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Preliminary investigation to combustion in a SI engine with direct ethanol injection and port gasoline injection (EDI+GPI)

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Abstract

Ethanol fuel, as a renewable fuel can play an important role in addressing the critical issue of energy resources if it is used in a proper way. Ethanol direct injection plus gasoline port injection (EDI+GPI) is such a new way to enable substantial improvement in engine efficiency and emission reduction in spark ignition engines. This paper reports our preliminary investigation to the combustion and emissions in this new dual fuel injection system. Experiments were conducted on a single-cylinder spark ignition engine equipped with EDI+GPI. In the experiments, the ethanol/gasoline volumetric percentage (EVP) was varied from 0% (gasoline fuel only) to 71%. Mass burnt fraction and indicated mean effective pressure (IMEP) were calculated from the measured cylinder pressure for analysing the combustion process. The variance of IMEP, reduced with the increased EVP, showed that the combustion stability was improved by the direct injection of ethanol fuel. The effect of EVP on initial, early and major combustion time periods showed that ethanol fuel's higher combustion velocity and low ignition energy might contribute to accelerating the flame propagating, shortening the combustion periods and reducing the combustion temperature when EVP was less than 48%. However further increase of EVP when it was over 48% resulted in a negative effect on combustion which might be caused by the ethanol fuel's over cooling effect. Hydrocarbon and carbon monoxide emission increased and nitric oxide emission decreased with the increase of EVP.

Introduction

Ethanol has long been regarded as an alternative and renewable fuel for spark-ignition (SI) engines. It has the potential to effectively improve engine efficiency by allowing greater compression ratio and to reduce the pollutant emissions by providing more oxygen in combustion process and lowering the in-cylinder temperature. However, the current method of pre-mixing ethanol and gasoline fuels cannot fully exploit ethanol's merits. The ethanol's low lower heating value, low volatility and other properties may play a negative role to engine performance by reducing the vehicle coverage, making the engine cold start more difficult and so on. Moreover, due to the fixed the ethanol/gasoline ratio, the ethanol's potentials in emissions decreasing and knock suppression cannot be fully exploited. Because previous research had already proved that different engine loads require different ethanol/gasoline ratios to produce the best emissions reduction results and effectively reducing engine knock [1, 2].

EDI+GPI provides a new way of using ethanol fuel to improve engine efficiency whilst maintaining a relatively low emissions. Because of the individual ethanol fuel direct injection system, the use of ethanol fuel can be independently controlled for the purposes of engine knock mitigation, combustion optimization and emission reduction based on different engine operating conditions. Thus the engine is capable to be greatly downsizing and to meet the more stringent emission standards.

The new technology has brought new challenges which require good understanding of the engine combustion in EDI+GPI. In previous research on ethanol fuel used in direct injection (DI) engine, it was found that the ethanol's high latent heat of vaporization could greatly reduce the in-cylinder temperature before combustion took place, resulting in longer combustion initiation period and enhanced engine anti-knock ability [3, 5]. It was found that the laminar burning velocity of the ethanol fuel was higher than that of the gasoline fuel, reducing the combustion duration and the time for heat loss through the cylinder wall [6, 7]. It was also noticed that the combustion temperature of ethanol fuel was lower than that of the gasoline fuel. The NO_X emissions were reduced by this lower combustion temperature as well as the charge cooling effect enhanced by the ethanol fuel [8]. Furthermore, the ethanol's mole fraction of combustion product and volumetric calorific value of ethanol/air mixture was higher than that of gasoline and this contributed to the increase of cylinder pressure [9, 10].

EDI+GPI is a new combustion module to enable substantial improvement in engine efficiency and emission reduction in internal combustion engines. To develop this new engine technology, investigation to its combustion characteristics is required. This paper reports our experimental investigation to the effect of ethanol/gasoline volumetric ratio on combustion characteristics and exhaust gas emissions using a self-developed EDI+GPI single cylinder research engine.

Experimental apparatus and methods

A 250cc motor cycle engine YBR250 was selected and modified in this study. It is a four-stroke single-cylinder SI gasoline engine originally equipped with port fuel injection and electronic control of engine speed. Its specifications are listed in Table 1.

