Combustion, performance and emissions characteristics of a spark-ignition engine fueled with isopropanol-n-butanol-ethanol and gasoline blends

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1 **Abstract**

2 Among primary alcohols, bio-n-butanol is considered as a promising alternative fuel candidate.
3 However, relatively low production efficiency and high cost of component recovery from the
4 acetone-n-butanol-ethanol (ABE) or isopropanol-n-butanol-ethanol (IBE) fermentation processes
5 hinders industrial-scale production of bio-n-butanol. Hence it is of interest to study the intermediate
6 fermentation product, i.e. ABE and IBE as a potential alternative fuels. However, for fuel
7 applications, the IBE mixture appears to be more attractive than ABE due to more
8 favorable properties of isopropanol over acetone, such as being less corrosive to engine part, higher
9 energy density and octane number. In this study, an experimental investigation on the performance,
10 combustion and emission characteristics of a port fuel-injection SI engine fueled with IBE-gasoline
11 blends was carried out. By comparisons between IBE-gasoline blends with various IBE content (0
12 vol.%-60 vol.% referred to as G100-IBE60) and more commonly used alternative alcohol fuels
13 (ethanol, n-butanol and ABE)-gasoline blends, it was found that IBE30 performed well with respect
14 to engine performance and emissions, including brake thermal efficiency (BTE), brake specific fuel
15 consumption (BSFC), carbon monoxide (CO), unburned hydrocarbons (UHC) and nitrogen oxides
16 (NO_x). Then, IBE30 was selected to be compared with G100 under various equivalence ratio
17 ($\Phi=0.83\sim 1$) and engine load (300 and 500 kpa BMEP). Overall, higher BTE (0.04-4.3%) and lower
18 CO (4%), UHC (15.1-20.3%) and NO_x (3.3-18.6%) emissions were produced by IBE30 compared
19 to G100. Therefore, IBE could be a good alternative fuel to gasoline due to the environmentally
20 benign fermentation process (from non-edible biomass feedstock and without recovery process) and
21 the potential to improve energy efficiency and reduce pollutant emissions.

22 **Keywords:** Isopropanol-n-butanol-ethanol; Performance; Combustion; Emissions; Alternative fuel;
23 SI engine

24 **1. Introduction**

25 Depleting fuel resources and increasing environmental problems have driven the development
26 of biofuels all over the world. Among various biofuels, alcohols have been extensively investigated
27 as alternative engine fuels because of their great potential for improving engine performance and
28 reducing pollutant emissions [1-6].

29 n-butanol is considered as a more promising alcohol compared to methanol and ethanol
30 because of its numerous advantages over short-chain alcohols including higher energy density,
31 higher viscosity and better blending ability [7]. Also, n-butanol is much less hygroscopic and
32 corrosive, therefore it is known as a “drop-in” fuel that would be compatible with the current fuel
33 distribution infrastructure [8]. n-butanol is a second-generation biofuel and can be produced from
34 non-edible biomass. A well-to-wheels analysis of corn-based n-butanol as a transportation fuel
35 showed that, on a life-cycle basis, the use of corn-based n-butanol could result in fossil energy
36 saving of 39-56% over gasoline while reducing greenhouse gas emissions by up to 48% [9].
37 Motivated by the potential of n-butanol being a viable alternative fuel, Zheng et al. experimentally
38 compared combustion and emission characteristics of a two-stage injection engine fueled with pure
39 diesel, diesel/gasoline, diesel/n-butanol and diesel/gasoline/n-butanol, respectively [10]. It was
40 found that blending gasoline and/or n-butanol in diesel improved smoke emissions while induced an
41 increase in maximum pressure rise rate. They also investigated the effects of four butanol isomers,
42 including iso-butanol, sec-butanol, n-butanol and tert-butanol, on conventional and low temperature
43 combustion in a single-cylinder diesel engine [11]. The results indicated that the addition of butanol
44 isomers retarded combustion phasing, improved thermal efficiency, and reduced soot emissions
45 compared with diesel. Liu et al. observed the spray and flame natural luminosity of various
46 oxygenated biofuels in a constant volume chamber using laser diagnostics [12, 13]. Compared to

