The Effect of Volume Ratio of Ethanol Directly Injected in a Gasoline Port Injection Spark Ignition Engine

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1 Abstract

Ethanol direct injection plus gasoline port injection (EDI+GPI) represents a more efficient and flexible way to utilize ethanol fuel in spark ignition (SI) engines. The greater cooling effect and higher octane number of ethanol fuel make it possible to implement engine downsizing technologies while avoiding knock issue in SI engines. In this paper, experiments were conducted on a single-cylinder 0.25L-displacement SI engine equipped with an EDI+GPI dual-injection fuel system. The engine was run at medium load (IMEP 6.3-7.0 bar) and stoichiometric fuel/air ratio. The ethanol ratio by volume varied from 0% (GPI only) to 100% (EDI only). Experimental results showed that the IMEP increased with the increase of ethanol ratio up to an ethanol ratio of 69% at 3500 RPM and 76% at 4000 RPM. With ethanol ratio greater than 69% or 76%, the IMEP reduced with the increased ethanol ratio. For engine exhaust gas emissions, the CO and HC emissions increased and NO decreased with the increase of ethanol ratio from 0% to 100%.

2 Introduction

The fossil fuel depletion and greenhouse effect are the main concerns of the modern human society. Bio-fuel is promising to address this issue. Ethanol can be produced from biomass and is considered as renewable fuel. Compared with gasoline fuel, ethanol has greater enthalpy of vaporization, larger octane number, higher flame speed and smaller stoichiometric air/fuel ratio. Ethanol is usually used as a substitute and octane-enhancing additive for gasoline fuel in spark ignition (SI) engines [1]. Currently, ethanol is mostly used via blending with gasoline fuel, such as the widely used E10 (gasoline containing 10% of ethanol by volume). Many studies have been carried out to investigate the effect of ethanol/gasoline blending ratio on the engine performance. Turner et. al. [2] investigated the combustion performance of a direct injection (DI) SI engine with various ethanol/gasoline blending ratios. Their results showed that blending ethanol with gasoline reduced emissions and increased efficiency, and the impact changed with the blending ratio. The combustion and emission characteristics of a single cylinder engine were studied with ethanol/gasoline blends at a constant mass fuel rate [3]. The experimental results showed that gasoline blended with 10% ethanol had marginal effects in combustion rates when compared to non-oxygenated fuels, but combustion process slowed down and cyclic dispersion increased with 20% ethanol. The experiments on a flexible-fuel vehicle showed that E85 and E75 reduced NOx emission but increased the emissions of CO, CH_4, formaldehyde, acetaldehyde and ethanol compared to the E5, E10 and E15 blends did [4]. The experiments on a DI SI engine showed that E20 improved combustion stability and reduced particle emissions than gasoline did [5].

However, blending ethanol with gasoline at a fixed ratio does not fully take the advantages of ethanol fuel, such as its greater enthalpy of vaporization for potentially increasing the compression ratio and consequently the thermal efficiency. To make the use of ethanol fuel more flexibly and efficiently, a new combustion system, ethanol direct injection plus gasoline port injection (EDI+GPI), has been investigated [6-9]. EDI+GPI enables the engine to be operated at any ethanol ratios according to the engine conditions. Moreover, the high octane number of ethanol fuel and the great cooling effect of EDI allow a higher compression ratio without the knock issue, leading to increased thermal efficiency. Wu et al. [7] tested the performance of a single cylinder SI engine equipped with gasoline port injection and bio-fuels direct injection. The potential of ethanol fuel with dual-injection to suppress knock in an SI engine was investigated [10, 11]. The leveraging effect of using ethanol fuel on replacing the gasoline fuel by EDI+GPI was experimentally demonstrated [9].

