Effect of water-containing acetone-butanol-ethanol gasoline blends on combustion, performance, and emissions characteristics of a spark-ignition engine

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1 **Abstract**

2 Bio-butanol has proved to be a promising alternative fuel in recent years; it is typically
3 produced from ABE (acetone-butanol-ethanol) fermentation from non-edible biomass feedstock.
4 The high costs for dehydration and recovery from dilute fermentation broth have so far prohibited
5 bio-butanol's use in internal combustion engines. There is an interesting in studying the
6 intermediate fermentation product, i.e. water-containing ABE as a potential fuel. However, most
7 previous studies covered the use of water-containing ABE-diesel blends. In addition, previous
8 studies on SI engines fueled with ABE did not consider the effect of water. Therefore, the
9 evaluation of water-containing ABE gasoline blends in a port fuel-injected spark-ignition (SI)
10 engine was carried out in this study. Effect of adding ABE and water into gasoline on combustion,
11 performance and emissions characteristics was investigated by testing gasoline, ABE30, ABE85,
12 ABE29.5W0.5 and ABE29W1 (29 vol.% ABE, 1 vol.% water and 70 vol.% gasoline). In addition,
13 ABE29W1 was compared with gasoline under various equivalence ratios ($\Phi=0.83\sim1.25$) and
14 engine loads (3 and 5 bar BMEP). It was found that ABE29W1 generally had higher engine torque
15 (3.1-8.2%) and lower CO (9.8-35.1%), UHC (27.4-78.2%) and NO_x (4.1-39.4%) than those of
16 gasoline. The study indicated that water-containing ABE could be used in SI engines as an
17 alternative fuel with good engine performance and low emissions.

18 **Keywords:** Water-containing ABE; Alternative fuel; Performance; Combustion; Emissions; SI
19 engine

20 **1. Introduction**

21 The shortage of fossil fuels, continued deterioration of the global environment and increasing
22 demand for energy security and diversity are driving the utilization of bio-fuels in engines [1-3].
23 Due to being biodegradable, less detrimental to ground water, and having a positive impact on
24 emissions reduction and performance improvement, ethanol has been commonly used in
25 spark-ignition (SI) engines [4, 5]. However, the use of ethanol as an alternative fuel in SI engines
26 needs to be examined due to the following critical issues: Ethanol is fully miscible in water, and
27 thus is corrosive to the existing distribution pipelines [6]; Compared to gasoline, the lower energy
28 content of ethanol has a negative effect on fuel economy [7]; The higher latent heat of vaporization
29 of ethanol results in issues during cold start [8]; Some bio-ethanol productions rely on edible
30 resources, which may lead to food shortages [9].

31 Due to its properties being relatively similar to gasoline, great attention has been paid to
32 butanol as a more promising fuel candidate in SI engines [10-12]. Butanol has several advantages
33 over ethanol, including being less hydrophilic and corrosive, having higher energy content and
34 flame velocity, and having relatively low latent heat of vaporization [13]. Szwaja and Naber [14]
35 investigated the combustion characteristics of n-butanol in a single cylinder CFR engine under
36 different spark timings, compression ratios and n-butanol volumes. The results showed that the
37 knocking tendency of n-butanol was similar to that of gasoline with a research octane number
38 (RON) of 90. Combustion stability did not vary significantly for pure n-butanol as measured by the
39 coefficient of variation (COV) of indicated mean effective pressure (IMEP). Wallner et al. [15]
40 compared the engine performance of pure gasoline, 10% ethanol and 10 % butanol-gasoline blends
41 (E10 and B10). Brake specific fuel consumption (BSFC) showed an increase of approximately 3.4%
42 for B10 and 4.2% for E10 compared to gasoline, while a similar brake thermal efficiency (BTE)

43 was presented between the three fuels. Dernet et al. [16] quantified the influence of butanol
44 addition on the emissions of unburned hydrocarbons (UHC), carbon monoxide (CO) and nitrogen
45 oxides (NO_x). They found that B20 and B40 produced lower UHC emission than gasoline. It was
46 also found that B80 was the only butanol-gasoline blend that did not produce lower CO emission
47 than gasoline. At stoichiometric and slightly lean conditions, NO_x levels were similar for all blends
48 except for B80.