Engine type	Single cylinder, air cooled, 4 stroke, SOHC.
Displacement	249.0 cc
Bore × stroke	74.0mm x 58.0 mm
Compression ratio	9.8:1
Lubrication system	Wet sump
Maxim power	15.4KW at 7500 RPM

Table 1 - Specifications of YBR250

The engine was modified by adding an ethanol fuel direct injection system and a new ECU which replaced the original ones and provided the flexibility of manual adjustment of parameters in engine operation. Figure 1 is a schematic of the research engine system. The engine was coupled to an eddy current dynamometer to maintain the engine speed. Air flow was stabilized by a 80L air butter and controlled by adjusting the throttle position. A Kistler 6115B measuring spark plug pressure transducer was used to measure the cylinder pressure and samples were taken at 0.5 crank angle degree (CAD) intervals for 100 consecutive cycles. The exhaust gas emissions were measured using a Horiba MEXA-584L gas analyser. The sample exhaust gas was taken at a position 0.4 meter from the exhaust valve and upstream of the three-way catalyst converter. More details about the test engine are reported in [11].

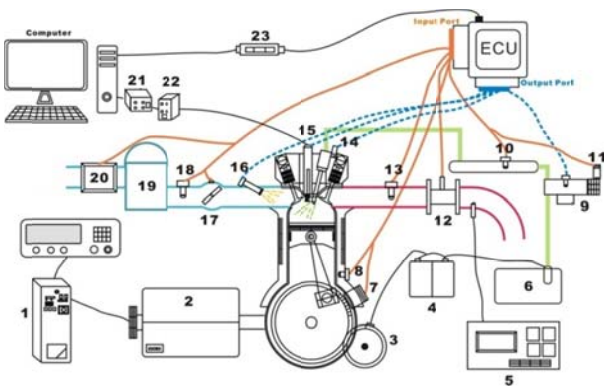

Figure 1- Schematic of the research engine system
1. Dynamometer controller 2. Dynamometer 3. Start motor 4. Battery 5. Horiba MXEA-584L gas analyser 6. Ethanol fuel tank 7. Encoder on crankshaft 8. Temperature sensor 9. High pressure fuel pump10. Common rail pressure sensor 11. Encoder on high pressure pump shaft 12. Bosch wide-band lambda sensor 13. Temperature Sensor14. 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 analyser 22. Charge amplifier 23. CAN Communication module

Experiments were conducted at stoichiometric air/fuel ratio (λ=1) in two engine conditions, Case 1 and Case 2. The test matrix is shown in Table 2. Case 1 is light load engine condition and Case 2 is medium load engine condition. The engine speed was 3500rpm and the spark timing was set to be the maximum brake torque (MBT) timing. The engine was started and warmed-up with gasoline fuel only. Once the lubricant oil temperature was in the range of 90±5°C, the quantity of the gasoline fuel was decreased and the ethanol fuel with equivalent energy was injected directly into the combustion chamber to compensate the reduced gasoline fuel. Three samples of data were recorded in each test.

Number of test	Case 1					Case 2					
Volume percentage of ethanol fuel[a] (%)	0	42	48	55	60	0	34	55	60	66	71
MBT spark timing (°CA BTDC)	45	44	47	48	50	30	31	32	33	35	35
Injection pressure[b] (Bar)	40	40	40	60	60	40	40	40	60	60	60

Table 2 - Engine test matrix

[a] Limited by the minimum injection pulses of both port fuel injector and direct fuel injector.
[b] Injection timing was 300° BTDC.

Results and Discussion

This section presents the experimental results in two parts, combustion and emissions. In each part, the effect of ethanol/gasoline volumetric percentage on engine performance is described and discussed.

