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Combustion characteristics of Butanol/n-Heptane blend fuels in an HCCI engine

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1. Abstract

Application of alternative fuels in high-efficiency combustion modes offers the potential to reduce demand for petroleum resources. Combustion of Butanol as a promising alternative for both diesel/gasoline fuels is studied in Homogeneous Charge Compression Ignition (HCCI). Combustion characteristics, efficiency, and emissions of Butanol blends in a single cylinder experimental engine are analyzed at over 50 lean-boosted steady-state operating points. The engine is run with several blends of Butanol and n-Heptane at different octane values at a range of equivalence ratios. The engine load range is measured and changes of HCCI heat release and ignition timing to variation of Butanol volume percent are investigated.

2. Introduction

Ultra low NOx and negligible PM emissions combined with a modest fuel economy penalty make HCCI a promising alternative to conventional spark/diesel combustion [1]. HCCI has a high fuel flexibility and can be applied for a wide range of fuels with different octane/cetane numbers. Alcohols, biomass based fuels, are the most important renewable, alternative fuels for internal combustion engines [2]. Butanol is a promising alcohol that combines the advantages of high energy density, high cetane number, strong hydrophobic properties and good atomization characteristics compared to other known oxygenate fuels like Ethanol. These make Butanol a viable alternative either single or blended with conventional based fuels to help decrease the demand for non-renewable petroleum as fuels.

Butanol can be produced from a wide range of biomass resources such as straw, corn husk, fiber plants and other agricultural wastes [3]. Production of Butanol is based on the anaerobic digestion of biomass by bacteria and does not consume food resources unlike some other alternative fuels. In addition transportation of Butanol via pipeline is relatively straight forward compared to Ethanol which is fully miscible in water [4] and compared to Ethanol, Butanol is not corrosive to metal parts [3].

Different studies [2, 4, 3] show promising results when Butanol is blended with either diesel fuel or gasoline. In diesel, fuel atomization characteristics are improved by blending the fuel with Butanol because of lower viscosity, density, and surface tension of butanol compared to diesel fuel [2]. In SI, Butanol blended gasoline requires no necessary modifications in the engine fuelling system to burn the fuel [3]. Compared to Ethanol blended gasoline, Butanol blended gasoline contains about 15% more energy density [4].

Several studies [3, 4, 5] have investigated Butanol as a fuel for compression ignition engines or spark ignition engines, however little research [6] has been done to investigate Butanol as a fuel for HCCI engines. The purpose of this paper is to experimentally explore the combustion properties of a Butanol blended with n-Heptane fuel in HCCI combustion. The emission and ignition timing characteristics at different operating conditions are measured and compared.

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3. Engine Setup

The experimental single cylinder Ricardo Hydra Mark III engine fitted with a VVT Mercedes E550 cylinder head is depicted in Figure 1 and the configuration of the engine is given in Table 1. Two separate fuel systems with 3-bar fuel pressure are used to ensure injection on closed intake valves. One fuel system is used to inject n-Heptane and the other is used to inject Butanol. The separate flow rate control of each of these two fuels allows any desired Butanol Volume Percentage (BVP) to be obtained. Both n-Heptane and Butanol injectors are aimed directly at the back of the intake valves. The fresh intake air entering the engine is first passed through a laminar air-flow meter for flow rate measurement. Then, a supercharger driven by a variable speed electric motor adjusts the intake manifold pressure and a 600W electrical band-type heater sets the mixture temperature to a desired value using a closed-loop controller. Finally the exhaust gases exiting the cylinder are sampled for emission analysis. A 5-gas emissions test bench is used to collect emissions data – NO_x is measured with 1 ppm resolution using Horriba CLA-510SS emission analyzer and Horiba FIZ-510 emission analyzer is used to measure HC with 10 ppm resolution. CO is measured with 0.01% resolution using Siemens ULTRAMAT6 emission analyzer.

