Preliminary Study of Hydrous Ethanol as a Fuel for a Spark Ignition Engine on Performance and Combustion

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Abstract

Fuel crisis during the last few decades has encouraged the use of alternative fuels available in Thailand. Recently, the government has issued a renewable energy plan to increase ethanol production. This has emboldened ethanol to be used as a fuel for transportation. Initially, anhydrous ethanol has been blended with gasoline in different amounts for the current spark ignition (SI) engines. However, the anhydrous ethanol production needs water removal at a cost. Therefore, the use of hydrous ethanol in a SI engine is a choice to promote the policy and also save energy for ethanol production. To investigate the engine performance and combustion characteristics, this work studies the effects on an unmodified 4-cylinder port fuel injection Honda engine fuelled with gasohol (E10), anhydrous ethanol (E100) and hydrous ethanol (5% water content, Eh95). The hydrous ethanol fuelled engine can operate on low to mid loads with lower performance than that of gasohol. E100 and Eh95 consume more fuel than E10. Thermal efficiencies from both ethanol combustions are lower than those of gasohol, especially at low load. Hydrous ethanol combustion shows the lowest maximum pressure and heat release rate among the others. It is suggested that the possibility to calibrate for better engine performance and emission can be achieved.

Keywords : Anhydrous Ethanol, Combustion, Engine performance, Gasohol, Hydrous Ethanol

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1. Introduction

Due to fuel crisis within the last few decades, alternative fuels such as ethanol and biodiesel which domestically produced have been increasingly used in Thailand. The government has issued 15-year renewable energy plan to increase an ethanol production to 9 million liters per day by 2022 [1]. As the results, the use of ethanol for transportation has been continuously encouraged. Initially, the ethanol blended with gasoline in different amounts named as "Gasohol" (90 % by volume of gasoline and 10% of anhydrous ethanol), E20 (80% by volume of gasoline and 20% of anhydrous ethanol) and E85 (15% of gasoline and 85% of anhydrous ethanol) is employed for the current spark ignition (SI) engines.

Ethanol is one of the environmentally-friendly fuels which contributes to carbon dioxide (CO_2) reduction in the atmosphere as it can be produced from renewable resources, for example sugar cane, cassava, corn and etc. The process of ethanol production depends on raw materials, e.g. grain starch or molasses. Basic ethanol production from grain starch consists of 5 processes [2]. Firstly, the raw materials (grain) are grinded to be the starch by milling method. This process increases its surface area of starch to mix with water. Then, the starch mixed with the enzyme in the cooking process is converted to sugar (glucose). Fermentation process is the third step to convert sugars into CO_2 and yield ethanol 8-12% by volume. Fermented ethanol is purified to the ethanol 95 % by volume (hydrous ethanol) by means of distillation method. The final process is dehydration which removes water from hydrous ethanol to high purity ethanol (99.5%) called "anhydrous ethanol".

With a high cost and energy consumption, the production of anhydrous ethanol involves the water removal process from hydrous ethanol. The significant properties of hydrous ethanol are similar to anhydrous ethanol, except for the water content as shown in Table 2. Therefore, the use of hydrous ethanol in SI engines is a possible approach to promote the government policy and also to save energy consumption of ethanol production process.

The main advantage properties of ethanol compared to gasoline are higher octane number and heat of vaporization. This results to increased anti-knock capability. The engine is allowed to perform with higher compression ratio or advanced ignition timing. Therefore, the engine performance is improved. However, its lower heating value leads to higher fuel consumption than that of gasoline.

Costa and Sodre [4] showed their comparative study between hydrous ethanol with 6.8% of water content and blended fuel (78% of gasoline and 22% of ethanol) on performance and emissions with a in-line four cylinders, 1.0 liter, 4-stroke SI engine. The results showed that torque and brake mean effective pressure (BMEP) were increased when hydrous ethanol was used at high engine speed. The hydrous ethanol produced higher thermal efficiency and specific fuel consumption than those of gasoline blend. Moreover, the exhaust emissions of hydrous ethanol decreased carbon monoxide (CO) and hydrocarbon (HC), but increased CO_2 and oxides of nitrogen (NO_X).

