Improved SI Engine Efficiency using Acetone-Butanol-Ethanol (ABE)

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Abstract

Alcohols, especially n-butanol, have received a lot of attention as potential fuels and have shown to be a possible alternative to pure gasoline. The main issue preventing butanol's use in modern engines is its relatively high cost of production. ABE, the intermediate product in the ABE fermentation process for producing bio-butanol, is being studied as an alternative fuel because it not only preserves the advantages of oxygenated fuels, but also lowers the cost of fuel recovery for individual component during fermentation. With the development of advanced ABE fermentation technology, the volumetric percentage of acetone, butanol and ethanol in the bio-solvents can be precisely controlled. In this respect, it is desirable to estimate the performance of different ABE blends to determine the best blend and optimize the production process accordingly. In this paper, pure ABE fuels with different component volumetric ratio, (A: B: E of 3:6:1, 6:3:1 and 5:14:1), were combusted in a naturally aspirated, port-fuel injected spark ignited engine. The performance of these blends was evaluated through measurements of in-cylinder pressure, and various exhaust emissions. In addition, pure gasoline and neat n-butanol were also tested as baselines for comparison of ABE fuels. The tests were conducted at an engine speed of

1200 RPM and loads of 3 and 5 bar brake mean effective pressure (BMEP) under different equivalence ratios. On the basis of the experimental data, the combustion characteristics and emission behavior of these fuels have been presented and discussed. It was found that in terms of thermal efficiency, ABE(6:3:1) might be much better suited for use as an alternative fuel, relative to ABE(3:6:1) or n-butanol.

Introduction

In recent years, growing public concern over the economic and environmental viability of gasoline, diesel and other fossil fuels, has prompted the investigation into oxygenates as fuel additives [1]. Oxygenated compounds previously considered include butanol, ethanol, methanol, and methyl or ethyl esters or ethers [2]. Many research studies into n-butanol have been conducted due to its properties that closely resemble those of gasoline [2-4]. These properties include ease of transportation through pipelines due to its hydrophobic nature thereby resulting in a lower tendency to separate from the base fuel (i.e. diesel or gasoline) when mixed with water; an air fuel ratio that closely resembles that of gasoline allowing for greater percentages of butanol to be mixed with gasoline; and an energy content that is 30% more than ethanol [5, 6]. Furthermore, n-butanol, when used as a transportation fuel, can save 39-56% fossil energy while reducing greenhouse gas emissions by up to 48% on a lifecycle basis [6]. The main issue that prevents butanol's use in modern engines however is its relatively high cost of production, which has been the subject of many other research studies [5-16].

Acetone-butanol-ethanol (ABE) fermentation primarily involves bacterial fermentation of biomass feedstock to produce acetone, n-butanol and ethanol at volume percentages of approximately 22-33%, 62-74%, and 1-6% respectively (roughly a 3:6:1 ratio [16, 17]). Due to the depletion of fossil fuels, and subsequent rise in oil prices, interest in ABE production by fermentation as a viable alternative to the petroleum process has been renewed. Bio-butanol usually uses a strain of bacteria from the Clostridia Class (Clostridium Family). Clostridium acetobutylicum is the most well-known strain, although Clostridium beijerinckii has also been used for this process with good results. Clostridial species show promise for ABE fermentation using lignocellulosic (e.g., bagasse, barley straw, wheat straw, corn stover and switchgrass etc.) and non-cellulosic (e.g., glucose, corn, sago, and sugarcane) feedstocks. The United States

produces the largest amount (approximately 280 million tons per year) of corn in the world, whereas China comes in second with approximately 131 million tons per year. From the availability point of view, the lignocellulosic materials have been marked as a cheaper and more abundant feedstock for biofuels [3]. If the intermediate product of fermentation, the ABE mixture, could be used for clean combustion, the separation costs would be mitigated. This would save an enormous amount of time and money in the production chain of bio-butanol [8]. It should be noted that the actual fermentation product contains a relatively small amount of water (< 0.5% by weight) [11], which has not been included in this study. This study of water-containing ABE is being carried out and will be discussed in a future paper. However, this level of purity is sufficient for full miscibility with gasoline. ABE fuel properties can be adjusted to suit internal combustion engine requirements, by changing the ratio of the ABE components through fermentation. As mentioned earlier, the typical ratio of acetone, butanol and ethanol is 3:6:1 during the formation process, but this is adjustable. Modification of bacterial strains at the genetic level is the common method for researchers to optimize production components. At the same time, fermentation products and the ratio of their formation also vary with the fermentation conditions (pH, temperature, nutrients) [17]. The goal of this study is to investigate the combustion characteristics of ABE in a spark-ignited engine estimate the performance of different ABE blends to determine the best blend and optimize the production process accordingly. In the future, ABE mixture could be used directly, with the components ratios controlled during the ABE fermentation process [18, 19]

