Investigations on the noise and vibration characteristics of SI engine powered with single and dual alcohol blends - performance, combustion, and emission studies

Bhavin K Bharath

NIT-Trichy: National Institute of Technology Tiruchirappalli

Arul Mozhi Selvan V (✉ arulmozhi@nitt.edu)

National Institute of Technology Tiruchirappalli <https://orcid.org/0000-0002-8900-3894>

Research Article

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Abstract

Significant growth in renewable fuel output is required to effectively remove fossil fuel dependence in the transportation sector. Alcohol is regarded as an excellent renewable gasoline substitute. This study investigates the noise and vibration characteristics of a spark-ignition (SI) engine in relation to engine performance, combustion, and exhaust emissions evaluation using methanol-based single and dual alcohol (Ternary) blends. The tests were carried out on a single-cylinder, 4-stroke SI engine utilizing an eddy current dynamometer to investigate the behaviors of single and dual alcohol blended fuels. This study compared the blended alcohol fuels (M10, M15, M20, iBM10, and iBM15) with baseline gasoline at various engine loads. Because of the natural oxygen levels of alcohols, alcohol blends had higher BTE, peak in-cylinder pressure, and heat release rate (HRR). At 2500 rpm and 75% load, the iBM15 had a higher BTE of 33.58 percent. The blended fuels significantly reduced HC and CO emissions compared to conventional gasoline. The noise level increases as the load increases, and blends had a slightly higher but comparable noise level to gasoline. Compared to gasoline, alcohol blends vibrate less, with iBM15 exhibiting a 25.2% and 51.12% drop in vibration level in the Z and Y direction, respectively. It's worth noting that the higher-order alcohol isobutanol can be used to improve the quality of methanol-gasoline blends and as a partial replacement for fossil fuel in the SI engine.

1. Introduction

In practically every part of the world, internal combustion (IC) engines are the most popular prime movers for passenger and freight transportation. Even while passenger transportation is likely to be provided by a combination of propulsion systems, including partial electrification and battery electric vehicles (BEV), it will take at least 20 years to transition the transportation sector to EVs [1]. However, EVs have several serious drawbacks, including driving range, storage batteries, shortage of recharging stations, overall vehicle weight, and expensive costs [2]. The transportation sector using IC engines, which accounts for 23% of worldwide carbon dioxide (CO₂) emissions, is a critical area where greenhouse gas (GHG) emissions must be reduced [3]. The need for transportation energy will surely climb as the world's population and GDP increase, underlining the importance of decarbonizing this sector. The contribution of freight transportation to CO₂ emission is predicted to rise and eventually surpass that of private transportation [2]. Nowadays the petroleum supplies are quickly diminishing globally and may not be sustained beyond 2050. The growing global demand for fossil fuels and the harmful pollutants emitted by petroleum-fueled IC engines. Alcohol fuels have various advantages over other alternative fuels, including compatibility with IC engines and the capacity to minimize greenhouse gas emissions [4, 5]. Methanol (CH₃OH) is a naturally occurring alcohol that exposes humans and other living things to very low doses over their lifetimes, mainly through dietary exposure. From 1988 until 2000, M85 (85% methanol and 15% gasoline) was commercially available in California, with no recorded adverse health effects [6]. Similarly, no known detrimental health consequences have been observed for the 5% methanol blends that were widely marketed in the eastern United States from 1981 to 1986. The benefits of reformulating gasoline with oxygenates such as methanol are immediate, and they apply to all classes

and ages of cars that use the cleaner-burning gasoline and which is often results in rapid reductions in air pollution without the need to wait for new emission reduction technologies to penetrate the fleet population. Alcohol improves fuel distillation temperatures, and it helps in the addition of oxygen and octane to the gasoline fuel, which further reduces numerous ozone precursor emissions from cars, such as carbon monoxide (CO), hydrocarbons (HC), and nitrogen oxides (NO_x) [6]. Renewable fuels are important for decarbonizing the transportation sector with IC engines. Alcohols, such as ethanol (C₂H₅OH) and methanol, can be used as a drop-in fuel in gasoline engines when blended. Still, they cannot be used as a drop-in fuel in compression ignition engines because of their high auto-ignition resistance [7, 8]. Methanol (CH₃OH), ethanol (C₂H₅OH), and butanol (C₄H₉OH) are currently considered clean fuels and have undergone substantial research, and these are the primary alcohol fuels used in the IC engines nowadays [4, 9, 10]. Adding isobutanol, higher carbon alcohol, to a methanol-gasoline blend improves the blend's properties. Compared to methanol-gasoline blends, isobutanol-methanol-gasoline minimizes the likelihood of phase separation [1, 4]. The following are some of the social advantages of alcohol alternative-fuel types according to Goktas et al. [10]: 1) Alcohol production can be an excellent source of revenue for nations with a diverse agricultural landscape, and 2) countries without crude oil deposits can meet their energy requirements without relying on oil. Alcohols have a high octane rating, making the fuel more resistant to knocking [5]. Compression ratios in engines can be increased to improve engine performance in this case.

