Proceedings of the Australian Combustion Symposium
December 7-9, 2015, The University of Melbourne

Effects of EDI on Combustion and Emissions performance of Small Spark Ignition Engine at lean conditions

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Abstract

Running the spark ignition engine under lean mixture operating conditions can improve the engine efficiency and reduce the specific emissions. However, the cycle-to-cycle variation can limit the lean burn spark ignition engine improvement. This experimental work investigates the effect of the ethanol direct injection strategy on the combustion and emissions performance running under lean mixture conditions. Three equivalent air/fuel ratios (λ=1.0, 1.1 and 1.2) and five ethanol direct injection percentages (EDIP= 0%, 25%, 50%, 75% and 100%) were adopted. The experiments were carried out on a small spark ignition engine equipped with a dual fuel injection system. The experimental results show that the ethanol direct injection can significantly improve the combustion performance and thus engine thermal efficiency by taking the advantages of ethanol fuel properties such as high laminar burning velocity and great latent heat. The EDI demonstrates a superior potential in the engine emissions reduction with the lean burn conditions than that of stoichiometric mixture. The coefficient of variation of the indicated mean effective pressure at λ=1.2 is also reduced considerably when ethanol starts to be injected into combustion chamber. Furthermore, the engine volumetric efficiency substantially improved with the increase of air/fuel ratio combined with the increase of ethanol direct injection percentages.

Keywords: Lean Burn, Ethanol Direct Injection Engine, Combustion.

1. Introduction

Recently, engine performance improvement and emissions reduction have become a main concern among the researchers and the automotive companies worldwide. Reducing the pumping and heat losses can be an interested outcomes as a consequence of lean mixture operating conditions [1, 2]. Diluting the stoichiometric air/fuel ratio with extra amount of air (lean mixture) can significantly improve the combustion quality and then reduce the emissions [1, 3]. Consequently, the engine thermal efficiency and volumetric efficiency can be significantly improved. However, when the conventional naturally aspirated gasoline engines are adopted, the highly diluted mixture will lower the laminar burning velocity and mixture homogeneity increasing the difficulties to start the combustion and enlarging the magnitude of cycle-to-cycle variations [4, 5]. This undesirable combustion behaviour will obviously limit the engine performance development. Lately, ethyl alcohol (Ethanol fuel) as a green environmentally friendly biofuel has been targeted as an alternative fuel that can enhance the gasoline fuel performance [3, 6-8]. Adopting an ethanol fuel to the spark ignition engines may promote more complete and stable combustion that may result in better engine thermal efficiency, engine volumetric efficiency and thus less emissions production [6, 9-11]. This can be mainly attributed to the ethanol fuel properties such as laminar burning velocity, the latent heat of vaporization and octane number when it is compared with gasoline fuel as shown in table1. A partial-burn and misfire occurring possibility can be increased due to the deceleration of flame propagation with lean burn engines [12]. The greater ethanol flame speed (~39cm/sec) compared with gasoline one (~33cm/sec) can substitute the reduction in the air/gasoline mixture combustion velocity due to adopting lean combustion mode. Employing ethanol direct injection (EDI) strategy to the lean burn engine can enhance the combustion stability and thus engine performance by taking better of the advantages of ethanol properties. Therefore, this paper aims to investigate the leveraging effect of the EDI combined with the lean burn on a small spark ignition engine (SSIE) performance. Three engine parameters that were examined which are the combustion and emissions performance in addition to coefficient of variation of the indicated mean effective pressure (COV_{IMEP}).

Properties	**Units**	**Gasoline**	**Ethanol**
Chemical formula	----	C_2-C_{12}	C_2H_5OH
Molecular weight	kg/kmol	114.15	45.07
H/C	Atom ratio	1.795	3
O/C	Atom ratio	0.7-0.78	0.794
Density (at 288.15K)	kg/m^3	750-765	785-809.9
Stoichiometric air-fuel ratio	w/w	14.2:1-15.1:1	8.97:1
Octane number	----	91	108.61
Higher heating value (HHV)	MJ/kg	44.0	26.9
Laminar flame speed at 100kPa, 325K	cm/s	~33	~39
Latent heat of vaporization	kJ/kg	298	948
Saturation vapour pressure	kPa	28.828	8.773

Table1. Gasoline and Ethanol Properties [3, 7, 13-15]

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2. Engine and Experimental Procedure

2.1 Research Engine

The experiments were conducted on a small naturally aspirated spark ignition engine. This single-cylinder four-stroke engine was originally designed with a port fuel injection (PFI) only before it was modified to be equipped with an adaptable EDI system. A new electronic control unit (ECU) was developed by Hents Technology and used in order to provide a wide rage control capability to the relevant engine operating parameters such as throttle position and the mount of injected fuel in the intake port (GPI) or directly into the cylinder (EDI). In order to set the required engine speed and measure the engine torque, an eddy current dynamometer with control system was coupled to the engine. A Kistler 6115B spark plug cylinder pressure transducer was used to monitor and record the in-cylinder pressure. More details about the research engine can be found in [8].