Combustion

Indicative mean effective pressure (IMEP) is a parameter evaluating the energy transferred from the heat released in the combustion to net work per engine volume. Figure 2 displays the variation of IMEP with EVP at 3500rpm. As it can be seen, the IMEP increases with the increase of EVP except slightly dropping at EVP of 48% in Case 1 and at EVP of 54% in Case 2. Factors contributing to the increase of IMEP may include charge cooling effect associated with fuel injection and ethanol's high latent heat of vaporization [8], high energy content of stoichiometric mixture per unit mass of air, mole multiplier effect [9] and ethanol's high combustion velocity [10].

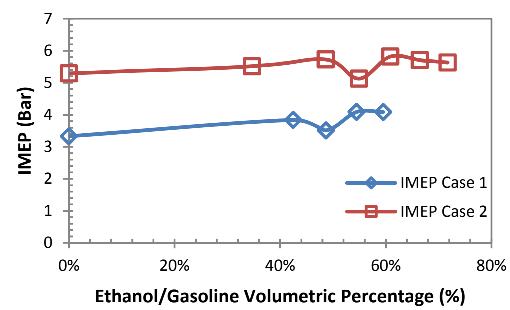

Figure 2- Variation of IMEP with EVP

To examine the stability of the combustion, cycle-by-cycle variation is presented by the coefficient of variation (COV). Figure 3 shows the COV of IMEP. As it can be seen, the COV decreases with the increase of EVP. However, in Case 1, the COV drops more quickly from 9.1% at EVP of 0% to 4.0% at EVP of 60%. In Case 2, the COV reduces relatively slowly from 7.1% at EVP of 0% to 5.4% at EVP of 48% and then becomes quite stable until it is 4.9% at EVP of 71%. It is assumed that the shortened combustion duration would contribute to the reduced cyclic variation [12]. The higher laminar combustion velocity and better low temperature combustion stability of ethanol fuel may also contribute to the decrease of COV in this study.

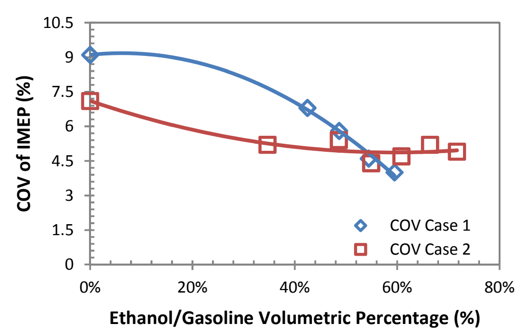

Figure 3- Variation of COV of IMEP with EVP

The combustion initiation duration, CA0-5%, defined by the crank angle degrees starting from the spark timing and ending when 5% of the fuel mass is burnt, is illustrated in Figure 4. It can be seen that the CA0-5% first decreases with the increase of EVP until the EVP reaches 42% in Case 1 and 34% in Case 2. It then gradually increases with the further increase of EVP. The decrease of CA0-5% may be attributed to the low ignition energy and higher flame velocity of ethanol fuel, which permits the ethanol/gasoline fuel mixture to be more easily ignited and faster combusted than the pure gasoline. The increase of CA0-5% with the further increase of EVP may be because of the ethanol's greater latent heat of vaporization which decreases the in-cylinder temperature and offsets the speed of flame growth. This analysis can be further supported by comparing the CA0-5% result of Case 1 and Case 2. As it can be seen, the CA0-5% in Case 1 increases more quickly than that in Case 2 when the EVP is greater than 48%. The engine load in Case 2 is higher than that in Case 1. Increasing engine load can increase in-cylinder temperature which can partially compensate the reduction of in-cylinder temperature caused by ethanol direct injection. Thus the slower increase of CA0-5% in Case 2 may be due to the rising in-cylinder temperature.

