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ORIGINAL ARTICLE

Effects of Lean Combustion on Bioethanol-Gasoline Blends using Turbocharged Spark Ignition Engine

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ABSTRACT – Liquid alternative fuels have been utilised as engine fuel since the 19th century. For several alternative fuels, bioethanol is well-known as the most suited friendly, alternative-product based and renewable for use in spark-ignition (SI) engines. In addition, it is well known that bioethanol has higher evaporation of heat, research octane number and flammability of temperature; therefore, it has a greater influence on performance and lower emission. In this study, the effect of gasoline fuel RON95 (G) was blended into bioethanol fuel (E10, E20 and E30) to investigate the engine combustion, performance and emission. The engine used was 1.8L Mitsubishi, four-cylinder, four-stroke, multipoint port injection and turbocharger SI. The engine speed used was 1000-3000 rpm at 10-40% load with wide-open throttle (WOT). The results showed that bioethanol addition to gasoline increases the brake torque at a higher load. The mass fraction burn (MFB) and coefficient of variation (COV) blend fuel and main fuel are comparable to each other. The brake specific fuel consumption (BSFC) significantly increases when engine speed increases. The emission of nitrogen oxide (NOx), carbon monoxide (CO), and hydrocarbon (HC) emissions reduced dramatically compared to gasoline fuel. Indeed, bioethanol-gasoline fuel allows the engine utilised in low proportion to increase engine performance and lower engine emission.

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Performance;
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NOMENCLATURE

RON	research octane number	CO_2	carbon dioxide
PI	port injection	CO	carbon monoxide
BMEP	brake mean effective pressure	NOx	nitrogen oxide
VE	volumetric efficiency	HC	hydrocarbon
BSFC	brake specific fuel consumption	RPM	radian per minute
SOHC	single overhead cam	ECU	engine control unit
MPFI	multiport fuel injection	GHG	greenhouse gas
PPM	particle per million	SI	spark ignition
CV	calorific value	Bp	brake power
ICE	internal combustion engine	BT	brake torque
H_2O	water		

INTRODUCTION

Today, growing emission is one of the significant problems in developed countries. The emissions from the vehicles have the main role in inducing greenhouse gas (GHG) and pollution. The design of the vehicle to follow legal regulations is not sufficient; thus, it necessary to use alternative fuel as modern energy. It is important that alternative fuel is produced from renewable sources and use directly without requiring any major engine modification. Alcohols fuel have been used for the engine since the 19th century to run the engine. Bioethanol is one of among alternative fuels that are well known suitable for the spark ignition (SI) engine. The properties of bioethanol most attractive for use in SI engines are, it can be produced from renewable energy sources such as barley, corn, sugar, cane or waste biomass [1]–[10]. Iodice and Senatore [11] described that in their study on National Interest Priority Sites (NIPS) with two categories (road traffic and industry power plans). From the anthropogenic activities, the carbon monoxide, nitrogen dioxide, particulate matter and volatile organic compounds are 83%, 28%, 39% and 41%, respectively.

The range of operating parameters for spark-ignition engines is limited by cyclic fluctuation, particularly in lean and highly diluted circumstances. Pressure parameters, combustion-related parameters, and flame-front related parameters can all be used to characterise cyclic combustion fluctuations. The cyclic variability in indicated work per cycle is defined by the coefficient of variation (COV) in Indicated Mean Effective Pressure (IMEP). Zhao et al. [12] described that the

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engine cylinder pressure and heat release increase when the engine run under lean combustion. It was shown that higher combustion efficiency and acceptable COV could be obtained when excess ratios are 1.2 and 1.3. While Mass Fraction Burned (MFB) chart illustrate how in-cylinder combustion progresses as a function of crank angle. The rate of combustion within a cylinder has a significant impact on the engine's thermal efficiency, peak cycle temperature and pressure, and exhaust emissions. The burn crank angles at which the MFB achieves a given value are commonly used to calculate this combustion rate. On the basis of cumulative heat release, the mass fraction burned functions are derived in Eq. (1). CA05, CA50, and CA95 are the locations of characteristic points characterising combustion advance, i.e. 5%, 50%, and 95% MFB, as read from MFB curves. Only the MFB50 was used in this study. According to MFB graphs by Ilhak et al. [13], the combustion times of gasoline and ethanol at 25% load are nearly identical; however, ethanol's combustion time at 50% load is greater than gasoline. Furthermore, ethanol has a CA50 value that is higher than gasoline at 50% load. In this situation, ethanol combustion, conversion to heat, and pressure effects take longer, resulting in higher ethanol variation values from cycle to cycle.

$$MFB = \frac{\sum_0^i \Delta Pc}{\sum_0^N \Delta Pc} \tag{1}$$

where, 0 is the start of combustion and N is the end of combustion. The mass fraction consumed after the considered i-th interval can be determined by assuming that the pressure rise ΔPc is proportional to the heat contributed to the in-cylinder medium during the crank angle period.

