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Effects of direct injection timing of ethanol fuel on engine knock and lean burn in a port injection gasoline engine

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Abstract

Ethanol is a promising alternative fuel for internal combustion engines due to its renewable feature. To make the use of ethanol fuel more effective and efficient, ethanol direct injection plus gasoline port injection (EDI + GPI) has been investigated in recent years. By directly injecting ethanol into the engine, the advantages of ethanol fuel such as high latent heat of vaporization, fast laminar flame speed, wide flammability and better low temperature combustion stability can be well utilized to enhance engine anti-knock ability and improve lean burn performance. For an engine equipped with direct injection (DI) system, start of injection (SOI) timing is an important control parameter which directly affects the engine performance. This paper reports the investigation to the effect of ethanol fuel SOI timing on knock mitigation and lean burn. Experiments were conducted on a 250 cc single cylinder spark ignition (SI) engine equipped with EDI + GPI system. Ethanol fuel SOI timing before and after the inlet valve closing, defined as early and late injection timings (EEDI and LEDI) were investigated in engine conditions at knock limited spark advance (KLSA) and lean burn limit. The experimental results showed that LEDI was effective on suppressing engine knock and permitting more advanced spark timing. EEDI was less effective than LEDI on mitigating knock due to the increased heat transfer from cylinder wall to gases. The mixture quality may be deteriorated in LEDI conditions which resulted in low engine efficiency and high emissions. Volumetric efficiency was increased and combustion duration was reduced in EEDI conditions. The combined effects of improved volumetric efficiency, reduced combustion duration and moderately advanced spark timing resulted in increased engine thermal efficiency in EEDI conditions. In lean burn, EEDI was more effective on extending lean burn limit. The maximum lambda achieved in EEDI condition was 1.29 when ethanol energy ratio (EER) was 24% and SOI timing was 290 CAD BTDC. LEDI only slightly increased lean burn limit which was just over stoichiometric air–fuel ratio (AFR). In EEDI conditions, IMEP was greater and combustion stability (COV) was better than that in LEDI conditions. The emissions in EEDI conditions were also lower than that in LEDI conditions.

Abbreviations: ATDC, after top dead center; BTDC, before top dead center; COV, covariance of variation; DI, direct fuel injection; EDI, ethanol fuel direct injection; EEDI, early ethanol direct injection; ERR, ethanol/gasoline energy ratio; ECU, electronic control unit; ISHC, indicated specific hydrocarbon; ISNO, indicated specific nitric oxide; IMEP, indicated mean effective pressure; GPI, gasoline port injection; KLSA, knock limited spark advance; LHV, low heat value; LEDI, late ethanol direct injection; SOI, start of injection; HE, heating energy; HC, hydrocarbon; MBT, maximum brake torque; NO, nitric oxygen; PFI, port fuel injection; Lambda (λ), air/fuel equivalence ratio

Keywords: ~~EDI + GPI;~~ Ethanol direct injection; Gasoline port injection; Fuel injection timing; Engine knock; Lean burn

1 Introduction

Improving engine efficiency and reducing its emissions are the major tasks of recent development in internal combustion technology [1]. This is driven by the society concerns about the global warming and the depletion in supply of fossil fuels. One of the feasible short-to-midterm solutions for addressing the concerns is to use renewable fuels such as ethanol. Many countries and areas have enacted legislations [2] or incentive policies to promote the use of ethanol and other bio/renewable fuels [3,4]. The use of ethanol and other bio/renewable fuels has brought new challenges to automotive sector to develop new technologies. Ethanol direct injection plus gasoline port injection (EDI + GPI) is one of the new technologies in recent development.

Previous studies on SI engines found that using ethanol fuel could help to reduce unburned emissions such as carbon monoxide (CO) and hydrocarbon (HC) [5–7], enhance engine anti-knock ability [8,9], and improve lean burn performance [10,11]. In the current applications, ethanol is externally blended with gasoline at a specific ratio. The applications of this method decreased pollutant emissions and increased engine efficiency in certain engine conditions [12,13]. However, the vehicles may face problems of reduced vehicle coverage, difficult cold start [14], corrosion and reduced lubrication [15]. Moreover, due to the fixed ethanol/gasoline ratio, the ethanol's potential in reducing engine emissions and improving thermal efficiency cannot be fully exploited. Research has already shown that the optimal ethanol/gasoline ratio for maximizing engine efficiency and minimizing emissions was varied with the engine operation condition [6,7]. EDI + GPI dual-fuel injection, on the other hand, provides an opportunity to solve the problems and meet the requirements. It offers greater flexibility of using ethanol fuel because the ethanol and gasoline mixing ratio can be instantly altered according to the engine operation condition and fuel availability. Through adjusting GPI and EDI, the advantages of GPI in forming homogeneous air–gasoline mixture and of EDI in charge cooling can be integrated. The engine performance can therefore be improved.

In SI engines, spark timing is usually selected at its minimum advance for the maximum brake torque (MBT). However, this MBT spark timing is often prevented by the onset of knock. Thus, SI engine's efficiency is limited by the engine knocking. The knocking tendency is related to fuel properties, compression ratio, injection strategies (in direct injection spark ignition (DISI) engines) and engine operating conditions such as spark timing and turbo-charging. In an EDI + GPI engine, its anti-knock ability is strengthened by both the charge cooling enhanced by ethanol's great latent heat and ethanol's high octane number. Stein et al. [16] and Daniel et al. [17] evaluated the effect of ethanol direct injection on knock mitigation in dual-injection engines. In their study, the ethanol fuel SOI timing was fixed before the inlet valve closing. They found that by directly injecting ethanol, the engine knock was substantially suppressed and the engine thermal efficiency was increased when compared with GPI only condition. Although, the effect of dual-injection on knock suppressing has been studied by several pioneer researchers, the relation between SOI timing and knock mitigation has not been investigated yet. The numerical simulation results of Cohen et al. [18,19] indicated that the ethanol fuel SOI timing could significantly affect the dual-injection engine anti-knock ability. It was predicted that if the ethanol was injected before the inlet valve closing, the boost pressure up to 2.4 Bar could be applied without knocking. If the ethanol was injected after the inlet valve closing, greater boost pressure could even be allowed. Moreover, SOI timing also affects mixture homogeneous quality and ultimately the engine efficiency and emissions. Thus, finding the relation between SOI timing and engine knock, efficiency and emissions is necessary in EDI + GPI engine development.

With the rapid development of engine control techniques, lean or stratified combustion has become more applicable to modern DISI engines. In this combustion mode, the greater throttle opening reduces the pumping losses associated with stoichiometric conditions at partial engine load. Lean burn can also reduce (nitrogen oxides) NO_x emissions which are temperature dependent, and HC emissions which are related to the availability of oxygen [20,21]. However, when the mixture is leaned, the combustion may become unstable. It requires precise control of engine parameters such as SOI timing and spark timing, where small variation in these parameters can lead to misfire or in-completed combustion [20]. In EDI + GPI engine, DI may help to form ignitable mixture in adjacent to the spark plug. The use of ethanol may enhance lean burn stability and reduce the time for fuel droplets to evaporate due to its high laminar flame speed and low molecular weight properties. The effect of ethanol on engine lean burn has been studied in port fuel injection (PFI) engine. It was reported that by using ethanol, the engine lean burn limit (maximum achievable lambda, denoted as λ) was increased by a value of 0.2 and the coefficient of variation (COV) of IMEP was reduced by about 2% [22,23]. As a new technology, the study on lean burn in dual-injection engine has not been reported so far and therefore requires investigation.

In a DI engine, SOI timing directly affects the heat transfer and mixture temperature. Earlier fuel injection cools the gas at an earlier time, which results in improved volumetric efficiency but also increases the heat transfer from the wall to the gases. Thus, the overall cooling effect on the charge is compromised. For late injection, the cooling effect due to fuel evaporation can be well preserved, leading to a lower knocking tendency. However, the mixture quality may be reduced and the combustion may be deteriorated due to less time for forming homogeneous mixture [24,25]. The volatility of ethanol is higher than that of gasoline when the temperature is over 410 K due to its single component [20]. This may help to produce more homogeneous mixture and reduce the time for fuel evaporation in DISI engine. Therefore, the SOI timing for ethanol fuel can be retarded to enhance anti-knock ability while maintaining the high quality of the mixture [26,27].

As reviewed above, in the development of ED + GPI engine, investigation to the effect of SOI timing on knock mitigation and lean burn is required. This study was aimed to meet the requirement. Experimental work was carried out on a self-developed EDI + GPI engine. The results presented and discussed include effects of ethanol fuel SOI timing on KLSA, lean burn limit, combustion and emissions.