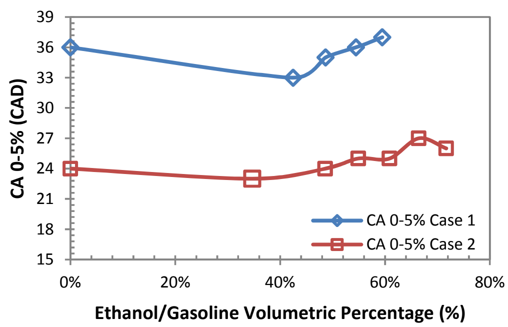

Figure 4- Variation of CA0-5% with EVP

The early combustion duration, CA5-50%, defined as 5-50% mass burnt fraction duration, is shown in Figure 4. This duration is presented because the timing/crank angle for the 50 % mass burnt fraction is often used to locate the combustion phasing. As illustrated in Figure 5, the CA5-50% first decreases with the increase of EVP. When EVP is greater than 48%, it starts to increase with the increase of EVP. As previously stated, the ethanol fuel has a faster laminar flame speed than that of gasoline, so a decrease of CA5-50% from EVP of 0% to EVP of 48% is expected. However, due to the increased cooling effect of ethanol fuel, further increase of EVP (greater than 48%) would result in a lower in-cylinder temperature which reduces the flame speed.

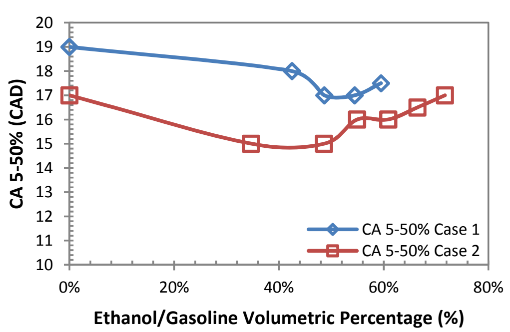

Figure 5- Variation of CA5-50% with EVP

The major combustion duration, CA10-90%, is the crank angle range starting with 10% of fuel mass burnt and ending with 90 % of fuel mass burnt. It directly affects to the engine thermal efficiency. The longer the combustion duration is, more heat will be lost through the cylinder wall. As it is shown in Figure 6, the CA10-90% decreases with the increase of EVP when EVP is less than 48%. When EVP is greater than 48%, CA10-90% increases with the EVP. This result may also be related to ethanol fuel's faster laminar flame speed and greater cooling effect. When EVP is in a certain range (less than 48%), the reduction of in-cylinder temperature due to directly injecting ethanol fuel may not be so significant to influence CA10-90%, while the high laminar combustion velocity of ethanol fuel may do so. However, when the EVP is greater than 48%, the cooling effect of ethanol fuel direct injection may be over effective, so that it may impede the growth of the ethanol/gasoline fuel mixture flame speed. As a combination of positive and negative effects, the CA 10-90% becomes to decrease when EVP is further increased.

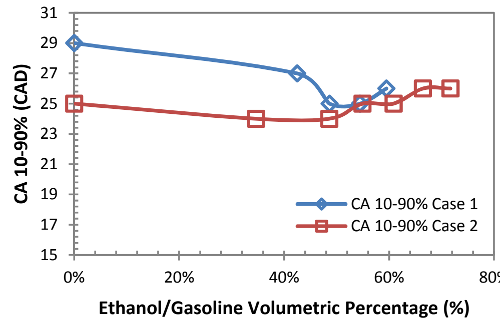

Figure 6- Variation of CA10-90% with EVP

Emissions

The variation of brake specific hydrocarbon (BSHC) emission with EVP is shown in Figure 7. As it can be seen, the BSHC emission increases with the increase of EVP. This may be caused by three factors. The first one is the poor mixture quality and wall-wetting effect caused by ethanol fuel direct injection. The second one is that the increase of cylinder pressure (IMEP) may make more hydrocarbons to be trapped in the crevice volumes. The third factor is that the lower in-cylinder temperature caused by ethanol direct injection results in less oxidation to take place when the trapped hydrocarbons get released (in the exhaust stroke) from the crevice volumes.