Currently, bioethanol is still used in light SI engines with gasoline fuel at low proportions. It allows the engine use of bioethanol without modification. This method was recommended and will be improved on performance, cold start and anti-knock. In addition, another advantage of the bioethanol flame is colourless in the natural burning process [2], [14]. Since the lower energy content of bioethanol is lower than gasoline, it needed to increase the mass of fuel during spray to achieve the same output power [15]. Bioethanol contains an oxygen atom, and it is miscible with H_2O in all mixtures. Then water contains in bioethanol contribute to the corrosion problem on the mechanical part. In order to avoid this problem on the fuel delivery system, some materials, especially copper, brass and aluminium, are not used. Therefore, among the material that can use is fluorocarbon rubber or stainless steel.

Many researchers have investigated the effects of bioethanol-gasoline fuel blends on engine performance and emission. Feng et al. [16] and Gravalos [17] mentioned that the difference of performance and emission alcohol fuel to reference fuel is due to the higher fuel vaporisation and combustible charge formation at high engine speeds and higher oxygen content. Deng et al. [18] used hydrous ethanol to blend gasoline in 10% and 20% proportions. They ran the engine at 1500-5000 rpm and used full load conditions. The blend of 10% hydrous ethanol-gasoline (E10W) exhibit enhanced performance, while NOx, CO and HC significantly decrease. Yusoff et al. [19] investigated the effect of ethanol-isobutanol-gasoline blends fuels on engine performance and emission at 1000-5000 rpm. The results showed that the fuel blends have slightly higher torque compared to gasoline, where E10, E20, iB10, iB20, E10iB10 recorded at 2.33, 3.88, 2.56, 3.06, 1.06 and 3.77%, respectively. Meanwhile, higher blends, i.e. E20, iB20, and E10iB10, give higher brake power compared to gasoline, with an average increase of 2.6, 2.57 and 2.22%, respectively. In terms of emissions, they concluded the E20 and E10iB10 have the lowest CO and HC, respectively. However, all the tested fuel blends showed higher CO_2 at an average of 10.67% compared to gasoline. The NOx emission rises dramatically when the engine uses higher speed and torque but blends fuel shows lower than gasoline. Basically, NOx formation is through three pathways, (1) fuel-nitrogen conversion, (2) Zeldovich thermal activation, and (3) fuel-rich prompt formation [20], [21]. The two-wheel vehicle shows the effect of fuel consumption and emission using three ways catalyst and belongs to Euro 3 legislation exhaust emission. As a result, the statistical obtained the medium size motorcycle under real driving behaviour [22].

He et al. [1] concluded that the fuel comprising 30% of ethanol by volume induces lower HC, CO, and NOx emission at idle, but unburned ethanol introduces increased emission. The equivalent ratio and mass of ethanol play a significant role in the combustion, as mentioned by Wu et al. [23]. The effect of using gasoline-ethanol blends with a different percentage on performance and emission SI engine was investigated by Al-Hasan [2]. They mentioned that the blends of ethanol to gasoline increase the volumetric efficiency, brake power, brake thermal and fuel consumption, while it decreases the brake specific fuel consumption and equivalence air-fuel ratio. The concentration of CO and HC emission decreased, whereas the CO_2 increased dramatically. Yucesu et al. [24] used gasoline (E0) as base fuel and gasoline-ethanol blends (E10, E20, E40 and E60) into engine one cylinder, four-stroke, spark ignition and adjustable compression ratio. As a result, blends fuel slightly increased the brake torque with increasing compression ratio without detonation and reduced HC and CO emission. The higher surface to volume ratio and higher compression ratio is due to higher HC formation [25] [26].