Tian et al. [9] discovered that mixing alcohol fuels with gasoline boosts the engine's brake torque (BT) and heat release rate (HRR). By mixing alcohol fuels with gasoline and adjusting the ignition duration and flame kernel radius, engine torque can be increased while fuel consumption and emissions are reduced. Wu et al. [11] studied methanol and gasoline combustion characteristics in a SI engine at idle conditions. The results showed that the methanol-fueled engine's indicated thermal efficiency (ITE) was higher than that of the gasoline-fueled engine. Methanol fuel increased the cylinder's combustion rate and made it more stable. With a peak value of 24.7% at $\phi = 1.4$, methanol fuel permitted a thermal efficiency boost under lean-burn circumstances. It reduced HC, CO, and NO_x emissions also. Liu et al. [12] investigated the performance of a SI engine using a methanol-gasoline blended fuel. Under wide-open throttle (WOT), the results showed that brake torque (BT) reduced as the methanol ratio increased, whereas BT would be improved if the spark ignition timing was advanced. The CO and HC emissions from the methanol-gasoline blend were lower than gasoline. Elfasakhani [13] investigated the performance of iso-butanol-bio-methanol/gasoline and n-butanol-bio-ethanol/gasoline blends in spark-ignition engines. He noticed that using ternary fuel blends like iBM and nBE without modifying the engine resulted in a slight performance loss. CO, HC, and CO₂ emissions were lower in all fuel blends than in gasoline.

Vibrations in IC engines cause premature wear on the engine's internal components, limiting efficiency and service life while also creating discomfort to the occupants [14–16]. Unbalanced rotational and reciprocating masses, inertia forces of various parts, misfire, and cyclic fluctuations in gas pressure caused by the combustion process contribute to engine vibrations [15, 17]. The critical parameters of engine vibration and noise include the mechanical structure of the engine, engine load, engine speed,

engine oil qualities, and fuel properties [18–21]. Gravalos et al. [22] studied the vibration behavior of a spark-ignition engine using three different fuels: unleaded gasoline, ethanol-gasoline blend, and methanol-gasoline blend. They discovered that the fuel directly impacts the vibration behavior since the vibrations produced rely on the explosion inside the combustion chamber. They proposed a simple, reliable method or procedure for determining the fuel type utilized in a SI engine. The authors examined vibration at 1000, 1300, 1600, and 1900 rpm and found that the maximum vibration frequency from a gasoline-fueled engine was 1500 Hz.

In contrast, the highest vibration frequency from a gasoline-fueled engine was 2400 Hz. Sharma et al. [21] examined the combustion, noise, and vibration characteristics of a single-cylinder GDI engine running on oxygenated test fuels. They observed that in comparison to baseline gasoline, gasohol had increased maximum in-cylinder pressure (P_{max}), HRR, cumulative heat release rate (CHR), and rate of pressure rise (RoPR). The authors also discovered that gasohol produced lower vibrations than gasoline and that these vibrations increased in vertical and longitudinal directions as engine speed rose. The influence of ethanol-gasoline and propanol-gasoline blends on the emission and noise characteristics of a four-stroke SI engine was investigated by Keskin et al. [23]. They discovered a 65.56% drop in HC and a 33.92% increase in NO and CO₂ emissions for propanol-gasoline and ethanol-gasoline blends, respectively. Compared to gasoline fuel, there was a modest reduction in noise level under low load test circumstances, but noise levels increase with blend concentration.

Although numerous publications cover the performance, combustion, and emission characteristics of SI engines fueled with alcohol-gasoline blends, there is a significant gap in studies investigating SI engines' noise and vibration characteristics with dual alcohol-gasoline blends (ternary blends). In this study, isobutanol, higher carbon alcohol, is used to make ternary blends, which act as an additive to improve the qualities of the methanol-gasoline blends. The authors conducted a thorough investigation into an SI engine's performance, combustion, and emission characteristics using methanol-based single and dual alcohol blends. The authors compared the noise and vibration characteristics of engines that ran on gasoline and alcohol blends, with respect to combustion and performance parameters.

2. Test Fuels

The reference fuel for this investigation was commercially available motor spirit or gasoline (G100) obtained from a local fuel station in Tiruchirappalli, India. The blending alcohols methanol (lower-order alcohol) and higher-order alcohol, isobutanol ($(CH_3)_2CHCH_2OH$) were purchased from Sisco Research Laboratories (SRL), Mumbai, India. The lower heating value (LHV) of alcohols typically increases as the number of carbon atoms increases. As a result, isobutanol has higher LHV than methanol and is closer to gasoline. Furthermore, because butanol isomers have a nearer stoichiometric air-fuel ratio of base gasoline than methanol, they can be blended at larger ratios without necessitating vehicle system changes [24]. As a result, isobutanol is selected and added to methanol-gasoline blends in this study to improve their properties. The blends prepared and used for the experiments were 10% 15% and 20% v/v of methanol-gasoline (single alcohol blends) and 10% and 15% methanol-isobutanol(1:1)-gasoline (dual

alcohol blends). The alcohol percentage in the gasoline was limited (up to 30% by volume) in all trials to prevent modifying the current engine and fuel line. Alcohol blend ratios in gasoline are represented as volumetric percentages; this convention is used throughout this paper unless otherwise noted. All the samples were prepared in laboratory conditions and labeled M10, M15, iBM10, and iBM15. The fuel properties are given in Table 1.