Butanol has been widely investigated as an alternative fuel for both gasoline and diesel engines. Zheng et al. investigated the effects of n-butanol and its isomers on combustion and emissions of a diesel engine, and found that the alcohol blends showed a retarded combustion phasing, higher combustion efficiency and lower soot emissions. However, gaseous emissions were not affected obviously [20]. They also studied combustion and emission of blends of diesel, gasoline and n-butanol, and found that the ITE was slightly increased with the blended fuels [21]. Liu et al. studied the combustion of neat n-butanol and soybean biodiesel in a constant volume chamber and found that n-butanol was more effective in soot suppression relative to biodiesel [22]. They also studied n-butanol and biodiesel dual-fuel combustion in a diesel engine. A slightly higher ITE and significantly reduced NO_x, soot emissions were observed [23]. Liu et

al. also investigated the effect of adding various oxygenated fuels (20% by volume) to diesel fuel and found that among n-heptane, iso-octane, n-butanol and methyl octynoate, n-butanol showed the largest soot reduction, however, they found that fuel properties and oxygenated structures had minor effects on gaseous emissions and ITE [24]. As for SI engines, Masum et al. [25] studied the combustion and emissions of methanol, ethanol, butanol and pentanol blended with 80% vol. gasoline. They found that all alcohol blends displayed better engine torque and lowered emissions relative to gasoline. Costagiola et al. [26] studied performance and emissions of various gasoline/alcohol blends. They found an increase in global efficiency and a reduction in emissions using the blends. Williams et al. [27] investigated a series of conventional and alcohol fuels and concluded that thermal efficiency, combustion, and emissions were not adversely affected as a result of adding any butanol to gasoline. Dernet et al. [28] evaluated the combustion and emissions characteristics of butanol-gasoline blends in a port fuel injection (PFI) SI engine. The results demonstrated that a 40% butanol/60% gasoline blend by volume minimized HC emissions and no significant change in NO_x emissions were observed with the exception of the 80% butanol/20% gasoline blend. The addition of butanol improved combustion stability and reduced ignition delay (0–10% MFB). The change of specific fuel consumption of B40 blend was within 10% of that of pure gasoline for stoichiometric mixture. Wigg et al. [29] showed that blends containing below 40% volume of butanol offered similar unburned hydrocarbon (UHC) emissions to gasoline, but higher hydrocarbons (HC) levels than pure gasoline at higher butanol concentrations. The results also indicated a slight increase in brake specific fuel consumption (BSFC) with the butanol addition. Venugopal et al. [30] studied engine performance with simultaneous injection of butanol and gasoline, as well as blended fuels. On the whole, at all operating conditions, simultaneous injection results in reduced HC levels and improved or similar performance as compared with B50 (injection of fixed blend). Gu et al. [31] studied combustion in a spark-ignition engine fueled with gasoline–n-butanol blends. It was found that, HC, carbon monoxide (CO) and NO_x emissions fueled with gasoline and n-butanol blends are lower than those of gasoline. Pure n-butanol increased the HC and CO while decreased the NO_x; these tendencies were similar to [28]. Yacoub et al. [32] performed several studies on application of straight-chain alcohols C1–C5 (methanol to pentanol) as fuels blended with gasoline. The study showed that all alcohol–gasoline blends showed reduction in CO emissions, and total hydrocarbons (THC) emissions were also reduced at optimized operating

conditions. However, all blends had a higher unburned alcohol emission than gasoline, with the highest emissions coming from those with the highest alcohol content. Aldehyde emissions were higher for all blends with formaldehyde as the main constituent and the NO_x emissions may increase or decrease depending on different operating conditions. Szwaja and Naber [33] investigated the combustion characteristics of n-butanol in a single cylinder engine and results indicated that the highest peak pressure advanced with the increase of n-butanol ratio due to a faster combustion and the crank angle degree (CAD) of 50% mass fraction burn (MFB) for n-butanol was approximately 2° earlier when compared to gasoline. Wallner et al. [34] investigated the combustion, performance, and emissions of pure gasoline, 10% ethanol (E10) and 10% butanol (Bu10) blends in a direct-injection (DI) four-cylinder SI engine. Results showed that the burning velocity of the Bu10 was higher than those of both E10 and gasoline. Their further study [35] demonstrated that addition of alcohol to the fuel blend results in a consistent reduction in NO_x emissions regardless of operating point. Both formaldehyde and acetaldehyde emissions increased with the addition of butanol, whereas formaldehyde did not increase significantly with addition of ethanol. Propene, 1, 3-butadiene, and acetylene emissions, which are required for carbon growth processes leading to benzene, also increased only with the addition of butanol.