2 Experimental apparatus

2.1 Test engine and instrumentation

Fig. 1 is a schematic diagram of the test rig. The engine was modified from a Yamaha product YBR250. It is a 4-stroke, air cooled single cylinder motorcycle SI engine with a displacement of 250 cc which represents the cylinder capacity of a light duty passenger car. Table 1 lists the engine's main specifications. The engine was originally equipped with an electronic PFI system operated at a constant fuel pressure of 2.5 Bar and an onboard electronic control unit (ECU). It was modified by adding an ethanol fuel direct injection system and a new ECU which replaced the original one and provided the flexibility of manual adjustment of spark timing, lambda (λ) value, ethanol injection timing and other parameters relevant to the engine operation. The ethanol fuel direct injection system consisted of a six hole injector which has a 34° spray cone angle and a 17° bent axis and a returnless high pressure pump providing fuel pressure in a range of 30 ~~Bar to~~ –130 Bar. The injector was side mounted on the same side as the spark plug opposite to the sprocket of camshaft to avoid interference. There is a slop angle of 15° between the axis of the injector and horizontal surface which is the interior surface of cylinder head and a 12° between the axis of the injector and the vertical surface. Two of the spray plumes were directed toward the spark plug with electrodes located near the edge of the spray, so that a locally rich mixture in adjacent to the spark plug can be formed. The position of the injector in cylinder head and relative position of spark plug are shown in Fig. 2.

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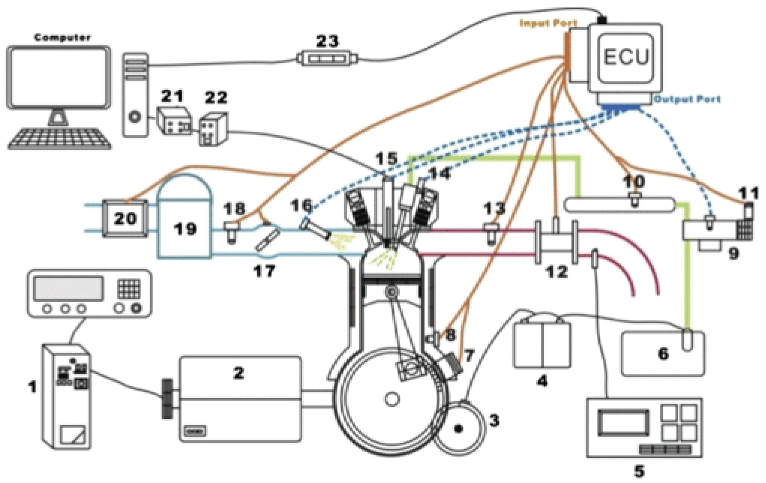

1. Dynamometer controller 2. Dynamometer 3. Start motor 4. Battery 5. Horiba MXEA-584L gas analyzer 6. Ethanol fuel tank 7. Encoder on crankshaft 8. Temperature sensor 9. High pressure fuel pump 10. Common rail pressure sensor 11. Encoder on high pressure pump shaft 12. Bosch wide-band lambda sensor 13. Temperature Sensor 14. Direct fuel injector 15. Kistler spark plug pressure transducer 16. Port fuel injector 17. Throttle valve position sensor and driving motor 18. Temperature sensor 19. Inlet air regulator 20. Air flow meter 21. Combustion analyzer 22. Charge amplifier 23. CAN Communication module

Fig. 1 Schematic of the engine system.

Table 1 Engine specifications.

Engine type	Single cylinder, air cooled,
	4-stroke, SOHC
Displacement	249.0 cm^3
Bore × stroke	74.0 mm × 58.0 mm
Compression ratio	9.80:1
Intake valve timing	Opening: 45° BTDC
	Closing: 60° ABDC
Exhaust valve timing	Opening: 87° ATDC
	Closing: 21° ATDC

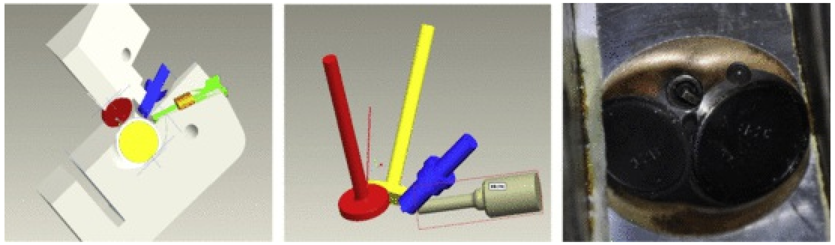

Fig. 2 Relative position of injector and spark plug in cylinder head.

The engine was coupled to a DC dynamometer for speed control and torque measurement. In-cylinder pressure was measured using a Kistler 6115B measuring spark plug pressure transducer via a Kistler 5011 charge amplifier. The cylinder pressure data was recorded from 100 consecutive cycles with a sample rate of 2 per angle degree. Engine body temperature, lubricant oil temperature, intake and exhaust air temperature were measured through K-type thermal couples with resolution of 0.1 K. A 80 L intake buffer tank (approximately 320 times the engine's displacement volume) was used to stabilize the intake flow. The intake air flow was measured via a ToCeil20 N hot-wire thermal air-mass flow meter. A Bosch wide-band lambda sensor which was mounted on the exhaust pipe was used to monitor the lambda when the engine was running with gasoline fuel only. In EDI + GPI, dual-fuel injection condition, the lambda was calculated from gasoline and ethanol fuel flow rates and the inlet air flow rate. This was because the fuel's chemical composition such as hydrogen to carbon (H/C) and oxygen to carbon (O/C) ratios, as well as stoichiometric AFR varied when EER was changed. Thus, the fuel composition data was unable to be real-time inputted into the lambda sensor or gas analyzer which required manual entering. Real-time lambda control was impossible if it was based on the data from lambda sensor or gas analyzer. The gasoline and ethanol fuel flow rates were measured through metering the injection pulse width of the injectors. The fuel injectors were calibrated before the experimental investigation started.

The exhaust gas emissions were measured using a Horiba MEXA-584L gas analyzer. In this gas analyzer, HC and CO emissions were measured through a Non-Dispersive Infra-Red (NDIR) detector and NO emission was detected via a chemi-luminescence (CLD) sensor. The sample exhaust gas was taken at a position 0.4 m~~eter~~ from the exhaust port before it flew into the three-way catalyst converter. Data were recorded at a sample rate of 1 Hz.

2.2 Test fuels

The gasoline fuel used in this study was BP Australia No. 91 gasoline with an octane number of 91. It was used to provide a benchmark to the ethanol fuel in the experimental investigation. The ethanol fuel was provided by Manildra Group. Properties of ethanol and gasoline fuels are listed in Table 2.

Table 2 Test fuel properties.

	Ethanol	Gasoline
Chemical formula	C_2H_6O	C2–C14
H/C ratio	3	1.795
O/C ratio	0.5	0
Gravimetric oxygen content (%)	34.78	0
Density@20 °C (kg/m^3)	790.9	744.6
Research octane number	106	91
Stoichiometric air/fuel ratio	9.0:1	14.79:1
LHV (MJ/kg)	26.9	42.9
LHV (MJ/L)	21.3	31.9
Enthalpy of vaporization (kJ/kg)	840	373
Temperature at boiling point (°C)	78.4	32.8

2.3 Experimental procedures

The engine was started and warmed up with gasoline fuel only. Once the oil temperature was stabilized at 368 ± 5 K, the quantity of the gasoline fuel was decreased and the ethanol fuel with equivalent energy was injected directly into the cylinder. All the engine tests were carried out at engine speed of 3500 rpm. Three samples were taken in each record and the average was used in analyzing the results. Error bars were used for some measurements with noticeable variation during the tests, as shown in Figs. 7 (volumetric efficiency) and 13 (lean burn limit) to show the variance.

To investigate the effect of SOI timing on engine knock mitigation, experiments were conducted at two engine load conditions, light load (IMEP 4.5–6.5 Bar) and medium load (IMEP 6.5–9 Bar) with EER (defined by Eq. (1)) of 24% and ethanol injection pressure of 40 Bar. The EER of 24% and direct injection pressure of 40 Bar were chosen because they were the conditions set for the best engine efficiency and emissions in the tested engine [30]. In each of the tests, the SOI timing was first adjusted to reach the designated value, then the spark timing was advanced from 15 CAD BTDC which was the original engine spark timing, until knock was detected. The engine knock was detected by monitoring the real-time cylinder pressure trace. If more than 10 of 100 consecutive cycles were detected with noticeable pressure wave oscillations (peak to peak oscillation over 1 Bar) around the peak cylinder pressure, this engine condition was regarded as knock condition. Fig. 3 illustrates a sample knock condition which was regarded with noticeable pressure wave oscillation. A MATLAB code was developed to detect the pressure wave oscillation. Once the knock condition was found, the spark timing was then retarded 2 CAD and marked as KLSA by adopting the method reported in [31]. The throttle position was fixed during each of the tests and AFR was kept at stoichiometric by adjusting the injection pulse-widths for gasoline and ethanol at each EER.

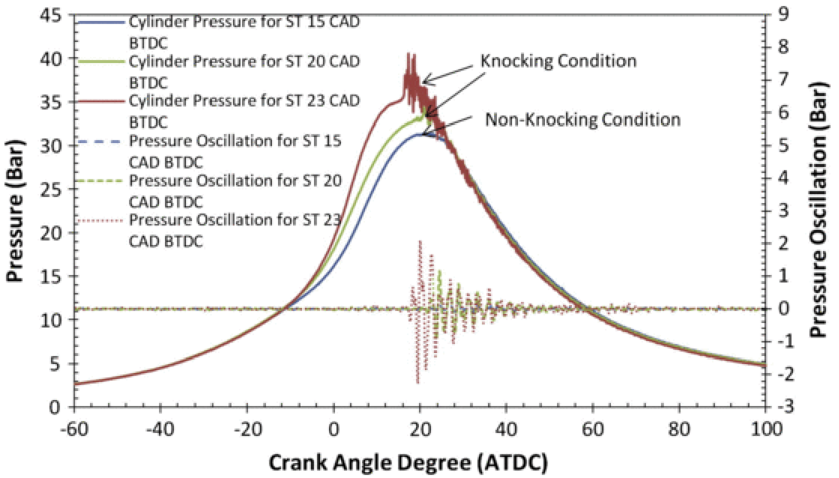

Fig. 3 Original cylinder pressure trace and 3–20 kHz band pass filtered pressure for knocking and non-knocking conditions at different Spark Timing (ST) (3500 rpm, throttle = 30% and gasoline fuel only).

