, the injected fuel mass was higher for the single injection compared to the double injection strategy when a similar power output was achieved. Figure 5 shows the net indicated power, which is a simple conversion of IMEP data shown in Fig. 4, and the estimated net indicated engine efficiency and indicated specific fuel consumption (ISFC). It is clearly seen that the overall engine efficiency is nearly two fold higher for the double injection strategy than that of the single injection. This means much higher fuel consumption as shown in the ISFC plot. These significant differences were thought to be due to the over penetration of gasoline that impinged on the cylinder liner wall – i.e. wall wetting [13]. It was noted that the liquid length of gasoline fuel at early crank angles of the single injection strategy could be long enough to reach to the cylinder liner of the light-duty engine used in the present study. This wall wetting issue was resolved when only a half of the fuel was injected at the early timing and the other half was injected near TDC. The double injection strategy of the present study therefore was effective to form a partially premixed charge as it was intended for the realisation of GCI combustion while controlling the combustion phasing as well as reducing the potential wall-wetting issue in a small-bore engine.

Conclusion

Partially pre-mixed combustion realising gasoline compression ignition (GCI) in a naturally aspirated single-cylinder diesel engine has been successfully tested. The results show that the double injection causes a good balance between the premixed charge and the ignition/combustion phasing control and thereby achieving much higher engine efficiency and lower fuel consumption. When the early first-injection timing is fixed, the advanced second injection results in the increased in-cylinder pressure and rate of heat release, which leads to the improved engine efficiency and fuel consumption. The potential wall-wetting issue could also be resolved using the double injection strategy.

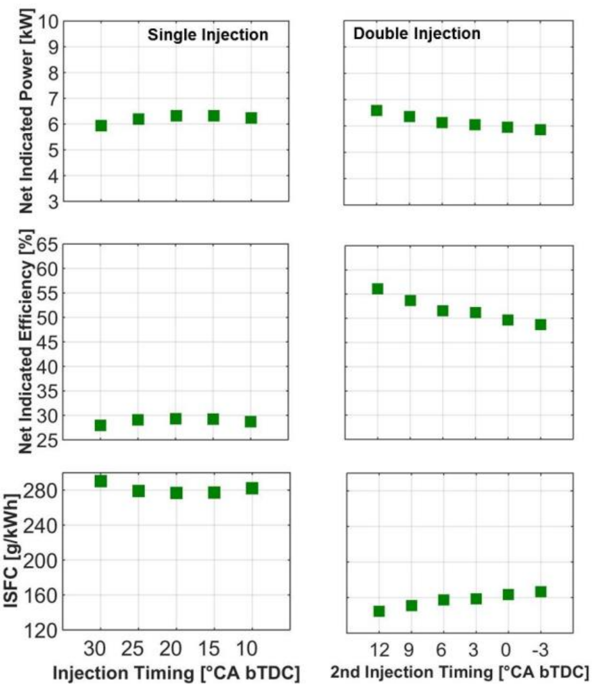

Figure 5. Effect of injection timings on the indicated power, net indicated engine efficiency and ISFC on GCI engine combustion for the single and double injection strategies

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