Recent studies on ABE include ABE-diesel blends combustion in diesel engines (showing simultaneous reductions in PM and NO_x emissions) [8] and kinetic modeling of ABE combustion [36]. Wu et al. [37-42] performed various optical studies on combustion characteristics of ABE and ABE-diesel blends in a combustion chamber, and found that ABE provides tremendous soot and NO_x reduction capabilities. It was also found that ABE(6:3:1) showed the highest combustion efficiency, while maintaining phasing close to that of pure diesel. The authors [43-47] have also studied the performance and emissions of ABE-gasoline blends on an SI engine, and found that a small amount of ABE addition could enhance efficiency and reduce emissions. However, to date, combustion characteristics of pure ABE fuels in SI engines have not been investigated. This is the primary motivation for this study.

Experimental Setup

Engine Setup

Most of the experimental setup details have been reproduced from [29], since the same test bench was used for this study. Experiments were conducted using a single cylinder engine with identical cylinder geometry to the V8 engine used in a 2000 Ford Mustang Cobra. The peak power output of the original V8 engine was 239 kW (329 HP) and 407 Nm (300 lb-ft) of torque resulting in a peak output for the single cylinder engine of slightly less than 30 kW (40 HP) and 52 Nm (38 lb-ft) as a result of increased frictional losses. The bottom end is composed of two iron castings produced by Ford. The lower casting houses the crankshaft bearings and the upper casting consists of a single cylinder bore which aligns with cylinder two on the head. The cylinder head is from the left bank of the production V8 engine featuring double overhead camshafts and 4 valves per cylinder with a centrally located spark plug. In order to reduce frictional losses, the rocker arms were removed from cylinders one, three and four. The engine is coupled to a GE type TLC-15 class 4-35-1700 dynamometer capable of delivering up to 14.9 kW (20 HP) and absorbing up to 26.1 kW (35 HP) at a maximum rotational speed of 4500 RPM. The dynamometer is controlled using a DyneSystems DYN-LOC IV controller and a DyneSystems DTC-1 digital throttle controller. In-cylinder pressure is measured using a Kistler type 6125B pressure transducer and an AVL 3057-AO1 charge amplifier and indexed against a crankshaft position signal from a BEI XH25D shaft encoder.

The engine is controlled through the use of a Megasquirt II V3.0 Engine Control Unit (ECU) which allows the adjustment of fuel through volumetric efficiency tables and adjustment of ignition timing (spark advance) as functions of engine speed (RPM) and engine load (manifold air pressure, (MAP)). The fuel injector used was a Bosch injector # 0 280 150 558 rated at 440 cm³/min at a fuel pressure of 3 bar.

Exhaust Gas Analyzers

NO_x and λ (and Φ) measurements were conducted using a Horiba MEXA-720 NO_x non-sampling type meter in the exhaust manifold of the engine. The measurement range for NO_x is 0-3000 ppm with ± 30 ppm accuracy for 0-1000 ppm, $\pm 3\%$ accuracy for 1000-2000 ppm, and $\pm 5\%$

accuracy for 2000-3000 ppm. The measurement of range for lambda is 0.65-13.7. To enhance accuracy and ensure reliable data, a LabVIEW code was written to collect and average NO_x and lambda measurements over a 60 second period at 10 samples per second (600 samples total). Measurements of unburned hydrocarbons and carbon monoxide were made using a Horiba MEXA-554JU sampling type meter. A probe was fabricated to fit in the exhaust manifold of the engine that allowed the sampling tube to transport the exhaust gases to the meter. The measurement range is 0-10,000 ppm for unburned hydrocarbons, 0.00-20.00% by volume for carbon dioxide, and 0.00-10.00% by volume for carbon monoxide. Exhaust gas temperature measurements were made using a type-K thermocouple located in the exhaust manifold. It should be noted that the analyzer used to measure emissions of unburned hydrocarbons uses a non-dispersive infrared analyzer (NDIR). Both NDIR and flame ionization detection (FID) measurements of emissions exhibit low responses to oxygenated hydrocarbons. Engine tests in [35] examined the differences between a Horiba FIA-23A FID analyzer and an MKS 2030 Fourier Transform Infrared (FTIR) analyzer, which can speciate hydrocarbons and more accurately measure oxygenated hydrocarbons. Comparisons between FID and FTIR showed that, for oxygenated fuels, FID consistently underestimated the amount of unburned hydrocarbons although the observed trends were preserved between the two analyzers. However, for the NDIR measurements reported here, using an alcohol fuel does not alter the substance of the results.