The engine out Air Fuel Ratio (AFR) value is measured by ECM AFRecorder 1200 UEGO and intake temperature is measured with 2°C resolution using a K-type thermocouple positioned in the intake manifold before the charge entering into the cylinder. The exhaust temperature is measured using a 1/32″ sheathed J-Type thermocouple which is placed in the exhaust as close as possible to the exhaust valve. Crank angle measurement with 0.1° resolution, is done using a BEI optical encoder connected to the crankshaft on the front of the engine. Measurement of the cylinder pressure is done using a Kistler water-cooled ThermoCOMP (model 6043A60) piezoelectric pressure sensor that is flush mounted in the cylinder head.

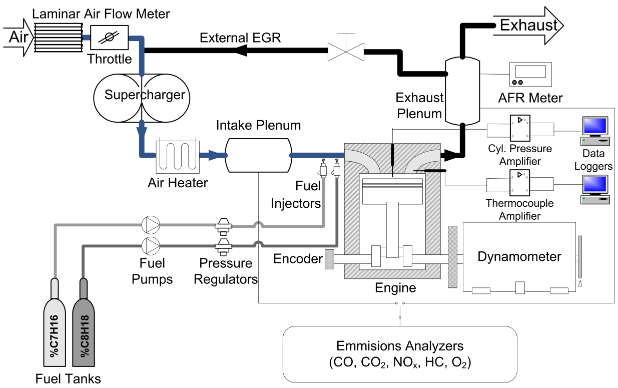

Figure 1: Single cylinder engine testbench schematic

Table 1: Single cylinder engine configuraton.(IVO: Intake Valve Opening, IVC: Intake Valve Closing, EVO: Exhaust Valve Opening, EVC: Exhaust Valve Closing, aBDC: after Bottom Dead Center)

Parameters	**Values**
Bore × Stroke [mm]	97 × 88.9
Compression Ratio	12
Displacement [L]	0.653
Number of Valves	4
IVO, IVC [aBDC]	151°, 21°
EVO, EVC [aBDC]	−100°, 130°

Table 2: Operating conditions range of the 57 steady-state data points

Variables	**Values**
Fuel (BVP[a])	6 - 48.5
Engine speed [rpm]	1021
Intake manifold temperature [^oC]	80 - 82
Equivalence ratio	0.3 - 0.43
Intake manifold pressure [kPa]	122 - 125
EGR [%]	0
$T_{coolant}$ [^oC]	69 - 71

[a]Butanol Volume Percentage

Table 2 details the experimental conditions of 57 steady-state points used in this study. The amount of injected fuel energy and the fuel mixing ratio (Butanol to n-Heptane) is varied while all other engine variables are maintained constant. Pressure traces from 300 consecutive engine cycles with 0.1 CAD resolution are recorded for each engine test. The start of combustion in HCCI is determined using the third derivative of the pressure trace criteria [7]. The Rassweiler method [8] is used to calculate CA_{50} which is the crank angles for 50% burnt fuel. The net heat release rate is determined using the usual heat release method [9] that applies the first law analysis on the engine charge assuming ideal gas properties. Other engine variables are measured at a constant sample rate of 10Hz during each test.

4. Results

Butanol can be blended with diesel like fuels to produce viable alternative fuels [10]. Here, experimental results of HCCI combustion using Butanol blended with a diesel like fuel (n-Heptane) are analyzed. Pressure-related parameters (IMEP and in-cylinder pressure), combustion-related parameters (ignition timing, heat release rate and thermal efficiency) and recorded engine-out emissions are used to study characteristics of an HCCI engine operating with this Butanol blend fuel.

Engine operation in HCCI mode is limited between misfire and excessive knock. Maintaining all engine operation parameters constant except BVP and mixture equivalence ratio (ϕ) the misfire/knocking limits HCCI operating points are measured and the results are shown in Figure 2. The engine can run in HCCI mode with BVP up to 48.5% and the operating ϕ range decreases for higher BVP. For all operating points IMEP (the engine load) is low ranging between 2.5 and 4 bar. Low IMEP is due to ultra lean operation and also lower energy content of butanol fuel compared to n-Heptane. Higher IMEP at higher BVP occurs because of a richer mixture (higher ϕ) at higher BVP results in injected fuel energy that is larger at higher BVP.