49 Bio-butanol is typically produced during the acetone-butanol-ethanol (ABE) fermentation
50 process from biomass. Its production efficiency, however, is quite low, even though the
51 fermentation process has been improved a lot [17]. Compared to butanol production, the yeast
52 ethanol fermentation process has a 10-30 times higher production rate [18]. In addition, the
53 dehydration and recovery of butanol from ABE fermentation mixture expends a lot of money and
54 energy. As a result, the cost of butanol is high, which has hindered its extensive use.

55 Given that the issue related to the high cost of butanol would be avoided if ABE could be
56 directly used for clean combustion, the interest in ABE as an alternative fuel has recently increased.
57 Chang et al. [19] investigated a series of water-containing ABE-diesel blends in an engine. It was
58 found that a diesel mixed with 20 vol.% ABE and 0.5 vol.% water enhanced the BTE by
59 3.26-8.56%. In addition, the emissions of particular matter, polycyclic aromatic hydrocarbons and
60 NO_x could be simultaneously reduced compared to diesel. They also reported the use of
61 water-containing ABE-biodiesel-diesel blends and found that both biodiesel-diesel blends and
62 ABE-biodiesel-diesel blends had a lower PM than regular diesel, but only the latter simultaneously
63 reduced NO_x emission [20]. Zhou et al. [21, 22] carried out a study on the combustion
64 characteristics of ABE-diesel blends in a constant volume chamber. The longer ignition delay and
65 soot lift-off length of ABE-diesel blends allowed more air entrainment upstream of the spray jet,

thus providing better air-fuel mixing. Meanwhile ABE20 presented close-to-zero soot luminosity with better combustion efficiency compared to diesel at a low ambient temperature of 800 K and ambient oxygen concentration of 11%. Wu et al. [23] compared the spray and combustion characteristics between pure ABE, n-butanol and diesel. The longer flame lift-off length, shorter liquid penetration and stronger premixed combustion indicated that ABE had a high potential to reduce soot. Through the study of ABE-diesel blends with different ABE component volumetric ratio (A:B:E of 6:3:1; 3:6:1; 0:10:0), the effect of acetone on the spray combustion of ABE-diesel blends was investigated [24, 25]. ABE20 (6:3:1) displayed combustion characteristics similar to those of diesel with a shorter ignition delay. Zhao et al. [26, 27] proposed a phenomenological soot model of ABE with the modification of the fuel pyrolysis process. Nithyanandan et al. [28, 29] investigated the performance and emissions of ABE-gasoline blends in an SI engine and found that the addition of 40 vol.% ABE or less did not change the combustion characteristics significantly.

The research done on ABE as an alternative fuel in engines is still at an early stage. To date, the use of water-containing ABE-gasoline blends in SI engines has not been investigated. Therefore, the objective of this study is to evaluate the use of water-containing ABE-gasoline blends in a port fuel-injected SI engine based on performance, combustion and emissions characteristics of various fuels, including pure gasoline (G100), ABE30, ABE85, ABE29.5W0.5 and ABE29W1 (29 vol.% ABE, 1 vol.% water and 70 vol.% gasoline).

2. Experimental methods

2.1. Fuel preparation

In this study, pure commercial winter gasoline with a RON of 90 was selected as the baseline fuel. The ABE mixture was first prepared with the typical volumetric concentration of 30% acetone, 60% butanol and 10% ethanol. In addition, less than 1% water was added to the ABE mixture to

89 simulate the raw ABE fermentation product [30]. Analytical grade acetone (99.5%), butanol
90 (99.5%), ethanol (99.8%) and distilled water were mixed with pure gasoline (G100) using a
91 temperature-controlled magnetic stirrer to make ABE30, ABE85, ABE29.5W0.5, ABE29W1 (29
92 vol.% ABE, 1 vol.% water and 70 vol.% gasoline) as shown in Table 1. The stability of fuel blends
93 was tested using a gravitational test. The prepared fuels were deposited in test tubes at 25 °C and 1
94 atm for 14 days. The fuels remained a clear single phase throughout the stability test.