The above studies have shown improvement in the performance of the engine equipped with EDI+GPI. Although the effect of ethanol/gasoline blending ratio has been extensively studied on the conventional single-injection engines, the effect of ethanol ratio on the performance of engine equipped with EDI+GPI dual-injection fuel system still needs more investigation. In this paper, experiments were conducted on a single cylinder SI engine equipped with EDI+GPI in a full range of ethanol ratio from 0% (GPI only) to 100% (EDI only).

3 Experimental Apparatus and Procedure

The experiments were conducted on a four-stroke single-cylinder SI engine equipped with an EDI+GPI dual-injection fuel system. Figure 1 shows the schematic diagram of the engine test rig and Table 1 lists the engine specifications. The original engine was an SI engine with gasoline port injection. It was modified to EDI+GPI engine by adding an EDI fuel system. The EDI+GPI fuel system offers the ability to operate the engine at any ethanol/gasoline ratios. More details about the engine test rig can be found in [9].

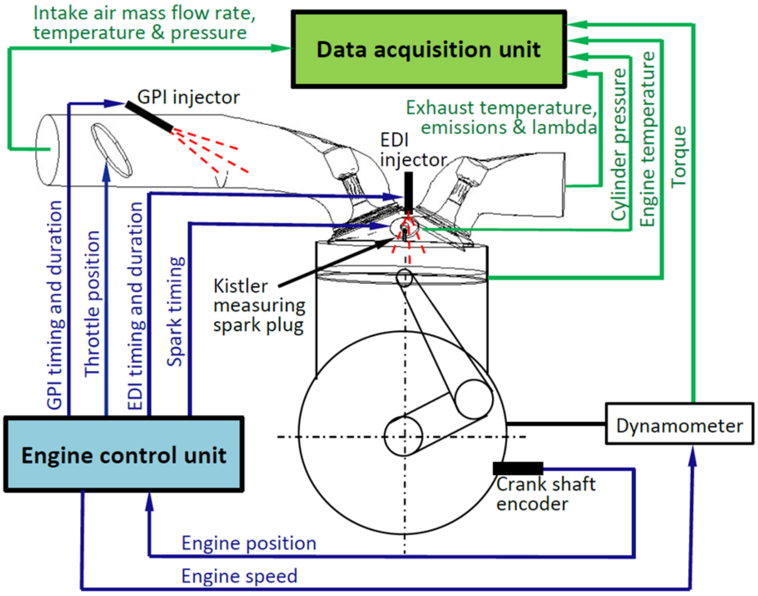

Figure 1. Schematic of the EDI+GPI research engine.

Table 1: Specifications of the EDI+GPI research engine

Engine type	Single cylinder, air cooled, four-stroke
Displacement	249.0 cc
Stroke × Bore	58.0 mm × 74.0 mm
Compression ratio	9.8:1
Valve timings	IVO: 22.20° BTDC; IVC: 53.80°ABDC EVO: 54.60° BBDC; EVC: 19.30° ATDC
Ethanol delivery system	Direct injection
Gasoline delivery system	Port injection

The gasoline fuel used was unleaded gasoline with an octane number of 91. The ethanol fuel was provided by the Manildra Group. The experiments were conducted at medium load (IMEP 6.3-7.0 bar) with engine speeds of 3500 rpm and 4000 rpm. At each engine speed, the lambda was kept at 1.0 and the ethanol ratio was varied from 0% (GPI only) to 100% (EDI only), including E0, E46, E58, E69, E76, E85 and E100 (E'X' means X% ethanol by volume. e.g. E46 is 46% via EDI + 54% via GPI). The throttle was 36% open for 4000 rpm and 34% open for 3500 rpm in order to keep the lambda at 1.0. Since the air/fuel ratio of ethanol (9.0) is smaller than that of gasoline (14.8), the total mass of injected fuels was increased with the increase of ethanol volume ratio to maintain the stoichiometric air/fuel ratio at the same engine speed and throttle position. The GPI pressure was kept constant at 0.25 MPa and the EDI pressure was 6 MPa. The spark timing was 15 CAD BTDC which was the spark timing set in the original engine control system. The GPI timing was 410 CAD BTDC and EDI timing was 300 CAD BTDC. During the experiments, the in-cylinder pressure, torque, intake and exhaust temperatures, cylinder head temperature and emissions were recorded.