Test Fuels

Ethanol-free Gasoline (Research Octane Number (RON) = 90) was selected as the baseline fuel in this study. The ABE solution was first prepared at volume ratios of A:B:E = 3:6:1, 6:3:1 and 5:14:1 using analytical grade acetone (99.5%), butanol (99.5%) and ethanol (99.8%). A gravitational test for stability was carried out as samples of the prepared blends were deposited in a test tube at 25 °C and 1 atm for 90 days. The blends displayed a clear single phase after the stability test. The n-butanol was supplied by Fisher Scientific while acetone and ethanol meeting USP specs were supplied by Decon Laboratories, Inc. The ABE blend with ABE ratio of 3:6:1 will be referred to as ABE(3:6:1); that with ABE ratio of 6:3:1 will be referred to as ABE(6:3:1), and so on, in the remainder of the text. Pure gasoline will be referred to as ABE0. The ABE ratio of 3:6:1 is studied because it is the most commonly produced in the fermentation

product [16,17]. The ratio of 6:3:1 is used as it helps in understanding the effect of increasing acetone and decreasing butanol. It has also been found in [39, 45] that ABE(6:3:1) showed great potential for improving combustion efficiency. ABE(5:14:1) is investigated as it was studied in [8] and showed promising results in diesel engine combustion. Finally, pure n-butanol is studied for comparison purposes and to further understand the effects of different components.

The properties of individual fuels are listed in Table 2. The difference in the latent heats of vaporization between the fuels is worth noting, as are the different laminar flame speeds (LFS). Note that the latent heat of vaporization of acetone is slightly higher than that of gasoline; however, those of ethanol and butanol are nearly 50-75% higher than that of gasoline. As far as the LFS is concerned, gasoline has the lowest value among the individual fuels. The LFS plays an important role in the early phase of combustion [33]. The LFS for the components is as follows: Butanol > Ethanol > Acetone > Gasoline [Table 2]. The LFS of ABE(3:6:1) was predicted by Van Geem et al. to be 37 cm/s (298 K, Stoichiometric) [36]. This is about 3 cm/s higher than gasoline. It is expected that the LFS of the fuels would increase with increasing n-butanol content. However, it is to be noted that a small addition of a component with higher LFS does not show any major effects on the LFS of the blend [50, 51]. Another parameter of importance is the latent heat of vaporization which would cause a charge-cooling effect. Alcohol fuels have been shown to have higher charge-cooling relative to that with gasoline [52]. The LFS strongly depends on temperature. It was shown that the effect of temperature on LFS dominated over the effect of 50% vol. addition of ethanol to iso-octane [51].

Therefore, it is expected that the combustion phasing of ABE blends would be determined by the balance between the increase in LFS due to the addition n-butanol, and the decrease in LFS due to the reduction in temperature at ignition by charge-cooling [44]. Finally, acetone with a boiling point of 56 °C could help the blend's spray collapse significantly, as a small portion of a low boiling point substance within a fuel may be the catalyst to spray collapse, even if the majority constituent of the fuel is not in the region of flash boiling in terms of temperature [53]. Table 3 shows properties of the blends calculated using simple mixing rules.

Test Conditions

In this study the engine load was set to 3 bar and 5 bar (BMEP) and the speed at 1200 RPM. The throttle plate was fully opened (Wide Open Throttle). The fuels were first tested under stoichiometric conditions with the same ignition timing as gasoline's MBT (18° Before Top Dead Center (BTDC) at 3 bar and 24° BTDC at 5 bar BMEP, the default values for gasoline on this engine), to perform an analysis of ABE use in SI engines without any modifications. Next, each fuel was tested at its own MBT (Table 5) and an air/fuel ratio sweep was performed. It should be noted that the engine started up immediately using all fuels except n-butanol, for which the engine had to be warmed up beforehand. This is likely due to the enhanced volatility of ABE fuels due to the presence of acetone. Measurements of brake torque, lambda, and NO_x were averaged of a 60-second period while UHC, CO and exhaust gas temperature (EGT) measurements were recorded directly from the emissions analyzer. In addition, in-cylinder pressure traces were taken for all fuels to examine the differences in peak cylinder pressure. The experiments were performed 3 times and these datasets were then averaged. The tests were performed in a temperature-controlled laboratory; so the effects of humidity were assumed to be negligible. The intake air pressure was controlled using an electronic regulator which provides precise control (Accuracy: +/- 0.1 kPa). Furthermore, the engine was allowed to run at every operating condition for an extended period of time to ensure steady state measurements. The conditions used in this test are summarized in Table 4.

Results and Discussion

The results under stoichiometric conditions and gasoline MBT are first presented and discussed in detail, after which, results over the range of equivalence ratios tested at each fuel's MBT are presented for completeness and briefly discussed. Although the tests were performed at both 3 and 5 bar BMEP under the fuels' MBTs, only the 3 bar results are shown for brevity's sake. The results under 5 bar BMEP showed similar trends. The in-cylinder pressure traces are first presented to compare differences in peak cylinder pressure between the fuels. Next, emissions measurements are presented starting with UHC. UHC emissions are presented first since they provide insight into how well the fuel mixes with the air and is consumed during the combustion process. Emissions of carbon monoxide are then shown to estimate the completion

of combustion, followed by NO_x emissions to analyze the effect of ABE's lower energy content on NO_x production. The error bars in the figures show the standard error.