In the experiments aimed to investigate the effect of SOI timing on lean burn, two test conditions with EERs of 24% and 48% were selected. The direct injection pressure for EER of 24% was 40 Bar and 90 Bar for EER of 48%. This was to keep the same EDI duration and to minimize the influence of end of injection timing on engine performance. Spark timing was set at 21 CAD BTDC and 25 CAD BTDC for EERs of 24% and 48% respectively, to provide more time for fuel evaporation and ensure stable combustion.

During the test of lean burn, the engine conditions of load and SOI timing were first set at the designated values with stoichiometric AFR. Then, the energy flow rates of ethanol and gasoline were fixed, and the throttle opening was increased to make the mixture increasingly lean. When the AFR was increased, the IMEP was first raised and reached the maximum with increase of AFR, then decreased with further increase of AFR. If the COV of IMEP was less than 10% during this process, the AFR for the IMEP which was 2% dropped from the maximum was regarded as the lean burn limit. This method of defining lean burn limit was suggested by Albert and Hedrick [32]. If the COV of IMEP was above 10%, the AFR would be decreased until the COV of IMEP was within 10% again. Then the corresponding AFR was recorded as the lean burn limit. Using 2% drop in IMEP as a marker to lean burn limit is because exploiting the penitential of EDI + GPI in improving engine efficiency is one of the major purposes in this study. The total energy input of both ethanol and gasoline fuels was kept unchanged in each of the tested condition, as previously described, thus the condition at which the IMEP reached the maximum was the condition with the maximum thermal efficiency. In order to ensure the combustion stability in the acceptable range, the upper combustion stability limit of 10% COV of IMEP was used as another marker to the lean burn limit. It should be noted that using 10% of COV of IMEP as upper limit is because the engine used for this test is a single cylinder air cooled engine with a relatively short stroke of 58 mm. Previous studies on similar small engines also used the 10% of COV of IMEP as the acceptance [33].

In each test of this investigation, the ethanol fuel SOI timing was advanced from 50 CAD BTDC to 110 CAD BTDC (late ethanol fuel direct injection, defined as LEDI) and 270 CAD BTDC to 330 CAD BTDC (early ethanol fuel direct injection, defined as EEDI) with 20 CAD intervals. Doing so was to compare the effects of SOI timing on knock mitigation before and after the inlet valve was closed. The ethanol fuel SOI timing from 120 CAD BTDC to 250 CAD BTDC was avoided due to the fluctuation of the IMEP during the transition of valve closing, as shown in Fig. 4. COV of IMEP and emissions of HC and CO were also found remarkably increased in this range. This result might be due to the turbulent flow passing the inlet port when the inlet valve was closing. 110 CAD BTDC was selected as the initial timing for LEDI, because the inlet valve was closed at 120 CAD BTDC. 50 CAD BTDC was chosen as the end point of LEDI to give minimum sufficient time for ethanol fuel to vaporize before the ignition.

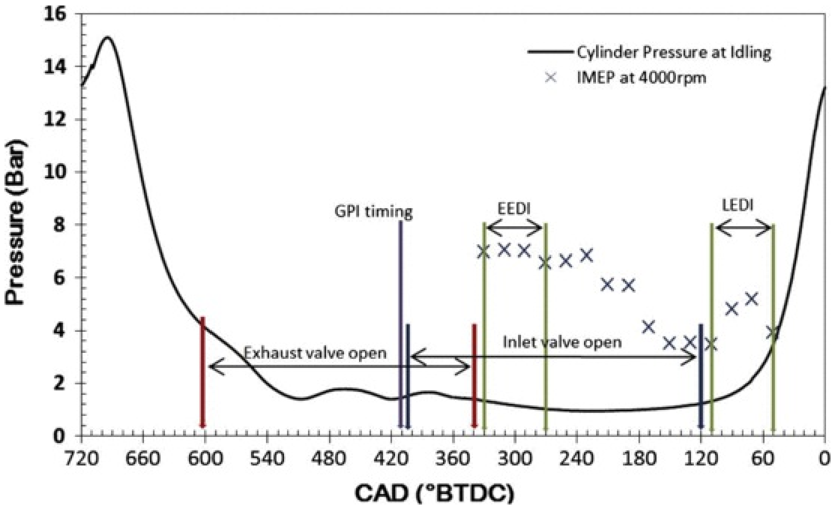

Fig. 4 Experimental engine SOI timing windows.

The EER is described in Eq. (1).

$$\text{Ethanol/gasoline energy ratio (EER)} = \dot{H}E_{\text{Ethanol}}/(\dot{H}E_{\text{Ethanol}} + \dot{H}E_{\text{Gasoline}}) \quad (1)$$

where ($\dot{H}$ *E*, rate of heating energy, kJ/s) = fuel mass flow rate (g/s) × LHV. The denominator in Eq. (1) is the rate of the total heating energy of the two fuels.

3 Results and discussion

In this section, the experimental results will be presented and discussed in two subsections: the effect of ethanol fuel SOI timing on knock mitigation and the effect of ethanol fuel SOI timing on lean burn. Results will include engine performance, combustion and emissions.

3.1 Effect of SOI timing on knock mitigation

As reviewed above, different SOI timing results in differences in charge cooling and mixture quality. So the effect of SOI timing on knock mitigation should be studied to find out which SOI timing range is more effective on suppressing engine knock and which SOI timing range can result in less pollutant emissions. At each tested SOI timing, the spark timing was advanced until the knock was detected from the cylinder pressure trace with the method described in Section 2.3. Gasoline only conditions were also tested to provide the reference frame for comparison.

The effect of ethanol fuel SOI timing on KLSA is illustrated in Fig. 5. As shown in the figure, the LEDI demonstrates greater potential in mitigating engine knock than EEDI. KLSA in LEDI conditions is advanced from 20 CAD BTDC (in GPI condition) to 28 BTDC at light load and 18 CAD BTDC (in GPI condition) to 25 CAD BTDC at medium load, whilst in EEDI conditions it can only be advanced to 25 CAD BTDC at light load and 22 CAD BTDC at medium load. As reviewed in Section 1, EEDI cools the gas at an earlier time, increasing the heat transfer from the wall to the gases. Thus, the cooling effect of fuel evaporation on the final charge temperature is compromised and the engine anti-knock ability is reduced. For LEDI, the heat transfer rate before fuel injection is lower, because the air temperature is higher. When fuel is injected late and cools the charge, the cooling effect by fuel evaporation can be well conserved, which leads to a lower in-cylinder temperature at the time of ignition as well as lower knocking tendency. Therefore, LEDI is more effective on suppressing engine knock than EEDI. GPI is least effective on suppressing the knock, as shown in Fig. 5. This may be related to two mechanisms. Firstly, In GPI conditions, gasoline fuel is injected into the intake port. Most of the fuel impinges to the metal surfaces of the port and valves [34]. The evaporation of this fuel relies mainly on heat transfer from the hot surface of the port and valve to the fuel film, which is unlike the directly injected ethanol fuel that vaporizes mainly by absorbing the heat from the charge in cylinder. Therefore, the evaporation of the gasoline fuel in GPI condition cools the charge less effectively than that in EDI conditions. Secondly, gasoline’s latent heat of vaporization and octane number are lower than ethanol’s.

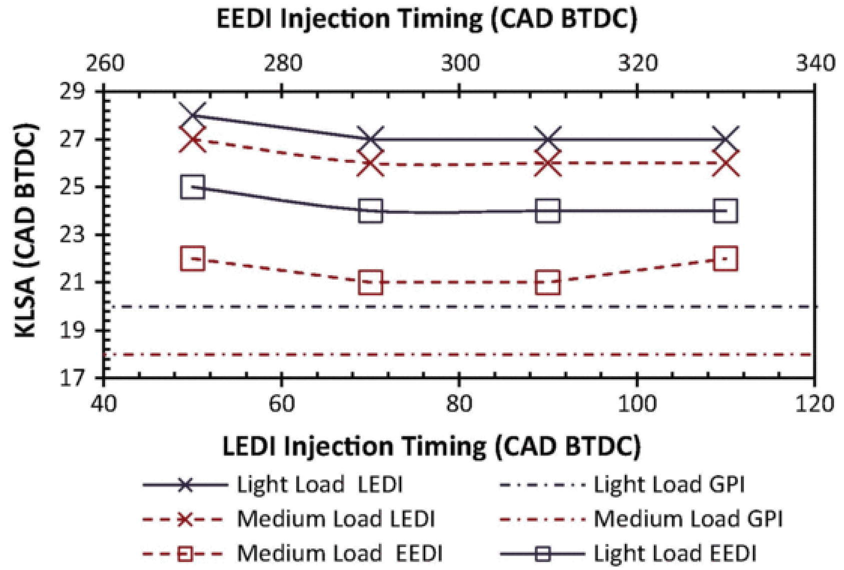

Fig. 5 Variation of KLSA with SOI timing.