4 Results and Discussion

This section presents and discusses the experiment results in two sub-sections, the effect of ethanol ratio on the engine performance and combustion and on the engine emissions.

4.1 Engine performance and combustion Characteristics

Figure 2 shows the effect of ethanol ratio on the indicated mean effective pressure (IMEP) at 3500 rpm and 4000 rpm. As shown in Figure 2, the IMEP increases with the increase of ethanol ratio until it reaches 69% for 3500 rpm and 76% for 4000 rpm. With further increase of ethanol ratio, the IMEP starts to decrease. This indicates that, compared with GPI only, EDI can help to increase the engine power output within a medium ethanol ratio. However, the engine power does not increase any more with ethanol ratio higher than 69% for 3500 rpm and 76% for 4000 rpm. Consistently, the indicated thermal efficiency increases when EDI is applied from 0% to 76% and decreases from 76% to 100%, as shown in Figure 3.

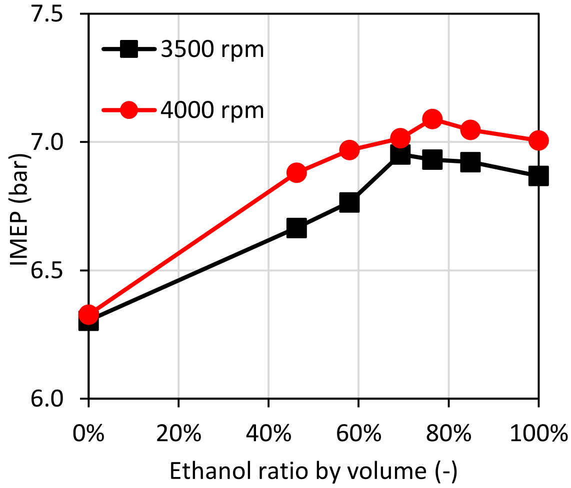

Figure 2. Effect of ethanol ratio on IMEP.

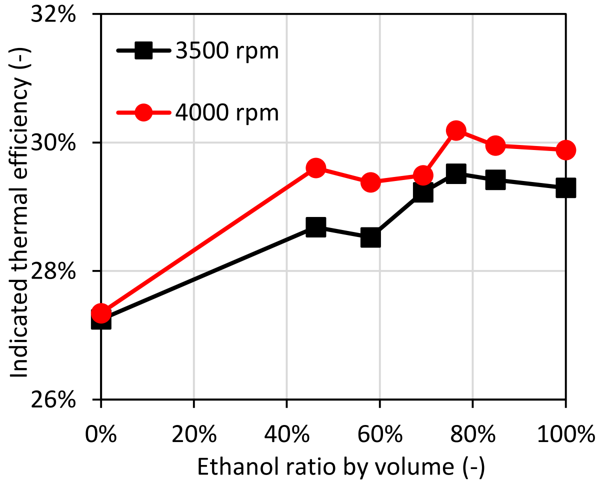

Figure 3. Effect of ethanol ratio on thermal efficiency.