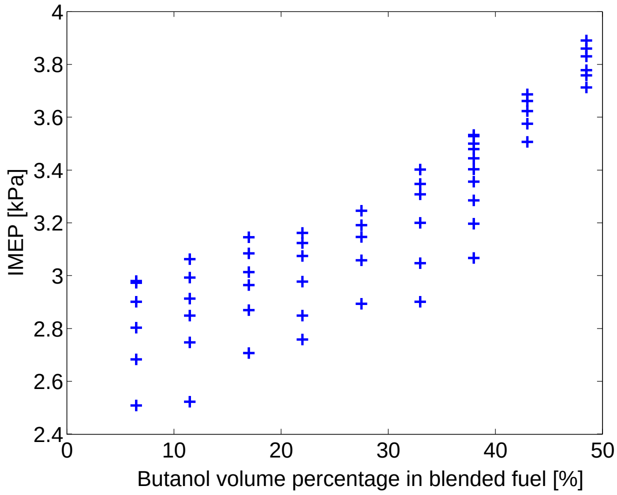

Figure 2: IMEP versus butanol volume percentage for all the data points in Table 2.

Indicated thermal efficiency as a function of ϕ for 5 BVP blends is shown in Figure 3. The thermal efficiency goes up to 37% and it is typically higher at higher BVP where richer air-fuel mixture is used. Higher thermal efficiency at higher BVP is partly due to HCCI combustion timing which is plotted in Figure 4-a. A higher thermal efficiency in Figure 3 is typically observed when CA50 occurs between 5 and 10 CAD aTDC. This can be seen by comparing the thermal efficiency of data points with BVP of 38% with those of 6.5% or 17%. Also thermal efficiency drops when CA50 retards after 10 CAD aTDC for data points with BVP of 48.5%. When the combustion occurs too late, the piston expansion rate is higher and this counteracts the temperature rise from combustion by the cooling from piston expansion. Thus the work output is reduced by releasing energy partway down the expansion stroke with the potential of partial burn and misfired cycles.

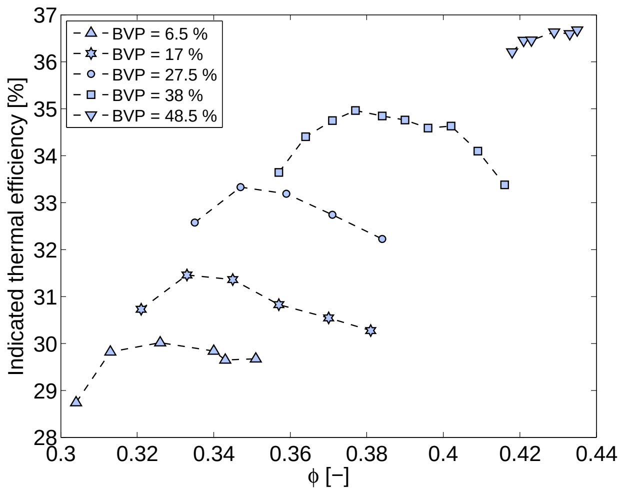

Figure 3: Variation of indicated thermal efficiency versus equivalence ratio at different Butanol Volume Percentage (BVP). (1021 RPM, T_m= 80±0.5 C, P_m= 122±0.5 kPa)

The thermal efficiency does not vary substantially ($\simeq$1-1.5%) in Figure 3 when the equivalence ratio varies, however CA50 changes significantly by changing the equivalence ratio as observed in Figure 4. The variation of CA50 versus ϕ is almost linear as seen in Figure 4-a. The linear slope approximation of $\Delta CA50/\Delta\phi$ is shown in Figure 4-a and indicates CA50 is more sensitive to ϕ variations at a higher BVP. This is partly because HCCI combustion is more sensitive to intake charge (e.g. ϕ or temperature variations) for late ignitions as discussed in [11, 12].