The test fuels for these studies were prepared shortly before the engine testing to avoid phase separation in the fuel blends and assure fuel blend homogeneity. Meanwhile, the test fuel samples (Fig. 1) were assessed using visual or gravitational observation.

The fuel blends were kept in glass bottles for several days at room temperature and found homogenous blends without phase separation even after several weeks.

Table 1
The specifications of the alcohol and gasoline [1].

Property	Unit	Methanol	Isobutanol	Gasoline	Test Standard
Molecular Formula	-	CH ₃ OH	C ₄ H ₉ OH	C ₈ H ₁₈	-
Density	g/cm ³	0.790	0.805	0.745	ASTM D402
Kinematic viscosity	mm ² /s	0.59	4.5761	0.51996	ASTM D445
Lower heating value	MJ/kg	19.6	35.64	43.46	ASTM D240
Oxygen content	%	50	21.6	0	-
Research octane number	-	107	105	95	-
Latent heat of vaporization	kJ/kg	1180	686.4	180-352	-
Reid vapour pressure	kPa	21.1	3.3	62.7	ASTM D5191
Auto-ignition temperature	°C	470	430	280	-

3. Experimental Setup And Procedure

The trials in this study were conducted on a single-cylinder, 4-stroke SI engine. Fig. 2 depicted the schematic representations of the experimental test setup. The test engine has an electronic fuel injection (EFI) system and an open electronic control unit (ECU) (MX1-i5, Nira Control AB, Sweden). Table 2 summarizes the parameters of this SI engine. The EFI system can be controlled by a closed-loop system with a lambda sensor on the exhaust gas. A 3.75 KW Eddy current dynamometer (TME10, TECHNOMECH, 1500-5000 rpm) was connected to the test engine to manage the engine load. A precision-strain gauge load cell is used to sense the torque developed by the engine. A precise Rotary encoder (Autonics, E50S8-360-3-T-1, Incremental type) was fitted to the engine crankshaft to detect the crank position, producing 1 to 8000 pulses per revolution. Rotary encoders convert shaft rotation into electrical pulses and use them

to track the position of an engine crankshaft. A data acquisition unit counts and evaluates the encoder's output pulses to estimate engine speed and TDC, which gives remarkable precision. A cam position sensor (PG3.9, Bosch) is also attached to the engine setup. A combustion pressure sensor (PCB 113B22 ICP, Piezotronics, Inc) is precisely instrumented in the cylinder head to measure the gas pressure during each cycle. The frequency dynamic pressure sensor utilized in the system is built with naturally piezoelectric, stable quartz sensing elements that are well-suited to measuring pressure changes across a wide amplitude and frequency range. Microsecond response times and a high resonance frequency distinguish them. A high impedance measurement cable is used to transmit the electrical charge from the pressure sensor. The NI-6210 data acquisition unit was integrated into the system for combustion analysis. The fuel line in the test rig is fitted with a piezoresistive fuel pressure transmitter (Dawyer 628 Series), a solenoid valve (Uflow DAN14Z), and a burette for accurate fuel measurements. A number of thermocouples were installed correctly to measure engine temperatures.

Table 2
Test engine specifications

Item	Specification
Engine make and model	Honda GX 200
Engine Type	Air-cooled, 4-stroke, OHV petrol engine with EFI
Cylinder number	1
Displacement	196 cc
Bore × Stroke	68 x 54 mm
Compression ratio	8.5 : 1
Max. net power	4.1 kW (5.5 HP) @ 3600 rpm
Max. net torque	12.4 Nm @ 2500 rpm

The AVL Di-gas 444N analyzer was used to measure exhaust pollutants. Table 3 describes the technical details of the emission analyzer. A high-speed combustion data acquisition and analysis system was utilized to gather and analyze engine combustion data and calculate in-cylinder combustion noise data. The noise data was recorded using an Amprobe SM-20A noise meter, while the vibration acceleration data was collected using a Lutron VB 8203 piezoelectric accelerometer. Engine tests on all test fuels were conducted at various engine loads to study engine behavior and emission patterns. All fuel samples were tested in a steady-state condition, with no alterations to the engine. Before each sample testing, the engine was warmed up with regular gasoline. This study assessed the usability of single alcohol blends and ternary blends using a conventional single-cylinder SI engine with no modifications to the engine or fuel system. The engine's load conditions were set to function without knocking during the tests.