From literature, bioethanol-gasoline can be used effectively as blends fuel in a SI engine. Bioethanol has a higher research octane number (RON) and the high evaporation of heat and contains over 30% oxygen by weight. If the amount of bioethanol increased in blends, the heating value of fuel blends decreases. Thus, the lower heating value allows for increasing the fuel mass during the intake suction process. Among the influence on engine performance, the timing of ignition, air-fuel ratio, heat of vaporisation, compression ratio and flame speed are considered. This study evaluated engine efficiency and emission on lean conditions with the presence of bioethanol in gasoline as a low proportion. The test was conducted in low load and low engine speed as used in the urban area. In this study, the effects of bioethanol-gasoline blends on engine performance and emission was investigated using a turbocharged light SI engine. The engine speed was used at 1000-3000 rpm, and fuel blends were set at 10%, 20% and 30% (E10, E20 and E30) with gasoline (G) as reference. The engine load was applied at 10-40% wide open throttle (WOT).

EXPERIMENTAL APPARATUS AND PROCEDURE

Experimental Apparatus

The set-up of experimental consists of a Mitsubishi turbocharged spark ignition (SI) engine, four-stroke, four-cylinder, and multipoint port injection. It was coupled with a 100 kW eddy current dynamometer to measure engine performance. A strain gauge based load cell attached to the dynamometer was used for load measurement. The schematic view of the experimental setup is shown in Figure 1. The general specifications of the test engine are given in Table 1. A fuel flow meter (AIC-1204 HR 2000 model) was used to measure air delivery into the engine cylinder. A gas analyser (Kane Auto-plus 5-2) was used to analyse the engine emission constituents, for example, NOx, CO, and HC and air-fuel proportions. All the devices were calibrated and tested regularly. The accuracies of the devices and uncertainties in the calculated results are shown in Table 2.

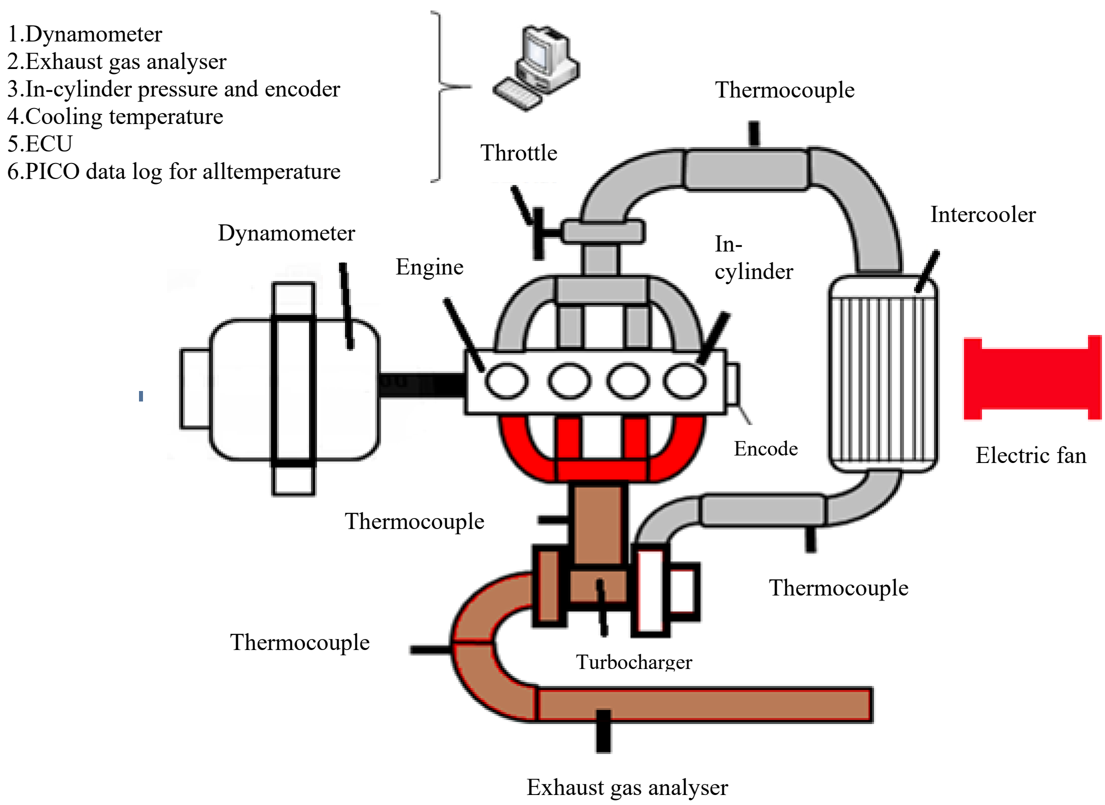


Figure 1. Schematic diagram engine test rig.

