Effects of Direct Injection Timing Associated with Spark Timing on a Small Spark Ignition Engine Equipped with Ethanol Dual-Injection

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Abstract

Dual injection of ethanol fuel (DualEI) has been in development. DualEI has the potential in increasing the compression ratio and thermal efficiency of spark ignition engines by taking the advantages of ethanol fuel properties and the direct injection. This paper reports an experimental investigation of the effect of direct injection (DI) timing associated with spark timing on the performance of a small DualEI engine. Experiments were conducted with fixed port injection timing and varied DI timing before (early) and after (late) the intake valve closed at 3500 RPM and two load conditions. Results show that the engine performance is enhanced by early DI timing, although the variation of IMEP and indicated thermal efficiency with DI timing is not significant either with early DI timing or in most of the tested conditions with late DI timing. Only in the medium load condition when the DI timing is retarded from 80 to 60 CAD bTDC, the IMEP and thermal efficiency significantly reduced by about 16% due to the increased initial combustion duration, resulting in reduced flame speed and increased combustion instability. The results also show different effects of early and late DI timing associated with the spark timing on engine emissions. With late DI timing, the engine emissions of CO and NO increase with the advance of late DI timing and spark timing. With early DI timing, the engine emissions increase with the advance of spark timing. However, the variation of engine emissions with early DI timing is more complicated than that late.

Keywords

Ethanol dual-injection (DualEI); Ethanol direct injection; Injection timing; Spark timing; Combustion; Emissions

Highlights

- Effect of DI timing was stronger than the spark timing on DualEI engine performance.
- IMEP and indicated thermal efficiency were improved with early DI timing compared to that of late DI timing.
- ISCO was decreased when DI timing was advanced from late to early DI timing.
- ISNO was decreased by enhanced charge cooling effect when DI timing was retarded from early to late DI timing.

Abbreviations

bTDC	before top dead centre
CAD	crank angle degree
CA10-90%	the major combustion duration
CA0-10%	the minor combustion duration
CA50	combustion phase when 50% of the fuel is burnt.
DI	direct injection
DIT	direct injection timing
DualEI	ethanol dual-injection
EDI+GPI	ethanol direct injection plus Gasoline port injection
EPI	ethanol port injection
GDI	gasoline direct injection
HRR_{max}	maximum heat release rate
IMEP	indicated mean effective pressure
ISCO	indicated specific carbon monoxide
ISFC	indicated specific fuel consumption
ISHC	indicated specific hydrocarbon
ISNO	indicated specific nitric oxide
MBT	minimum spark advance for best torque
P_{max}	maximum cylinder pressure
ST	spark timing
λ	air/fuel equivalent ratio

1 Introduction

Replacing fossil fuel by renewable or alternative fuels has been driven by the concerns of sustainability and environmental protection. Moreover, the ever increasingly tightened legislations are enforced to reduce emissions produced by internal combustion (IC) engines. Compared to Euro 3, Euro 6 lowered the nitric oxide (NO), carbon monoxide (CO) and hydrocarbon (HC) emission limits by 60%, 57% and 50% for light-duty cars respectively [1]. These legislations highlighted the significance of environmental protection. Ethanol fuel has been used widely as an alternative fuel in IC engines to address the issue of sustainability [2]. Ethanol is a renewable fuel produced from a variety of bio-resources. Compared to gasoline, ethanol has a greater flame speed, octane number, and heat of vaporisation, leading to potentially cleaner combustion [3]. Furthermore, the lower adiabatic flame temperature and greater cooling effect could result in a lower combustion temperature and thus less convective heat losses [4]. Nevertheless, ethanol has a lower heating value, which consequently increases the fuel consumption compared to gasoline. The slow evaporation rate of ethanol fuel is also reported at low ambient temperature [5].

Injecting fuel directly into the combustion chamber is an approach to improve the engine performance and reduce the pollutant emissions in gasoline engines. The market of vehicles equipped with gasoline direct injection (GDI) engines is continuously increasing globally [6]. However, GDI comes with some drawbacks such as particular matter (PM) and greater hydrocarbon (HC) emissions due to fuel impingement [7]. Therefore, it is important to understand the effect of operating parameters on combustion in a GDI engine, such as direct injection (DI) timing and spark timing. Wang et al. [8] investigated the effect of injection strategy on PM emissions in a GDI engine fuelled with ethanol or gasoline. It was found that using high DI pressure and ethanol fuel could potentially reduce the PM emissions, due to the

finer droplets and less fuel impingement of higher injection pressure, and greater combustion speed and oxygen content of ethanol fuel.

Blended fuels (e.g., ethanol-gasoline and ethanol-diesel blends) were extensively studied in engine research [9, 10]. Bielaczyc et al. [10] tested a spark ignition (SI) engine under a wide range of ethanol-gasoline blending ratios from 5% to 85%. Their results showed that the combustion performance was enhanced due to the greater combustion speed and the oxygen component of ethanol fuel. The higher auto-ignition temperature, the greater latent heat of vaporisation and the larger research octane number of ethanol fuel could mitigate the engine knock. Greater ethanol percentages were commonly used in Brazil and the United States aiming to reduce the dependence on fossil fuels and improve the engine performance [11, 12]. The effect of E85 (15% gasoline plus 85% ethanol) on the SI engine performance was investigated [13-16]. Although ethanol is renewable, adopting ethanol as an alternative fuel to gasoline can come with challenges [17]. The lower vapor pressure and heating value of ethanol are two challenges for SI engines [18]. As a result, flex-fuel systems have been used as an auxiliary system to help SI engine in cold starting by using gasoline fuel [19]. Nakata et al. and Taniguchi et al. [20, 21] investigated the feasibility of running the SI engine with ethanol fuel only and concluded that ethanol could improve the engine thermal efficiency. It was reported that 100% ethanol direct injection (EDI) could improve the engine performance and reduce the NO emission [11, 22]. However, the over-cooling effect and severe fuel impingement of pure EDI were represented as possible issues to the SI engines [23].

Zhuang et al. [24, 25] investigated the effects of EDI on a gasoline port injection (GPI) engine. It was reported that the combustion performance was significantly improved when EDI was used. Huang et al. analysed the combustion process numerically of the same engine [5, 23]. Their results showed that the engine performance was improved when EDI ratio was within 60%. However, further increase of EDI ratio could deteriorate the combustion quality

due to the ethanol impingement and over-cooling effect. The greater laminar flame speed and the oxygen content of ethanol enhanced the combustion performance. A significant reduction of NO emission occurred due to the lower adiabatic flame temperature of ethanol and the cooling effect of EDI strategy. However, HC and CO emissions increased considerably due to the rich mixture regions created inside the combustion chamber when EDI was used. These ethanol rich regions were attributed to ethanol film formed on the combustion chamber walls. Zhu et al. [26] tested the effect of different ethanol-gasoline dual-injection strategies on the combustion characteristics, including GPI plus GDI, GPI plus EDI and ethanol port injection (EPI) plus GDI. The best combustion performance was recorded for GPI+EDI, in which the indicated mean effective pressure (IMEP) increased by 2% at light load.