Pressure trace and heat release rate for the data points with a constant BVP of 27.5% from Figure 4 are shown in Figure 5. HCCI combustion of Butanol blend fuel exhibits a two-stage combustion. The results in Figure 5 show that the first stage of combustion (low temperature reactions) is almost insensitive to ϕ variations as no significant

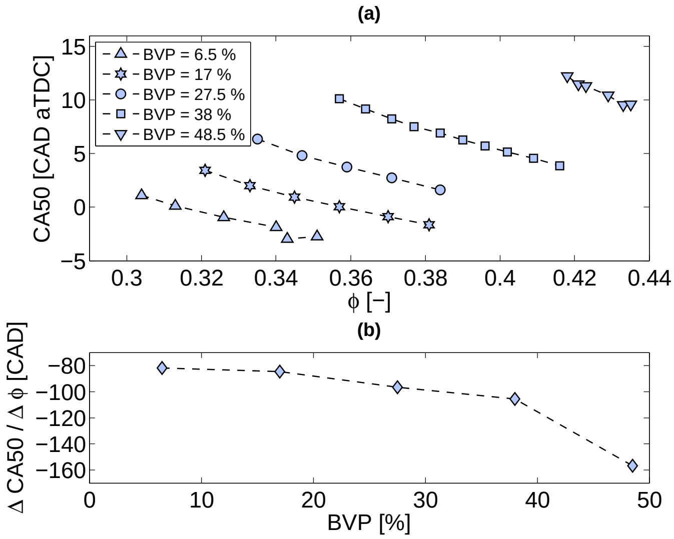

Figure 4: Variation of CA50 versus equivalence ratio at different Butanol Volume Percentage (BVP). (Data points are the same as those in Figure 3.)

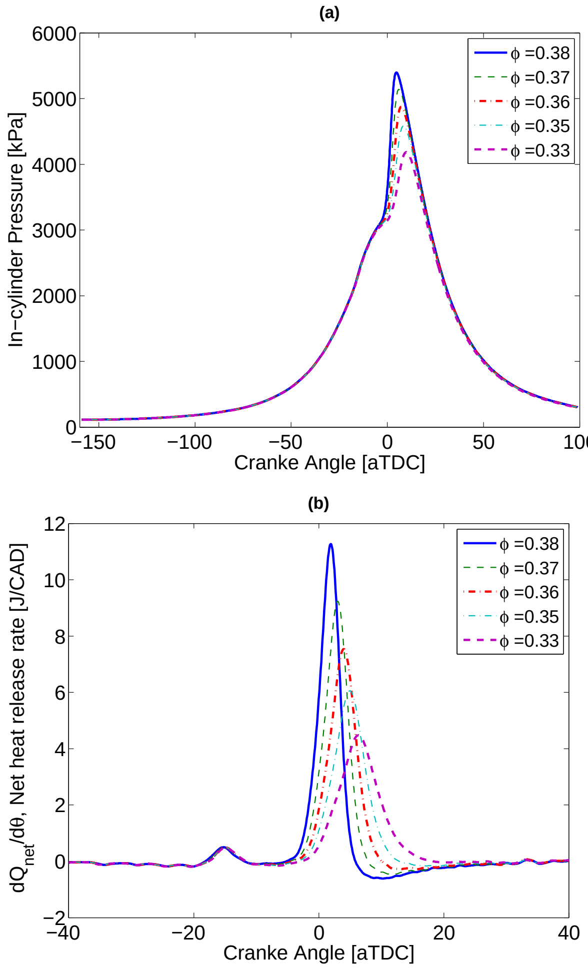

Figure 5: Variation of pressure trace and heat release versus equivalence ratio at a constant BVP of 27.5 % based on the data points from Figure 4. (1021 RPM, T_m= 80 C, P_m= 122 kPa)