Table 1. Specifications of SOHC 16 V MPI Mitsubishi engine.

Item	Specification	Item	Specification
Number of cylinders	4 straight line	Maximum torque	220@ 3000 (Nm)
Displacement	1.8(L)	Intake valve open	261 degree BTDC
Chambers combustion type	Pentroof Type	Intake valve close	209 degree ATDC
Cylinder bore	81(mm)	Intake valve lift	9.7mm
Piston stroke	89(mm)	Exhaust valve open	257 degree BTDC
Compression ratio	9.5:1	Exhaust valve close	204 degree ATDC
Maximum output	118@ 6000 (kW)	Exhaust Valve lift	9.5mm

Table 2. The accuracies of the devices and uncertainties in the calculated result by manufacturer.

Measurement	Range	Accuracy
In-cylinder pressure	40-80 Bar	±0.4%
Fuel consumption	1.2–130 kg/h	±0.1%
Temperatures	80-90 ºC	±1 ºC
Air delivery	0.5–100 slpm	±0.8%

Test Cases Examined

All tests were conducted at three different engine speed (1000, 2000 and 3000 rpm) at 10-40% wide open throttle after engine warm-up. At each engine speed, three different fuels (E10, E20 and E30) were compared to gasoline. The properties of the fuels are shown in Table 3, where bioethanol has a purity of 99.5%. The test was carried out for brake torque, engine speed, fuel consumption, air delivery into the cylinder, spark timing, and emission (CO, HC and NOx). For each experiment, three times was replication to average data. The condition of the engine was allowed to reach a stable then measurements were verified. Then, for each operating condition, the access of air-fuel ratio was maintained during all experiments. Table 4 shows the specification of the exhaust gas analyser using for this study.

Table 3. Properties of gasoline and bioethanol [27] [28][29].

Properties	Test Standard	Bioethanol	Gasoline	E10	E20	E30
Stoichiometry (weight)	Calculated	9	14.7	13.5	12.6	11.7
Octane Number	ASTM D2699	106.6	95	98.1	100.7	102.4
LHV (MJ/kg)	ASTM D240	26.9	44	41	39.4	37.8
Density (kg/m^3)	ASTM D4052	785[30]	737	761	765	768

Table 4. Specification of exhaust gas analyser.

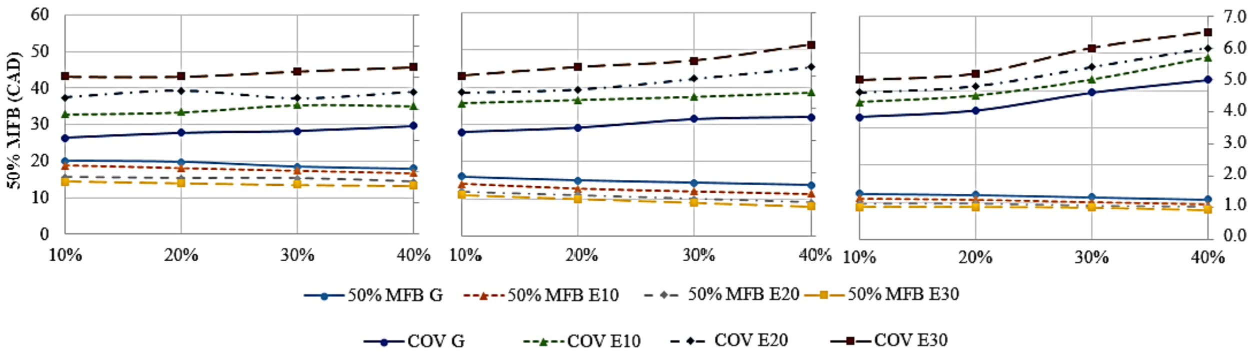
Description	Specification
Model	Autoplus 5-2
Measurement principle	Flow rate (Minimum 2 l/min, normal 2.5 l/min)
Power supply	6 volt DC
Flue temp	0-1200 °C with suitable probe
Inlet temp	0-50 °C
Oxygen (O_2)	Range 0-21% @ ±5% accuracy
Carbon monoxide (CO)	Range 0-10% @ ±5% accuracy
Carbon dioxide (CO_2)	Range 0-16% @ ±5% accuracy
Nitrogen oxides (NOx)	Range 0-5000ppm @ ±5% accuracy