Table 1 Properties of the test fuels [31-36]

Parameters	Individual fuels				Fuel blends				
	Gasoline	Acetone	Butanol	Ethanol	ABE30	ABE29.5W0.5	ABE29W1	ABE85	ABE100
Chemical formular	C ₄ -C ₁₂	C ₃ H ₆ O	C ₄ H ₉ OH	C ₂ H ₅ OH					
Octane number	90		87	100					
Oxygen content (wt.%)		27.6	21.6	34.8	7.4	7.7	8.1	21.0	24.7
Density (kg/m ³)	715-765	791	813	795					804.6
Lower heating value (MJ/kg)	43.4	29.6	33.1	26.8	39.6	39.3	38.9	33.1	31.4
Boiling Temperature (°C)	38-204	56	118	78					
Latent heat at 298 K (kJ/kg)	380-500	518	582	904					595
Stoichimometric AFR	14.7	9.5	11.2	9.0	13.44	13.38	13.34	11.1	10.5
Auto-ignition temperature (°C)	228-470	465	343	420					
Laminar flame speed (cm/s)	33-44 ^A	34 ^B	48 ^C	48 ^C					37 ^B

Note: ^Ap=1 atm, T =298~358K, Φ =1; ^Bp=1 atm, T = 298 K, Φ =1; ^Cp=1 atm, T =343 K, Φ =1

96 2.2. Test engine

97 Experiments were conducted using a single cylinder SI engine with identical cylinder geometry
98 to a 2000 Ford Mustang Cobra V8. The general specifications of this engine are given in Table 2.
99 The engine was coupled with a GE type TLC-15 class 4-35-1700 dynamometer capable of
100 absorbing a maximum load of 26 kW at a speed of 4500 rpm (see Fig. 1). A DYN-LOC IV
101 controller and a DyneSystems DTC-1 controller were used to control the dynamometer and
102 throttle position, respectively. The engine was controlled by a Megasquirt V3.0 engine control unit,
103 which allowed the adjustment of the fuel injection and spark timing. In-cylinder pressure was
104 measured by a Kistler type 6125B pressure transducer and recorded by a LabVIEW code. The crank

105 angle position was acquired with a BEI XH25D shaft encoder. The intake air system provided
106 compressed, filtered air from the building to the engine. The flow of air from the building supply
107 system was controlled by an electronic regulator which provided precise control (Accuracy: +/- 0.1
108 kPa). From there the air flowed through a surge tank and then into the intake manifold. An intake air
109 temperature sensor and a manifold absolute pressure (MAP) sensor were installed at the intake
110 manifold directly to measure intake air temperature and pressure. The air then flowed past a throttle
111 body where the throttle valve was controlled with a throttle position sensor. From there the air was
112 mixed with the injected fuel and flowed into the engine via the intake valves. The fuel injector used
113 was a Bosch injector rated at 440 cm³/min at a fuel pressure of 3 bar. A Horiba MEXA-720
114 instrument was used to measure air-fuel ratio (AFR) and NO_x emission with an accuracy of 5% for
115 AFR, and 30 ppm, 3% and 5% for NO_x emission in range of 0-1000 ppm, 1000-2000 ppm and
116 2000-3000 ppm, respectively. The measurements of CO and UHC emissions were taken using a
117 Horiba MEXA-554JU with an accuracy of 12 ppm Vol. for UHC and 0.06% Vol. for CO. Water
118 vapor in the exhaust gas was condensed before emissions measurements. All instruments were
119 calibrated before use.