Optimising the combustion phase is a vital method that could potentially improve the engine performance [27]. CA50 defines the crank angle degree (CAD) at which 50% of the mass fuel is burnt [28, 29]. The effect of CA50 on the SI engine performance was extensively investigated [28-31]. The cylinder volume at the position of CA50 and heat released rate (HRR) are factors that can significantly influence the engine performance. Ayala et al. examined the correlation between CA50 and IMEP at a range of equivalence and compression ratios [29]. Their results showed the importance of CA50 to the peak in-cylinder pressure (P_{max}) and its phase (Θ_{Pmax}). It is well known that retarding or advancing the spark timing (ST) from the optimal position of CA50 reduces the engine power [32, 33]. This is caused by the P_{max} reduction and the shift of Θ_{Pmax} from the optimal phase.

Controlling the DI timing is another approach to optimise the combustion phase and improve the engine performance. The significance of DI timing comes from its effect on the quality of the mixture. Advanced DI timing gives the fuel more time to atomise, evaporate and mix with air. However, fuel impingement could increase due to the smaller combustion chamber and lower cylinder pressure, resulting in higher HC emission [24]. Retarding DI timing could

reduce fuel film, but reduces the time available for air-fuel mixing as well. Davy et al. [34] experimentally investigated the effect of DI timing on the fuel spray development inside a GDI prototype engine. They found that the fuel impingement, especially to the piston surface, reduced significantly due to larger cylinder volume at the retarded DI timing. This might enhance combustion efficiency and reduce the HC and CO emissions simultaneously. However, the NO emission could increase due to greater combustion temperature caused by higher HRR [3].

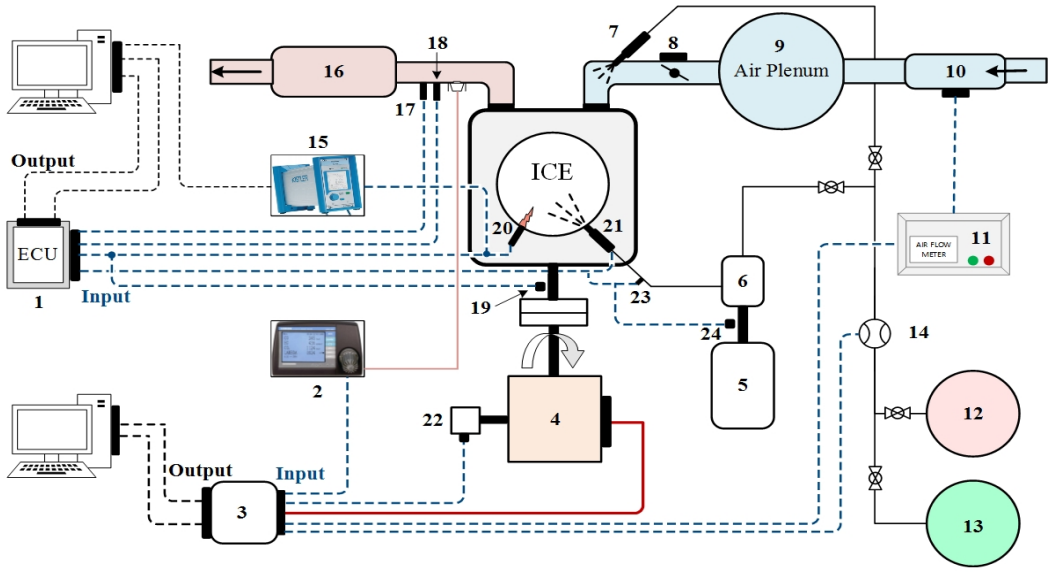
As discussed above, adopting ethanol fuel in SI engine development could present new challenges. Different technologies were used to exploit the benefits of ethanol, although the fossil fuel like gasoline is still the main fuel for SI engines. In this paper, a DualEI (EDI plus EPI) was used to provide a better method to fully utilize the properties of ethanol fuel in improving the performance of SI engines. Experiments were conducted to investigate the effect of DI timing associated with spark timing on the DualEI engine performance. The relevant results of the combustion and emissions characteristics were presented and analysed.

2 Experimental Apparatus and Procedure

2.1 Engine Setup

The experiments were performed on a modified Yamaha YBR250 motorcycle engine as shown in Figure 1. The YBR250 engine is a single cylinder four-stroke air-cooled naturally aspirated SI engine. This engine was chosen because it offers the flexibility for engine modification and is suitable for engine downsizing research. Table 1 shows the major specifications of the engine. The engine was originally equipped with a GPI system. It was modified to be equipped with a DualEI system by Hents Technology to meet the research needs. The engine modification included the installation of a DI system and an electronic control unit (ECU). The ECU was used to control simultaneously the throttle position, spark

146 timing, DI timing and pressure, and DI and PI durations. The DI system comprised of a
147 returnless high-pressure pump and a six-hole high-pressure injector [35]. The injector was
148 side mounted between the intake valve seat and the spark plug [25]. The slope angles of the
149 injector are 15° from the horizontal surface of the cylinder head to the axis of the injector and
150 12° from the vertical surface of the cylinder head. Table 2 shows the major specifications of
151 the injector. More details about the spray plume bend angles and their distribution can be
152 found in [18]. A super-micro LSN39 fuel flow meter with ±3% accuracy was used to measure
153 ethanol flow rates of PI and DI. An eddy current dynamometer was used to control the engine
154 speed and measure the torque. A Kistler 6115B spark plug pressure transducer and a Kistler
155 5011 charge amplifier were used to measure the in-cylinder pressure. K-type thermocouples
156 were used to measure the cylinder head temperature and exhaust gas temperature with a
157 resolution of 0.1 °C and uncertainty of 0.35%. A MEXA-584L Horiba exhaust gas analyser
158 was used to measure the exhaust gas emissions of CO, CO_2, HC and NO, and to calculate the
159 equivalence ratio (λ). The H/C and O/C ratios were manually set in the gas analyser to be 3.0
160 and 0.5 respectively. The intake airflow was stabilised in an 80L intake buffer tank and
161 measured using a ToCeiL20N thermal air-mass flow meter.



1. Electronic Control Unit (ECU) 2. Horiba MXEA-584L Exhaust Gas Analyser 3. NI-Module and Dynamometer Controller 4. Eddy Current Dynamometer 5. Electrical Motor 6. High Pressure Fuel Pump 7. Low Pressure Fuel Injector 8. Step Motor and Throttle Valve Position Sensor 9. Intake Air Stabiliser (Air Plenum) 10. Intake Air Flow Sensor 11. TOCEIL Air Flow Meter 12. Unleaded Gasoline Tank 13. Pure Ethanol Tank 14. Super Micro M-39 Fuel Flow Meter 15. KISTLER-5015A Charge Amplifier 16. Exhaust Gas Catalyser 17. K-Type Thermocouple 18. Bosh Wide-Band Lambda Sensor 19. Crank Shaft Encoder 20. KISTLER-6115B Spark Plug Pressure Transducer 21. High Pressure Fuel Injector 22. S-Type Load Cell 23. Common Rail High Pressure Sensor 24. High Pressure Fuel System Encoder

Figure 1. Ethanol dual-Injection research engine.