Gupta and et al. [5] carried out a study on effects of water content (10% and 20% of water contents) with ethanol for a single cylinder, 125 cm^3 4-stroke SI engine. The results showed an increased thermal efficiency for hydrous ethanol with higher water content. The brake specific fuel consumption (BSFC) increased with increasing water content. HC and CO increased when water was added up to 20%. Then, they were found to be lower than those of gasoline. NO_X produced by hydrous ethanol is relatively low.

Donovan [6] concluded in the initial tests conducted in Europe and confirmed that hydrous ethanol and gasoline could be blended as HE15 (15% of hydrous ethanol and 85% of gasoline) without phase separation or other problems. An unmodified Volkswagen Golf 5FIS was successfully operated on HE15, conforming European exhaust emission standard in the test conducted by the Netherlands Research Organization (TNO) Automotive and SGS Drive Technology Center of Austria.

The present work aims to preliminary study the use of hydrous ethanol (5% of water content) in a SI engine. Hydrous ethanol (Eh95) and anhydrous ethanol (E100) are employed and compared with commercial gasohol (10% of ethanol, E10). The engine performance and combustion characteristics are investigated and analysed. Afterwards, the results of this work will be a database for engine modification and calibration in the future work.

2. Experimental setup and procedure

Experiments were carried on an unmodified MY 2010 Honda 4-stroke L15A7 SI engine. Its technical specifications are described in Table 1.

Table 1 Engine specifications

Parameter	**Description**
Number of cylinder	4 in-line
Displaced volume	1,496 cm^3
Bore x stroke	73.0 mm × 89.4 mm
Compression ratio	10.4 : 1

The engine was coupled and loaded with a Land & Sea DYNO-mite 012-200-1K eddy current dynamometer with maximum brake power of 200 H.P. The engine torque and speed, intake air, exhaust gas and ambient temperature were recorded via National Instruments NI USB-6218 data acquisition system in corporate with an in-house developed LabView software code. The A/F ratio, ignition timing, cooling water temperature and injection duration were recorded by On-Board Diagnostics Generation II (OBD II) of G-Scan. On mass basis, the fuel consumption was measured by OHAUS PA4102 digital weight indicator with the accuracy of ±0.1 g. For the analysis of combustion characteristics, the cylinder pressure traces were measured via piezoelectric pressure transducer, Kistler model 6052C coupled with a Dewetron DEWE-30-4 charge amplifier. An incremental shaft encoder, Leine Linde RHI530, 3,600 ppr was used to collect crank angles corresponding to the cylinder pressure traces. However, due to optimized resolution of the measurement system, the encoder was set to 720 ppr for all tests. The cylinder pressures with corresponding crank angle signals were recorded in real time data acquisition DEWEtron via software DEWEsoft 6.5.1. For each test condition, the cylinder pressure data from 100 consecutive engine cycles were acquired, and averaged values are presented as typical representatives. A schematic diagram of experimental set up is depicted in Fig. 1.

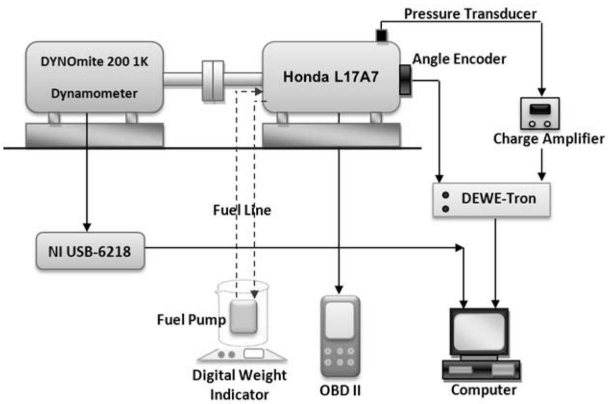

Fig. 1. Schematic diagram of the test engine

Table 2 Fuel properties

Properties	Gasohol Octane 95	Anhydrous Ethanol 99.5%	Hydrous Ethanol 95%
Formula*	$C_{6.62}H_{15.0}O_{0.23}$	C_2H_5OH	$C_{1.71}H_{5.52}O_1$
Stoichiometric air/fuel ratio	14.34	8.950	8.400
Density (kg/l) @ 30 °C	0.734	0.781	0.798
Lower heating value (MJ/kg)	36.84	27.77	26.52
Viscosity (sCt)	0.473	1.090	1.204

*calculate for 1 mole of fuel for give volume ratio of content (using measured value of densities and standard value of molecular weight) [5]

There were three fuels used throughout the test: gasohol with octane number 95 (10% of ethanol and 90% of gasoline), anhydrous ethanol, and hydrous ethanol. Their properties are listed in Table 2.