RESULTS AND DISCUSSION

Mass Fraction Burn (MFB) and Coefficient of Variation IMEP (COVimep)

Figure 2 shows the location mass fraction burn (MFB) or maximum heat release rate (MHRR) for G, E10, E20 and E30 using engine speed at 1000-3000 rpm. MFB advance when fuel blends increase, engine speed and engine load increase. The advancing of MFB due to fuel blends show faster flame speed during lean combustion. Elik et al. [31] described that MFB is a process of chemical energy released as a function of crank angle. Iodice et al.[32] pointed that faster flame speed by blends ethanol induces the increase of brake power and increased volumetric efficiency.

While the COV imep shows stability in engine combustion, the graph of COV imep shows an increasing trend when the engine uses gasoline-bioethanol blends fuel compared to gasoline. It means that the engine is slightly rough when using ethanol-gasoline blends. The range of COVimep in this study is 3-7%. Then the E30 shows higher COV compared to other fuels. The COV increase due to occurring lean combustion of blended fuels and occurring misfire in the engine cylinder. Park et al. [33] described that an increased air access ratio (lean) causes an increase in the cycle of variation and results in the deterioration of thermal efficiency. Heywood [34] mentioned that when the engine is below 10% of COV, the engine run smooth and stable. Wang et al. [35] also pointed out that the engine is stable if the COV range is below 10%.



(a) (b) (c)

Figure 2. The location MFB 50% and COV as a function using engine load at 10-40%.

Brake Torque

Figure 3 shows the brake torque as a function engine load at 10-40%. The brake torque for engine speed at 1000 rpm was recorded around 52-62 Nm, and Gasoline (G) is higher than all fuel. At 2000 rpm was illustrated that the brake torque for gasoline and bioethanol-gasoline blends is comparable to each other at around 45-82 Nm and close with each other. The brake torque at engine speed 3000 rpm early start with 40 Nm. The trend was showed that bioethanol-gasoline blends increase compared to gasoline. The brake torque for blends bioethanol is higher than gasoline and was recorded at 105 Nm. Below are several reasons that can be related to brake torque bioethanol, which is higher than gasoline. Although bioethanol has lower heating value, the bioethanol has oxygen in the atom allows more to complete combustion, thereby increasing the torque and power. In addition, a larger spray fuel for some volume was injected into the cylinder due to the higher density of bioethanol [19]. The hydroxyl radical occurring due to complete combustion induces advanced flame speed [36], and subsequent heat release occurs early. The average raising in brake torque for bioethanol fuel blends compared to gasoline was about 1%, 1.8% and 2.3% for E10, E20 and E30, respectively.

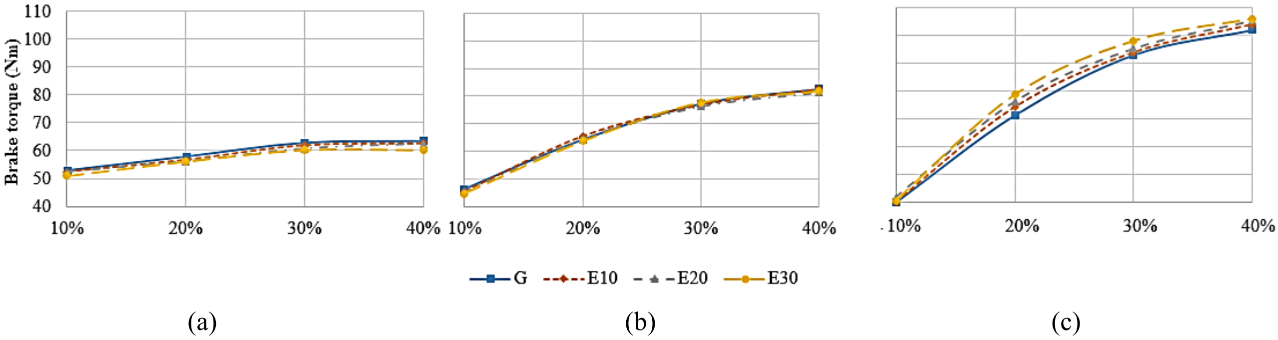


Figure 3. Brake torque as a function engine load (a) 1000 rpm, (b) 2000 rpm and (c) 3000 rpm.