Table 1. Specifications of the ethanol dual-injection research engine

Engine Type	Single cylinder, four-stroke, air-cooled, naturally aspirated, SI
Displacement	249.0 cc
Bore	74.0 mm
Stroke	58.0 mm
Compression Ratio	9.8:1
Intake Valve Open (IVO)	382.2 CAD bTDC
Intake Valve Close (IVC)	126.2 CAD bTDC
Exhaust Valve Open (EVO)	594.6 CAD bTDC
Exhaust Valve Close (EVC)	340.7 CAD bTDC

Table 2. Specifications of the direct fuel injector [18, 36]

Manufacturer	Bosch
Operating pressure	Up to 500 bar
Number of holes	6 holes
Hole diameter	110 µm
Flow rate @ 100 bar	Up to 1640 g/min
Spray angle single beam	17º
Operating temperature range	-31 to 130 ºC

2.2 Experimental Procedure

The engine was started and warmed up using GPI until the cylinder head temperature became stable at around 200 ºC. Then GPI was switched to EPI and the air/fuel ratio was kept at stoichiometric point. Table 3 shows the tested engine conditions. The volumetric ratio (DI%) of the ethanol DI to PI was kept at 50% at light load and 56% at medium load because they were the optimal DI ratios for achieving effective cooling with minimal fuel impingement [23]. The fuel heating value was fixed at around 420 J/cycle at light load and 610 J/cycle at medium load. The spark timing was swept from 19 to 25 CAD bTDC (denoted by ST'XX' hereafter) at light load and from ST28 to ST34 at medium load, as they were the spark timings to produce the maximum output power [37]. At each spark timing, DI timing was varied from 330 CAD bTDC to 240 CAD bTDC (denoted by DIT'XXX' hereafter) before the intake valve was closed (early DI timing), and from DIT120 to DIT60 after the intake valve closed (late DI timing). The range of the late DI timing was chosen to eliminate the effect of the hot residuals, hot combustion chamber walls and intake flow velocity and then to address the effect of DI timing more independently. The range of early DI timing was chosen to include the impact of the previously mentioned factors and the time on engine performance. As shown in Figure 2, the early injection was swept with 30 CAD increment, and the late injection was swept with 20 CAD increment. The DI timing between DIT240 and DIT120 was avoided due to the pressure fluctuation when the intake valve was closing [24]. The throttle position was set at 23% for light load and 34% for a medium load. The air/fuel equivalence ratio (λ) was kept at around the stoichiometric condition (λ=1) and the engine speed at around 3500 RPM.

Table 3. Experimental operating conditions.

Engine loads	Light load (23% throttle opening) Medium load (34% throttle opening)
Engine speed (RPM)	3500
Spark timing (CAD bTDC)	Light load: 28 30, 32 and 34

	Medium Load: 20, 24, 26 and 28
DI ratio	Light load: 50% Medium load: 56%
PI timing	410 CAD bTDC
DI timing (CAD bTDC)	Early DI timing: 240, 270, 300 and 330 Late DI timing: 60, 80, 100 and 120

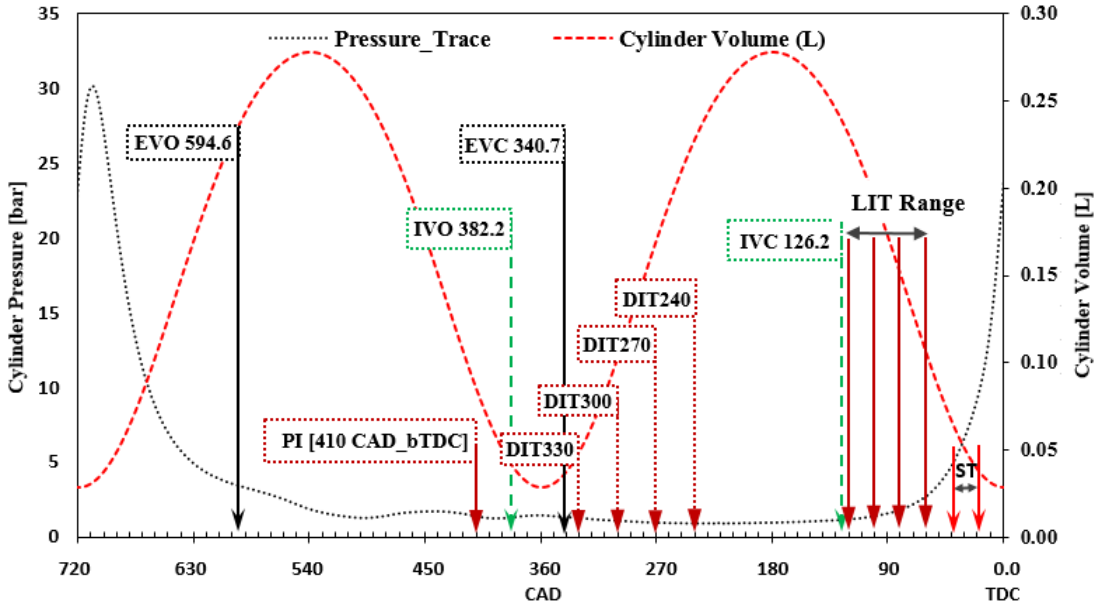


Figure 2. Valve, spark and injection timings set in the experiments.

A LabVIEW code was home-developed to record the engine performance and emission data. Five samples were recorded at a sampling rate of 1 Hz. The average values were used in the calculations and analyses. The in-cylinder pressure data was recorded at 0.5 crank angle degree resolution with 100 consecutive cycles in each sample. The ensemble average of the in-cylinder pressure was used to calculate the IMEP, combustion duration and heat release rate. The maximum standard deviations of the emissions measurements were 6.8% for ISCO, 7.1% for ISNO and 3.7% for ISHC, which showed an acceptable quality of the experimental data. The uncertainties of the emission measurements of the Horiba exhaust gas analyser was particularised by the manufacturer in reference [38]. They are 10 ppmvol for HC, 25 ppmvol for NO, 0.04 %vol for CO_2 and 0.03 %vol for CO.

3 Results and Discussion

The experimental results will be presented and discussed as follows. Section 3.1 reports the effect of early DI timing associated with spark timing on the IMEP and indicated the thermal

efficiency of DualEI engine. The combustion characteristics will be analysed and discussed to understand the mechanism behind the change in engine performance. In Section 3.2, the effect of early DI timing on emissions will be discussed and analysed. Sections 3.3 and 3.4 will discuss the effect of late DI timing on engine performance, combustion, and emissions.