Table 3
Details of measuring technique, resolution and range of emission analyzer.

Emission	Measurement Technique	Resolution	Range
HC	NDIR Principle	1-10 ppm	0-20000 ppm
CO	NDIR Principle	0.01% by vol.	0-10% by vol.
CO ₂	NDIR Principle	0.1% by vol.	0-20% by vol.
NO _x	Electrochemical measurement	1 ppm	0-5000 ppm

4. Results And Discussions

The findings and discussions based on the most recent literature on the performance, combustion, and emission characteristics of single alcohol blends (methanol-gasoline) and dual alcohol or ternary blends (isobutanol-methanol-gasoline) under various loads and speeds are presented in the following sections.

4.1 Combustion analysis

The indicated mean effective pressure (IMEP) is often used in internal combustion engine tests to calculate the HRR and analyze combustion characteristics. Heat release analysis, based on the first law of thermodynamics using in-cylinder pressure vs. crank angle (CA) measurements over the compression and expansion strokes, is a useful technique for examining engine performance [25]. Figure 3(a) - 3(c) shows the variation of peak pressure, rate of pressure rise, heat, HRR, cumulative heat release rate (CHR) with different loads at different speeds for gasoline fuel (G100). These are the base plots for the engine fueled with neat gasoline for comparison. Integrating the HRR curve yield the cumulative heat release (CHR). The appropriate crank angle position for MBF of 10% and 90% was then established, and the 'combustion time' was estimated based on the variations in crank angle positions for these MBF values. Combustion factors such as cylinder pressure rise and heat release influence engine noise and vibration investigated and discussed in the following sections.

4.1.1 In-cylinder pressure

Figure 4 (a) and (b) represent the in-cylinder pressure versus the crank angle for the blended fuels and gasoline at no load and loading conditions, respectively. The maximum in-cylinder pressures for the alcohol blends appeared to be higher than the pure gasoline, as seen in Figure 4. Similar results were obtained by Yaman et al. [26] and Geo et al. [27]. The obtained peak pressure values at 2500 rpm and no load are 6bar, 5.94 bar, 5.9 bar, 5.7 bar, 5.53 bar, and 5.34 bar, respectively. Peak pressure values for iBM15, iBM10, M20, M15, M10, and G100 at 2500 rpm, 50% load, and 4.6 bar IMEP are 27.07bar, 25.6bar, 23.49bar, 20.63bar, 19.61bar, and 18.78 bar, respectively. This is related to the increasing amount of fuel delivered into the combustion chamber and the combustion chamber's volumetric efficiency [28]. The ternary blends had the highest peak in-cylinder pressure among the test fuel samples, as seen in Figure 4.

This is due to the presence of normal oxygen molecules in the chemical bonds of alcohol, which results in a better combustion response when radicals are produced during combustion [27].

Because of the fast flame velocity, which leads to higher cylinder pressure, the combustion process starts quickly with alcohol fuels when the spark plug ignites the fuel. This scenario offers a more convenient zone for developing the combustion process in SI engines. The work done in the piston and in-cylinder gas pressure increases more than gasoline in this scenario.

4.1.2 Heat release rate

The HRR calculation is a method for learning about the combustion process in IC engines [26]. Its heating value and combustion efficiency determine the heat release rate of fuel, and it was computed using the first law of combustion [27]. For determining apparent HRR, a typical single-zone model based on the first law of thermodynamics was used. The following equation (Eq. (1)) is used to find the CA-dependent HRR [25, 26, 29].

$$\frac{dQ}{d\theta} = \frac{\gamma}{\gamma - 1} P \frac{dV}{d\theta} + \frac{1}{\gamma - 1} V \frac{dP}{d\theta}$$

1

Where,

- Q : Total heat input
- P : In-cylinder pressure
- V : Volume of the cylinder
- γ : Ratio of specific heat values (taken as 1.35 [25, 26])
- θ : Crank angle

The HRR of a SI engine is affected by various fuel parameters such as calorific value and latent heat of vaporization (LHoV), auto-ignition temperature, and air-fuel mixing rate [30]. Fig. 5 **(a)** and **(b)** show the apparent heat release rate data related to the crank angle at no load and 50% engine load conditions. Hydrocarbon combustion begins with a hydrogen abstraction process, which produces radicals capable of reacting. The activation energy required to break the bonds controls the rate of hydrogen abstraction (rate of hydrogen atom transfer (HAT)) and subsequent radical generation reactions. Fuels with lower activation energy have a higher hydrogen abstraction and radical production rate, which results in a faster start of combustion and reduced combustion phasing [27]. The rate of hydrogen atom transfer (HAT) and subsequent radical production reactions are controlled by the activation energy required to break the bonds.