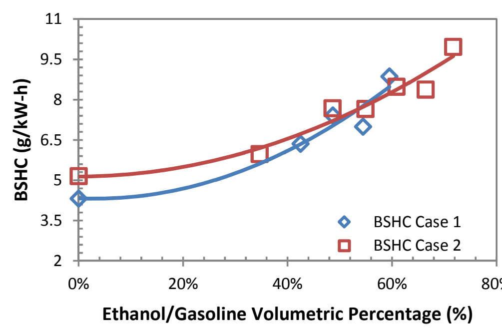

Figure 7- Variation of BSHC with EVP

The variation of brake specific carbon monoxide (BSCO) emission with the EVP is shown in Figure 8. As show in the figure, the BSCO emission in Case 1 increases with the increase of EVP while in Case 2, it first reduces slightly at EVP of 34%, then increases with the increase of EVP. CO emission is product of incomplete combustion. The decrease of BSCO emission at EVP of 34% in Case 2 may be due to the combustion improved by the ethanol fuel's fast laminar combustion speed and oxygen content property. The increase of BSCO may also be caused by three factors. The first two are the same as that causing the increase of BSHC, poor ethanol fuel mixture quality and wall-wetting effect and low in-cylinder temperature. The third one is that the advanced spark timing (see Table 2) reduces the time for ethanol fuel/air mixing and vaporization process.

The variation of brake specific nitrogen oxides (BSNO) emission with EVP is shown in Figure 9. As it can be seen, the BSNO emission, in Case 1, decreases with the increase of EVP while in Case 2, it first increases with the increase of EVP, then decreases when the EVP is greater than 34%. It is well know that the level of NO emission increases exponentially with the increase of in-cylinder temperature. As previously discussed, the increase of EVP may increase the combustion speed but also increase the combustion temperature which is the necessary condition to form BSNO emission. However, further increase of EVP would decrease the in-cylinder temperature due to the charge cooling effect. Thus the BSNO emission decreases.

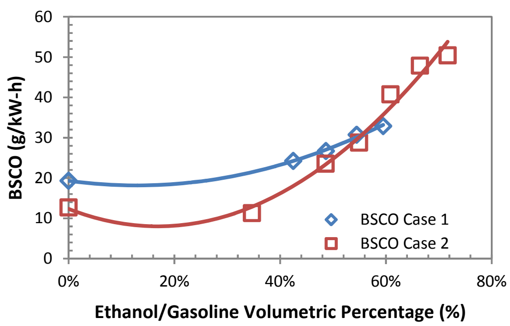

Figure 8- Variation of BSCO with EVP

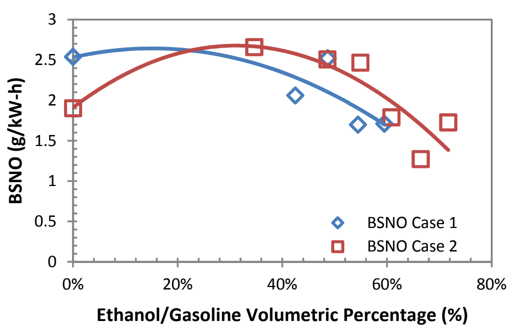

Figure 9- Variation of BSNO with EVP

Conclusions

Experiments were conducted on an EDI+GPI single cylinder research engine to investigate the effect of ethanol/gasoline volumetric percentage on engine combustion characteristics and emissions. The engine was tested at two load conditions with engine speed of 3500rpm and stoichiometric air/fuel ratio. Based on the analysis of experimental results, the following conclusions can be drawn.

1. The IMEP increased and its COV reduced with the increase of EVP. The reduced COV of IMEP indicated that the combustion stability was improved by direct injection of ethanol fuel.

2. In a certain range of EVP (< 48%), the initial combustion period (CA0-5%), early combustion period (CA5-10%) and major combustion period (CA10-90%) decreased with the increase of EVP. This may be mainly due to the ethanol fast laminar combustion speed. However, they were increased with the increase of EVP when EVP was greater than 48%. The longer combustion duration when EVP is greater than 48% may be caused by the in-cylinder temperature reduced by the over cooling effect of the increased percentage of ethanol fuel. In this case, the in-cylinder temperature might be too low during the combustion.

3. Direct injection of ethanol fuel effectively reduced the in-cylinder temperature, resulting in lower BSNO emission. However, it could also influence the fuel vaporization and oxidization process and lead to higher BSCO and BSHC emissions.

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