The increase of IMEP can be attributed to the increased volumetric efficiency due to the cooling effect of EDI and the faster flame speed of ethanol fuel. However, the mixing process at high ethanol ratio becomes poor due to ethanol's large enthalpy of vaporization and low evaporation rate. Numerical simulation was carried out to investigate the cooling effect and mixture formation processes of the EDI+GPI engine at different ethanol ratios at 4000 rpm. The in-cylinder flows were simulated by the Realizable k-ε Turbulence Model. The spray breakup and evaporation processes were simulated by the WAVE Model and Convection/Diffusion Controlled Model. It simulated the process starting from GPI injection and ending just before ignition. More details about the simulation can be found in [12]. Figure 4 shows the distributions of in-cylinder temperature and equivalence ratio by spark timing predicted for E46, E58 and E85. As shown in Figure 4, the cooling effect is demonstrated by the decreased in-cylinder temperature with the increase of ethanol ratio. Particularly, when the ethanol ratio is higher than 58%, the near-wall region under the exhaust valve (right hand side in the figure) is cooled to a very low temperature (~500 K) compared to the mean cylinder temperature 700 K. This region is over cooled. Meanwhile, because of ethanol's low evaporation rate in low temperature environment before combustion takes place, the equivalence ratio in the spark gap becomes smaller than 0.5 which is out of the ignitable limit range of $0.5<\Phi<1.5$ when ethanol ratio is higher than 58%. The lean mixture around the spark plug causes difficulty for the ignition process which would consequently lead to incomplete combustion and increased instability of combustion. High ethanol ratio at E85 shows stronger cooling effect than that at lower ethanol ratios. However the mixture of E85 becomes over-cooled and too lean. These deteriorate the combustion process and lead to the decreased IMEP when ethanol ratio is higher than 69% for 3500 rpm and 76% for 4000 rpm.

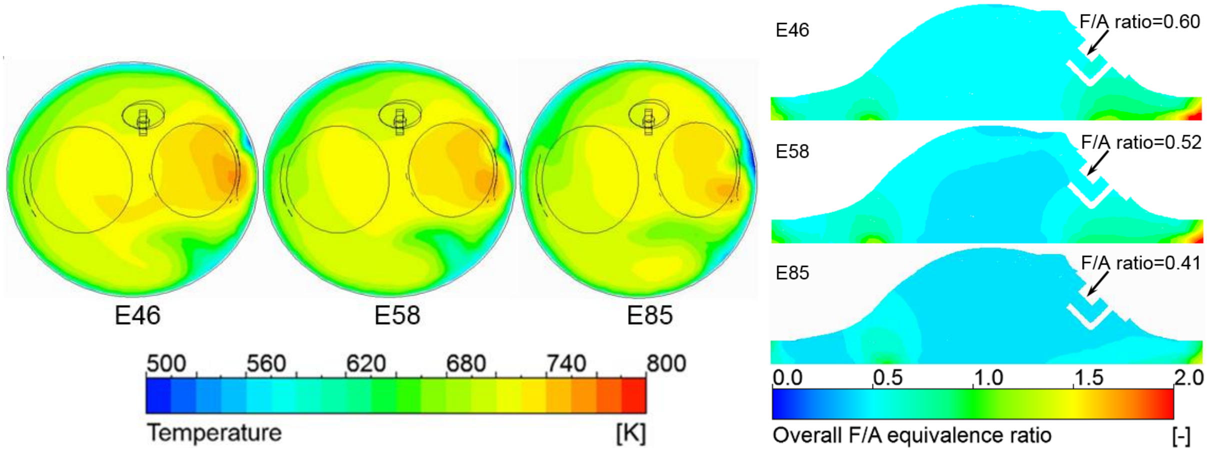

Figure 4. The predicted spatial distributions of in-cylinder temperature (left) and equivalence ratio (right) by spark timing for E46, E58 and E85 at 4000 rpm.

Figure 5 shows the major combustion duration (CA10-90%) varied with the ethanol volume ratio, as derived from the cylinder pressure measured in experiments. The major combustion duration, denoted by CA10-90%, is defined as the crank angle degrees from 10% to 90% mass fraction burnt. The shorter is the CA10-90%, the closer the combustion process is to the constant volume and consequently the higher the thermal efficiency will be. As shown in Figure 5, the major combustion duration decreases with the increase of ethanol ratio from 0% to 69% at 3500 rpm and 0% to 76% at 4000 rpm, indicating improved combustion. The decrease of CA10-90% can be attributed to the faster flame speed of ethanol than that of gasoline. However the CA10-90% increases when the ethanol ratio is further increased. This is because the mixture is over-cooled and becomes too lean when ethanol ratio is too high, as discussed in Figure 4. The lean mixture slows down the combustion and causes the decreased thermal efficiency at high ethanol ratios shown in Figure 3.