3.1 Effect of Early DI timing on engine performance

The effect of early DI timing on IMEP and thermal efficiency of DualEI engine at different spark timing is shown in Figure 3. The coefficient of variation of IMEP (COV_{IMEP}) at various DI and spark timings is shown in Figure 4. COV_{IMEP} is derived from in-cylinder pressure and measures the combustion stability (cyclic variability) [27]. As shown in Figure 4, the COV_{IMEP} is almost independent of DI and spark timings' effects at both engine loads. At light load, COV_{IMEP} remains below its maximum of 4.46% at DIT300 and ST34. At medium load, the COV_{IMEP} approaches its minimum value of 1.99% at DIT330 and ST25. The results of COV_{IMEP} shown in Figure 4 exhibit an acceptable combustion stability (less than 5%) in the full range of DI and spark timings.

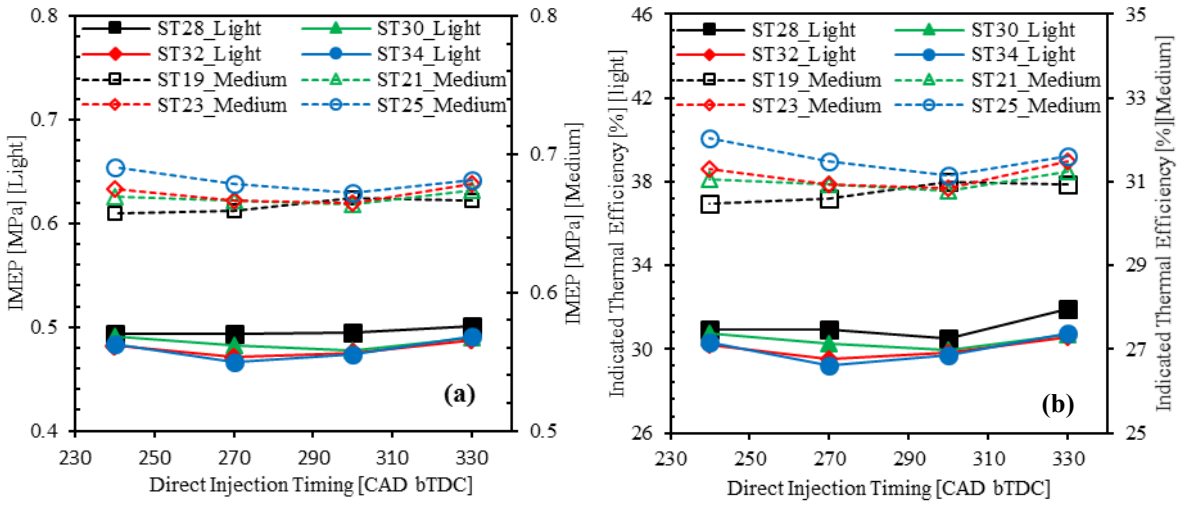


Figure 3. Variation of IMEP (a) and indicated thermal efficiency (b) with early DI at different spark timing.

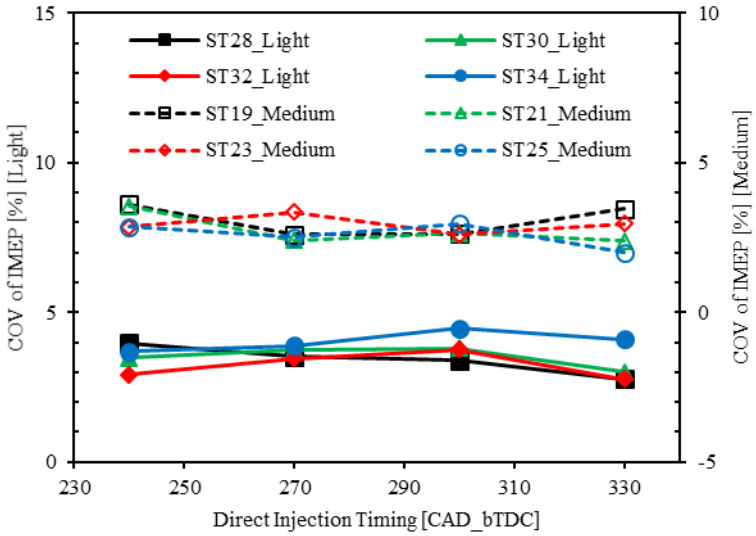


Figure 4. Variation of COV_{IMEP} with DI and spark timings.

As shown in Figure 3, the IMEP and thermal efficiency slightly decrease when DI timing is advanced from DIT240 to DIT300 at different spark timings, except ST19 and ST28 at medium and light loads respectively. Further advancing of DI timing from DIT300 to DIT330 slightly increases IMEP and thermal efficiency of DualEI engine. The reduction in IMEP and thermal efficiency might be attributed to rich air/fuel regions formed due to the impingement of ethanol that injected directly with combustion chamber walls [23] when the piston is moving upward to reduce cylinder volume, as shown in Figure 2. The increment of IMEP and thermal efficiency could be due to the longer time available at DIT330 compared with DIT300. This could potentially improve the mixing process of air with fuel before spark timing. Moreover, hot residuals might heat ethanol fuel that is directly injected at a few degrees after the exhaust valve is closed. This could increase the evaporation rate of ethanol inside the cylinder and thus improves the quality of mixture and combustion. At ST19 and ST28, IMEP and thermal efficiency increase with advanced DI timing from DIT240 to DIT330. Figure 3 shows the greatest IMEP and efficiency occur at the latest spark timing, ST19 for medium load and ST28 for a light load. The maximum increment of the IMEP occurs 1.67% at medium load and ST19, and 1.51% at light load and ST28 when the DI timing is within the range from DIT240 to DIT330.

To understand the combustion characteristics in DualEI engine, the results of the initial (CA0-10%) and major (CA10-90%) combustion durations at different DI and spark timings conditions are presented and analysed. CA0-10% is the initial combustion duration defined by the crank angle degrees starting from the spark timing to the time with 10% heat release. Early flame development and propagation have a strong effect on combustion stability [27]. As shown in Figure 5 (a), at both engine loads, CA0-10% significantly decreases with advanced early DI timing from DIT240 to DIT300, and then it slightly increases when DI timing is further advanced to DIT330. Spark timing shows a weak effect on CA0-10% at medium load condition. However, in the light load, CA0-10% significantly decreases with retarded spark timing from ST34 to ST28. The CA0-10% reduction could partially explain the improved combustion stability which is shown by the COV_{IMEP} in the light load condition in Figure 4. As shown in Figure 4, the COV_{IMEP} is in the range of 2%-5% when the early DI timing is in the range of 240-330 CAD bTDC. It decreases from 3.5% to about 2% when the spark timing is retarded from 34 to 28 CAD bTDC at DIT330. CA10-90% is the main combustion duration defined by the time from 10% of the fuel burnt to 90% burnt. Figure 5 (b) does not show a close link between the CA10-90% and the IMEP and the indicated thermal efficiency. However, CA10-90% significantly decreases when the DI timing is advanced from DIT240 to DIT330 in the light load condition. Figure 6 shows the combustion phase (CA50) related to the results that shown in Figure 3. CA50 is defined as the crank angle degree when 50% of the fuel is burnt. As shown in Figure 6, CA50 reduces when DI timing is advanced from DIT240 to DIT300 in the full set of spark timing, and at the two engine loads. Then, CA50 slightly increases when DI timing is advanced further, from DIT300 to DIT330. The advanced CA50 (near or even before the engine's top dead centre (TDC)) means a negative indicated power could be generated. Therefore, IMEP and indicated

thermal efficiency slightly increase as shown in Figure 3 when CA50 decreases with advanced DI timing from 300 to 330 CAD bTDC, as shown in Figure 6.