Although the KLSA is more advanced in LEDI conditions than that in EEDI conditions as shown in Fig. 5, the IMEP does not increase with the advance of KLSA. As shown in Fig. 6, the IMEP values in EEDI conditions are greater than that in LEDI and GPI conditions, and almost independent of the SOI timing. IMEP in LEDI conditions decreases with the retard of SOI timing. When the SOI timing is retarded to be later than 90 CAD BTDC, the IMEP at both engine loads is lower than that in GPI conditions. This indicates that the effect of EEDI on engine power output improvement is stronger than that of LEDI, although its effect on knock mitigation is weaker than LEDI's. Advanced spark timing, increased volumetric efficiency (Fig. 7) and more homogeneous mixture which improves the combustion (Figs. 8 and 9) may contribute to the increase of IMEP in EEDI conditions. Among these factors, the mixture quality improved in EEDI conditions may play an important role in increasing the IMEP.

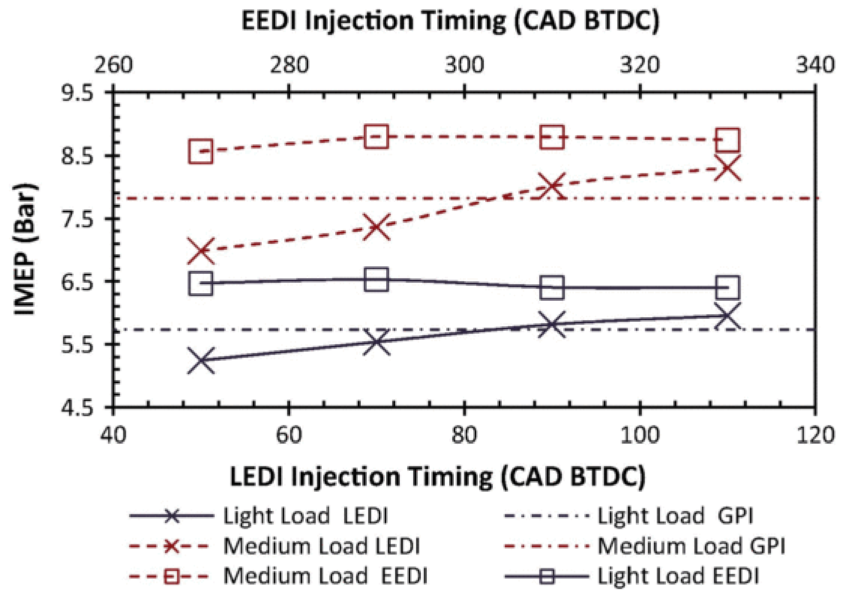

Fig. 6 Variation of IMEP with SOI timing at KLSA.

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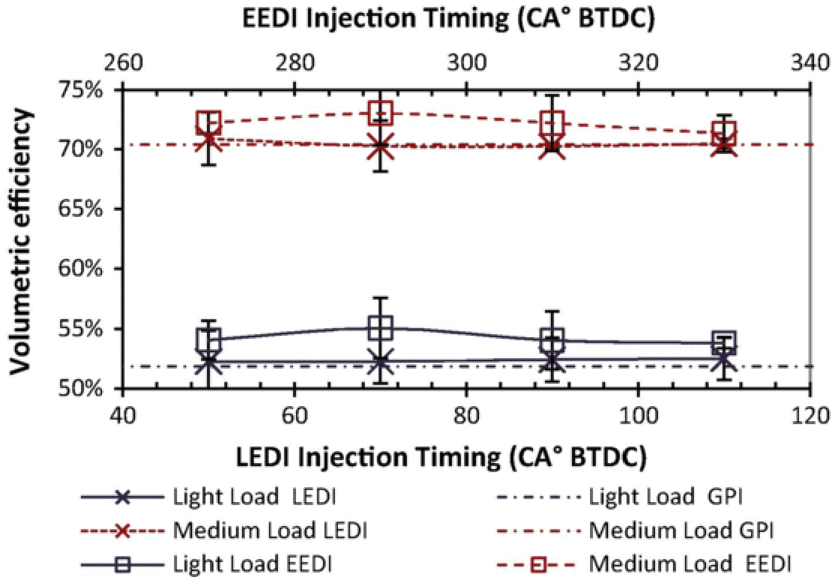

Fig. 7 Variation of volumetric efficiency with SOI timing at KLSA.

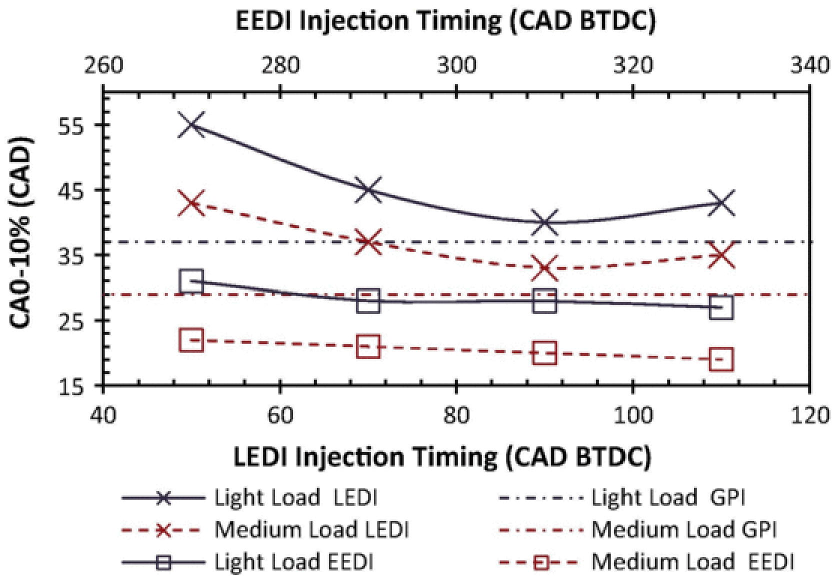

Fig. 8 Variation of CA0–10% with SOI timing at KLSA.

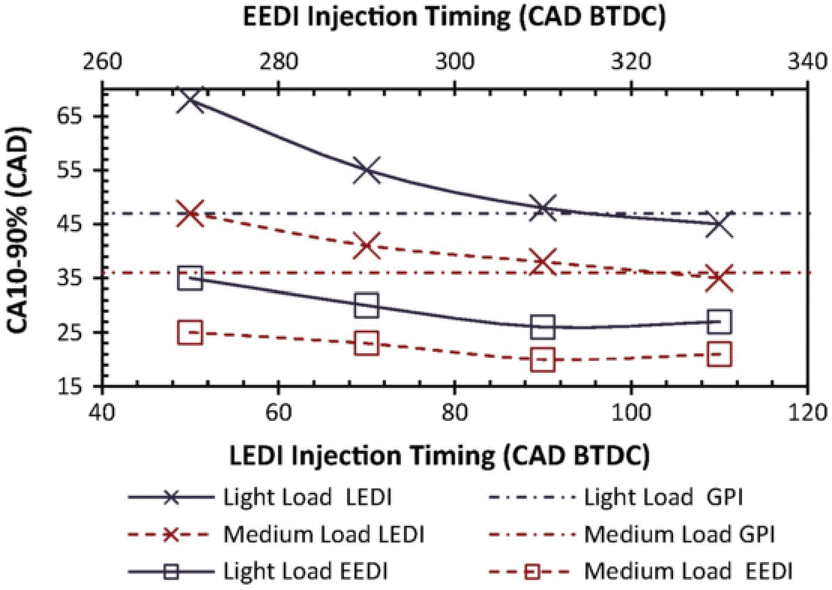

Fig. 9 Variation of CA10–90% with SOI timing at KLSA.

Fig. 7 shows the variation of volumetric efficiency with SOI timing at KLSA. The charge cooling effect of EDI on suppressing engine knock is realized mainly through reducing fresh charge temperature [24]. The charge density is also affected by the reduced temperature during this process which may ultimately influence volumetric efficiency. It can be seen in Fig. 7 that in both engine loads, the volumetric efficiency in EEDI conditions is higher than that in LEDI and GPI conditions. In EEDI conditions, the ethanol fuel is injected during the intake stroke. Volumetric efficiency can be increased by the reduced temperature and volume of the fresh charge due to fuel's evaporation. Moreover, the evaporation of ethanol requires more heat transferred from the fresh charge and the cylinder wall than evaporation of gasoline does. Thus the charge temperature can be further reduced, resulting higher volumetric efficiency in EEDI condition.