In-Cylinder Pressure Traces

Figure 2(a) shows the pressure traces of all tested fuels at 3 bar BMEP and gasoline MBT at $\Phi = 1$. The traces shown are the mean traces of several 25 consecutive engine cycle samples recorded over a 60 second period. The peak cylinder pressures of all ABE fuels except ABE(6:3:1) are higher compared to that of gasoline. ABE(6:3:1) shows the most retarded phasing due to it having the lowest LFS. ABE(3:6:1) shows a slightly advanced peak pressure because its' higher flame speed which causes the combustion to initiate faster and approach completion, leading to a higher combustion peak and advanced CAD position with respect to gasoline. Figure 2(b) shows the pressure traces of all tested fuels at 3 bar BMEP and each fuel's own MBT (shown in Table 5), under stoichiometric conditions. It appears that at their MBTs, ABE(3:6:1), ABE(5:14:1) and n-butanol show similar combustion phasing, whereas, ABE(6:3:1) and gasoline behave similarly. However, ABE(6:3:1) has a marginally retarded phasing.

Mass Fraction Burnt (MFB) Profiles

Normalized MFB plots, which can express heat release from combustion, were determined from each of the pressure traces and illustrated in Fig. 3. In this analysis the heat transfer to walls and fuel flow into crevices were neglected. Therefore, the apparent heat release rate was calculated from the pressure trace using the first law of thermodynamics as expressed in Eq.(1).

$$\frac{dQ_n}{dt} = \frac{\gamma}{\gamma - 1} p \frac{dV}{dt} + \frac{1}{\gamma - 1} V \frac{dp}{dt} \quad (1)$$

where, γ is the specific heat ratio, p is the in-cylinder pressure, V is the cylinder volume, and Q_n is apparent heat release.

From Fig. 3, quantities such as 10% MFB, 50% MFB and 90% MFB can be determined. These values correspond to 0.1, 0.5 and 0.9 of normalized MFB, respectively. The ignition delay

shown in Fig. 4 (a) is calculated as the difference between the spark timing and 0.1 MFB timing; combustion duration presented in Fig. 4 (c) is calculated as a difference between 0.9 MFB and 0.1 MFB. They are expressed in degrees of crankshaft angle.

Fig. 4 (a) shows the ignition delay for the different fuels at both 3 and 5 bar BMEP, at gasoline MBT at $\Phi = 1$. During this period of early combustion, the combustion rate is impacted by the laminar flame speed of the fuel–air mixture. At later times which are in the fully developed bulk burn, the combustion is dominated by turbulent flame propagation [33]. ABE(3:6:1), ABE(5:14:1) and n-butanol show similar, relatively shorter ignition delays, followed by gasoline, and ABE(6:3:1) having the longest ignition delay. This trend is in accordance with the LFS of the fuels, and this effect is further pronounced at 5 bar BMEP. Fig. 4 (b) shows the CA50 location for the different fuels under the same conditions. The CA50 trends are similar to those seen for ignition delay. Under 5 bar, the difference in CA50 between ABE0 and ABE(3:6:1), ABE(5:14:1) is about 1.2° (advanced), and that between ABE0 and ABE(6:3:1) is about 1° (retarded). The 50% MFB represents the center of combustion and it has been shown that the engine torque strongly depends on the location of 50% MFB. From Fig. 4(c), we can see that the combustion duration trends at 3 bar BMEP are similar to those observed for CA50. At 5 bar BMEP, it appears that all ABE fuels show slightly longer combustion duration than that of pure gasoline.

Fig. 5 shows the ignition delay (top), CA50 (middle) and combustion duration for the different fuels at 3 bar BMEP, at the fuels' MBT at varying equivalence ratios. It is apparent that the fuels behave most similarly near stoichiometric and richer conditions. At leaner equivalence ratios, the combustion behavior is more widely separated. A shorter ignition delay is observed at richer equivalence ratios. This is due to the fact that flame development is slower at leaner equivalence ratios [49]. It can be seen that throughout the equivalence ratio range tested, combustion phasing is consistent with the amount of butanol in the fuel. As butanol content increases, phasing gets advanced. ABE(3:6:1), ABE(5:14:1) and n-butanol behave quite similarly, whereas gasoline and ABE(6:3:1) have similar phasing. Fuels with higher n-butanol show shorter ignition delays and combustion durations due to higher LFS. ABE (6:3:1) behaves similar to ABE0 for the most part, likely because these two fuels would have almost similar LFSs. As such, the trends observed are similar to those in [27, 28, 32, 33], where the addition of

butanol advanced combustion phasing. In summary, ABE(6:3:1) does not cause an appreciable change in phasing, however, ABE(3:6:1) and ABE(5:14:1) advance combustion phasing.

Brake Specific Fuel Consumption & Efficiency

Figure 6 shows the brake specific fuel consumption (BSFC) of different fuels. The brake specific fuel consumption (BSFC) represents the fuel consumption as a function of engine power.