47 biodiesel, the soot concentration of n-butanol was lower and restricted within the downstream of the
48 spray jet. By comparing five different fuels, including n-heptane, iso-octane, n-butanol, 2-butanol
49 and methyl octynoate, it was concluded that oxygenated structures have little effect on nitrogen
50 oxides (NO_x), carbon monoxide (CO), unburned hydrocarbons (UHC) emissions and gross
51 indicated thermal efficiency [14]. Ref. [15] reported n-butanol/biodiesel dual-fuel injection in a
52 diesel engine. n-butanol was injected into the intake port, while soybean biodiesel was directly
53 injected into the cylinder.

54 However, n-butanol is currently less competitive with ethanol and gasoline in term of costs,
55 mainly because of the relatively low production efficiency and high recovery costs in the
56 acetone-n-butanol-ethanol (ABE) or isopropanol-n-butanol-ethanol (IBE) fermentation processes,
57 which are current bio-n-butanol production methods [16]. If the ABE or IBE mixtures could be
58 directly used for clean combustion, the costs of recovery and dehydration processes would be
59 eliminated. In this respect, ABE has been tested in several studies as a green fuel. By the
60 investigation of diesel engine generator and diesel engine dynamometer fueled with
61 water-containing ABE and diesel blends, it was found that the addition of 20 vol.% ABE and 0.5
62 vol.% water enhanced the brake thermal efficiency (BTE) and reduced the emissions of particulate
63 matter (PM), NO_x , polycyclic aromatic hydrocarbons (PAHs) and the toxicity equivalency of PAHs
64 (BaP_{eq}) [17]. The spray and combustion characteristics of ABE-diesel blends were studied in a
65 constant volume chamber [18, 19]. ABE-diesel blends showed a high potential to increase thermal
66 efficiency and decrease soot emission based on its shorter combustion duration and lower natural
67 flame luminosity. In addition, the combustion, performance and emissions characteristics of
68 ABE-gasoline blends were also investigated [20, 21].

69 However, for fuel application, the IBE mixture appears to be more attractive than ABE [22,

23]. An important reason for this is that the acetone in ABE is potentially corrosive to the engine parts composed of rubber or plastic [24-26]. Meanwhile, isopropanol shows a higher energy density than acetone (23.9 MJ/L vs 22.6 MJ/L). It also has been reported that isopropanol can be used as a fuel additive for the preparation of high-octane gasoline [27]. Therefore, the objective of this study is to evaluate the use of IBE-gasoline blends in a port fuel-injected SI engine based on the investigation on performance, combustion and emissions characteristics.

2. Experimental methods

2.1. Fuel preparation

In this study, pure commercial summer gasoline with research octane number (RON) of 92 was selected as the baseline fuel. Analytical grade acetone (99.5%), n-butanol (99.5%), isopropanol (99.5%) and ethanol (99.8%) were first mixed using a temperature-controlled magnetic stirrer to provide ABE and IBE mixtures with a volume ratio of 3:6:1 (A: B: E or I:B:E). This ratio was used to simulate the composition of the ABE and IBE fermentation product. The properties of individual fuels and fuel blends are listed in Table 1 [8, 28-34]. The properties of the fuel blends were calculated using simple mixing rules [35]. The stability of fuel blends was tested using a gravitational test. The prepared fuels were deposited in test tubes at 25 °C and 1 atm for 14 days. The fuels displayed a clear single phase throughout the stability test.