The engine performance tests were operated at 25%, 50%, and 75% of wide open throttle (WOT) positions. The engine speeds were varied from 1,500 to 3,500 rpm for all engine loads. For the combustion analysis, the engine load was fixed at 70 Nm at the speeds of 2,500 and 3,000 rpm.

3. Results and discussion

3.1 Performance tests

The engine performance tests were comparatively studied on torque, power, BSFC and thermal efficiency when using different fuels. The brake power (P_b), BSFC, and thermal efficiency (η_{th}) are defined by Eq. (1) to (3), respectively.

$$P_b = 2\pi N T_b \quad (1)$$

$$BSFC = \frac{\dot{m}_f}{P_b} \quad (2)$$

$$\eta_{th} = \frac{1}{BSFC \times Q_{HV}} \quad (3)$$

where,

T_b = Brake torque measured by dynamometer (N.m)

$\dot{m}_f$ = The fuel consumption rate (g/s)

N = Engine speed (rps)

Q_{HV} = Heating value of fuel (J/kg)

3.1.1 Torque and power

Fig. 2 illustrates the influences of different fuel on brake torque and power of the unmodified engine under 25%, 50%, and 75% loads. From Table 2, the heating value of neat ethanol is lower than E10. This affects engine performance, especially hydrous ethanol. As shown in Fig. 2(a) and 2(b), E100 showed slightly lower torque than that of E10 around 0.4% - 3.9% and 0.12% - 5.0% at 25% and 50% loads, respectively. In the case of Eh95, the torque was significantly lower than E10 about 7.4% -10.8% and 6.2% -10.1% at 25% and 50% load, respectively. The brake power output was calculated from torque and engine speed. Therefore, the trends of brake power of both loads were similar to the trends of torque.

The Electronic Control Unit (ECU) of the engine limits the longest injection duration and the injectors were designed to use with gasoline or gasohol. In addition, the heating value of ethanol (E100 and Eh95) is lower than that of gasohol. Therefore, at the high load in which the engine required the high energy input, the engine could not operate with the limited amount of ethanol for all test conditions. As shown in Fig. 2(c), E100 could operate for all engine speeds but maximum performance could not be obtained. In the case of Eh95, the engine failed to operate in some speeds.

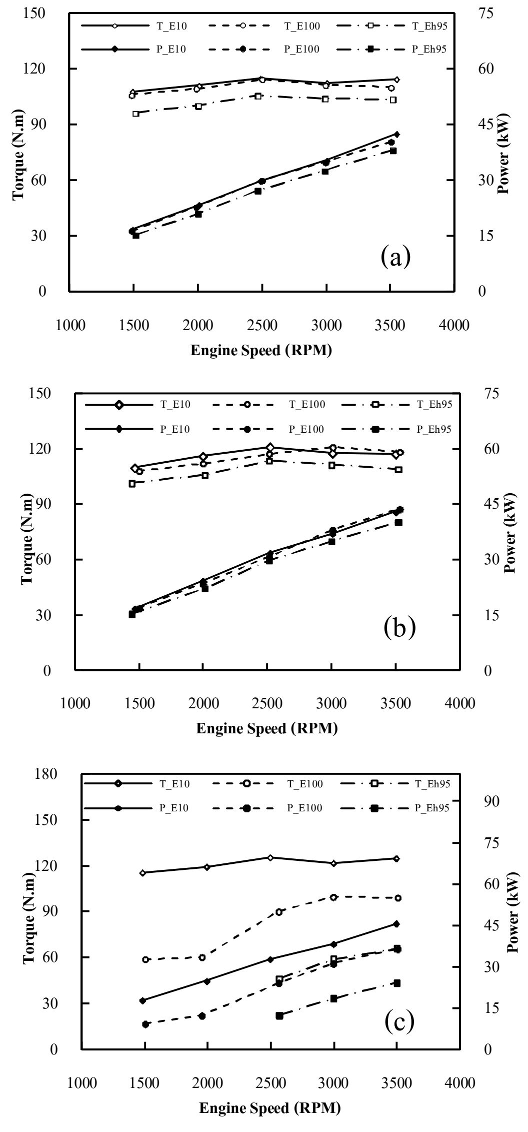

Fig. 2. Fuel influence on engine torque and power at (a) 25%, (b) 50%, and (c) 75% loads