All test blends have higher BSFC than gasoline because of their lower LHV (seen in Table 3). ABE(6:3:1) has the lowest LHV among the fuels, so its BSFC is the highest at the same output torque. At $\Phi=1$, the BSFC of the blends was roughly 10-30% higher than that of ABE0.

The fuel energy input changes with the fuel properties, mainly based on the ratio of lower heating value to stoichiometric air demand. To ensure an unbiased comparison and to better evaluate the fuel economy of alternative fuels, the Gasoline Equivalent BSFC is used in this study, which is defined as follows:

$$\text{Gasoline Equivalent BSFC} = \text{BSFC}_{\text{blends}} \times \frac{\text{LHV}_{\text{blends}}}{\text{LHV}_{\text{gasoline}}} \quad (2)$$

Fig. 7 (a) shows the Gasoline Equivalent BSFC of the fuels. The differences in Gasoline Equivalent BSFC among all test fuels are much less compared to those in BSFC. The relatively lower Gasoline Equivalent BSFC of ABE(6:3:1) indicates a higher BTE (Fig. 7 (b)). ABE(6:3:1) shows relatively higher efficiency than all other fuels at stoichiometric and richer air/fuel ratios, whereas butanol shows highest efficiency at leaner air/fuel ratios. The other fuels show slightly higher efficiency relative to that of gasoline. For instance, at $\Phi=1$, the BTE of ABE(6:3:1) is 1.6% higher than that of ABE0, whereas n-butanol shows an increase of 1% relative to ABE0. The relatively lower efficiency of the blends with higher butanol content is likely due to their retarded spark timing and incomplete combustion as seen by a spike in HC and CO emissions (shown later).

The higher thermal efficiency of ABE(6:3:1) can be attributed to the fact that blends with low carbon numbers contain more oxygen than those with high carbon numbers. As a result,

combustion is improved, thereby enhancing thermal efficiency. Secondly, from Fig. 2(b), it can be seen that ABE(6:3:1)'s compression work is quite reduced compared to the other fuels. From Fig. 5, it can be observed that ABE(6:3:1) shows relatively similar CA50 as that of gasoline, however, it has a longer combustion duration. This reduced compression work increases the net work output, and the slower rate of combustion leads to comparatively lower heat losses, and better post-flame HC and CO oxidation, which ultimately leads to a higher thermal efficiency. Acetone's higher volatility is likely to make it easier to burn thus approaching complete combustion, as indicated by ABE(6:3:1)'s reduced HC and CO emissions (shown later). In Direct Injection (DI) engines, fuel is vaporized in the compression stroke when latent heat of vaporization is high. Given that fuel absorbs heat from the cylinder during vaporization, the air-fuel mixture is compressed more easily, thus improving thermal efficiency [25]. This indicates the potential for consumption reductions in DI spark ignition engines. The higher acetone content would also enhance spray collapse and mixing and make the fuel much more volatile. Therefore, acetone addition could improve combustion quality significantly. In summary, the typically produced fermentation product, i.e., ABE(3:6:1) may not be ideal for engine combustion. In terms of efficiency, fermentation products with higher acetone content, such as ABE(6:3:1) would be much better suited as alternative fuels for SI engines.

Exhaust Gas Temperature

EGT provides insight into the combustion process by measuring the temperature of the burned gases directly after they exit the engine. With ignition timing fixed, the differences in EGT should be proportional to the combustion temperature of the fuels. Fig. 8 shows the EGT of different at $\Phi=1$. It can be seen that there is no real change in the exhaust temperature. Even though the blends have a higher latent heat of vaporization and a lower heating value than gasoline, their pressure traces and combustion phasing were relatively similar, and hence the EGTs are not affected significantly.

Emission Behavior

All reported emissions are raw emissions without the use of a catalytic converter. UHC emissions provide direct insight into the combustion process by measuring how much fuel is left over after the combustion of the fuel-air mixture. Engine-out THC emissions are primarily a

result of engine configuration, fuel structure, oxygen availability, and residence time. It might be hypothesized that the addition of an alcohol such as ethanol or butanol to gasoline would improve THC oxidation due to the higher oxygen content in the cylinder and exhaust. However, note that the engine is operated at the stoichiometric air fuel ratio for each specific fuel blend and thus, excess oxygen is not available [34].

Figures 9(a) and 9(b) show the UHC emissions at gasoline MBT ($\Phi=1$) and fuels' MBT, respectively. The HC emissions are reduced with the use of ABE(6:3:1), which shows the lowest HC emissions. This is due to higher oxygen content, higher volatility and better post-flame oxidation of the fuel, likely due to the impact of acetone. However, with higher butanol content, the HC emissions rise steeply, as shown by ABE(5:14:1) and n-butanol. From Fig. 9(b), it is seen that similar behavior is observed at all equivalence ratios under the fuels' MBT. The increase of HC with increasing butanol addition is consistent with the literature [26, 28]. This is due to butanol's high latent heat of vaporization, which could reduce the combustion temperature and lead to unburnt fuel.