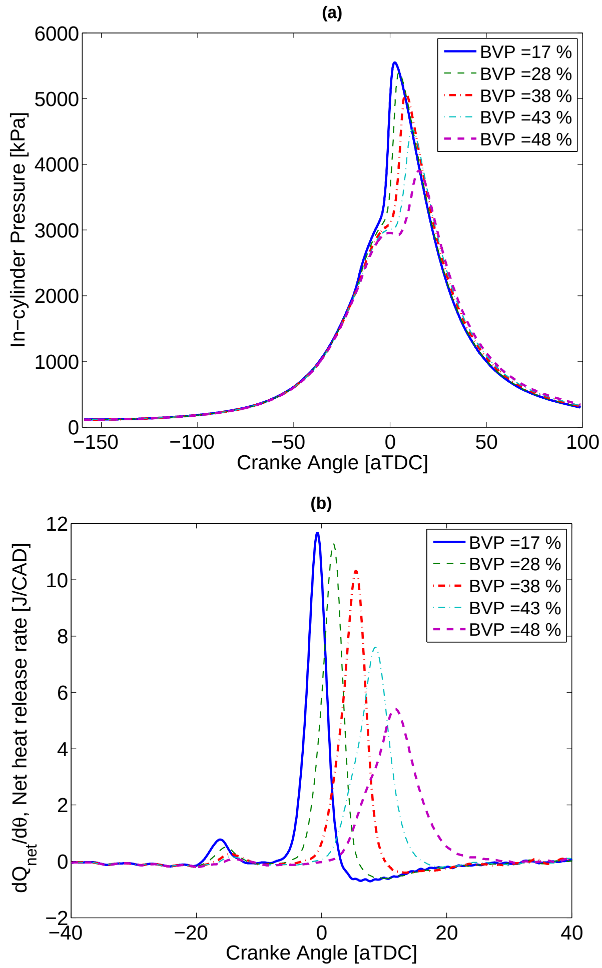

Figure 6: Variation of pressure trace and heat release rate versus BVP at conditions with the same fuel injection energy of 0.662±0.005 kJ. (1021 RPM, T_m= 80 C, P_m= 122 kPa)

change is observed in the amount and location of heat release as ϕ is varied. But the second stage of combustion is very sensitive to variations of ϕ (high temperature reactions). As expected, the main heat release is delayed and the total heat release decreases when ϕ is reduced.

In-cylinder pressure trace and the heat release rate are shown in Figure 6 for the data points with the same fuel injection energy for 5 BVP ratios. Since Butanol has less energy content than n-Heptane, more fuel is injected for higher BVP to provide consatnt fuel injection energy compared to the lower BVP cases. Here, both first and second combustion stages are influenced by changing BVP. Both combustion stages retard and total heat release decreases as BVP increases. This results in lower peak pressure and lower IMEP at higher BVP for a same fuel injection energy. Results in Figures 4-6 show changing BVP and ϕ can be an effective way to adjust HCCI combustion timing.

Emissions measurements plotted versus CA50 – ϕ is varied for different BVP values – are shown in Figure 7. A strong relation is observed between CA50 and amounts of THC, CO and NOx emissions. Both THC and CO emissions generally increase with retarding CA50 and this can be explained by higher in-cylinder temperature during HCCI combustion at earlier ignitions. High combustion temperature which is well characterized by the peak cylinder cycle temperature is the main requirement for CO and THC oxidation in HCCI engines [1, 13, 14]. However, higher peak combustion temperature at earlier ignitions causes higher NOx emissions as observed in Figure 7-c. Both NOx and CO emissions are lowest at the highest BVP in Figure 7. This is because when BVP increases the combustion occurs late which results in a lower peak temperature but a higher exhaust temperature. Thus Nox decreases due to lower peak temperature and CO decreases because of possibility for post oxidation at higher gas temperature in expansion and exhaust strokes. However measured THC in Figure 7-a is higher for data points at higher BVP. This is attributed to higher equivalence ratio (more fuel concentration) for the data points at higher BVP as shown in Figure 4-a.