2.2. Test engine

The engine used in this study was a single cylinder SI engine with identical cylinder geometry to a 2000 Ford Mustang Cobra V8. The general specifications of the test engine are given in Table 2. The engine was connected to a GE type TCL-15 class 4-35-1700 dynamometer controlled by a DYN-LOC IV controller. A DyneSystems DTC-1 controller was used to control throttle position. A Megasquirt V3.0 electronic control unit system was used to control air-fuel ratio (AFR) and spark

timing. In-cylinder pressure was measured by a Kistler type 6125B pressure transducer and recorded by a LabVIEW acquisition system. The crank angle position was acquired with a BEI XH25D shaft encoder. The measurements of AFR and NO_x emission were conducted using a Horiba MEXA-720 analyzer. A Horiba MEXA-554JU analyzer was used to measure UHC and CO emissions. Water vapor in the exhaust gas was condensed out before emissions measurements. The measuring range, accuracy and resolution of the experimental apparatus are listed in Table 4. The picture and schematic diagram of the engine setup are shown in Fig. 1.

2.3. Test conditions and parameters

In this study, the engine speed was fixed at 1200 rpm. The throttle plate was fully opened and the intake manifold pressure was fixed at 60kPa and 90kPa by regulating the compressed air, which corresponded to engine loads of 300 kPa BMEP (Brake Mean Effective Pressure) and 500 kPa BMEP. The engine was running at the spark timing corresponding to gasoline's MBT at stoichiometric condition. Equivalence ratio varied over a range of lean and stoichiometric conditions, i.e. Φ varying from 0.83 to 1. Measurements of engine torque, equivalence ratio and NO_x emission were averaged in a 60-second period, while UHC and CO emissions were recorded directly from the emissions analyzer. The tests of each fuel were performed 3 times on a single day, and the datasets for each fuel were then averaged. The experiments were performed on several consecutive days in a temperature and humidity-controlled laboratory. The test conditions mentioned above are summarized in Table 3. In each test, the investigated parameters for combustion, performance, and emissions characteristics of the engine are normalized heat release (NHR), initial combustion duration (ICD) given by 0-10% NHR, major combustion duration (MCD) given by 10-90% NHR, brake mean effective pressure (BMEP), brake thermal efficiency (BTE), brake specific fuel consumption (BSFC), and emissions of CO, UHC and NO_x, respectively.

116 3. Results and discussion

117 3.1. Comparisons between IBE-gasoline blends with varying IBE content

118 Figs. 2 and 3 show the combustion characteristics of IBE-gasoline blends with varying IBE
119 content. It can be seen from Fig. 2(a) that IBE60 (60 vol.% IBE mixtures blended with gasoline)
120 showed the most advanced combustion phasing, resulting in the highest peak cylinder pressure. To
121 further evaluate the combustion phasing of different fuels, the ICD, MCD and 50% NHR location
122 were studied based on the NHR profiles shown in Fig. 2(b). During the ICD, the combustion rate
123 was mainly impacted by laminar flame speed (LFS) [36]. The combination of previous
124 investigations in Ref. [29, 33, 37] indicated that the components of IBE had a higher LFS than that
125 of gasoline, and therefore it can be seen from Fig. 3 that a 1.9%, 3.9% and 8.1% shorter ICD were
126 obtained by IBE10, IBE30 and IBE60, respectively, compared to that of G100. The later period of
127 MCD involves fully developed bulk burn, and the combustion is dominated by the turbulent flame
128 [36]. Combustions of the fuels followed a similar sequence with that of the ICD. It is because the
129 ICD can influence the consequent combustion rate. The short ICD caused a relatively high pressure
130 due to more energy released in the same time [38]. Then, the higher temperature and pressure built
131 during the ICD promoted the combustion rate in the following flame propagation. Furthermore, it
132 has also been shown in Ref. [36] that a higher LFS and hence a shorter ICD leads to a higher peak
133 pressure. Based on the ICD and MCD, 0.2, 0.7 and 1.2 °CA advanced 50% NHR location were
134 attained for IBE10, IBE30 and IBE60 when compared to G100.