CO production is primarily controlled by the air/fuel ratio in the cylinder. Mixtures richer than stoichiometric produce high levels of CO and are sensitive to small changes in air/fuel ratio. Mixtures at stoichiometric and leaner produce little CO emissions and are relatively insensitive to air/fuel ratio changes [34]. Figures 10(a) and 10(b) show the CO emissions at gasoline MBT ($\Phi=1$) and fuels' MBT, respectively. From Fig. 10(a), it is observed that CO emissions for the alcohol fuels are consistently higher than that for gasoline. ABE(3:6:1) has the highest CO emissions, while that for ABE(6:3:1) is very close to that of ABE0. This is also reflected in the increased BTE exhibited by ABE(6:3:1) and the reduced BTE of ABE(3:6:1). ABE(3:6:1), ABE(5:14:1) and n-butanol show relatively similar CO emissions. From Fig. 10(b), it is seen that the CO emissions are slightly improved at the fuels' MBTs; however, increasing butanol content results in increasing CO emissions at $\Phi=1$. Decreases in both CO post-flame oxidation and engine power output are responsible for this.

It appears that ABE(6:3:1) can improve both power output and emissions behavior. Also, ABE(6:3:1) and gasoline show roughly similar emissions behavior. This makes sense as fuels with similar CA50 for a given start of ignition (SOI) are known to show similar emissions

behavior [54]. The effect of acetone is to keep ABE(6:3:1)'s phasing close to that of gasoline, while improving efficiency and reducing emissions.

Figures 11(a) and 11(b) show the NO_x emissions at gasoline MBT ($\Phi=1$) and fuels' MBT, respectively. From Fig. 11(a), no major changes are seen in NO_x emissions between gasoline, ABE(6:3:1) and ABE(3:6:1). With increasing butanol content, NO_x decreases slightly for ABE(5:14:1) and n-butanol. This agrees with the results in [26, 28, 31] where no major changes in NO_x emissions were observed for blends of gasoline and n-butanol (i.e., relatively lower butanol content). Similar behavior is observed across the equivalence ratio range tested, as shown in Fig. 11(b), where n-butanol consistently shows lowest NO_x emissions. Combustion approaches close to completion under stoichiometric conditions, which leads to higher peak combustion temperatures, and hence the highest NO_x emissions are seen at stoichiometric conditions with a decrease as the equivalence ratio gets relatively richer or leaner. The lower NO_x measured with increasing butanol is in accordance with higher vaporization heat, typical of oxygenated compounds, giving lower air/fuel mixture temperature at intake and, as a consequence, lower peak temperatures in the combustion chamber. The mixture temperature at intake valve closing, measured during experiments, is progressively lower with alcohol increasing in blends [28].

Conclusions

Blends of Acetone-Butanol-Ethanol (ABE) (6:3:1, 3:6:1 and 5:14:1 vol. % ratio), were combusted in a PFI SI engine with ethanol-free gasoline and pure n-butanol as baselines for comparison, and the combustion performance and emission behavior were analyzed. The fuels were combusted at 1200 RPM, and 3 bar and 5 bar BMEP (medium load) and measurements such as brake torque and emissions were made along with in-cylinder pressure data. Each fuel was tested across a range of equivalence ratios, from lean to rich. The blends' MBT timings were also found at the tested conditions.

In-cylinder pressure data showed that under gasoline MBT, the peak pressure of the ABE blends was slightly higher than that of gasoline, while ABE(3:6:1) also showed an advanced peak relative to gasoline. Under their MBT timings, ABE(3:6:1), ABE(5:14:1) and n-butanol

showed similar phasing, whereas ABE(6:3:1) and gasoline behaved similarly. Increasing n-butanol showed advanced combustion phasing (CA50) which is attributed to the increase in laminar flame speed due to butanol's higher flame speed. ABE(6:3:1), however, displayed combustion phasing similar to that of ABE0, although slightly retarded. Similar trends were observed at all equivalence ratios.

The BSFC increased steadily with the ABE blends, due to the lower energy content of the blends and thus more fuel was required to match the power output of gasoline. The BTE of ABE(6:3:1) was higher than that of ABE0, whereas other fuels showed similar or slightly higher efficiency relative to ABE0.

Emission data showed that increasing n-butanol showed increased HC emissions and increased CO emissions, due to incomplete combustion. On the other hand, ABE(6:3:1) showed reduced HC emissions. CO for ABE(6:3:1) was roughly the same as that of ABE0. With respect to NO_x, no major changes were observed between gasoline and ABE(6:3:1), ABE(3:6:1), which was supported by the minor variations in exhaust gas temperature.