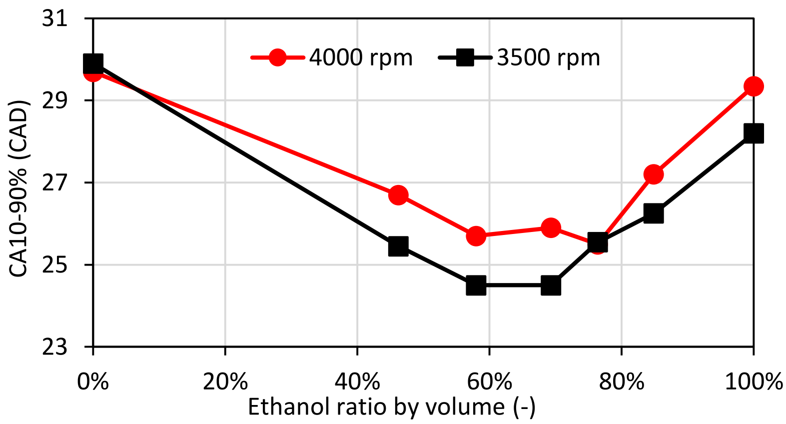

Figure 5. Variation of CA10-90% with ethanol ratios.

4.2 Engine emissions

Figure 6 shows the variation of indicated specific nitric oxide (ISNO) emission varied with ethanol ratio. As shown in Figure 6, the NO emission decreases with the increase of ethanol ratio due to the decreased in-cylinder temperature caused by the following three main factors. Firstly, the cooling effect is stronger in EDI+GPI condition than that of GPI only condition due to ethanol's large enthalpy of vaporization. Secondly, the adiabatic flame temperature of ethanol (2144 K) is lower than that of gasoline (2300 K) [13]. Thirdly, the mixture is leaner in EDI+GPI condition than that in GPI only condition due to ethanol's low evaporation rate, which decreases the combustion temperature significantly [14]. All these three factors contribute to the decrease of combustion temperature in EDI+GPI and become stronger with the increase of ethanol ratio. According the Zeldovich NO mechanism, the NO formation is less intensive in EDI+GPI condition than that in GPI only condition.

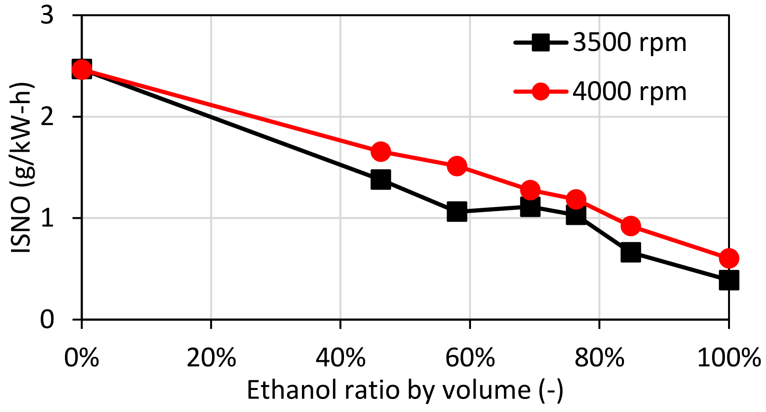

Figure 6. Effect of ethanol ratio on ISNO.