135 Table 5 shows the engine performance of G100, IBE10, IBE30 and IBE60, including brake
136 thermal efficiency (BTE) and brake specific fuel consumption (BSFC). The BTE indicates how well
137 an engine can convert the chemical energy in fuel into mechanical energy. IBE30 had a similar BTE
138 relative to that of G100 and a 1.0% and 1.4% higher BTE than that of IBE10 and IBE60,

139 respectively. This could be explained by the advanced combustion phasing after adding IBE which
140 could lead to a substantial and rapid cylinder pressure increase before the end of the compression
141 stroke and the subsequent increase of work lost in the compression process and decrease of net
142 useful work [39]. However, the fuel-borne oxygen in IBE could improve combustion quality and
143 increase BTE. In addition, the reduced combustion duration due to IBE addition was also beneficial
144 to improve BTE [40]. For the higher BTE of IBE30, it could be explained that the positive effect of
145 fuel-borne oxygen and reduced combustion duration offset the side effect of improper combustion
146 phasing on the BTE. For the BSFC, G100 showed the minimum value due to its higher lower
147 heating value (LHV). The increases of BSFC were approximately 3.1%, 8.9% and 20.6% for IBE10,
148 IBE30 and IBE60, respectively.

149 Table 6 shows CO, UHC and NO_x emissions of different IBE-gasoline blends. A sudden
150 increase of CO emission occurred for IBE60. Generally, a higher CO emission level could be
151 caused by locally rich conditions, insufficient oxidizer or low combustion temperature. After adding
152 IBE, the lack of oxygen should not lead to increased CO emission. Previous studies explained that
153 alcohol-containing fuels can produce more products in terms of the heat capacity of combustion
154 products, which lowers combustion temperature and further slows down the oxidation process of
155 CO emission [41-43]. Additionally, IBE-gasoline blends had a shorter combustion duration, and
156 thus the insufficient oxidation of CO can also cause the increase in CO emission [44]. UHC
157 emission is mainly influenced by the combustion quality. The oxygen content in IBE was beneficial
158 to improve combustion quality, resulting in lower UHC emission of IBE-gasoline blends compared
159 to that with G100. The lower AFR of IBE60 led to the use of higher mass of fuel (higher injection
160 durations), which resulted in higher amounts of fuel getting into the crevice volumes or absorbed in
161 oil layers and deposits, and as a consequence there was an increase in UHC emissions compared to

those of IBE10 and IBE30 [45]. Zeldovich thermal activation is the predominant mechanism for NO_x emission formation from engines. The higher combustion temperature and local oxygen concentration in the peak temperature zone were in favor of NO_x emission formation [46]. By using IBE, a higher oxygen concentration was caused due to IBE self-provided oxygen, but it could also result in a lower combustion temperature. In terms of heat capacity of combustion products, alcohol-containing fuels can produce more products. The stoichiometric chemical reactions of gasoline, ethanol, isopropanol and n-butanol are listed below. It can be seen that ethanol, isopropanol and n-butanol produced more triatomic molecules in the combustion products than gasoline by 26%, 17% and 13%, respectively, leading to a larger heat capacity and thus a lower combustion temperature. Based on the competition between them, 0.5%, 5.5% and 14.7% lower NO_x were finally obtained using IBE10, IBE30 and IBE60, respectively, compared to that of G100.

3.2. Comparison with common alternative alcohol fuels

From the results mentioned above, IBE30 performed well in the cases of engine performance and emissions. In this section, IBE30 is compared with more commonly used alternative alcohol fuels, including E30, B30 and ABE30 (30 vol.% ethanol, n-butanol and ABE blended with gasoline). Figs. 4 and 5 show the combustion characteristics of these four fuels. It was apparent that E30 presented the most advanced combustion phasing with a 2.0%, 4.1% and 3.6% shorter ICD, a 1.5%, 2.9% and 2.4% shorter MCD, and 0.5°C, 1°C and 0.8°C advanced 50% NHR location than those of B30, ABE30 and IBE30, respectively. It should be noted that E30 had the same LFS as that of n-butanol (see Table 1), but had a the shorter ICD, which also occurred in Ref. [47, 48]. It is