2.2 Experimental Procedure

Five volume-based ethanol direct injection percentage (EDIP) were set to be started from 0.0% (GPI only), 25%, 50%, 75% and EDIP100% (ethanol only). In order to achieve lean mixture operating conditions at each EDIP, the throttle opening percentage (throttle position) was set to be changed from 32% at λ=1.0 to 43% at λ=1.2 as are shown in table 2. During the test, the engine was set to a medium load, a constant engine speed of 3500 RPM, EDI timing of 300 crank angle degree before top dead centre (CAD BTDC) and fixed spark timing of 15CAD BTDC. In different EDI percentages and lean burn conditions, the total amount of fuel heating energy (~580J) is kept the same. The indicated mean effective pressure (IMEP) and COV_{IMEP} were calculated based on the mean value of 100 consecutive engine cycles of the recorded in-cylinder pressure data at rate of 0.5CAD. Each cylinder pressure recording was repeated three times and the average of the IMEP and COV_{IMEP} was used into the result analysis. With regard to other output data such as engine torque and emissions, five samples were taken and the theoretical calculations were based on the mean value.

Engine speed	3500 RPM
EDI percentage (v/v)	0%, 25%, 50%, 75%, 100%
Percentage of GPI (v/v)	100%, 75%, 50%, 25%, 0%
Equivalence Air/Fuel ratio (λ)	1.0, 1.1, 1.2
Throttle position	32%, 37%, 43%
EDI pressure	40 bar
GPI pressure	2.5 bar

Table2. Engine operating conditions.

3. Results and Discussion

The main aim of the present paper is to investigate the effect of leveraging of EDI combined with lean burn (λ≤1.0) on the engine and emissions performance of the naturally aspirated SSEI fueled originally with gasoline by PFI system.

3.1 Engine Performance

Figure 1 shows the variation of the IMEP with EDI percentages at different values of air/fuel ratios. As it shown in Fig. 1 the general behavior of the IMEP is increased with the increase of EDIP at different input of air/fuel ratios comparing with GPI only. This trend is possibly attributed to the greater laminar burning velocity of ethanol fuel than gasoline. At λ=1.1 the IMEP slightly improved at all EDIP whereas at λ=1.2 the IMEP is slightly decreased compared to stoichiometric conditions. Firstly, at λ=1.1, the IMEP improvement could be attributed to the combined effect of ethanol fuel laminar burning speed, oxygen content and the charge cooling effect with pumping loss reduction and excess amount of air that provide more oxygen promoting more complete combustion process [3, 6]. These factors could improve the value of the IMEP considerably [16]. Secondly, at λ=1.2, even though the extra amount of air could enhance the chemical reaction of fuel leading to better fuel oxidation that will be noticed in the emissions performance section, the reduction of the mixture flame speed and the mixture quality might be a reasonable cause to the slightly decrease of the IMEP along the experimental test compared to stoichiometric condition [1, 2, 4].

Figure 2 shows the variation of the COV_{IMEP} with EDIP at different values of λ. As it is obviously seen in Fig.2, the cycle-to-cycle variation represents the main challenging to the lean burn engine improvement. It is assumed that the highly diluted mixture could lead to a slow burning rate that the combustion process cannot complete properly [4]. The COV_{IMEP} is dramatically decreased with EDIP increasing including most of the experimental tests, but the effect of the EDI is clearly seen to be more significant with the highly lean mixture (λ=1.2). The significant reduction of the COV_{IMEP} with the EDI at all engine operating conditions except 100% EDI might be attributed to the greater laminar burning velocity and better low temperature combustion stability of ethanol compared to that of gasoline [3]. At pure EDI operating conditions especially at λ=1.2, the cycle-to-cycle variation is substantially increased possibly due to the mixture quality deterioration. This might be attributed to the highly mixture dilution combined with lower ethanol fuel volatility compared with gasoline.