This investigation kept the ignition timing and engine speed constant, so the graph shows the heat release of a particular fuel type. It has been discovered that the heat release rate of alcohol blends is higher than that of gasoline. At 5Nm and IMEP 4.6 bar, the HRR for gasoline, M10, M15, M20, iBM10, and

iBM15 is 4.36 J/CAD, 6.37J/CAD, 6.59J/CAD, 7.51J/CAD, 8.03J/CAD, 10.29 J/CAD, respectively. At no-load conditions, the values are 1.46 J/CAD, 1.558 J/CAD, 1.6259 J/CAD, 1.596 J/CAD, 1.636 J/CAD, respectively. Compared to other oxygenate blends and gasoline, the most considerable heat release rate is recorded with iBM15. The increased hydrogen abstraction and radical production rate lead complete combustion and peak heat release earlier.

4.2 Performance analysis

4.2.1 Brake Thermal Efficiency (BTE)

The conversion of fuel burnt into useful energy is inefficient in most internal combustion engines. The efficiency with which they accomplish this is called "Thermal Efficiency." The power gain from the given energy input determines the brake thermal efficiency. The thermal efficiency determines how efficiently the combustion system recognizes the tested fuel and ensures similar assessments of how well the energy in the fuel is converted into mechanical output. [26]. Providing optimal operating conditions for the SI engine using open ECU technology and an appropriate testing procedure, according to Yaman et al. [7], is effective in raising engine efficiency. It can be seen from the graph (**Fig. 6**) that as the load increases, the BTE also increases.

At the 75% engine load, the maximum BTE for gasoline, M10, M15, M20, iBM10, and iBM15 occurred as 28.55, 29.22, 31.56, 32.3, 32.21, and 33.58, respectively. Alcohol-based fuels have a larger latent heat of vaporization (LHoV) and higher oxygen content than gasoline, and BTE values rise when alcohols are blended with gasoline [31]. Furthermore, under high engine loads, increased combustion temperatures and pressure were reported to increase BTE. [32, 33]. In this study, the ternary blend, iBM15 showed higher BTE at all the loading conditions. Because of their comparatively higher LHoV, alcohol blends absorb more heat from the engine cylinder during the vaporization process in the compression stroke than gasoline. As a result, the intake charge is significantly cooled, resulting in a significant rise in charge intensity, boosting the engine's volumetric efficiency for alcohol blends. Puli et al. [12] discovered that the BTE is higher for some alcohol blends because of the greater combustion efficiency. Geo et al. [34] also found a similar trend in SI engines using ethanol and benzyl alcohol gasoline blends. This outcome is consistent with the findings of earlier investigations. The addition of isobutanol to gasoline increased BTE for all compression ratios, according to Balki et al. [31]. According to Agarwal et al. [35], the BTE value attained with M10 and M20 fuels is higher than gasoline. The explanation for the larger BTE in comparison to gasoline was that methanol's laminar flame speed was faster than gasoline's. As a result of decreasing heat loss from the cylinder walls, rapid and full combustion occurred in the cylinder, and the oxygen content of the methanol allowed for a better combustion process. The BTE may be expected to decrease when a specific proportion of the alcohol level in the combination has been increased. The authors concur that more than 30% (by volume) of alcohol in the blends can reduce BTE because of the lower energy content. Alcohols' higher octane rating allows engines to operate at higher CRs, releasing more energy from the fuel. The faster flame speed of higher alcohols causes combustion to be completed earlier than base gasoline, reducing heat losses to the cylinder walls and thus requiring less work for

compression. Alcohols' high octane rating allows engines to operate at higher CRs, releasing more energy from the fuel; alcohols' high octane rating allows engines to operate at higher CRs [24].

4.2.2 Brake Specific Fuel Consumption (BSFC)

Figure 7 depicts the variation of BSFC with different engine loads at 2500 rpm. All of the blend fuels in this investigation had a higher BSFC than gasoline. The LHV of methanol and isobutanol account for the increased BSFC. The BSFC values increased as the alcohol concentration in the blend increased. This finding is consistent with prior research by Sayin et al.[36], Canakci et al.[37] and Sharudin et al. [38]. Isobutanol's lower heating value is higher than methanol's and is closer to gasoline's heating value. As a result, ternary mixes (iBM10 and iBM15) had a lower BSFC than M20. Despite having a lower LHV than base gasoline, methanol and isobutanol have higher density and kinematic viscosity. As a result, they could supply more fuel mass and have a greater effective power fuel conversion efficiency than base gasoline fuel [39].

The BSFC values increased by 0.56%, 1.72%, 3.02%, 3.73% and 2.46%, respectively, compared to gasoline by using M10, M15, M20, iBM10 and iBM15 blend at 75% load. Air/fuel ratio, Compression ratio, and modification in ignition advance can help reduce the increase in BSFC caused by increased BP value [10]. Despite the fact that BSFC values are rising as a result of blended fuel, the increase is less than 5% for all the blends at various loads.