As shown in Fig. 7, the volumetric efficiency first increases with the advance of SOI timing until it reaches the peak value at SOI timing of 290 CAD BTDC. Then it decreases gradually with further advance of SOI timing when it is earlier than 290 CAD BTDC. The low volumetric efficiency at SOI timing of 270 CAD BTDC may be due to the reduced time for the fuel evaporating, which results in less effective charge cooling. The decrease of volumetric efficiency with the advance of SOI timing earlier than 290 CAD BTDC may be due to the fuel impingement onto the piston surface. At SOI timings of 310 CAD BTDC and 330 CAD BTDC, the piston just passes the top dead center (TDC) and the ethanol fuel spray may easily impinge onto the piston surface. Thus, the evaporation of ethanol fuel may mainly take the heat away from the piston surface and cylinder wall rather than from the fresh charge. The volumetric efficiency decreases subsequently.

In LEDI conditions, the volumetric efficiency is almost independent of the variation of SOI timing. This is because the ethanol fuel is injected after the inlet valve is closed and DI timing has no effect on the fresh charge. Therefore, the volumetric efficiency stays stable. As shown in Fig. 7, the volumetric efficiency in LEDI condition is slightly higher than GPI condition. In EDI + GPI mode, part of the fuel energy is provided by directly injected ethanol, so the energy provided by port injected gasoline is reduced. The partial pressure of gasoline fuel decreases and the partial pressure of the fresh air increases due to the reduced gasoline volume fraction. This improvement contributes to the increase of volumetric efficiency. Thus, the volumetric efficiency in LEDI condition is slightly higher than that in GPI condition.

Fig. 8 shows the variation of combustion initiation duration with SOI timing at KLSA. The combustion initiation duration, denoted as CA0–10%, is defined by the crank angle degrees starting from spark ignition to the timing for 10% of fuel mass fraction burnt (MFB). The combustion initiation duration directly relates to the combustion stability and is sensitive to the mixture homogeneous quality and in-cylinder temperature before the ignition [26]. Normally, long combustion initiation duration indicates that the mixture is hard to be ignited. Engine efficiency can be negatively affected because of the poor ignition. Therefore, the present study is aimed to find an SOI timing range at which the engine knock is effectively mitigated and the combustion initiation duration is not largely elongated. As shown in Fig. 8, the CA0–10% in EEDI conditions is shorter than that in LEDI and GPI conditions. It gradually decreases with the advance of SOI timing in both EEDI and LEDI conditions except at SOI timing of 110 CAD BTDC at which the CA0–10% increases slightly. The minimal values of CA0–10% in LEDI conditions are 40 CAD at light load and 33 CAD at medium load when SOI timing is 90 CAD BTDC. The minimal values of CA0–10% in EEDI conditions are 27 CAD at light load and 19 CAD at medium load when SOI timing is 330 CAD BTDC. This result indicates that EEDI is more suitable than LEDI for the present study, because it results in better combustion. The decrease of CA0–10% with the advance of SOI timing may be because advancing SOI timing increases the time for ethanol fuel vaporization, which results in more homogeneous mixture and high in-cylinder temperature. The slight increase in CA0–10% at SOI timing of 110 CAD BTDC can be attributed to the weak in-cylinder flow during ethanol injection at this timing (inlet valve is just open), which decreases the mixture homogeneous quality and prolongs the combustion initiation duration.

Fig. 9 shows the variation of major combustion duration, CA10–90%, with SOI timing at KLSA. The major combustion duration, CA10–90%, is defined by the crank angle degrees starting from 10% of MFB and ending with 90% of MFB. It is one of the major parameters which characterize the combustion and affect the engine thermal efficiency. Normally, the longer the combustion duration is, more heat will be lost through the cylinder wall. As it can be seen in Fig. 9, CA10–90% in EEDI conditions is shorter than that in LEDI and GPI conditions. On average, the CA10–90% in EEDI conditions was 20 CAD shorted than that in LEDI conditions. This result indicates that although LEDI is effective on knock mitigation, the major combustion duration is substantially increased in LEDI condition. The longest CA10–90% is 68 CAD at SOI timing of 50 CAD BTDC and the shortest CA10–90% is 20 CAD at 290 CAD BTDC. CA10–90% generally decreases with the advance of SOI timing in all the tested conditions. This decrease in CA10–90% can be attributed to the improved fuels/air mixture quality when SOI timing is earlier. The CA10–90% result, here again, proves that the EEDI results in better combustion and this should be one of the main factors that contribute to the high IMEP in EEDI condition.

The effect of SOI timing on indicated thermal efficiency at KLSA is shown in Fig. 10. As shown in the figure, the peak indicated thermal efficiency in EDI + GPI condition is 33.5% at SOI timing of 290 CAD BTDC and medium load. Whereas, the indicated thermal efficiency in GPI condition is only 29% because the advent of knock constraints further advance of spark timing. The indicated thermal efficiency in EEDI conditions is almost independent with the SOI timing and is greater than that in LEDI and GPI conditions. Indicated thermal efficiency in LEDI conditions, gradually increases with the advance of SOI timing. It becomes greater than that in GPI condition when the SOI timing is earlier than 70 CAD BTDC. This result indicates that EEDI results in higher indicated thermal efficiency than LEDI or GPI does. The advanced spark timing shown in Fig. 5 and reduced combustion duration shown in Fig. 9 may be the main factors that contribute to the higher efficiency in EEDI conditions. In addition, the increased volumetric efficiency [34] and ethanol’s high energy content of stoichiometric mixture per unit mass of air [35] may also contribute to the increase of indicated thermal efficiency.

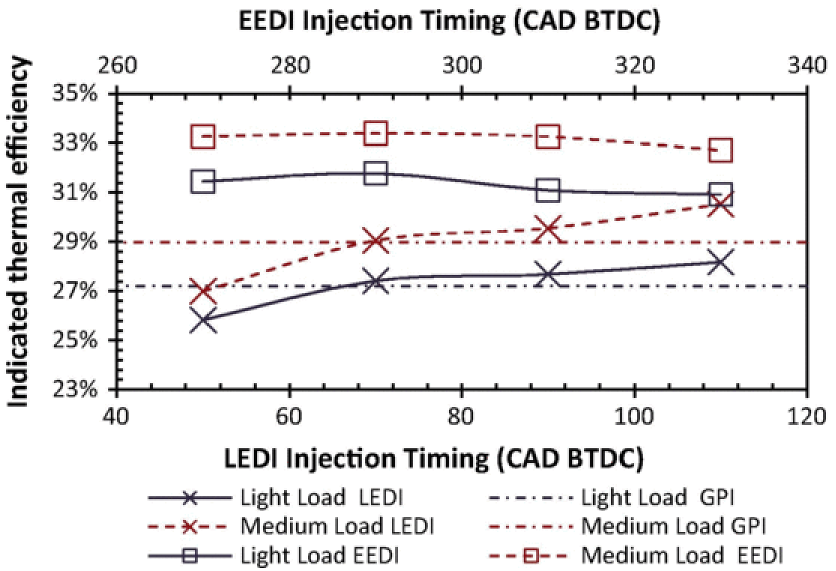

Fig. 10 Variation of indicated thermal efficiency with SOI timing at KLSA.

When the engine is running at KLSA, minimal change of spark timing can lead to moderate variation in engine power output but substantial changes in NO_x and HC emissions [36]. The variations of indicated specific nitrogen oxides (ISNO) and indicated specific hydrocarbon (ISHC) with SOI timing at KLSA are shown in Figs. 11 and 12, respectively. As shown in Fig. 11, ISNO in LEDI conditions is less than that in GPI and EEDI conditions. At both loads, ISNO gradually increases with the advance of SOI timing in EEDI and LEDI conditions.

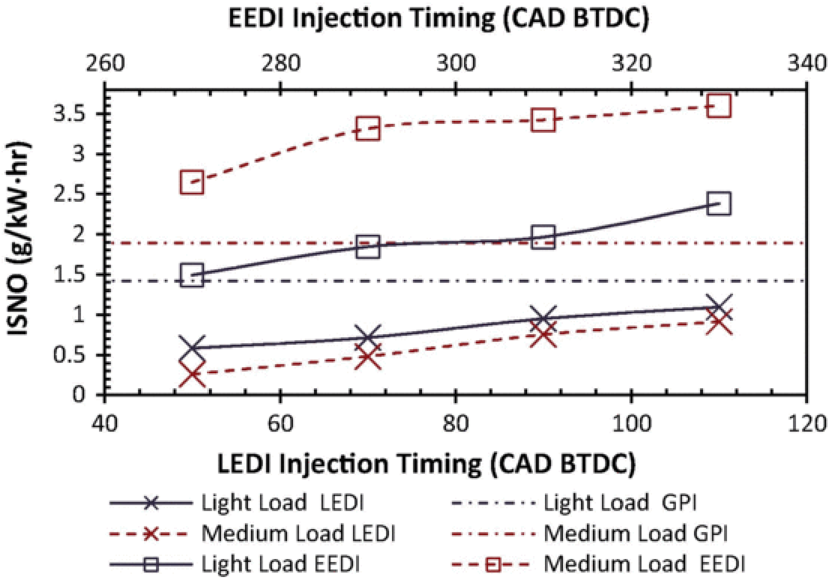

Fig. 11 Variation of ISNO with SOI timing at KLSA.