120

Fig. 1. Engine setup

Table 2 Engine specifications

Engine type	SI engine
Displaced volume (cm ³)	575
Stroke (mm)	90.1
Bore (mm)	90.3
Connecting rod length (mm)	150.7
Compression ratio	9.6:1
Number of valves	4
Number of cylinders	1

2.3. Test conditions

In this study, the engine speed was fixed at 1200 rpm. The throttle was adjusted to be fully open, and thus engine load was characterized by the MAP sensor. By regulating supply air, intake manifold pressures of 60 kPa and 90 kPa were used, which corresponded to engine loads of 3 bar BMEP (brake mean effective pressure) (low load) and 5 bar BMEP (high load), respectively. Maximum brake torque (MBT) is always used to optimize the spark timing to get maximum power and efficiency. Therefore, for each operating condition in this study, the engine was run at the fuel's MBT spark timing. Equivalence ratio was varied from 0.83 to 1.25, covering lean, stoichiometric and rich conditions. Measurements of engine torque, equivalence ratio and NO_x emission were averaged over a 60-second period, while UHC and CO emissions were recorded directly from the

emissions analyzer. Each experiment was performed three times, with each set of data taken on a single day. The three datasets were then averaged. The tests were performed on several consecutive days in a temperature and humidity-controlled laboratory. Furthermore, the engine was allowed to run at every operating condition for an extended period of time to ensure steady state measurements. The test conditions mentioned above are summarized in Table 3.

140

Table 3 Test conditions

Throttle position (%)	100
Engine speed (rpm)	1200
Load (bar BMEP)	3 and 5
Equivalence ratio	0.83-1.25
Fuel pressure (bar)	3
Spark timing	Each fuel's MBT

141 3. Results and discussion

142 3.1. Effects of ABE additive on performance, combustion and emission characteristics

Fig. 2 shows the effects of 30 vol.% and 85 vol.% ABE added to gasoline on combustion characteristic. As seen in Fig. 2(a), a higher peak pressure was caused by adding ABE due to the increased combustion rate which was discussed as follows. The combustion process of the fuels was considered in this study, thus, the mass fraction burnt (MFB) was of particular interest. The MFB can reflect combustion rate, and is defined as the ratio of M_F upon M_T , where M_F is the integrated heat release at (up to) each CAD and M_T is the total heat released in the cycle [14]. Fig. 2(c) compares initial combustion duration (ICD) given by 0-10% MFB and major combustion duration (MCD) given by 10-90% MFB between G100, ABE30 and ABE85 at $\Phi=1$ and 3 bar BMEP. During the ICD period, combustion rate was mainly impacted by the laminar flame speed (LFS) of fuel air mixture [14]. It was observed that the ICD decreased with increasing ABE blend ratio due to the higher LFS of ABE (see Table 2). However, a much lower ICD was not attained when ABE blend ratio increased to 85 vol.% probably because of another important parameter, liquid volatility. The

155 poor vaporization of butanol due to its relatively low vapor pressure meant that it needed a higher
156 temperature or longer time to completely vaporize, which could extend the ICD period [37, 38]. Ref.
157 [39, 40] also found a retard combustion phasing after adding butanol into gasoline. During the
158 MCD period, which was in the fully developed bulk burn, the combustion was dominated by the
159 turbulent flame. The higher pressure built during early combustion period promoted the mixing of
160 fuel and air due to the increase in turbulence, and it improved the combustion rate in the following
161 flame propagation [14]. Thus, the MCD for G100, ABE30 and ABE85 followed the same trend as
162 ICD.