known that the latent heat of vaporization and vapor pressure are two important thermodynamic properties that affect early combustion [49]. Latent heat of vaporization exerts a charge cooling effect on the pre-ignition temperature, especially for direct injection gasoline engines. The examination of the pressure traces at the crank angles closer to spark timing in Fig. 4(a) showed that there was very little difference among the four fuels, which indicates that the charge cooling effect was negligible. Actually, a PFI engine was used in this study. Fuel and air were mixed in the intake manifold and then introduced into the cylinder through a very hot intake valve, which would heat the mixture up and reduce the charge cooling effect. Apart from the latent heat of vaporization, vapor pressure is another important property affecting early combustion. It was found that vapor pressure of n-butanol was much lower than that of ethanol, which meant that n-butanol needed a higher temperature or a longer time to get completely vaporized [42].

Among the alcohol-gasoline blends, IBE30 had better results in term of BTE and BSFC as shown in Table 7. A relatively lower BTE was caused by E30 and B30 than that of ABE30 and IBE30 due to the extremely advanced combustion phasing causing more loss in the net work as discussed above. For the BSFC, E30 showed the highest value in comparison to the others due to it having the lowest LHV and BTE.

Table 8 compares the emissions of different alternative fuels. The CO emission of E30 was significantly increased by 25.9%, 47.8% and 41.7% relative to B30, ABE30 and IBE30, respectively. Yacoub et al. found that ethanol had a lower adiabatic temperature than gasoline and n-butanol, which would retard the oxidation of CO emission [1]. In addition, the shorter combustion duration of E30 can also cause the insufficient oxidation of CO emission. For the UHC emissions, ABE30 showed a 13.1%, 9.0% and 12.4% higher values compared to E30, B30 and IBE30, respectively. This could be attributed to the added acetone with a different reaction mechanism with

206 alcohols, which should be further studied. For NO_x emission, similar results were obtained with the
207 different fuels.

208 *3.3. Comparisons between IBE30 and G100 under various equivalence ratio and engine load*

209 In this section, the performance, combustion and emissions characteristics of IBE30 and G100
210 were compared under different equivalence ratios ($\Phi=0.83\sim 1$) and engine loads (300 and 500 kPa
211 BMEP). It can be seen from Fig. 6 that the ICD and MCD reduced with increasing equivalence
212 ratios and engine loads. This can be explained by the fact that a higher combustion temperature was
213 attained under higher engine load, which caused a faster combustion rate. Gauthier et al. [50] had
214 found that ICD decreased with increasing equivalence ratio. In addition, with the increase of
215 equivalence ratio and engine load, the differences in ICD and MCD between IBE30 and G100 were
216 reduced. These trends were consistent with those of Ref. [50, 51]. Overall, compared to G100,
217 IBE30 had a more advanced combustion phasing with 2.4-6.2% shorter ICD and 0.7-9.6% shorter
218 MCD due to the higher LFS of IBE components.

219 Fig. 7 shows the BTE and BSFC of IBE30 and G100 with respect to equivalence ratio and
220 engine load, in which the error bar of each data point indicates standard deviation. The BTE
221 increased with decreasing equivalence ratio and increasing engine load. With the decrease of
222 equivalence ratio, increased dilution improved isentropic efficiency by decreasing heat loss and
223 increasing the adiabatic exponent value due to the lower temperatures than those under
224 stoichiometric condition [52]. The higher combustion temperature caused by increasing engine load
225 resulted in an improved combustion quality and a higher BTE. IBE30 increased BTE by 0.04-4.3%
226 compared to that of G100 due to the improved combustion quality and shorter combustion duration,
227 and an apparent increase occurred at lean conditions. This could be explained by the fact that the
228 combustion phasing of fuels was retarded at lean conditions, but the engine was still running at

gasoline's MBT at stoichiometric condition, and thus the advanced combustion phasing of IBE30 was more suited, thus negating the impacts of improper combustion phasing on BTE (that were seen at stoichiometric conditions). Due to the improved BTE, the BSFC of fuels decreased with decreasing equivalence ratio and increasing engine load. Based on the lower LHV, IBE30 showed a 4.9-9.3% higher BSFC than that of G100.