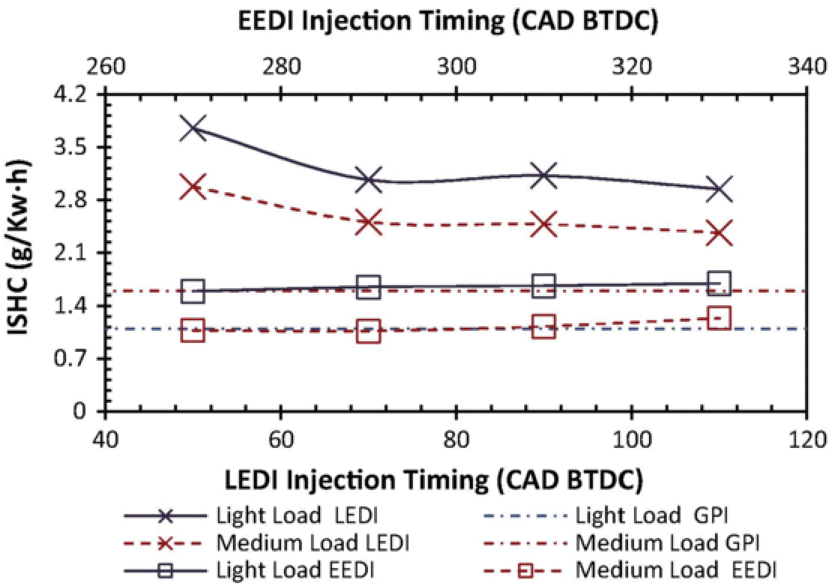

Fig. 12 Variation of ISHC with SOI timing at KLSA.

The formation of NO_x is related to in-cylinder temperature. Firstly, when the fresh charge is cooled at early time in EEDI condition, the time for heat transfer from the wall to the gases is increased and the cooling effect on the final charge is reduced. Thus, the combustion temperature in EEDI condition is higher than that in LEDI condition and the NO emission increases consequently. Secondly, Injecting ethanol when the piston just passes the TDC (at SOI timing from 300 CAD BTDC to 330 CAD BTDC) may lead to severe wall-wetting. Most of the injected ethanol fuel may be vaporized by absorbing heat from the piston head and cylinder wall, but not from the fresh charge. Thus the cooling effect on fresh charge may be reduced. NO emissions at SOI timing ranged from 300 CAD BTDC to 330 CAD BTDC increase. Finally, advancing SOI timing results in improved combustion as shown in Figs. 7 and 9 and increased in-cylinder temperature which is due to more time available for mixture heat recovery. The combustion temperature is therefore increased and NO emissions increase as well.

Fig. 12 shows the variation of ISHC with SOI timing at KLSA. As shown in Fig. 12, more HC emissions were generated in LEDI conditions than that in EEDI and GPI conditions. The result of high HC emissions supports the previous discussion about the poor mixture quality in LEDI condition, resulting in more unburned combustion products such as HC [7]. As shown in Fig. 12, at both loads, ISHC decreases with the advance of SOI timing in LEDI conditions but almost independent with the advance of SOI timing in EEDI conditions. At medium load, the ISHC in EEDI conditions is lower than that in GPI condition while at light load, the ISHC in EEDI conditions is slightly higher than corresponding GPI condition. The maximal value of ISHC is 3.75 g/kW h at SOI timing of 50 CAD BTDC at light load and the minimum is 1.07 g/kW h at SOI timing of 270 CAD BTDC at medium load. The decrease of ISHC with the advance of SOI timing in LEDI conditions may be because advancing SOI timing provides more time for fuel vaporization. Thus, the combustion is more complete and ISHC decreases.

3.2 Effect of injection timing on lean combustion

In order to further explore the potential in using EDI to improve GPI engine efficiency and reduce emissions, lean burn has been investigated in this study. According to the review in Section 1, stratified combustion with fuel directly injected later in the compression stroke (LEDI) has more potential in extending the lean burn limit than homogeneous lean combustion (EEDI and GPI). However, stable combustion in stratified condition is hard to achieve. The SOI timing needs to be carefully optimized to ensure stable combustion [20]. Therefore, the effect of SOI timing on EDI + GPI engine lean burn is worth to be investigated. In experiments aimed for investigating lean burn, the engine speed was kept at 4000 rpm and load was in a medium range (IMEP 6.5–9 Bar). Higher engine speed and load conditions may lead to unstable engine operation in lean burn as found by Küsell et al. [37]. Conditions with EERs of 24% and 48% were tested. GPI condition was also tested to provide comparison.

Fig. 13 shows the variation of lean burn limit with SOI timing at two EERs of 24% and 48%. As it can be seen in the figure, EEDI can keep the engine operate at leaner conditions than LEDI and GPI. The lean burn limit, defined by the maximum λ achievable, in EEDI conditions is greater than that in LEDI and GPI conditions. At both EER levels of 24% and 48%, the lean burn limit first reaches its peak at SOI timing of 290 CAD BTDC in EEDI condition and 90 CAD BTDC in LEDI condition. Then, it begins to slightly decrease with further advance of SOI timing. EER of 24% seems to be more suitable for extending lean burn limit than EER of 48% does. The lean burn limit reaches its maximum of 1.29 when EER is 24% at SOI timing of 290 CAD BTDC. The minimum lean burn limit exists at EER of 48% in LEDI conditions, which is only slightly greater than the stoichiometric AFR (λ = 1) while the other lean burn limits are all above λ = 1.1.

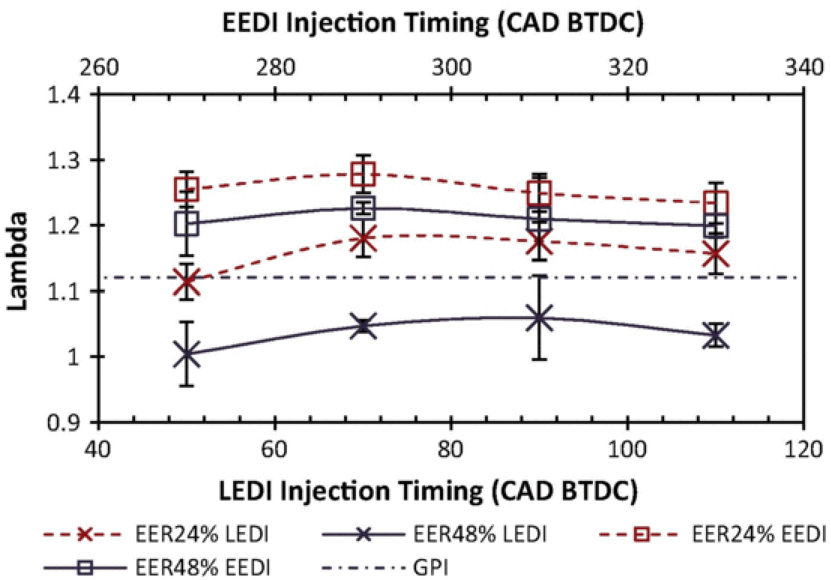

Fig. 13 Variation of lean burn limit with SOI timing at EERs of 24% and 48%.

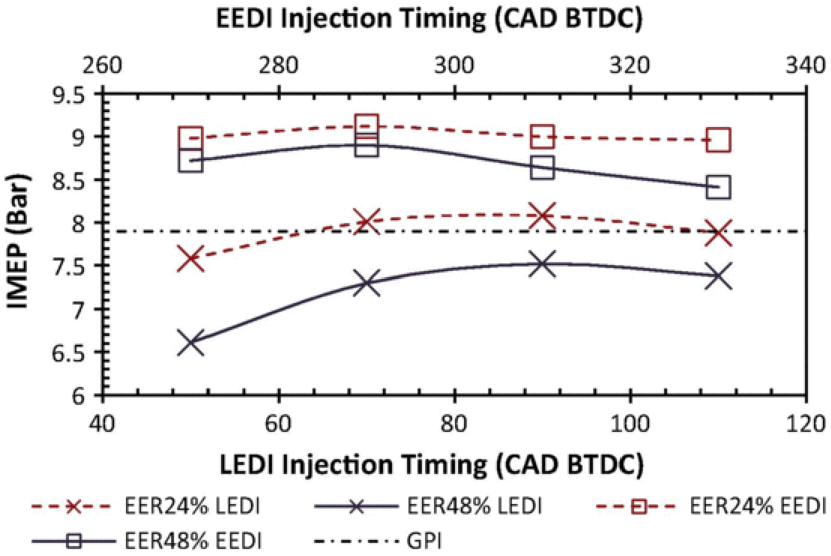

Fig. 14 Variation of IMEP at lean burn limit.

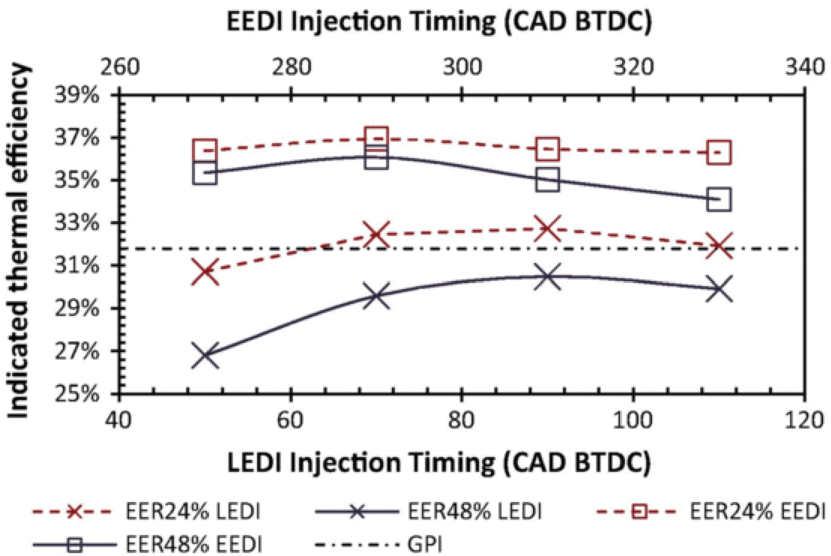

Fig. 15 Variation of indicated thermal efficiency at lean burn limit.