163

164

165 **Fig. 2.** Comparisons of combustion characteristics at $\Phi=1$ and 3 bar BMEP between G100, ABE30 and ABE85: (a)
 166 Pressure; (b) MFB; (c) ICD; (d) MCD

167 Fig. 3 shows the effects of 30 vol.% and 85 vol.% ABE added to gasoline on engine
 168 performance. As shown in Fig. 3(a), ABE-gasoline blends provided a higher engine torque.
 169 Compared to G100, engine torque was increased by 3.1% and 4.6% for ABE30 and ABE85,

170 respectively. This could be explained by several reasons. As an oxygenated fuel, ABE addition
 171 would improve combustion quality. The higher latent heat of ABE compared to G100 caused lower
 172 intake manifold temperature and increased volumetric efficiency [41]. In addition, the reduced
 173 combustion duration was beneficial for improving thermal efficiency [42]. As expected, the
 174 indicated specific fuel consumption (ISFC) increased as the ABE content increased due to the lower
 175 LHV of ABE as shown in Fig. 3(b). The ABE addition caused an 10.1% and 29.4% increase in
 176 ISFC of ABE30 and ABE85, respectively, when compared to G100.

177
 178 **Fig. 3.** Comparisons of engine performance at $\Phi=1$ and 3 bar BMEP between G100, ABE30 and ABE85: (a)
 179 Torque; (b) ISFC

180 Fig. 4 compares CO, UHC, NO_x emissions and maximum in-cylinder temperatures for G100,
 181 ABE30 and ABE85. The maximum in-cylinder temperatures were calculated based on the modified
 182 Woschni in-cylinder heat transfer model. From Fig. 4(a), a higher CO emission was observed in
 183 ABE30 and ABE85 compared to G100, respectively. The conversion from CO to CO₂ occurred

184 through the later stages of combustion. Oxygen concentration and combustion temperature were the
185 main factors affecting this reaction [43]. After adding ABE, the lack of oxygen should not lead to
186 increased CO emission. Previous studies explained that even though a high oxygen content in
187 alcohol-containing fuels offered a higher combustion quality, more products were produced in terms
188 of heat capacity of combustion products. This lowered combustion temperature (see Fig. 4(d)) and
189 further slowed down the oxidation process of CO emission [44, 45]. Additionally, ABE-gasoline
190 blends had a shorter combustion duration based on the analysis mentioned above, and thus
191 insufficient oxidation of CO caused the increased CO emission [46]. In comparison between ABE30
192 and ABE85, the higher oxygen content in ABE85 benefited the oxidation of CO emission, and thus
193 caused lower CO emission than that of ABE30. As shown in Fig. 4(b), a significant reduction in
194 UHC emission was observed for ABE-gasoline blends. This can be explained by the fact that the
195 carbon to hydrogen (C/H) fuel ratio was lower for ABE-gasoline blends, limiting the level of UHC
196 emission [42]. The oxygen content contained in ABE promoted more complete combustion, which
197 also led to the reduced UHC emission. NO_x emission are generally formed by three pathways: (1)
198 fuel-nitrogen conversion, (2) Zeldovich thermal activation, and (3) fuel-rich prompt formation [47].
199 Zeldovich thermal activation is the predominant mechanism for NO_x emission formation from
200 internal combustion engines. The higher combustion temperature and local oxygen concentration in
201 the peak temperature zone favors NO_x emission formation. In comparison with G100, ABE-gasoline
202 blends had lower NO_x emission, meaning that the addition of ABE was effective in inhibiting NO_x
203 emission formation (see Fig. 4(c)) as a result of the decreased combustion temperature, as shown in
204 Fig. 4(d).