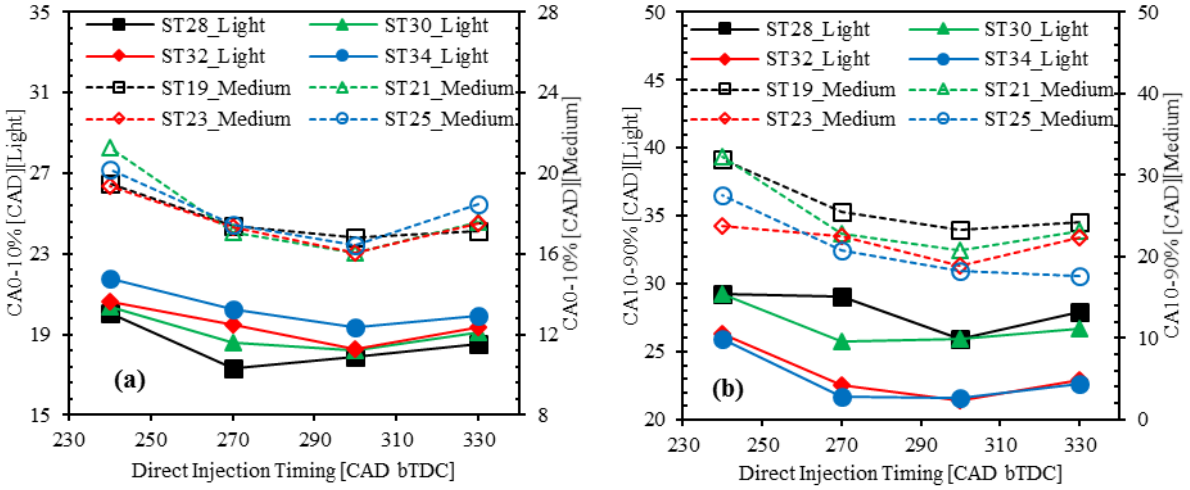


Figure 5. CA0-10% and CA10-90% vs. spark timing at DIT330, light load (a) and medium load (b).

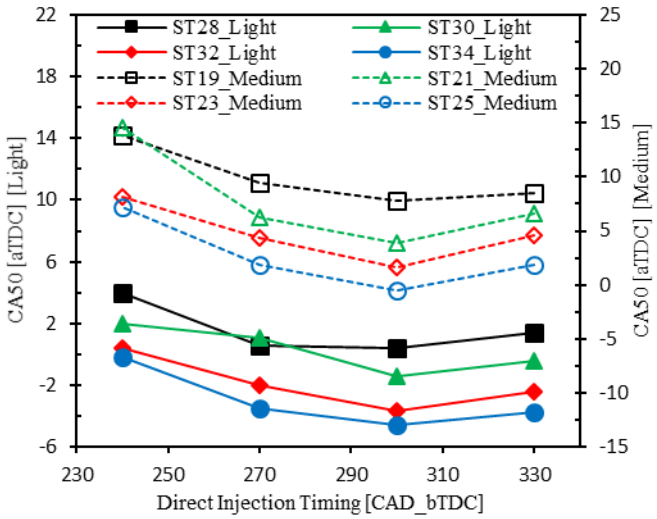
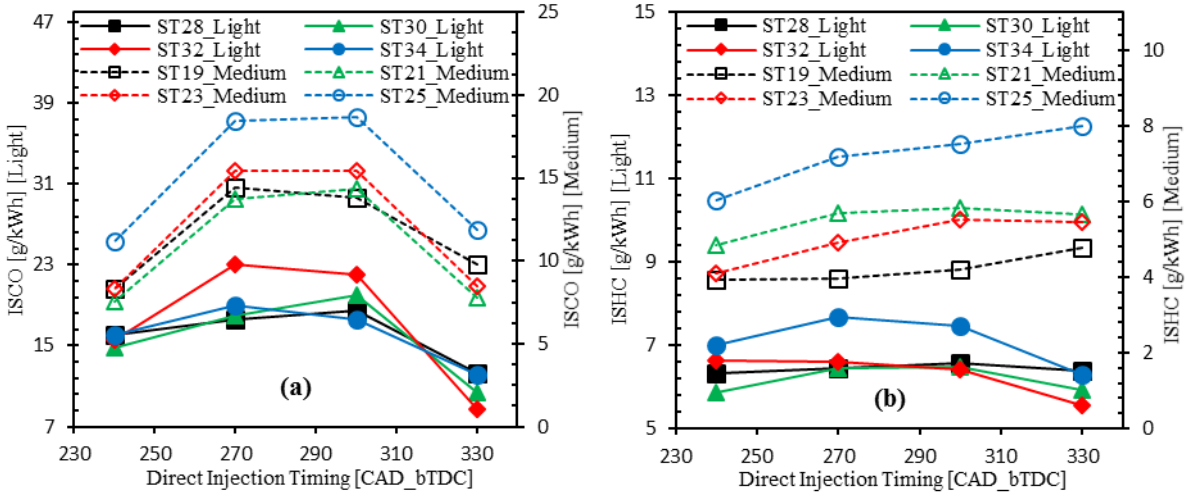


Figure 6. Variation of CA50 with an early DI and spark timings.

3.2 Effect of Early DI timing on emissions

Figure 7(a) shows the effect of early DI timing on ISCO at different spark timing. The lack of oxygen in combustion results in larger concentration of CO emission [39, 40]. As shown in Figure 7(a), the spark timing does not show a clear effect on ISCO emission when DI timing is in the range from 240 to 330 CAD bTDC. On the other hand, at both engine loads, the ISCO increases when the DI timing is advanced from DIT240 to DIT300. However, the ISCO significantly reduces when the early DI timing is further advanced to DIT330. The

increased CO emission could be attributed to the rich mixture regions caused by fuel impingement due to smaller combustion chamber volume with advanced DI timing, as shown in Figure 2. Sever fuel impingement could form a thick film, and slow down the fuel vaporization [41]. The decreased ISCO might be because the hot residuals heated the directly injected fuel since about half of fuel was injected at 10.7 degrees after the exhaust valve closed. This could accelerate the evaporation rate of ethanol directly injected into the cylinder [41-43]. Compared to DIT240, ISCO is increased by 85.9% at DI timings of DIT300 and 2.3% at DIT330, at ST23 and medium load. At light load condition and ST32, ISCO increases by 41% when DI timing is advanced from DIT240 to DIT300, but it decreases by about 43% with advanced DI timing to 330 CAD bTDC, as shown in Figure 7 (a). The ISCO results could partly explain the results of indicated thermal efficiency shown in Figure 3 (b).