As reviewed, LEDI produced stratified charge in adjacent to the spark plug, which theoretically has more potential in extending the lean burn limit than homogeneous lean burn (EEDI and GPI) [20]. However in the present study, the effect of LEDI on extending lean burn limit is weaker than that of EEDI. One possible explanation to this result is that LEDI might lead to severe fuel impingement on the piston crown and cylinder wall as reported in [21]. This wall-wetting negatively affects the mixture formation, combustion, decreasing combustion stability (Fig. 16) and elongating combustion duration (Fig. 18). Therefore, the lean burn limit in LEDI conditions is low. The top lean burn limit at SOI timing of 290 CAD BTDC may be due to the improved mixture quality. Similar engine performance improvement at this SOI timing is also shown in Figs. 6 and 7.

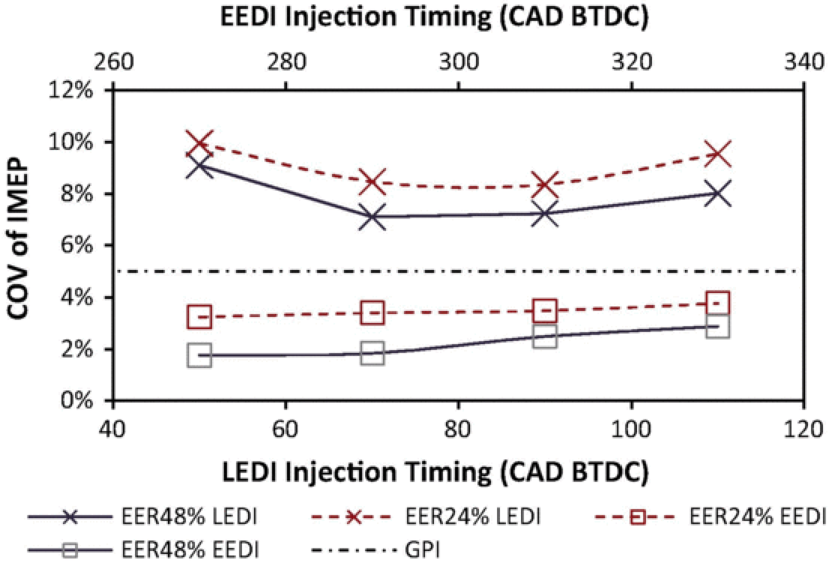

Fig. 16 Variation of COV of IMEP at lean burn limit.

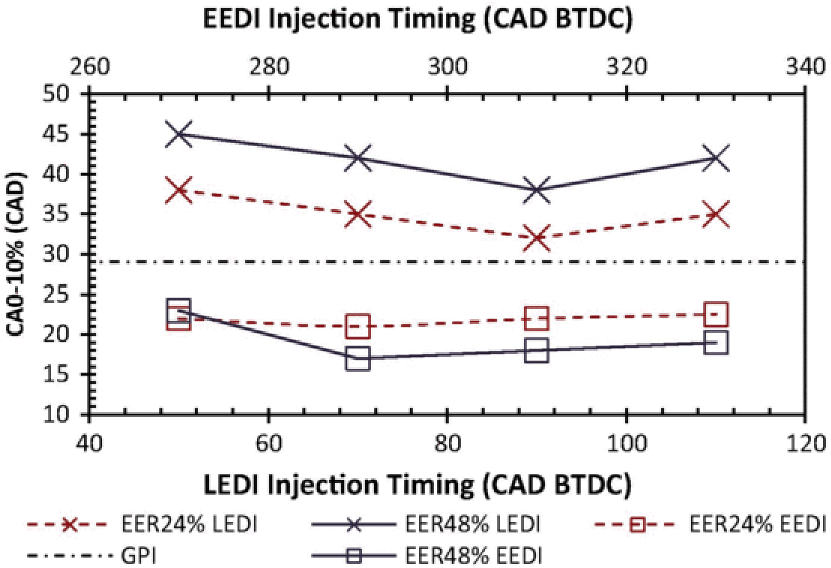

Fig. 17 Variation of CA0–10% at lean burn limit.

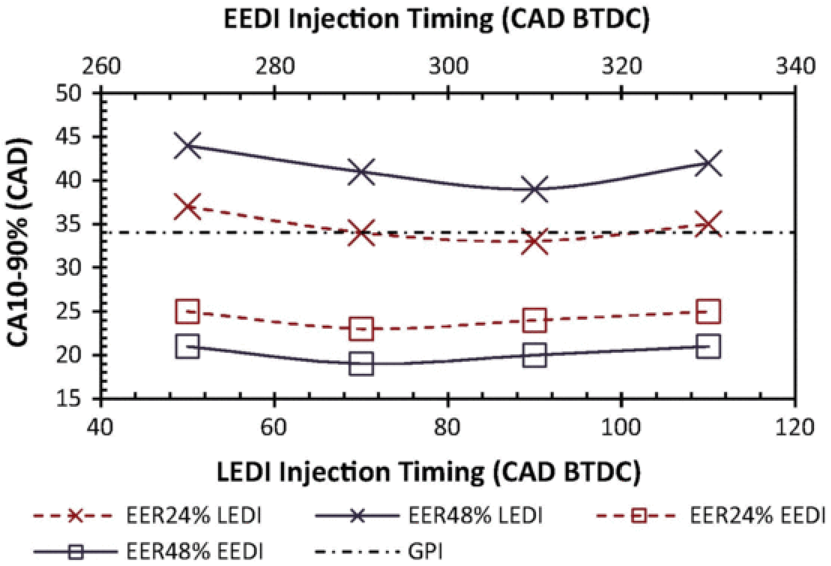

Fig. 18 Variation of CA10–90% at lean burn limit.

The variation of IMEP with SOI timing at lean burn limit is shown in Fig. 14. It can be seen that IMEP is greater in EEDI conditions than that in LEDI and GPI conditions. IMEP at EER of 24% is higher than that at EER of 48% in both LEDI and EEDI conditions. As shown in Fig. 13, IMEP is varied with lean burn limit. The IMEP increases when the lean burn limit increases, and decreases when the lean burn limit reduces. This result is mainly due to the 2% drop in IMEP from the maximum to define the lean burn limit.

The effect of SOI timing on indicated thermal efficiency at lean burn limit is shown in Fig. 15. As it can be seen in the figure, the indicated thermal efficiency is greater in EEDI condition than that in LEDI and GPI conditions. The peak indicated thermal efficiency is 36.8% at SOI timing of 290 CAD BTDC and EER of 24%. In EEDI conditions, the indicated thermal efficiency at both EER levels (24% and 48%) first slightly increases until it reaches the maximum at SOI timing of 290 CAD BTDC, and then decreases with further advance of SOI timing. In LEDI conditions, the indicated thermal efficiency first increases with the advance of SOI timing and then begins to decrease when the SOI timing is earlier than 90 CAD BTDC.

Results shown in Fig. 15 may be related to the quality of ethanol/air mixture. When the SOI timing is at 330 CAD BTDC and 270 CAD BTDC, the mixture quality may be negatively affected by wall-wetting and reduced time for fuel evaporation respectively, as stated in the discussion for Fig. 7. Therefore the combustion quality deteriorates and the indicated thermal efficiency reduces. In LEDI conditions, advancing SOI timing provides more time for ethanol evaporating and heat recovery. As a consequence, the mixture quality is improved and the in-cylinder temperature is increased. Thus the combustion is improved and indicated thermal efficiency is increased.

The variation of COV of IMEP at lean burn limit is shown in Fig. 16. When the mixture is leaned, the combustion becomes unstable. It is therefore necessary to monitor the COV of IMEP to keep the combustion stability in the acceptable range (10% in this study). As shown in Fig. 16, the COV of IMEP in EEDI conditions is almost independent with the SOI timing and stays around 3.8% at EER of 24% and 2.1% at EER of 48% when the SOI timing varies. In LEDI conditions, the COV of IMEP reaches its minimum in the SOI timing range between 70 CAD BTDC and 90 CAD BTDC. At SOI timing of 50 CAD BTDC, the COV of IMEP at EER of 24% is close to 10% and at EER of 48%, it approaches 9%. The minimum COV of IMEP, 1.7%, occurs at SOI timing of 270 CAD BTDC and EER of 48%. COV of IMEP in GPI condition is 4.8%, which is greater than that in EEDI conditions and lower than that in LEDI conditions.

Previous investigation showed that the COV of IMEP gradually decreased with the increase of EER at original spark advance (15 CAD BTDC) and stoichiometric AFR, possibly due to ethanol's high laminar combustion velocity and better low temperature combustion stability [38]. In present work, high EER level also shows the improvement to combustion stability in lean burn. As it shown in Fig. 16, in both EEDI and LEDI conditions, COV of IMEP at EER of 48% is lower than that at EER of 24%.