Distillation properties

Fuel volatility is one of the most critical fuel characteristics for optimal SI engine performance and driveability, and it's frequently tested using a distillation curve. The distillation curve plots the boiling temperature of a fluid mixture against the volume fraction distilled. The cold start, engine warm-up, evaporative emissions, and vapor lock are all measured by front-end volatility (T0 -T20). Gasoline fuel's warm-up, acceleration, and cold-weather performance ability can be interpreted using mid-range volatility (T20 - T90). The tail-end volatility (T90 - end-point) is used to estimate the likelihood of deposits forming and oil dilution [24, 40, 41]. The distillation temperature profile of gasoline is mainly responsible for its cold engine drivability. In general, when the temperature drops at the 10, 50, and 90 percent evaporated points of the distillation curve, the cold engine drivability of gasoline improves. Lowering the distillation temperature of gasoline enhances fuel vapourization and thus performance in a cold engine running. As a result, because methanol and other alcohols lower distillation temperatures, the cold drivability of alcohol blends should be comparable to or better than gasoline fuel [6]. Fig. 8 shows the distillation curves for single and dual alcohol-gasoline blends and neat gasoline. Due to the methanol and its azeotropes boiling off first, methanol-gasoline blends (single alcohol blends) have the most distorted distillation curves. Similar results were observed by Andersen et al. [41]. The graph shows that almost all methanol is evaporated after 70% distillation, and the distillation curve reverts to something near a neat gasoline curve for all the blends. Because its molecular structure is closer to gasoline hydrocarbons, isobutanol reacts with gasoline less adversely than methanol [41]. As a result, the fall in boiling point from the pure hydrocarbon is significantly larger with methanol-gasoline blends than with isobutanol-

methanol-gasoline blends (Ternary blends). M20 shows the lowest temperature distillation up to 70% volume among other blends and gasoline. The closest similarity to the base gasoline curve was obtained with blends containing methanol and butanol (dual alcohol blends) from front-end to tail-end volatility. M10 shows closest similarity of distillation curve after 50% distillation at the tail-end distillation.

.3 Exhaust emission analysis

4.3.1 Effect of hydrocarbon (HC) emissions

Figure 9 (a) depicts the influence of increasing load (25%, 50%, 75% and 100% load) on HC emission at 2500 rpm. Insufficient oxygen is available for combustion when the engine is operating in a rich air-to-fuel ratio, resulting in incomplete combustion and unburnt hydrocarbons in the exhaust [42]. As seen 6 (a), the HC emissions of the blended fuels are lower than the neat gasoline by at different loads. The HC emissions from different blended fuels are lower than the neat gasoline by 5.7%, 4.61%, 1.6%, 4.9%, 10.3% at 75% load and 4.61%, 6.4%, 5.1%, 5.4%, 5.7% at 100% load for M10, M15, M20, iBM10 and iBM15 respectively. The presence of oxygen in the alcohol molecule structures is related to the reduced HC emissions of alcohol blends when compared to neat gasoline. As shown in Table 1, the oxygen content of isobutanol and methanol is around 21.6 and 50 mass %, respectively, compared to 0.0 mass% for gasoline. Similar results of lowering HC emissions for alcohol fuels have been reported by Sharudin et al. [38], Elfasakhany [43] and Geo et al. [34].

4.3.2 Effect of carbon monoxide (CO) emissions

Figure 9 (b) shows the CO emissions of gasoline and the blended fuels that were evaluated. When looking at the graph, it's clear that alcohol blends lower CO emissions across the board at all load circumstances. The graph also shows that the CO emissions increased as the load increased. Under 75% load condition by using M10, M15, M20, iBM10 and iBM15 blends, the CO emission reduced by 23.21%, 24.47%, 24.05%, 24.47% and 24.89% respectively. Previous studies that used alcohol-gasoline mixtures yielded similar outcomes [31, 36, 38], The formation of CO is caused by lack of oxygen for complete combustion, resulting in a loss in engine power. As a result, the generation of CO emissions is increased when the engine is operated at higher loads due to poor mixing [42]. Because alcohol, such as methanol, has less carbon than gasoline, the amount of CO produced by blended fuels is reduced. Since this carbon easily converts to CO when burned, methanol reduces CO production and emissions [44]. Methanol also has a high oxygen concentration (50 wt.%), and the methanol-gasoline blend has more oxygen when methanol is added to gasoline. The "pre-mixed oxygen effect" allows the reaction to progress to a complete state, resulting in lower CO and HC emissions [44]. The maximum cylinder temperature was stated to have increased as a result of the alcohol fuels forming a more homogeneous mixture with air, lowering emissions. Another reason for decreased CO emissions has been attributed to the fact that methanol and isobutanol have a faster laminar flame speed than gasoline [38].