Acetone addition could improve combustion quality significantly. In summary, the typically produced fermentation product, i.e., ABE(3:6:1) may not be ideal for engine combustion in terms of thermal efficiency. Fermentation products with higher acetone content, such as ABE(6:3:1) would be much better suited as alternative fuels for SI engines. This study further affirms the potential of ABE to be used as an alternative fuel in SI engines. Future work will involve testing of ABE fuels containing water.

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Definitions/Abbreviations

ABE	Acetone-Butanol-Ethanol
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ASOI	After Start of Ignition
ATDC	After Top Dead Center
BMEP	Brake Mean Effective Pressure
BSFC	Brake Specific Fuel Consumption
BTDC	Before Top Dead Center
BTE	Brake Thermal Efficiency
CAD	Crank Angle Degree
COV	Co-efficient of Variation
CO	Carbon Monoxide
DI	Direct Injection
ECU	Engine Control Unit
EGT	Exhaust Gas Temperature
ETE	Engine Thermal Efficiency
FID	Flame Ionization Detection
GE BSFC	Gasoline Equivalent BSFC
HC	Hydro-Carbons
HoV	Heat of Vaporization
ICE	Internal Combustion Engine
IMEP	Indicated Mean Effective Pressure
LFS	Laminar Flame Speed
LHV	Lower Heating Value
MAP	Manifold Air Pressure
MFB	Mass Fraction Burnt
NDIR	Non-Dispersive Infrared
NO _x	Nitrogen Oxides
PFI	Port-Fuel Injection
RON	Research Octane Number
RPM	Revolutions per minute
SI	Spark Ignition
SOI	Start of Ignition

THC	Total Hydro-Carbons
UHC	Unburned Hydro-Carbons
Φ	Equivalence Ratio
λ	Lambda (= 1/ Φ)

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Table 1. Engine Specifications

Displaced volume	575 cc
Stroke	90.1 mm
Bore	90.3 mm
Connecting Rod	150.7 mm
Compression ratio	9.6:1
Number of Valves	4
Fuel Injection	Port Fuel Injection

Table 2. Properties of Individual Fuels [29, 44]

Parameter	Gasoline	Ethanol	Butanol	Acetone
Chemical Formula	C_4-C_{12}	C_2H_5OH	C_4H_9OH	C_3H_6O
Composition (C,H,O) (Mass %)	86, 14, 0	52, 13, 35	65, 13.5, 21.5	62, 10.5, 27.5
Lower Heating Value (MJ/kg)	43.4	26.8	33.1	29.6
Density (kg/m^3)	715-765	790	810	790
Energy Density (MJ/l)	32.20	21.17	26.81	23.38
Octane Number ((R+M)/2)	90	100	87	117
Boiling Temperature ($^{\circ}C$)	25-215	78	118	56.2
Latent Heat of Vaporization ($25^{\circ}C$) (kJ/kg)	380-500	904	716	518
Self-Ignition Temperature ($^{\circ}C$)	~300	420	343	465
Stoichiometric Air/Fuel Ratio	14.7	9.0	11.2	9.5
Laminar Flame Speed (LFS) (cm/s)	~33 ^a	~39 ^a	~48 ^b	~34 ^c
Mixture Calorific Value (MJ/m^3)	3.72	3.85	3.82	4.04
Ignition Limits in Air (vol. %) [Lower-Upper]	0.6 - 8	3.5 - 15	1.4 - 11.2	2.6 - 12.8
Solubility in Water at $20^{\circ}C$ (ml/100 ml H_2O)	<0.1	Fully Miscible	7.7	Fully Miscible
^a p = 1 atm, T = 325 K [29]; ^b p = 1 atm, T = 343 K [44]; ^c p = 1 atm, T = 298 K [44]				

Table 3. Properties of Fuel Blends (Calculated)

Fuel Type	Specific Gravity	Lower Heating Value (LHV) (MJ/kg)	Energy Density (MJ/l)	Stoichiometric Air/Fuel Ratio	Butanol Vol. %	Acetone Vol. %
Gasoline	0.739	43.44	31.68	14.65	0	0
ABE(6:3:1)	0.796	30.3	24.1	9.94	30	60
ABE(3:6:1)	0.802	31.45	25.22	10.36	60	30
ABE(5:14:1)	0.804	31.93	25.67	10.64	70	25
n-Butanol	0.810	33.1	26.81	11.06	100	0

Table 4. Test Conditions

Engine Speed	1200 RPM
Load (BMEP)	3 bar, 5 bar
Equivalence Ratio	0.83 – 1.25
Spark Timing	Gasoline MBT, Fuel MBT
Fuel Pressure	3 bar

Table 5. Fuel MBTs

Fuel	MBT @ 3 bar BMEP	MBT @ 5 bar
Gasoline	18° BTDC	24° BTDC
ABE(6:3:1)	17.7	23.6
ABE(3:6:1)	17.6	23.5
ABE(5:14:1)	17.5	23.4
n-Butanol	17.4	23.3