The variations of CA0–10% with SOI timing at lean burn limit is shown in Fig. 17. This result directly reflects the mixture ignition characteristic in lean conditions. It can be seen that CA0–10% in EEDI conditions is shorter than that in LEDI and GPI conditions at both EERs of 24% and 48%. In EEDI conditions, CA0–10% is almost independent with SOI timing except at 50 CAD BTDC where the CA0–10% at EER of 48% increases sharply. In LEDI conditions, CA0–10% first decreases with the advance of SOI timing and reaches the minimum at SOI timing of 90 CAD BTDC, then it increases with further advance of SOI timing. At both EERs of 24% and 48%, the CA0–10% in EEDI conditions is less than 24 CAD, and in LEDI conditions, it is greater than 30 CAD. CA0–10% in GPI condition is 29 CAD which lies between that in EEDI and LEDI conditions.

The variation of CA0–10% is related to the combustion stability (COV) [39]. Previous studies have found a correlation between COV of IMEP and CA0–10%. It was reported that when the CA0–10% was over certain range (30 CAD in their study), the COV began to increase substantially [39]. In present work, the CA0–10% of 30 CAD plays a similar role as a threshold. As it can be seen in Figs. 14 and 15, when CA0–10% is longer than 30 CAD, the COV of IMEP stays at high level which is greater than 6.5%. When the

CA0–10% is shorter than 30 CAD, the combustion stays stable and COV of IMEP is less than 4% (in EEDI conditions).

The variation of CA10–90% with SOI timing at lean burn limit is shown in Fig. 18. As it shown in the figure, CA10–90% generally follows the same trend as that of CA0–10%. In EEDI conditions, CA10–90% first decreases and reaches the minimum at SOI timing of 270 CAD BTDC, then it increases with further advance of SOI timing. In LEDI conditions, CA10–90% first slightly decreases, then raises with the advance of SOI timing. CA10–90% in GPI condition is 34 CAD which is longer than that in EEDI conditions but shorter than that in LEDI condition. It can be also seen from Fig. 18 that in LEDI conditions, CA10–90% at EER of 48% is higher than that at EER of 24%. This may be because high EER (48%) level leads to more fuel impingement onto the cylinder surface due to the increased DI fuel amount, which ultimately prolongs the combustion duration.

The effects of SOI timing on ISNO emission at lean burn limit is shown in Fig. 19. It can be seen that in EEDI conditions, ISNO at both EERs is almost independent with SOI timing. ISNO at EER of 48% is about 50% greater than that at EER of 24%. The high ISNO at EER of 48% in EEDI condition may be because at EER of 48% the engine lean burn limit is lower than that at EER of 24% (Fig. 13). Richer mixture may result in higher in-cylinder temperature which increases the formation of NO_x emissions. It can be also seen in Fig. 19 that in LEDI conditions, the ISNO is lower than that in GPI and EEDI conditions, and slightly increases with the advance of SOI timing. The increase of ISNO with SOI timing may be because the advance of SOI timing provides more time for heat transfer from cylinder wall to fresh charge, which finally contributes to the increase of combustion temperature.

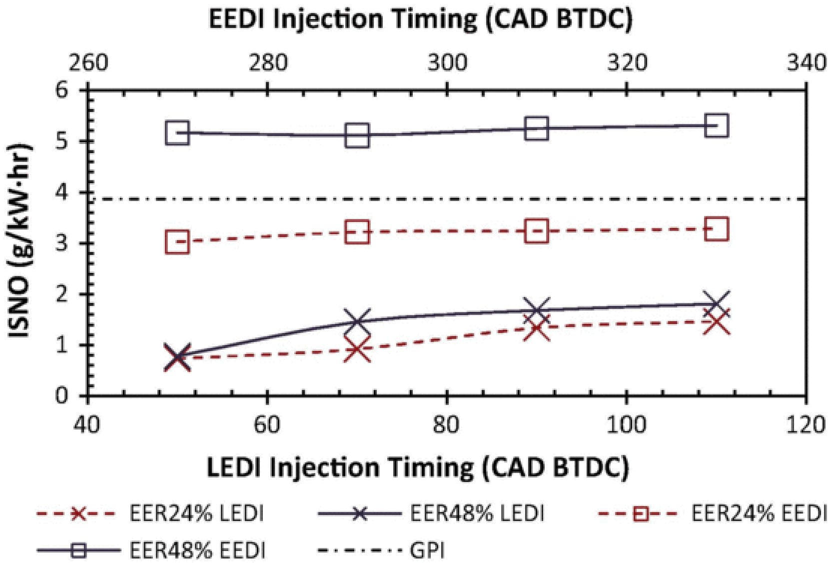

Fig. 19 Variation of ISNO at lean burn limit.

In certain AFR range (14.7–17.6) [20], lean mixture has the great potential in reducing HC emissions. The effect of SOI timing on ISHC at lean burn limit is shown in Fig. 20. It can be seen that at both EERs, ISHC in EEDI conditions is less than that in LEDI and GPI conditions. In EEDI conditions, ISHC is almost independent with SOI timing and is less than 0.62 g/kW h. The low ISHC in EEDI conditions can be attributed to high lean burn limit (Fig. 13) and better mixture quality. Leaner mixture can provide more oxygen for HC oxidization, thus resulting in less HC emissions. Early fuel direct injection (EEDI) enables more time for fuel evaporation which improves mixture quality and leads to more complete combustion. Therefore, HC emissions are decreased in EEDI conditions due to the above two reasons. In LEDI conditions, ISHC at EER of 24% is almost independent with SOI timing, and ISHC at EER of 48% gradually decreases with the advance of SOI timing. The decrease of ISHC with advance of SOI timing at EER of 48% may be because advancing SOI timing provides more time for improving mixture homogeneity.

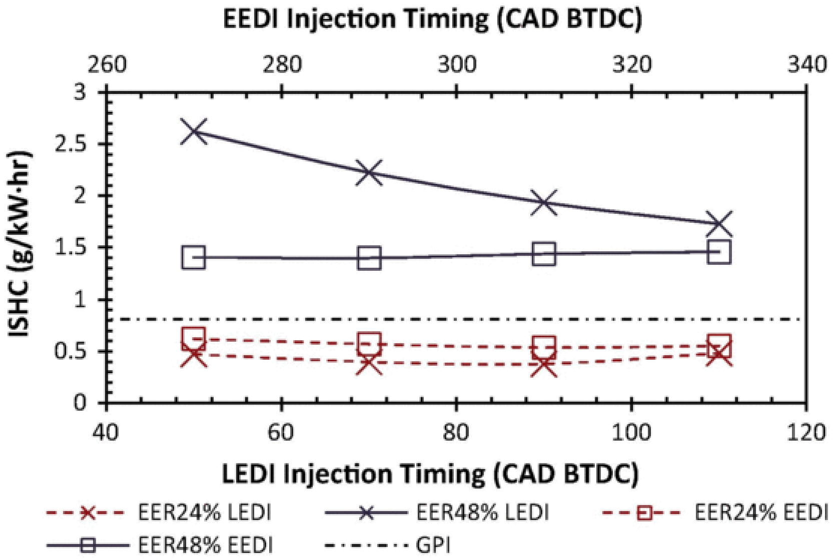

Fig. 20 Variation of ISHC at lean burn limit.

4 Conclusions

Experiments were conducted on a self-developed single cylinder SI engine equipped with EDI + GPI system. The effect of SOI timing on knock mitigation and lean burn was investigated. Results at two EER levels of 24% and 48% and injection strategies of LEDI and EEDI were compared. Based on the analysis of experimental results, the following conclusions can be drawn.

1. The engine knock could be effectively suppressed by LEDI. More advanced spark timing without knocking was allowed in LEDI conditions than that in EEDI conditions. IMEP and indicated thermal efficiency were low in LEDI conditions, possibly due to poor mixture quality which deteriorated the combustion.
2. EEDI was less effective than LEDI on knock mitigation. However, IMEP and engine thermal efficiency at KLSA in EEDI conditions were greater than that in LEDI conditions. When EER was 24%, SOI timing at 290 CAD BTDC was the optimum for minimum NO emissions and maximum engine efficiency.
3. EEDI was more effective on extending the lean burn limit than LEDI. The maximum lean burn limit (λ) achieved by EEDI was 1.29. COV of IMEP and emissions in EEDI conditions were less than that in LEDI conditions. LEDI only slightly increased lean burn limit which was just over stoichiometric AFR. Poor mixture quality may be the main reason to the low lean burn limit. IMEP in EEDI conditions was greater than that in LEDI conditions. EER of 48% resulted in lower IMEP and higher HC and NO emissions than that at EER of 24% in both EEDI and LEDI conditions.

5 Uncited references

[28,29].

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Highlights

- Ethanol direct injection + gasoline port injection was experimentally investigated.
- Late ethanol direct injection (LEDI) was effective on suppressing engine knock.
- Early ethanol direct injection (EEDI) resulted in higher efficiency than LEDI did.
- EEDI was more effective on extending the lean burn limit than LEDI was.

Queries and Answers

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