4.3.3 Effect of CO₂ emissions

CO₂ emissions for various fuel samples at different loads are shown in Fig. 9 (c). When looking at the graph, it's clear that CO₂ emissions are often higher with higher-concentration alcohol blends under higher-load conditions. The increasing trend may be due to increased oxygen content and a higher air-fuel ratio than gasoline. However, blended fuels produce less CO₂ at lower load conditions than gasoline. M 10 and M 15 showed less CO₂ emission than gasoline at all loading conditions. Up to 50% load all blended fuels showed less CO₂ emission. The greater CO₂ levels for blended fuels are due to improved combustion. The authors' prior study [1] similarly found that alcohol blends emit more CO₂. The earlier study conducted by Masum et al. [45] and Balki et al. [31] conducted investigations that support this increasing trend in CO₂ emissions for alcohol blended fuels. Under 50% load condition by using M10, M15, M20, iBM10 and iBM15 blends, the CO₂ emission reduced by 11.61%, 10.9%, 14.21%, 17.77% and 18% respectively. Compared to gasoline, the ternary blends iBM10 and iBM15 increase CO₂ emissions by 4.8% and 4.6%, respectively, at 100% loading. HC emissions increase with load, as previously mentioned (section 4.3.1), meaning that the higher the oxygen percentage in the alcohol blend fuel, the more it combines with carbon atoms to produce CO₂ [38].

4.3.4 Effect of NOx emissions

Figure 9(d) depicts NOx emissions for different alcohol blends and gasoline at varied load conditions. The NOx emissions from all test fuels increase with an increase in load, as seen in the graph. Alcohol blends produced higher NOx than gasoline. The increased NOx emissions are due to the increased combustion temperature and the high flame propagation speed. According to the study, alcohol-containing fuels increased BTE and BSFC values by introducing more oxygen into the combustion process. As a result, the maximum temperature of combustion increased. In comparison to gasoline, these factors may be the reason for increased NOx emissions.

4.4 Noise and vibration characteristics

Combustion and mechanical forces are the primary sources of noise and vibration. These forces are transmitted to the engine's exterior surface over a wide frequency range. The external surfaces of the engine are subjected to vibrations of various amplitudes due to the effect of these forces. [16]. Gravalos et al. [46] discovered that changing the characteristics of the test fuel affected engine block vibration because of differences in the combustion process. Two other investigators, Keskin [18] and Othman et al. [47], have reported similar behavior on various engines and fuel testing types. The combustion of the fuel-air mixture inside the combustion chamber, initiated by the heat input in each of the cylinders, causes vibration. The fuel type utilized will directly impact this form of vibration by imparting specific properties [14]. Several factors influence engine noise and vibrations, including gasoline type, lubrication, engine load, and engine speed [1, 48, 49].

4.4.1 Noise characteristics

In an engine, there are three significant sources of noise. Combustion noise, mechanical noise, and intake and exhaust noise are the three types. The quick rise in combustion-generated cylinder pressure is linked to engine noise. It is dependent on the engine load, speed, and combustion mode.

The majority of mechanical noise is caused by rotating and reciprocating engine components. It is caused by inertial forces such as gears, timing drives, valve trains, fuel injection systems, and bearings [29]. Total noise refers to the sounds produced by the engine, including combustion, exhaust, and mechanical noise from various components or accessories [48]. Fig. 10 shows the total noise produced by various test fuels under different loading conditions at 2500 rpm. The noise level increased with increased load for all the test fuels. Alcohol blends showed a slightly higher noise level than gasoline. Higher exhaust pressure corresponds to higher peak cylinder pressure, resulting in increased exhaust noise and total noise. Fig. 11 shows the octave band spectrum of test fuels at 2500 rpm and different load conditions. Here, 1/3 octave band analysis is done to understand the frequency distribution of sound pressure level. The 1/3 Octave Bands are created by dividing a 1/1 octave band into three equal halves in order to provide a more detailed look at noise levels across the frequency spectrum. The Lower Band Limit, Center Frequency, and Upper Band Limit are the three components of an octave frequency band [1]. In the 10,000 Hz frequency band, all of the test fuels had the highest sound pressure levels.

4.4.2 Vibration characteristics

The reciprocating and rotating masses of the engine are primarily responsible for engine vibrations. [50]. The engine's unbalanced forces change with fuel supply, speed, and combustion characteristics due to fluctuations in inertial forces induced by the combustion and compression of the piston-cylinder arrangement throughout its operation [1]. The authors of this study attempted to determine the vibration characteristics of a SI engine using various alcohol blended fuels and compare them to straight gasoline fuel because different fuel compositions will produce different vibrations due to variations in the chemical energy released during the mixture's explosion. The vibration is measured in terms of RMS values of acceleration in m/s^2 or g. The direction of piston movement between BDC and TDC correlates to the acceleration in the Z-axis direction in this study. The piston strikes the cylinder walls along the Y-axis, and the X-axis is the mutually perpendicular direction to the Y-axis and Z-axis. Fig. 12 shows the comparison of the vibration levels of different test fuels. It is clear from the plot that ternary blends showed lower vibrations in the Y and Z direction at all loading conditions than single alcohol blends and gasoline. When compared to gasoline, the reduction in Z direction vibration with blended fuels at high load is 18.1%, 15.9%, 14.6%, 19.9%, and 25.2% for M10, M15, M20, iBM10, and iBM15, and reduction in Y direction acceleration is 28%, 38.5%, 16.51%, 34.28% and 51.12% for M10, M15, M20, iBM10, and iBM15, respectively. Alcohol blends exhibit less vibration in almost all loads than gasoline, except for M20, which exhibits more vibration in the X-direction. Similar results have been obtained by Sharma et al. [21]. For alcohol blends, the in-cylinder pressure, HRR, and ROPR were all found to be higher; however, the vibration levels were comparatively lower than gasoline fuel. The Z-direction and Y-direction vibrations are higher than the X-direction vibration because these two vibrations represent the direct impact of piston motion