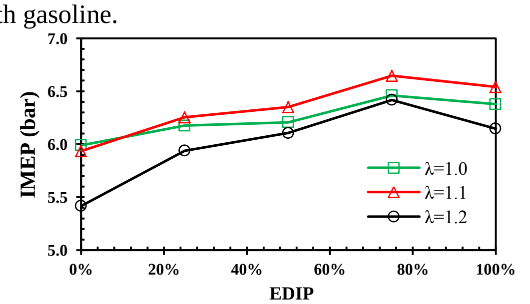

Figure 1. Variation of IMEP with EDIP and λ

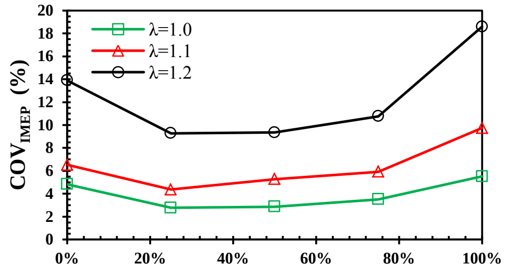

Figure 2. Variation of COV_{IMEP} with EDIP and λ

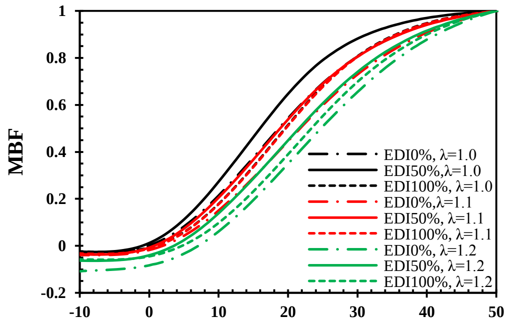

Figure 3. Variation of MBF with CAD

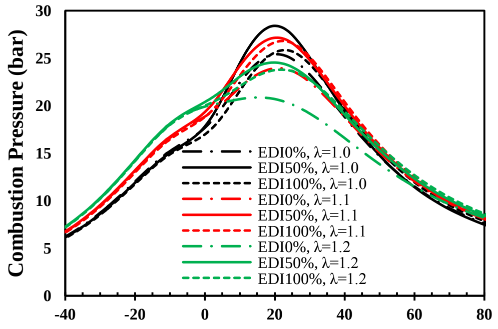

Figure 4. Variation of Combustion Pressure with CAD

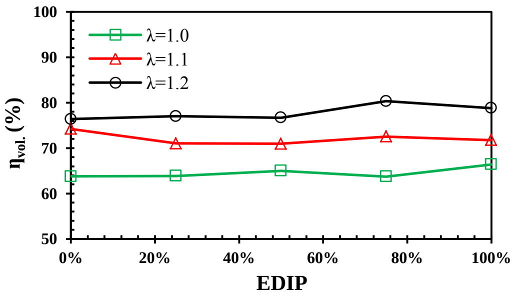

Figure 5. Variation of $\eta_{vol.}$ with EDIP and λ

The effects of the combination of EDIP and lean mixture on the combustion performance at each CAD are illustrated in Fig. 3. As shown in the figure, the EDI can considerably improve the combustion quality at different cases of λ. The combustion duration is shortened and thus more quantity of fuel is consumed at each CAD with increase of EDI percentage. Consequently, more heat is released from fuel resulting in HC and CO emissions reduction as it will be mentioned in the next section. This behavior might be attributed to the ethanol greater flame speed combined with the oxygen contents. However, pure EDI operating condition demonstrates a lower potential effect on the combustion quality than that of EDI50%. This tendency could be attributed into two main reasons. Firstly, injecting the entire amount of ethanol fuel that has around three times greater latent heat of vaporization than that of gasoline directly into the combustion chamber will significantly decrease the chemical reaction temperature. Consequently, the burning rate will be relatively decelerated and thus less stable combustion. Secondly, the combination of the ethanol lower volatility compared to that of gasoline with the high fuel impingement into the combustion chamber surfaces possibly deteriorate the air/fuel mixture quality and thus lower combustion performance compared to that of EDIP 50%. EDI 25% and 75% were omitted in order to demonstrate the effect of the EDI at different lean burn cases clearly.

The solid lines with different colors in Fig. 4 illustrate the combustion pressure of EDI 50% at various lambda operating conditions. It can be clearly observed that the 50% of the EDI demonstrates maximum in-cylinder pressure at different air/fuel ratios. This tendency might be attributed to the relatively greater laminar burning velocity of ethanol combined with the ethanol charge cooling effect.

Figure 3 shows the variation of the engine volumetric efficiency with the EDIP and different value of λ. As it can be observed in Fig. 3, the leveraging effects of the EDI combined with lean burn mixture on the engine volumetric efficiency result in an important improvement. This could be attributed to two main reasons. Firstly, regarding ethanol effect, the charge cooling effect due to the higher latent heat of vaporization compared with gasoline might play an important job to increase the fresh charge density and then improve the engine capacity. Moreover, the ethanol oxygen content is also assumed to play a significant role in the volumetric efficiency increment. Secondly, concerning lambda effect, run the engine with excess air could promote more complete and stable combustion which are also possibly effect positively on the engine capacity performance.