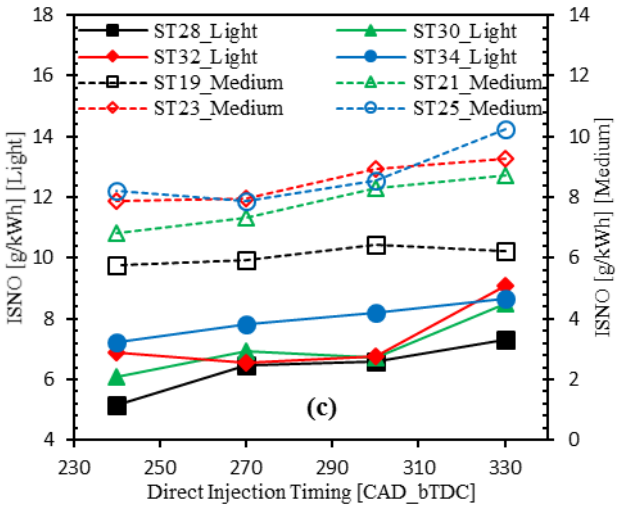


Figure 7. Variation of ISCO (a), ISHC (b) and ISNO (c) with an early DI and spark timings.

Figure 7 (b) shows the effect of early DI timing on HC emission at different spark timing. HC emission contains of unburned fuel from incomplete combustion due to local flame extinction that occurs when temperatures are lower and reaction times may become larger than mixing times [39, 40]. As shown in Figure 7 (b), at light load, ISHC increases when the early DI timing is advanced from DIT240 to DIT300, and then slightly decreases when DI timing is further advanced to DIT330. At medium condition, ISHC increases with advanced DI timing from 240 to 330 CAD bTDC. Spark timing exhibits a clear effect on ISHC emission that increases with advanced spark timing from ST28 to ST34 in light load, and from ST19 to ST25 in medium load. The effect of DI and spark timing is stronger in medium load than that in light load condition. Since the engine speed is fixed, the available time for the mixing process is constant for both engine load conditions. Nevertheless, the mass of fuel injected per cycle is about 45% greater (from 420 J/cycle for the light load to 610 J/cycle for the medium load) when the load increases from light to medium. This probably causes insufficient time available for forming a homogenous mixture at medium load compare to that in light load resulting in greater HC emission at the medium load condition.

The combination of nitric oxide (NO) and nitrogen dioxide (NO_2) forms the nitrogen oxides (NO_X) emissions which are one of the major pollutants produced by combustion [44]. In this study, only NO is measured. NO emission is formed from the N_2 and O_2 dissociation at the

high combustion temperature [45]. Figure 7 (c) shows that the ISNO increases with advanced spark and DI timings in the light and medium load conditions. As shown in Figure 7 (c), at ST21, the ISNO increases by 28% when DI timing is advanced from 240 to 330 CAD bTDC in medium load. At light load and ST30, it increases by 39% when DI timing is advanced. At medium load and DIT330, the ISNO increase by about 64% when the spark timing is advanced from ST19 to ST25. Likewise, at light load condition but with less significant, the ISNO increases when the spark timing is advanced 28 to 34 CAD bTDC.

3.3 Effect of late DI timing on engine performance

Figure 8 shows the effect of late DI timing on the IMEP and indicated the thermal efficiency of the DualEI engine at different spark timing. As shown in Figure 8, the IMEP and indicated thermal efficiency are almost independent of the spark timing effect. This may be because the mixture quality and condition are similar before the spark timing at each DI timing condition. As shown in Figure 8, the effect of DI timing on IMEP and indicated thermal efficiency is not strong in the light engine load condition. At medium load, the effect of DI timing is not so obvious until the DI timing is later than 80 CAD bTDC. The IMEP slowly decreases from 0.62 MPa to 0.6 MPa and indicated thermal efficiency from 26.6% to 25.9% when the DI timing is retarded from 120 to 80 CAD bTDC at ST19. However, the IMEP decreases from 0.6 MPa to about 0.5 MPa and the indicated thermal efficiency from 25.9% to about 21% when the DI timing is retarded from 80 CAD bTDC to 60 CAD bTDC in the medium load condition. As the engine speed is fixed at 3500 rpm, the time for mixture formation is equal in light and medium load conditions. In light engine condition, 50% of the fuel is injected directly in the cylinder. The time for forming the mixture of fuel and air looks sufficient as the IMEP and thermal efficiency are quite stable in the full range of DI timing, as shown in Figure 8. However, when the engine load is increased from light to medium condition, the ratio (56%) of ethanol fuel directly injected in cylinder and the fuel flow rate are increased.

When the DI timing is retarded from 80 to 60 CAD bTDC, the time for the fuel to evaporate and to mix with the fuel port injected and air may become insufficient at medium load, resulting in a significant decrease in IMEP and indicated thermal efficiency.

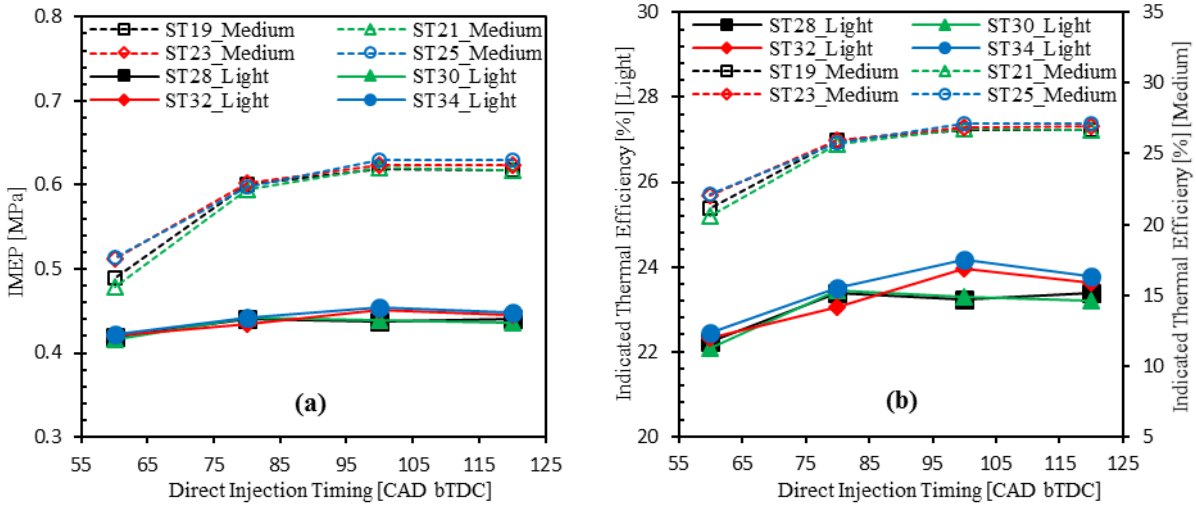


Figure 8. Variation of IMEP (a) and indicated thermal efficiency (b) with the late DI and spark timings.