Brake Specific Fuel Consumption

Figure 4 illustrates the brake specific fuel consumption (BSFC) of the engine using G, E10, E20 and E30 fuels with respect to engine loads at 10-40% and engine speed 1000-3000 rpm. As shown in graph BSFC for 1000 rpm, low load shows higher BSFC than higher load due to higher heat loss. It was occurred due to efficiency air suction at low load better than higher load. In addition, the BSFC is slight far from each other at the low load. It's recorded at a range of 560-750 g/kW.h. At higher load, the BSFC reduced due to choking at the throttle. At 2000 rpm, the BSFC shows an increased average of 20% for lower load compared to 1000 rpm. The range BSFC of 2000 rpm is 290-620 g/kW.h. At a higher load, it was reduced at 8% compared to 1000 rpm. BSFC at 3000 rpm recorded at 480-380 g/kW.h at low load. When higher load, it showed the lowest decrease at 350-390 g/kW.h. All BSFC shows that increasing trend when increased engine speed and engine load.

The lower heating value of bioethanol-gasoline fuel causes raising BSFC of the engine when it is used without any alter of the engine. The increment significantly depends on the percentage of bioethanol. Najafi et al. [30] mentioned that the BSFC of bioethanol fuel is higher than that of gasoline owing to its lower heating value, higher viscosity and density. The viscosity obstructs the vaporisation and atomisation and deteriorates the combustion, leading to an increased BSFC [37]. Hasan et al. [38] mentioned that using biofuel blend with gasoline increase BSFC due to higher density, then it contributes to increasing brake torque. The decrease of BSFC at higher load and higher engine speed is due to pumping losses. It contributes to occurring slow motion of air intake in the cylinder compared to lower load and lower engine speed due to some of the air sent back to the intake system [39].

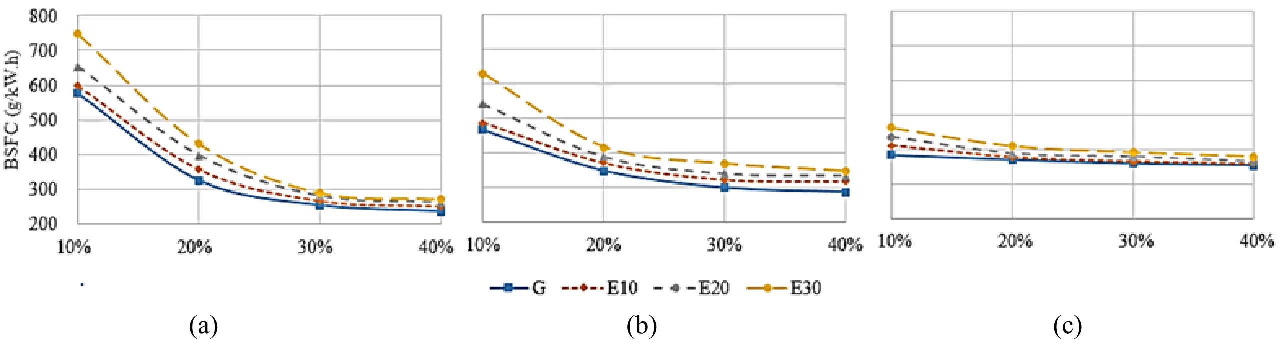
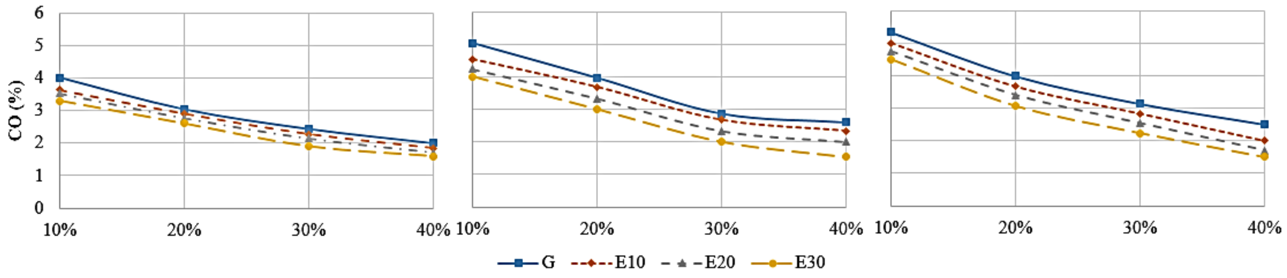


Figure 4. Brake specific fuel consumption as a function engine load at (a) 1000 rpm, (b) 2000 rpm and (c) 3000 rpm.