3.2 Emission performance

Operating the engine with lean mixture conditions can dramatically reduce the indicated specific carbon monoxide (ISCO) emissions as shown in Fig. 6. The more oxygen proportion in the mixture the more oxidation can be promoted. As a result of this, the amount of ISCO emissions are reduced dramatically with the lean combustion compared to the stoichiometric one [4, 16]. However, as it can be seen in Fig. 6, when EDIP is bigger than 50%, ISCO is relatively increased with the increase of the EDIP gradually possibly due to poor mixture quality and spontaneous local rich regions that might be caused by a spray impingement into the combustion chamber walls [11, 16].

Figure 7 shows the variation of the indicated specific hydrocarbon (ISHC) emissions with the EDI percentages at different values of lambda. For EDI percentages between 0% and 50%, the ISHC emissions are significantly decreased probably due to excess amount of air for both lean burn conditions which result in better combustion process. In contrast, when the EDI turn up into 75% and 100%, the ISHC emissions starts to be relatively increased. This tendency could be attributed to the increase of the amount of the directly injected ethanol fuel that impinged into the combustion chamber walls combined with lower saturation vapor pressure of ethanol compared to that of gasoline resulting in a poor mixture and combustion quality [8, 13].

Figure 8 demonstrates the effect of the EDI and lean burn conditions on the indicated specific oxides of nitrogen (NOx) emissions. Because the ISNOx emissions are highly related to the combustion temperature, the ISNOx emissions are substantially reduced with the gradual increase of EDI percentage at most of the operating conditions [7]. Start injecting ethanol fuel directly into the combustion chamber (EDI 25%) might advance the combustion possibly due the ethanol laminar burning velocity resulting in higher flame temperature and increasing the ISNOx slightly. When the EDI is between 25% and 100%, the charge cooling effect due to the greater ethanol latent heat of vaporization compared to that of gasoline overwhelms the effect of flame speed. Consequently, considerably reduction of ISNOx emissions are happened at EDI 100% compared with the stoichiometric conditions.

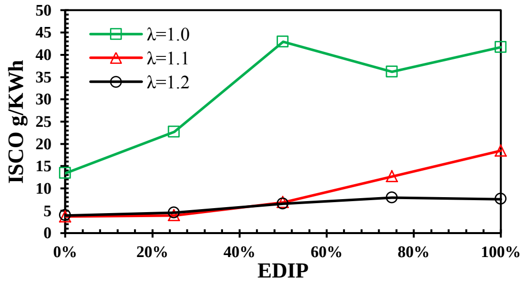

Figure 6. Varation of ISCO wtih EDIP and λ

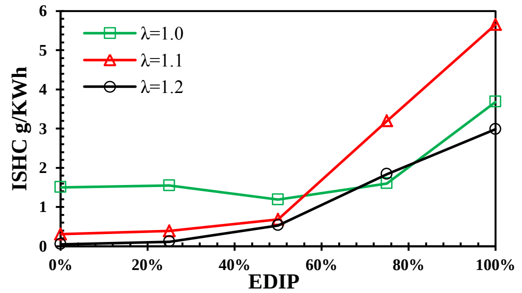

Figure 7. Variation of ISHC with EDIP and λ

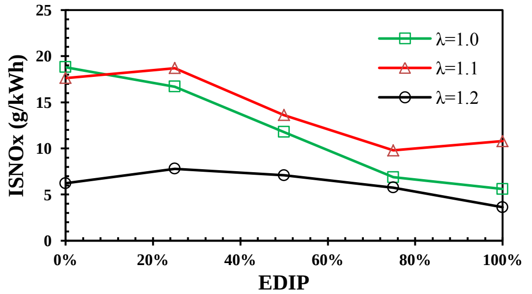

Figure 8. Variation of ISNOx with EDIP and λ

4. Conclusion

The effect of the EDI on the engine performance and emissions at lean burn conditions was examined. Experiments were conducted on a spark ignition engine equipped with dual fuel injection system (EDI+GPI). Five volumetric percentages of ethanol fuel were chosen as shown in table 2. Three lambda values of 1.0, 1.1 and 1.2 and fixed engine speed (3500RPM) were chosen.

The engine performance could be improved by adopting EDI strategy in to the gasoline port injection engine. EDI could substitute the flame speed reduction in the lean mixture due to greater burning velocity compared to gasoline. However, the IMEP is slightly decreased at EDI 100% probably due the overcharge cooling effects combined with a relatively poor mixture quality.

Once the EDI start working at 25%, the COV_{IMEP} was dramatically decreased especially at highly lean combustion (λ=1.2). This positive behaviour could be attributed to the greater laminar flame speed and better low temperature combustion stability compared to that of gasoline [3].

CO and HC emissions were reduced at leaner combustion possibly due to the excess amount of air that might improve the fuel oxidation. However, CO and HC emission were increased with EDIP over 50% probably due to the poor mixture quality and the oil film formation. In contrast, NOx emissions were decreased with EDI percentage possibly due to the charge cooling effects.

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