Fig. 8 shows the variations of CO, UHC and NO_x emissions with equivalence ratio and engine load for IBE30 and G100. It was observed that the equivalence ratio controlled CO emission until lean conditions were reached after which CO emission did not vary significantly. These low CO emission under lean conditions can be explained by the fact that there was more than enough oxygen available to carry on the oxidation process [53]. When compared to G100, IBE30 produced 15.1-20.3% lower UHC emission due to the improved combustion quality as a result of fuel-borne oxygen. It was observed from Fig. 8(c) that the highest NO_x emission was seen at $\Phi=0.9-1.0$ with a decrease as the equivalence ratio got relatively leaner because a peak combustion temperature was attained under $\Phi=0.9-1.0$ [54]. In comparison to G100, although a higher oxygen concentration was provided, a decreased combustion temperature was also caused so that the final 3.3-18.6% lower NO_x emission was produced for IBE30.

4. Conclusions

This experimental study revealed the potential of IBE-gasoline blends as a green fuel for SI engines. The performance, combustion and emissions characteristics between IBE-gasoline blends and conventional fuels were compared. Some conclusions are shown as follows.

By the comparison between IBE-gasoline blends with various IBE content under stoichiometric condition, it was found that the combustion phasing was advanced with increasing IBE content. The negative effect of improper combustion phasing was offset by improved

252 combustion quality as a result of fuel-borne oxygen and reduced combustion duration. As a result,
253 IBE30 showed a similar BTE relative to that of G100 and 1.0% and 1.4% higher BTE than that of
254 IBE10 and IBE60, respectively. For the emissions, IBE30 provided better results based on its lower
255 CO (4.2%), UHC (18.9%) and NO_x (5.5%) emissions than those of G100.

256 For the different IBE-gasoline blends mentioned above, IBE30 performed well with respect to
257 engine performance and emissions, and thus it was selected to be compared with some commonly
258 used alternative alcohol fuels, including E30, B30 and ABE30. E30 and B30 had a more advanced
259 combustion phasing, which also caused the relatively lower BTE than that of ABE30 and IBE30.
260 The significant increase of CO and UHC emissions occurred for E30 and ABE30, respectively. In
261 addition, similar NO_x emission was produced by the different fuels.

262 IBE30 was further compared with G100 under various equivalence ratios and engine loads.
263 IBE30 had a generally advanced combustion phasing, and the difference in combustion phasing
264 between IBE30 and G100 decreased with increasing equivalence ratio and engine load. For engine
265 performance and emissions, compared to G100, a higher BTE (0.04-4.3%) and lower CO (4%),
266 UHC (15.1-20.3%) and NO_x (3.3-18.6%) emissions were observed for IBE30 in some cases.

267 IBE could be a good alternative to gasoline based on the fact that it can be produced using an
268 environmentally benign fermentation process (from non-edible biomass feedstock and without
269 recovery process) and has the potential for improving energy efficiency and reducing pollutant
270 emissions.

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Figures

Fig. 1. Engine setup

Fig. 2. Pressure and NHR for G100, IBE10, IBE30 and IBE60 at $\Phi=1$ and 300 kPa BMEP

Fig. 3. Combustion phasing for G100, IBE10, IBE30 and IBE60 at $\Phi=1$ and 300 kPa BMEP

Fig. 4. Pressure and NHR for E30, B30, ABE30 and IBE30 at $\Phi=1$ and 300 kPa BMEP

Fig. 5. Combustion phasing for E30, B30, ABE30 and IBE30 at $\Phi=1$ and 300 kPa BMEP

Fig. 6. Combustion phasing for IBE30 and G100 under various equivalence ratio and engine load

Fig. 7. Engine performance for IBE30 and G100 under various equivalence ratio and engine load

Fig. 8. Engine emissions for IBE30 and G100 under various equivalence ratio and engine load

Tables:

Table 1 Properties of the test fuels

Parameters	Individual fuels					Fuel blends	
	Gasoline ^a	Acetone ^b	Ethanol ^c	Iso-propanol ^d	n-butanol ^e	IBE(361)	ABE(361)
Chemical formular	C ₄ -C ₁₂	C ₃ H ₆ O	C ₂ H ₅ OH	C ₃ H ₇ OH	C ₄ H ₉ OH		
Octane number	92	-	100	112	87		
Oxygen content (wt.%)	-	27.6	34.8	26.6	21.6	24.4	24.7
Density (kg/m ³)	736	791	795	786	813	803.1	804.6
Lower heating value (MJ/kg)	43.4	29.6	26.8	30.4	33.1	31.7	31.4
Boiling Temperature (°C)	38-204	56	78	84	118		
Latent heat (at boiling point) (kJ/kg)	349	518	904	758	582	667	595
Stoichimometric AFR	14.7	9.5	9.0	10.4	11.2	10.7	10.5
Auto-ignition temperature (°C)	257	465	420	399	343		
Laminar flame speed (cm/s)	33-44 ^f	34 ^g	48 ^h	45 ^h	48 ^h		37 ^g

Note: ^a Properties of gasoline are from [28, 34]; ^b Properties of acetone are from [30]; ^c Properties of ethanol are from [28, 29, 32]; ^d Properties of iso-propanol are from [32, 33]; ^e Properties of n-butanol are from [8, 32]; ^f p=1 atm, T=298-358K; ^g p=1 atm, T = 298 K; ^h p=1 atm, T=343 K.

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

Table 3 Test conditions

Throttle position (%)	100
Engine speed (rpm)	1200
Load (kPa BMEP)	300 and 500
Equivalence ratio	0.83-1
Fuel pressure (kPa)	300
Spark timing	Gasoline's MBT at $\Phi=1$

Table 4 Measuring range, accuracy and resolution of the experimental apparatus

Apparatus	Measuring range	Accuracy ($\pm$)	Resolution
Engine speed	1-5000 rpm	0.2%	1 rpm
Torque	0-300 Nm	0.5%	0.1 Nm
Exhaust gas temperature	0-900°C	1 °C	0.1 °C
CO emission	0-10% Vol.	0.06%	0.01% Vol.
HC emission	0-10000 ppm Vol.	12 ppm Vol.	1 ppm Vol.
CO ₂ emission	0-20% Vol.	0.5%	0.01% Vol.
NO _x emission	0-3000 ppm	3%	1 ppm
Lambda	0.65-13.7	0.3 %	0.01
Mass flow meter	0-1000 L/min	1%	0.1 L/min

Table 5 Engine performance for G100, IBE10, IBE30 and IBE60 at $\Phi=1$ and 300 kPa BMEP

Fuel	Engine Performance	
	BTE (%)	BSFC (g/kWh)
G100	21.6	384.7
IBE10	21.4	396.5
IBE30	21.6	418.8
IBE60	21.3	464.1

Table 6 Engine emissions for G100, IBE10, IBE30 and IBE60 at $\Phi=1$ and 300 kPa BMEP

Fuel	Engine emissions		
	CO (% Vol)	UHC (ppm Vol)	NO _x (ppm)
G100	0.25	346	1530
IBE10	0.26	329	1522
IBE30	0.24	291	1450
IBE60	0.40	330	1341

Table 7 Engine performance for E30, B30, ABE30 and IBE30 at $\Phi=1$ and 300 kPa BMEP

Fuel	Engine performance	
	BTE (%)	BSFC (g/kWh)
E30	21.3	443.7
B30	21.2	424.3
ABE30	21.4	424.7
IBE30	21.6	418.8

Table 8 Engine emissions for E30, B30, ABE30 and IBE30 at $\Phi=1$ and 300 kPa BMEP

Fuel	Engine emissions		
	CO (% Vol)	UHC (ppm Vol)	NO _x (ppm)
E30	0.34	289	1455
B30	0.27	300	1428
ABE30	0.23	327	1445
IBE30	0.24	291	1450