205

Fig. 4. Comparisons of emissions at $\Phi=1$ and 3 bar BMEP between G100, ABE30 and ABE85: (a) CO; (b) UHC; (c) NO_x; (d) Max. In-cylinder Temperature

3.2. Effect of water additive on performance, combustion and emissions characteristics

The effect of water addition to the ABE-gasoline blend on performance, combustion and emissions characteristics was studied in this section. Fig. 5 compares the combustion characteristics

212 between ABE30, ABE29.5W0.5 and ABE29W1. When looking at the combustion development in
213 different phases, the ICD and MCD of ABE29.5W0.5 were increased by 5.9% and 2.9%, while
214 those of ABE29W1 were decreased by 0.8% and 5.5%, when compared to ABE30, respectively.
215 Das et al. [48] investigated the influence of water addition on the combustion rate of H₂/CO-air
216 mixtures, and found that the combustion rate increased because the improvement in chain reactions
217 due to water addition was much more pronounced than the negative thermal effect for low
218 percentages of water addition. Dryer [49] and Rajan [50] also explained the improved combustion
219 rate as a result of the catalytic activity of water vapor due to increased OH radicals. In addition, the
220 water addition reduced combustion temperature and lowered the combustion rate. These competing
221 factors could explain why ABE29W1 had a shorter ICD and MCD compared to G100, but
222 ABE29.5W0.5 did not. However, the effect of water addition on combustion characteristics of
223 hydrocarbon fuels is not completely understood [51]. Further work on elucidating the physical and
224 chemical kinetic roles of water on the combustion in engines is still in progress.

Fig. 5. Comparisons of combustion characteristics at $\Phi=1$ and 3 bar BMEP between ABE30, ABE29.5W0.5 and ABE29W1: (a) Pressure; (b) MFB; (c) ICD; (d) MCD

Fig. 6 compares the engine performance for ABE30, ABE29.5W0.5 and ABE29W1. Compared to the engine torque of ABE30, ABE29.5W0.5 showed a similar value, while a 3.7% increase was observed with in ABE29W1. Ref. [52] also obtained an increased engine torque after adding water to

231 butanol-gasoline blends. Similar to the previous analysis, the increased engine torque can be a result
 232 of the increased volumetric efficiency and improved combustion quality due to the higher latent
 233 heat and catalytic activity of water vapor for water-containing ABE blends. From Fig. 6(b), it was
 234 found that though ABE29.5W0.5 and ABE29W1 had a lower LHV, there was a 1.5% and 2.2%
 235 reduction in ISFC than that of ABE30, respectively, based on the fact that ISFC was also affected by
 236 thermal efficiency in addition to LHV.

237
 238 **Fig. 6.** Comparisons of engine performance at $\Phi=1$ and 3 bar BMEP between ABE30, ABE29.5W0.5 and
 239 ABE29W1: (a) Torque; (b) ISFC

240 Fig. 7 presents the CO, UHC, NO_x emissions and maximum in-cylinder temperatures for
 241 ABE30, ABE29.5W0.5 and ABE29W1. It was observed that CO emission was reduced for
 242 ABE29.5W0.5 and ABE29W1 compared to ABE30 because of the water-gas shift mechanism
 243 (WGSM) describing the reaction of carbon monoxide and water vapor to form carbon

dioxide and hydrogen ($\text{CO} + \text{H}_2\text{O} \rightleftharpoons \text{CO}_2 + \text{H}_2$) over the temperature range of 600-2000 K [53].

The application of WGSM can effectively reduce the concentration of carbon monoxide and increase hydrogen production, and thus it has been widely used in the manufacture of ammonia, hydrocarbons, methanol, etc. For UHC emission, a trend opposite to that of CO emission was found as shown in Fig. 7(b). The stoichiometric lower AFR of ABE29.5W0.5 and ABE29W1 relative to ABE30 (see Table 1) led to a higher amount of fuel getting into the crevice volumes or absorbed in oil layers and deposits, which increased UHC emission. Additionally, UHC oxidation reactions were highly temperature sensitive. The reduced combustion temperature shown in Fig. 7(d) due to water addition retarded UHC oxidation and resulted in the increase of UHC emission for ABE29.5W0.5 and ABE29W1 [54]. From Fig. 7(c), it was apparent that 0.5% and 1% water contained in the ABE-gasoline blend resulted in decreased NO_x emission because of the reduced combustion temperature. Kadota and Yamasaki [51] had summarized that water addition played a critical role in suppressing NO_x formation processes.