Carbon monoxide

Figure 5 illustrates the influence of bioethanol-gasoline and gasoline on carbon monoxide (CO) emissions as a function engine load at 10-40%. The CO emission is a product of incomplete combustion due to an insufficient amount of air intake mixture in the combustion. The formation of CO emissions significantly depends on the engine operation and the air-fuel mixture. At 1000 rpm, the CO emission reduces due to complete combustion when the load increases. Thus, CO emission averagely increase 20% at 2000 rpm. At 3000 rpm, the CO emission averagely increased by 6% compared to 2000 rpm. Gasoline fuel showed that CO emission was higher than compared to all bioethanol-gasoline fuel blends. But all CO emission shows a decreasing trend when increases engine load. The reduction of CO emission due to the lean mixture effect is defined as enhancing the oxygen in the fuel mixture, which improves combustion efficiency in the engine cylinder [40][41]. In addition, it decreases with decreasing the temperature of combustion [34][25]. Besides that, bioethanol has a higher flame speed compared to gasoline fuel will assist in completing combustion, which induces lower CO emission [42][25]. The results were also concordant with findings of the previous researcher, which utilised bioethanol and gasoline blends [43][44].

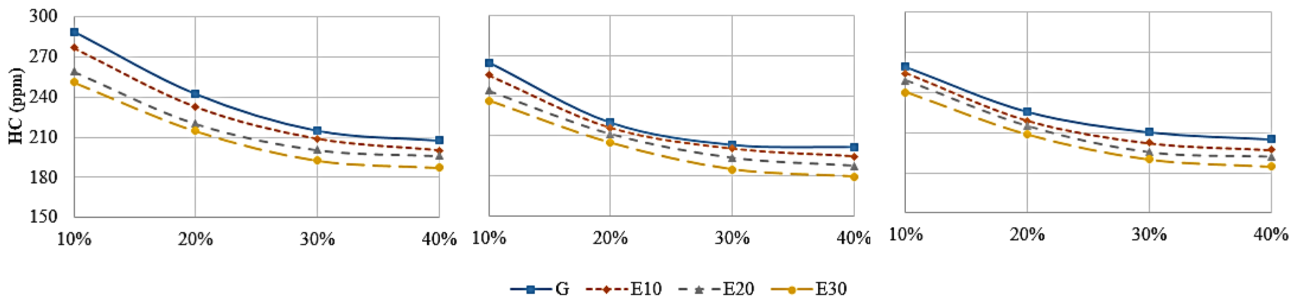


(a) (b) (c)

Figure 5. Carbon monoxide as a function engine load at (a) 1000 rpm, (b) 2000 rpm and (c) 3000 rpm.

Hydrocarbon

Figure 6 illustrates the hydrocarbon (HC) emissions as a function to 10-40% using engine load at 1000-3000 rpm. The HC emission significantly reduces when the engine uses bioethanol-gasoline fuel blends compared to gasoline and closely with each other. At 1000 rpm, it reveals the range of HC emission is 185-285 (ppm). At 2000 rpm, it was decreased averagely by 7% compared to 1000 rpm. HC emission for 3000 rpm reduces averagely at 1.7% compared to 2000 rpm. The heat of air suction from turbocharger boost induces complete combustion, thus the oxygen enrichment and leaning effect. It is determined that raising the bioethanol in gasoline improves the turbulence intensity during combustions. In addition, bioethanol blends provide extra complete combustion and induce a reduction in HC formation. Besides, the formation of HC emission is closely related to engine design and variable engine operation. Then, the lowest cylinder combustion temperature resulted in incomplete combustion of the residual fuel [25]. Results are also concordant with Najafi et al. [30]. They mentioned that oxygen content in bioethanol is shown to pre-oxidise the air-fuel mixture leading to reduce HC emissions.