3.1.2 BSFC and thermal efficiency

Fig. 3 shows the increased BSFC for both ethanol fuels when compared with E10 at 25% load (Fig. 3(a)) and 50% (Fig. 3(b)). At light load, the engine consumes more fuel for all tested than that of medium load (50% load). Hence, higher thermal efficiency at light load is observed in Fig.4.

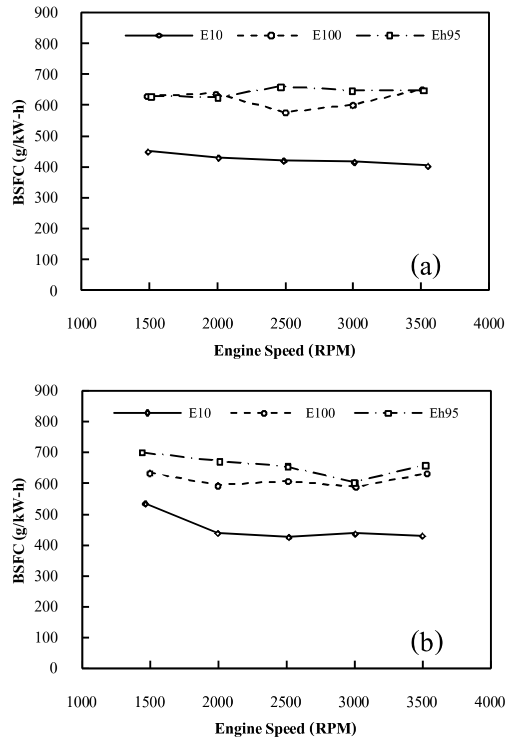

Fig. 3. Fuel influence on BSFC at (a) 25% and (b) 50% loads

The hydrous ethanol has the lowest heating value and tends to operate at low relative air-to-fuel ratios. Under low load condition of 25%, the BSFC of ethanol was greater than E10 fuels about 27.2% to 38.1% for E100 and 28.4% to 37.8% for Eh95. At 50% load, 15.6% to 31.9% of E100 was consumed more than E10 while Eh95 showed the increased BSFC in the range of

23.6% to 34.9%. These trends agree well with the previous work [4-7].

Fig. 4 shows thermal efficiencies when the engine was operated with the three fuels at light load (Fig. 4(a)) and medium load (Fig. 4(b)). It is clearly seen that both ethanol combustions result in lower thermal efficiency.

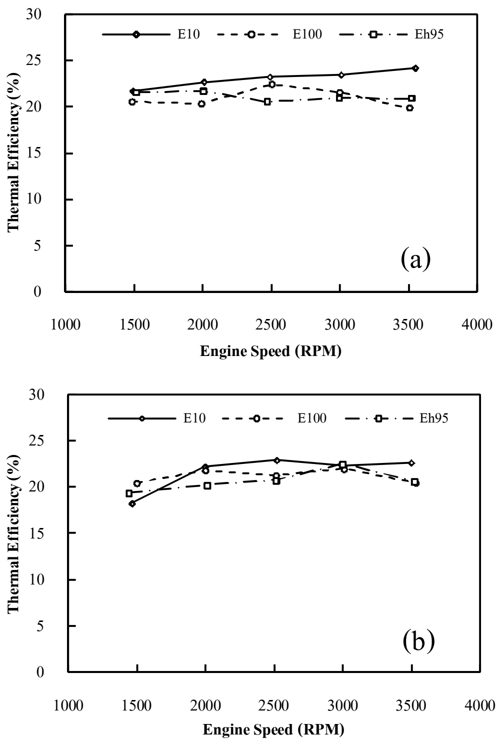

Fig. 4. Thermal efficiency at (a) 25% and (b) 50% loads

The thermal efficiencies from both ethanol fuel combustions were lower than that of gasohol in the range of 0.78% to 4.32% and 0.12% to 3.27% for E100 and Eh95 at 25% load (Fig. 4(a)). For 50% load (Fig. 4(b)), thermal efficiency of ethanol was lower than gasohol up to 2.19% to 2.2% and 1.11% to 2.2% for E100 and Eh95, respectively. These results differ from those obtained by the previous work [4-7] due to the fact that this work uses the un-calibrated engine.