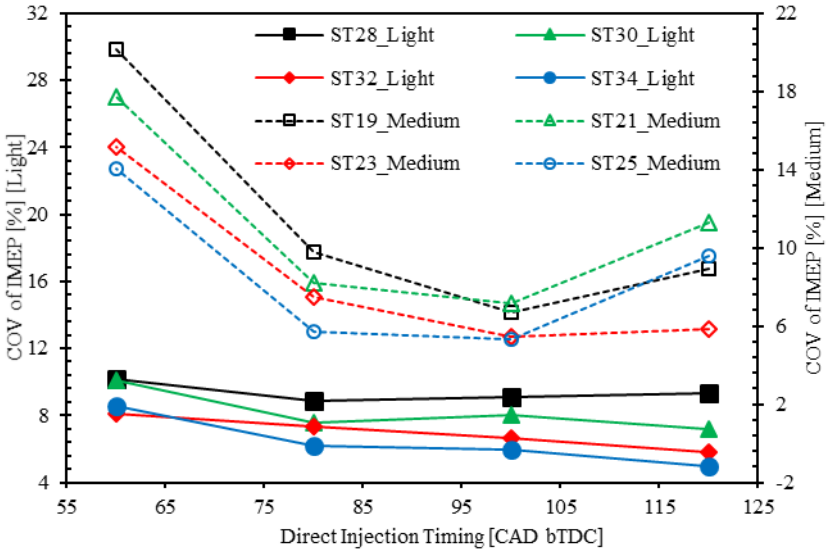


Figure 9. Variation of the COVIMEP with late DI and spark timing.

To further understand the effect of late DI timing on the engine performance, the combustion characteristics will be analyzed and discussed. Figure 10 shows the CA0-10% and CA10-90% which are corresponding to the results shown in Figure 8. Variation of IMEP (a) and indicated thermal efficiency (b) with the late DI and spark timings.. CA0-10% is the initial combustion duration defined by the crank angle degrees starting from the spark timing to the time with 10% heat release. Early flame development and propagation have a strong effect on

combustion stability [27]. As shown in Figure 10 (a), the effect of late DI timing on CA0-10% is not significant in the light load condition. However, in medium load condition, the CA0-10% is reduced with the DI timing advanced from 60 to 100 CAD bTDC and then slightly increased with DI timing further advanced to 120 CAD bTDC. It reduces more quickly when the DI timing is advanced from 60 to 80 CAD bTDC. The reduced CA0-10% means increased laminar flame speed and improved stability. This explains the strong effect of DI timing on the IMEP and indicated thermal efficiency when the DI timing is advanced from 60 to 80 CAD bTDC and then weak effect of DI timing with further advance of DI timing, as shown in Figure 8. Variation of IMEP (a) and indicated thermal efficiency (b) with the late DI and spark timings.. The improved combustion stability associated with reduced CA0-10% is also shown by the COV_{IMEP} in medium load condition in Figure 9. As shown in Figure 9, the COV_{IMEP} is in the range of 5%-20% when the DI timing is 60 CAD bTDC. It is decreased from 17.7% to 8.2% when the DI timing is advanced from 60 to 80 CAD bTDC at ST21. CA10-90 is the main combustion duration defined by the time from 10% of the fuel burnt to 90% burnt. Figure 10 (b) does not show a close link between the CA10-90% and the IMEP and the indicated thermal efficiency but a significant increase of CA10-90% when the DI timing is advanced from 80 to 120 CAD bTDC in the light load condition.

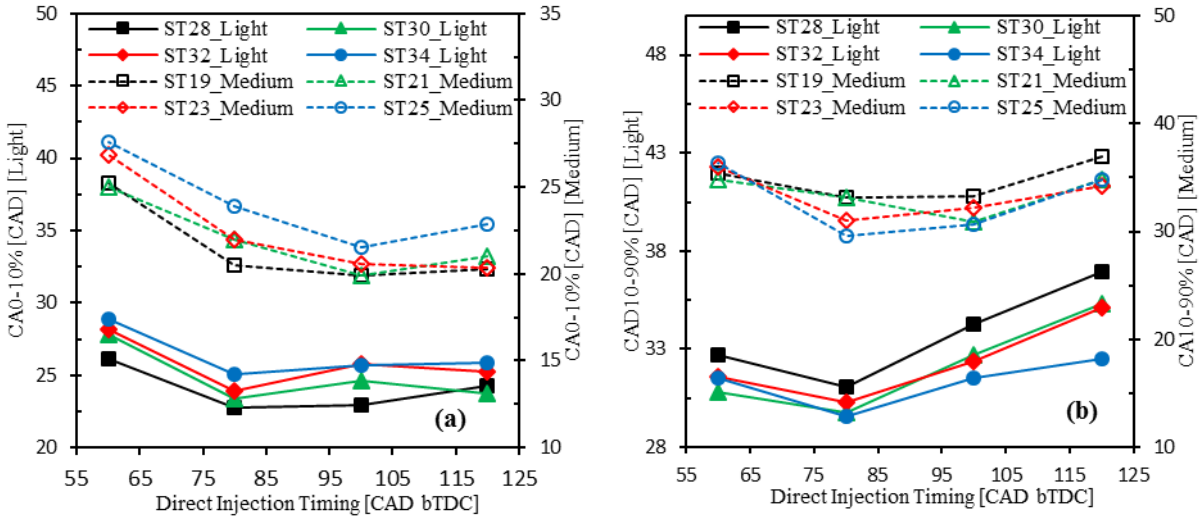


Figure 10. Variation of CA0-10% (a) and CA10-90% (b) with the late DI and spark timings.

3.4 Effect of late DI timing on emissions

The variation of the ISNO, ISCO, and ISHC with the DI and spark timings are presented in Figure 11. Fundamentally, the elevated temperature of combustion promotes NO formation. As shown in Figure 11 (a), the ISNO decreases noticeably when the late DI timing is retarded from 120 to 60 CAD bTDC. It is also decreased when spark timing is retarded from ST25 to ST19 in the medium load and from ST34 to ST28 in the light load condition. The cooling effect of DI strategy could play a significant role in the reduction of NO emission in the late DI timing condition. Advanced DI timing could reduce the mixture temperature at an earlier time, but this also increases the convective heat transfer from combustion chamber walls to the mixture. This could enhance the mixture quality and improve the engine performance, as shown in Figure 8 and Figure 10. When the DI timing is retarded, the cooling effect due to fuel evaporation associated with DI strategy could be well preserved. This could decrease the combustion temperature and then the NO emission, as shown in Figure 11 (a). Figure 11 (a) shows that the ISNO decreases with retarded spark timing in the light and medium load conditions. This could be attributed to the smaller gas temperature due to the phase change of combustion when spark timing is retarded [37].