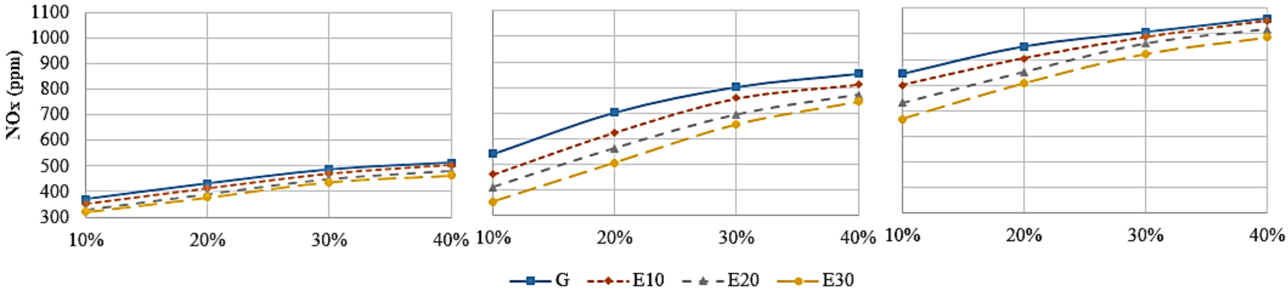


(a) (b) (c)

Figure 6. Hydrocarbon as a function engine load (a) 1000 rpm, (b) 2000 rpm and (c) 3000 rpm.

Nitrogen oxide

The formation of nitrogen oxide (NOx) emission significantly depends on the peak temperatures during combustion. It formed above the temperature of 1500 ºC. Figure 7 shows the NOx emission as a function of 10-40% engine load. NOx emission for 1000 rpm was recorded at 325-510 ppm. At 2000 rpm, it increased to 350-850 ppm. NOx emission at 3000 rpm was recorded at 650-1050 ppm range. When higher load, the graph close for each other due to complete combustion. All the NOx emission graphs showed an increasing trend when the engine load increased. NOx for gasoline was showed higher than all bioethanol-gasoline blends. The temperature of combustion is higher, and local oxygen concentration in the peak temperature zone was favourable to NOx emission formation in gasoline fuel. But, NOx emission reduces as a result of adding bioethanol to gasoline fuel. It was concordant with Li et al. [21] and concluded that use alcohol in their study contributes cooling effect in the cylinder combustion. In addition, the high latent heat of vaporisation of bioethanol lowers the flame temperature and induces NOX lower emission. So, NOx emission may change depending on the percentage of bioethanol blends in gasoline. Oxygen concentration and combustion temperature, and time are the main parameters affecting the NOx emission [45].



(a) (b) (c)

Figure 7. Nitrogen dioxide as a function engine load (a) 1000 rpm, (b) 2000 rpm and (c) 3000 rpm.

CONCLUSION

In this study, the effects of gasoline (G) and bioethanol-gasoline blends (E10, E20 and E30) on engine performance and emission were investigated in a turbocharged SI engine, four-stroke, four-cylinder at WOT.

i. It was demonstrated that bioethanol-gasoline fuel mixes provide comparable engine performance and reduced emissions without detonating the engine.
ii. While COV showed an increasing trend, that means slight vibration occurred during combustion when using ethanol blend compared to gasoline. The lean combustion also promotes MFB advance for 50% duration compared to gasoline. It shows that when fuel blends increase, a slightly advanced combustion occur but lower performance due to lower heating value.
iii. The research presented here has shown that two-component blends of gasoline and ethanol may be produced and used in existing gasoline automobiles. Such gasoline-ethanol mixes have constant volumetric energy content, constant octane values, and near-constant vaporisation enthalpies when blended with constant stoichiometry. The energy content in bioethanol-gasoline fuel is lower than gasoline caused the BSFC of the engine to increase and depends on the percentage of blends.
iv. An effect of lean combustion and complete combustion was showed a reduction in HC and CO emission due to oxygenated fuel by the bioethanol. Because to lower pumping and heat loss, as well as more thorough combustion, stratified lean burn characteristics enhanced engine fuel conversion efficiency.
v. The load, combustion efficiency, and net indicated efficiency of gasoline-ethanol were all similar. Because of the greater hydrocarbon ratio, total CO and HC emissions from ethanol blends were lower than gasoline. The peak bulk temperatures produced by the combustion of ethanol blends were lower than those produced by gasoline, resulting in a lower NOx emission index, which also accounted for the higher ethanol flow rates into the engine.

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