The lower thermal efficiency of both ethanol fuels can be explained by relative air-to-fuel ratio (λ) that is defined as the actual air-to-fuel mass ratio to theoretical air-to-fuel mass ratio. The test engine was initially designed to operate on gasoline or gasohol. Therefore, E10 was burned in near complete combustion condition (λ~0.99) that achieved the highest thermal efficiency in each condition. On the other hand, both ethanol fuels were operated in the slightly lean conditions, λ~1.05 and λ~1.10 for E100 and Eh95 as the engine was un-calibrated. In the future work, the λ will be focused to improve thermal efficiency and engine performance by means of engine calibration.

3.2 Combustion analysis

The combustion characteristics of the three types of fuels are compared in terms of cylinder pressure and heat release rate (*HHR*) which can be calculated by a single zone of the first law of thermodynamics [7] in Eq. (4)

$$HRR = \frac{dQ_{net}}{d\theta} = \frac{\gamma}{\gamma-1}P\frac{dV}{d\theta} + \frac{1}{\gamma-1}V\frac{dP}{d\theta} \qquad (4)$$

where:

Q_{net} = *Net heat Rate (J)*

θ = *Crank angle (degree)*

γ = *Specific heat ratio* (c_p/c_v)

P = *Cylinder pressure (bar)*

V = *Cylinder volume* (m^3)

Furthermore, mass burned fraction (X_b) profile is calculated to analyze the fuel-air mixture burn rate. It is calculated from cylinder pressure and cylinder volume, which developed by Rassweiler and Withrow [7]. This equation is based on assumption that, the actual pressure change (Δp) is assumed to be the sum of a pressure rise due to combustion (Δp_c) and a pressure change due to volume change (Δp_v):

$$\Delta p = \Delta p_c + \Delta p_v \tag{5}$$

The pressure rise from combustion is proportional to the heat added to the in-cylinder medium during the crank angle interval. The X_b at the end of the considered *i-th* interval could be calculated as [8]:

$$X_b = \frac{m_{b(i)}}{m_{b(total)}} = \frac{\sum_0^i \Delta p_c}{\sum_0^N \Delta p_c} \tag{6}$$

where:

0 = Start of combustion

i = Crank angle interval

N = End of combustion (the total number of crank intervals)

The crank angle position corresponding to the start of combustion for mass burn analysis is related to spark ignition timing and the end of combustion is where the X_b value reaches the unity.

The HHR and X_b are shown after the start of ignition timing (ASOI). The starts of ignition timing at 2,500 rpm were of 27.0°, 29.0° and 27.0° before top dead center (BTDC) and 30.5°, 33.0° and 31.5° BTDC at 3,000 rpm for E10, E100, and Eh95, respectively.

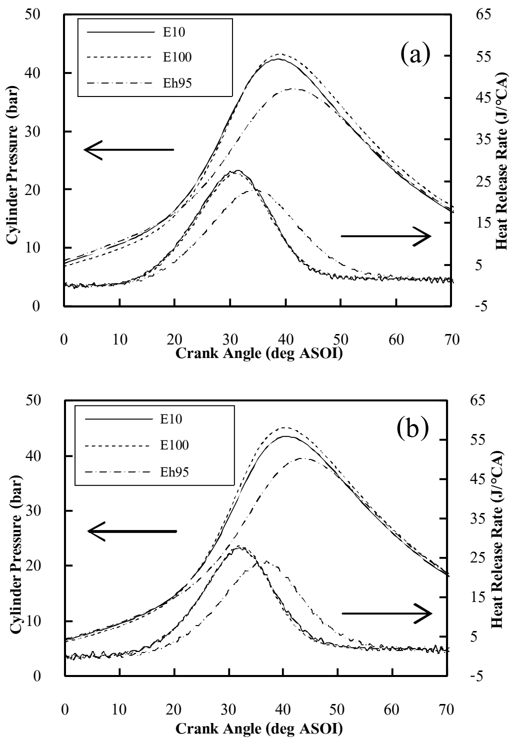

Fig. 5. Cylinder pressure and heat release rate at 70 Nm (a) 2,500 and (b) 3,000 rpm