Fig. 7. Comparisons of emissions at $\Phi=1$ and 3 bar between ABE30, ABE29.5W0.5 and ABE29W1: (a) CO; (b) UHC; (c) NO_x; (d) Max. In-cylinder Temperature

3.3. Comparisons of performance, combustion and emissions characteristics between ABE29W1 and G100 under various equivalence ratios and engine loads

As mention above, ABE29W1 performed well with respect to engine performance and emissions. In this section, the combustion, performance and emissions characteristics of ABE29W1

264 and G100 were compared under different equivalence ratios ($\Phi=0.83\sim1.25$) and engine loads (3 and
 265 5 bar BMEP). A higher engine load provided the conditions for improving combustion rate due to
 266 the increased cylinder temperature and pressure. As a result, a shorter ICD and MCD were obtained
 267 at 5 bar BMEP as shown in Fig. 8. It was also found that the ICD and MCD decreased with
 268 equivalence ratio, which was consistent with the findings of Ref. [55]. As the fuel mixture got richer,
 269 it readily ignited and had a higher combustion rate, leading to shorter ICD and MCD [57]. It was
 270 worth noting that both ICD and MCD were almost the same for different fuels under fuel-rich
 271 conditions ($\Phi=1.25$). It implies that the oxygen concentration played a predominant role over their
 272 individual fuel properties under this condition [37]. The ICD and MCD of ABE29W1 decreased by
 273 5.5-12.7% and 1.4-10.6% for lean and stoichiometric conditions, but increased by 2.1-13.2% and
 274 1.9-2.1% for rich conditions relative to G100, respectively.

275
 276 **Fig. 8.** Comparisons of combustion characteristics under various equivalence ratio and engine load between
 277 ABE29W1 and G100: (a) ICD; (b) MCD.

278 Fig. 9 compares both engine torque and ISFC for ABE29W1 and G100 at different equivalence

ratios and engine loads. It can be seen that the engine torque for ABE29W1 was increased by
 3.9-8.2% at 3 bar BMEP, as well as by 3.1-7.4% at 5 bar BMEP over the entire equivalence ratio
 range compared to G100. As for the ISFC, ABE29W1 showed a 5.6-8.9% and 6.1-9.6% higher
 ISFC than that of G100 at 3 and 5 bar BMEP, respectively, because of the decreased LHV from
 adding ABE and water to G100. Again, ISFC was also affected by thermal efficiency, a lower ISFC
 was attained at higher engine load and lean conditions due to higher thermal efficiency. The higher
 cylinder temperature caused by increasing engine load resulted in the higher thermal efficiency. In
 addition, combustion was incomplete under the rich conditions. With the further decrease of
 equivalence ratio, increased dilution improved isentropic efficiency by decreasing heat loss and
 increasing the adiabatic exponent due to the lower temperatures than that under stoichiometric
 conditions [56]. This contributed to the higher thermal efficiency under lean conditions.

Comparisons of engine performance under various equivalence ratio and engine load between ABE29W1
 and G100: (a) Torque; (b) ISFC

Fig. 10 shows the variation of CO, UHC, NO_x emissions and maximum in-cylinder

temperatures with respect to equivalence ratios and engine loads. It was observed that, at a given engine load, CO emission was relatively constant for lean conditions, however, the equivalence ratio affected CO emission until lean conditions were reached [57]. ABE29W1 had higher CO emission at stoichiometric condition, while 9.8-35.1% lower CO emission was observed for rich conditions compared to G100. This could be explained by the increased fuel-borne oxygen content that was available for CO emission oxidation at rich conditions. The positive factors, including improved reactions due to the increased OH radicals and increased oxygen concentration, led to more complete combustion and resulted in a 27.4-78.2% decrease in UHC emission for ABE29W1. In addition, UHC emission increased rapidly as the mixture became substantially richer owing to incomplete combustion. Because the thermal mechanism was a major source of NO_x formation, the higher NO_x emission was observed at stoichiometric condition and high engine load, where higher combustion temperature was obtained. The added water and ABE resulted in a lower combustion temperature, therefore producing the 4.1-39.4% lower NO_x emission for ABE29W1 compared to G100 over all equivalence ratios and engine loads. ABE29W1 showed higher NO_x emission under richer conditions at 5 bar BMEP, which possibly resulted from the fuel rich formation mechanism for NO_x formation. This meant that more hydrocarbon radicals were generated from ABE29W1 due to the lower molecular weight and increased amount of fuel injected, which increased the formation of HCN and led to higher NO_x emission than that of G100.