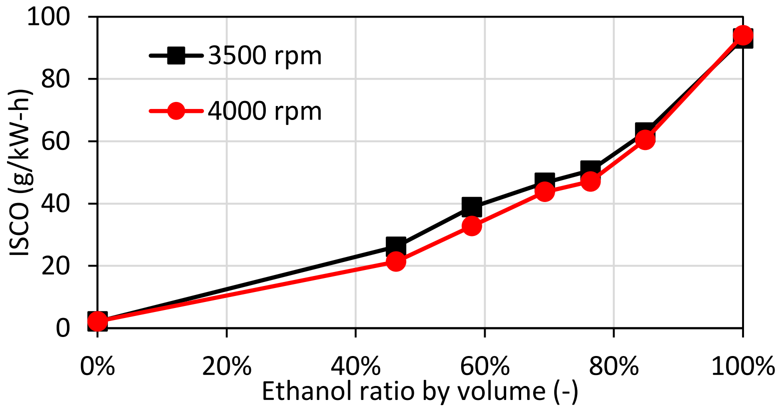

Figure 7. Effect of ethanol ratio on ISCO.

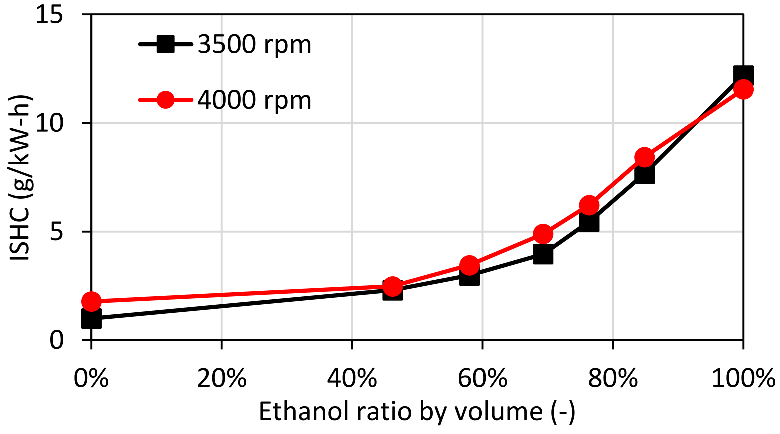

Figure 8. Effect of ethanol ratio on ISHC.

On the other hand, the indicated specific carbon monoxide (ISCO) and hydrocarbon (ISHC) emissions increase with the increase of ethanol ratio, as shown in Figures 7 and 8. CO and HC emissions are products of incomplete combustion. When EDI is applied, the cylinder temperature is lower and over-cooling occurs at high ethanol ratios. Moreover, the low evaporation rate of ethanol fuel leads to a large number of liquid ethanol droplets in the combustion chamber during the combustion process and the in-cylinder fuel/air mixture becomes more uneven in EDI+GPI than that in GPI only. Simulation results showed that the mixture became lean in the region around the spark plug but rich in some near cylinder-wall region with EDI injection [14], resulting in incomplete combustion and consequently the increase of CO and HC emissions.

5 Conclusions

Experiments were conducted to investigate the combustion and emission performance of a single-cylinder SI engine equipped with EDI+GPI dual-injection fuel system over the full range of ethanol volume ratio from 0% (GPI only) to 100% (EDI only). The engine was operated at medium load (IMEP 6.3-7.0 bar) and stoichiometric air/fuel ratio at engine speeds of 3500 rpm and 4000 rpm. The effect of ethanol ratio on the combustion and emission characteristics of the engine was discussed. The main conclusions can be drawn as follows.

1. The IMEP increased with the increase of ethanol ratio up to 69% at 3500 RPM and 76% at 4000 RPM. Further increase of ethanol ratio from 69% or 76% to 100% led to the decrease of IMEP.
2. The major combustion duration CA10-90% decreased with the increase of ethanol ratio from 0% to 69% for 3500 rpm and 0% to 76% for 4000 rpm, indicating an improved combustion efficiency. CA10-90% increased when ethanol content was further increased.
3. The NO emission decreased and CO and HC emissions increased with the increase of ethanol ratio. The NO emission was decreased due to the stronger cooling effect and lower combustion temperature of ethanol than that of gasoline. The CO and HC emissions were increased due to the over-cooling at higher ethanol ratio and low evaporation rate of ethanol at low temperature environment before combustion, which caused incomplete combustion.

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