The results of combustion characteristics in Fig. 5 confirmed the reduced engine performance. The cylinder pressure profiles of hydrous ethanol showed the lowest maximum cylinder pressure for both engine speeds. It is lower than E10 and E100 approximately 5.20 and 6.01 bar at 2,500 rpm and 5.48 and 4.00 bar at 3,000 rpm. It is consistent to the HRR profile in which the hydrous ethanol resulted in the lowest value. The maximum heat release rates of Eh95 were lower than E10 and E100 about 4.68 and 4.06 J at 2,500 rpm and 3.98 and 4.51 J at 3,000 rpm.

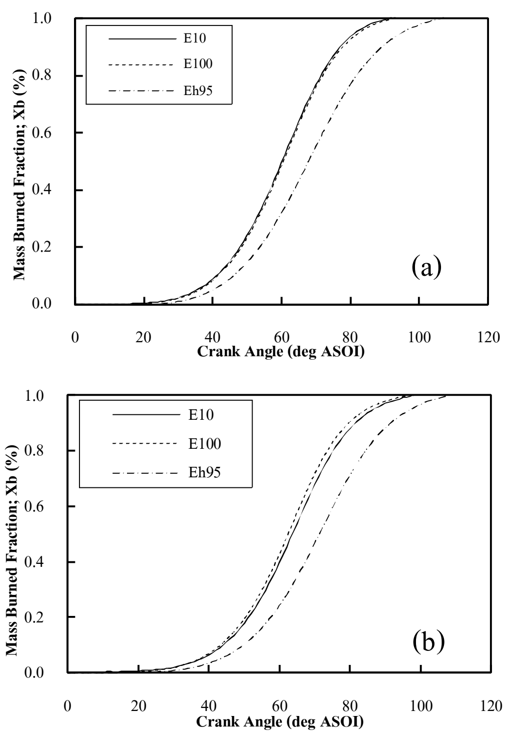

Fig. 6. Mass burned fraction rate (X_b) at 70 Nm (a) 2,500 and (b) 3,000 rpm

The slowest flame development is observed from the engine running on the hydrous ethanol (Fig. 6) although the spark timing ignited as the same time of E10. During 0 - 10% of X_b called flame-development period, Eh95 exhibited the latest start of flam propagation. In addition, Eh95 consumed the longest time in the rapid-burning combustion phase (10% to 90% of X_b) [7]. The water contained in hydrous ethanol mitigates combustion process and yields the mixture to ignite at unsuitable timing. These causes lead to the reduction in the engine performance when hydrous ethanol is fuelled. However, due to higher octane number of hydrous ethanol than that of gasohol, it is possible to improve the combustion process by increasing compression ratio or advancing ignition timing.

In the case of anhydrous ethanol, the results in Fig. 5 show the higher maximum cylinder pressure and maximum HHR than the baseline fuel. The advanced spark timing around 2 degree crank angle may be the cause. In addition, the faster laminar flame speed of the pure ethanol than that of gasohol rapidly rises up the cylinder pressure [4].

4. Conclusion

The unmodified engine used in the test successfully runs on both ethanol fuels at low and medium load conditions. The engine brake torque and power from hydrous ethanol combustion are lower than those of gasohol for all test conditions. BSFCs of E100 and

Eh95 are greater than E10 up to 38.1% and 37.8%, respectively. Thermal efficiencies from both ethanol combustions are lower than those of gasohol, especially at low load condition, up to 4.32% for E100 and 3.27% for Eh95.

The hydrous ethanol combustion shows the lowest maximum pressure and HRR when compared to other fuels. Hydrous ethanol starts its combustion at the latest, causing the reduced engine performance. However, it shows the possibility to improve engine performance by calibrating some engine parameters such as ignition timing and injection duration or even replacing injectors.

5. Acknowledgments

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