Fig. 10. Comparisons of emissions under various equivalence ratio and engine load between ABE29W1 and G100: (a) CO; (b) UHC; (c) NO_x; (d) Max. In-cylinder Temperature

4. Conclusions

The present study was designed to evaluate the potential of water-containing ABE gasoline blends as a green alternative fuel in SI engines. The ICD, MCD, engine torque, ISFC, and CO, UHC and NO_x emissions were analyzed to investigate the performance, combustion and emission

319 characteristics of an engine fueled with G100, ABE30, ABE85, ABE29.5W0.5 and ABE29W1. The
320 major results are as follows.

321 (1) The effect of adding ABE to gasoline was studied comparing G100, ABE30 and ABE85. It
322 was found that the ICD and MCD were decreased with increasing ABE ratio in ABE-gasoline
323 blends. For engine performance, ABE-gasoline blends show a higher engine torque and ISFC. In
324 addition, ABE-gasoline blends decreased UHC and NO_x emissions, but increased CO emission.

325 (2) The investigation of 0.5 and 1 vol.% water added to ABE-gasoline blend (ABE30) showed
326 that the ICD and MCD were increased for ABE29.5W0.5, while decreased for ABE29W1 relative
327 to ABE30. It should be noted that water-containing ABE-gasoline blends not only improved engine
328 torque, but also reduced ISFC, especially in the case of ABE29W1. Meanwhile, water-containing
329 ABE-gasoline blends showed decreased CO and NO_x emissions and increased UHC emission when
330 compared to ABE30.

331 (3) In the comparison of G100 and ABE29W1 under various equivalence ratios ($\Phi=0.83\sim1.25$)
332 and engine loads (3 and 5 bar BMEP), ABE29W1 presented a faster combustion rate than G100 for
333 leaner and stoichiometric mixtures, while the condition was reversed for richer mixtures. Generally,
334 ABE29W1 increased engine torque (3.1-8.2%) and reduced CO (9.8-35.1%), UHC (27.4-78.2%),
335 NO_x (4.1-39.4%) compared to that of G100.

336 (4) This study found that the direct use of water-containing ABE as a green alternative fuel in
337 SI engines, without dehydration and recovery processes, could be feasible based on the improved
338 engine performance and the reduced emissions.

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343 **Definitions/Abbreviations**

ABE	Acetone-Butanol-Ethnaol
ATDC	After Top Dead Center
BMEP	Brake Mean Effective Pressure
BTE	Brake Thermal Efficiency
BSFC	Brake Specific Fuel Consumption
CA	Crank Angle
CO	Carbon Monoxide
COV	Coefficient of Variation
ECU	Engine Control Unit
ICD	Initial Combustion Duration
IMEP	Indicated Mean Effective Pressure
ISFC	Indicated Specific Fuel Consumption
LHV	Lower Heating Value
LFS	Laminar Flame Speed
MFB	Mass Fraction burned
MCD	Major Combustion Duration
MAP	Manifold Absolute Pressure
MBT	Maximum Brake Torque
NO _x	Nitrogen Oxides
RON	Research Octane number
RPM	Revolutions Per Minute
SI	Spark Ignition
UHC	Unburned Hydrocarbon
WGSM	Water-Gas Shift Mechanism
Φ	Equivalence Ratio

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