and the direct cyclic imbalance of forces exerted in the cylinder by piston motion. They are more powerful than the X-direction vibration [21].

5. Conclusions

This article investigates the performance, combustion, emission, noise, and vibration characteristics of a single-cylinder EFI engine employing methanol-based single and dual alcohol blends. The following are the key conclusions reached as a result of the findings and discussion:

1. The oxygenated alcohol blends, particularly dual alcohol blends, outperformed neat gasoline in terms of performance, combustion, and vibration characteristics. Compared to gasoline and other alcohol blends, the ternary blend had a higher in-cylinder pressure and HRR.
2. The BTE of iBM15 was found to be the greatest under 75 percent load, followed by iBM10, methanol mixes, and gasoline. The BTE of the ternary blend was found to be 5% higher than that of gasoline.
4. At all load conditions, the HC and CO emissions of single alcohol-gasoline (M10, M15, M20) and dual alcohol – gasoline blends (iBM10, iBM15) were lower than gasoline. Under 75% loading, iBM15 shows a maximum decrease of 10% and 24.89% in HC and CO emissions, respectively. Blended fuels emit less CO₂ under low loading situations. NO_x emissions were found to be higher than gasoline for blended fuels due to the increased combustion temperature and the high flame propagation speed.
5. Vibrations in the Y- and Z-direction are higher than vibrations in the X-direction due to the direct influence of piston motion and the direct cyclic imbalance of forces exerted in the cylinder by piston motion. The use of iBM10 and iBM15 reduces Z-direction acceleration by 19.9% and 25.2% and Y direction acceleration by 32.28% and 51.12%, respectively, compared to gasoline.
6. For all test fuels, the noise level increased as the load increased. The noise levels of blended fuels were slightly higher than that of gasoline.
7. Ternary or dual alcohol blends (isobutanol-methanol-gasoline) are excellent in enhancing performance while lowering emissions and vibration.

Abbreviations

Nomenclature			
IC	Internal combustion	RoPR	Rate of heat release
SI	Spark-ignition	HRR	Rate of heat release
CI	Compression ignition	CHR	Cumulative heat release
EFI	Electronic fuel injection	BSFC	Brake specific fuel consumption
ECU	Electronic control unit	HC	Unburned hydrocarbon
G100	100% gasoline	CO	Carbon monoxide
M10	10% methanol & 90% gasoline (by vol.)	CO ₂	Carbon dioxide
M15	15% methanol & 85% gasoline (by vol.)	NO _x	Nitrogen oxides
iBM10	10% methanol & 10% butanol & 80% gasoline (by vol.)	CAD	Crank angle degrees
iBM15	15% methanol & 15% butanol & 70% gasoline (by vol.)	MFB	Mass fraction burned
BEV	Battery electric vehicle		
GHG	Greenhouse gases		

Declarations

Statements and Declarations

The authors have no competing interests to declare that are relevant to the content of this article.

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Figures

Figure 1

Fuel samples for stability test

Figure 2

Schematic of the experimental engine setup

Figure 3

- (a) Combustion analysis of SI engine fueled with gasoline at 2000 rpm at various loads
- (b) Combustion analysis of SI engine fueled with gasoline at 2500 rpm at various loads
- (c) Combustion analysis of SI engine fueled with gasoline at 3000 rpm at various loads

Figure 4

- (a). In-cylinder pressure vs. crank angle position for different test fuels at no load.
- (b). In-cylinder pressure vs. crank angle position plot of test fuels at various loading conditions.

Figure 5

HRR as a function of crank angle position for different test fuels at no load and load conditions.

Figure 6

Variation of BTE with varying engine load for different test fuels.

Figure 7

Variation of BSFC at different engine loads.

Figure 8

Distillation curve for alcohol blends and neat gasoline.

Figure 9

Variation of HC, CO, CO₂, and NOx emission at different load conditions.

Figure 10

Variations of total noise with varying engine load for different test fuels.

Figure 11

Octave band spectrum for test fuels at various loads

Figure 12

Variations in vibration levels at different engine loads for test fuels.

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