The effect of DI and spark timings on ISCO and ISHC are shown in Figure 11 (b). It can be seen that at all the tested spark timings, ISCO increases when the DI timing is advanced from DIT60 to DIT120. Although the DIT120 provides more time for fuel evaporating compared to DIT60, the in-cylinder pressure and then the temperature is lower than that of DIT60, as shown in Figure 2. Greater compression temperature may promote a faster evaporation rate of ethanol. Furthermore, the air/fuel equivalence ratio (λ) is slightly decreased when the DI timing is advanced from DIT60 to DIT120. Consequently, less amount of oxygen is available to the fixed amount of injected fuel per cycle, which result in greater ISCO and ISHC

emission. The ISCO and ISHC results do not show a clear link with IMEP or the results of the combustion characteristics, which are shown in Figure 8 and Figure 10.

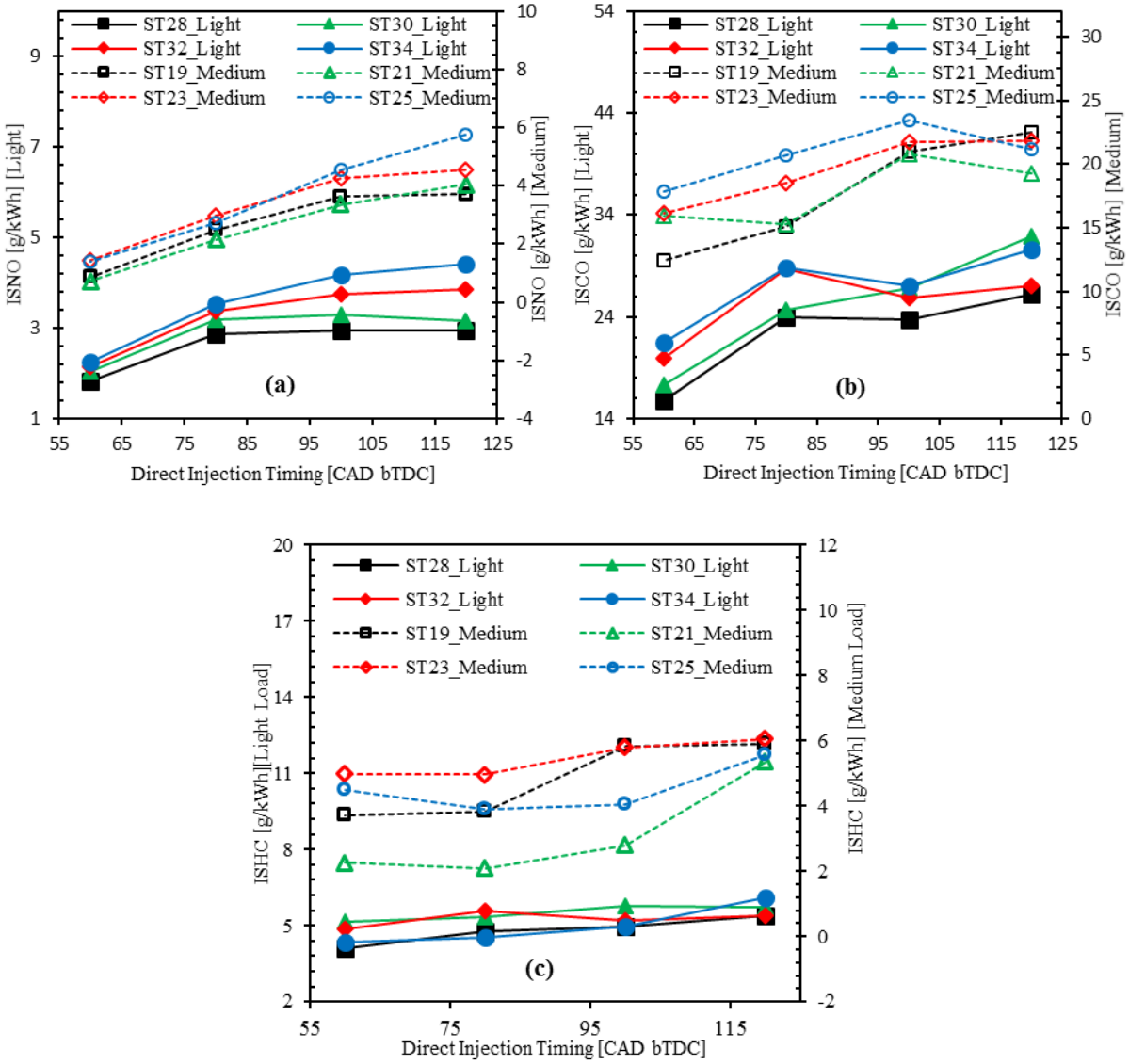


Figure 11. Variation of ISNO (a), ISCO (b) and ISHC (c) with the late DI and spark timings.

4 Conclusions

Experiments were carried out to investigate the effect of DI timing associated with the spark timing on the performance of a naturally aspirated SI engine equipped with a DualEI system. The DI timing was advanced from DIT240 to DIT330 (before the intake valve is closed, early DI timing) and from DIT60 to DIT120 (after the intake valve is closed, late DI timing). At

each DI timing, the spark timing was varied from ST28 to ST34 at light load and from ST19 to ST25 at medium load. The results of this study can be concluded as follows.

1. In early DI timing conditions, the effect of DI timing on IMEP is not strong, so is that of the spark timing. At medium engine load, the IMEP slightly decreases when the DI timing is advanced from 240 to 300 CAD bTDC with small differences caused by the spark timing. When the DI timing is further advanced from 300 to 330 CAD bTDC, the IMEP increases slightly or keeps constant. At light load, the IMEP increases slightly with the advanced spark timing but independently with the DI timing. As the intake valve is still open in early DI timing, the mixing process and the quality of the mixture are not significantly affected by the DI timing.
2. In late DI timing conditions, IMEP increases with the advanced DI timing. However, the effect of DI timing is not very strong when the DI timing is earlier than 80 CAD bTDC. When the DI timing is retarded from 80 to 60 CAD bTDC, the IMEP is decreased from about 0.6 MPa to 0.5 MPa. This is mainly because the DI timing of 60 CAD bTDC is too late and leaves insufficient time for the ethanol fuel to evaporate and mix with the air in the cylinder before it is ignited. This is evident with the longer initial combustion duration with DI timing of 60 CAD bTDC than that with more advanced DI timing. The ISNO and ISCO increase with the advanced DI timing as well as spark timing.
3. This DualEI engine performs better with early DI timing than with late DI timing, in terms of IMEP and indicated thermal efficiency. With early DI timing, the IMEP is in ranges of 0.60-0.68 and 0.47-0.50 MPa and indicated thermal efficiency is in ranges of 30-32% and at 29-31% medium and light loads, respectively. With late DI timing, the IMEP is in a ranges of 0.47-0.62 and 0.41-0.44 MPa and indicated thermal efficiency is in ranges of 21-27% and 22-24% at medium and light loads, respectively.

5 Acknowledgment

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