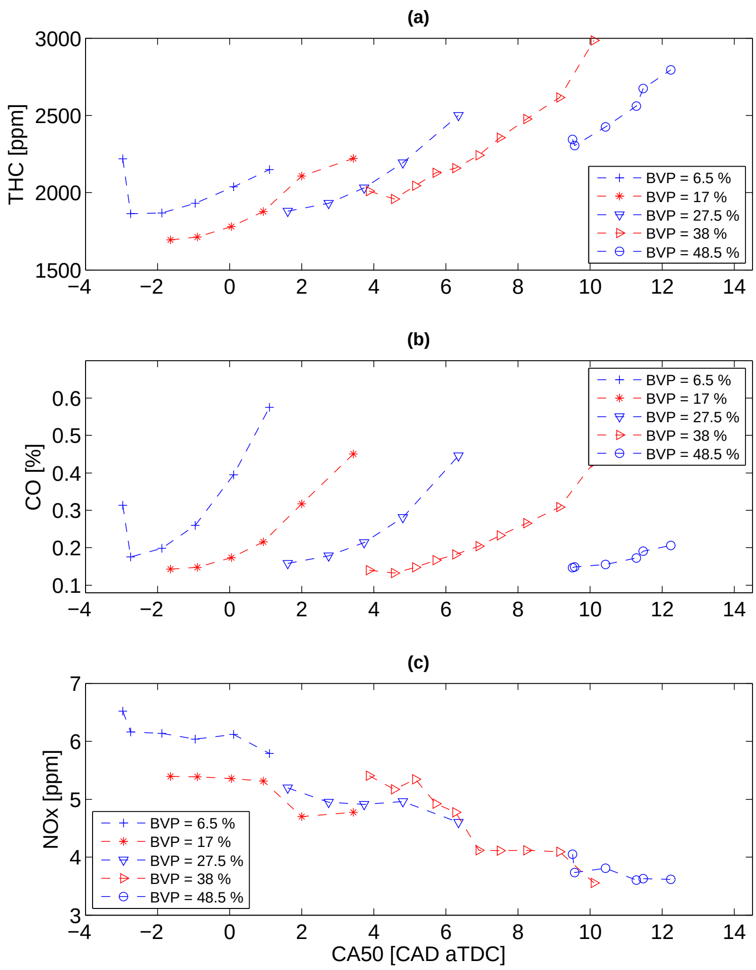

Figure 7: Variation of THC, CO and NOx emissions versus CA50 at different BVPs. (Data points are the same as those in Figure 3.)

5. Summary and Conclusions

An experimental study of an HCCI engine using a blend of Butanol and n-heptane is performed at one engine speed. The volume percent of Butanol and the equivalence ratio (ϕ) are varied while holding all other engine inputs constant and 57 HCCI operating conditions (between misfire and knock) are obtained. HCCI operation is possible with Butanol blends up to 48.5% for the engine and conditions studied. The thermal efficiency ranges from 28% to 37% with higher thermal efficiency occurs at higher BVP. Location of ignition timing affects the engine thermal efficiency and a 5-degree window of CA50 (5-10 CAD aTDC) is found to provide the highest thermal efficiency. Varying BVP is an effective means to adjust HCCI ignition timing since ignition timing is delayed by increasing BVP. A higher sensitivity of HCCI ignition to variation of ϕ is observed at higher BVP values. The HCCI combustion of Butanol and n-Heptane blends exhibits a two-stage heat release where the heat release from Low Temperature Reactions (LTR) is more substantial at a lower BVP with a same fuel injection energy. It is found that LTR is almost insensitive to ϕ variations but High Temperature Reactions (HTR) are very sensitive to variations of ϕ. The location of ignition timing (CA50) significantly influences engine-out emissions. Higher T_{exh} and higher CO and THC emissions but lower NOx emission are observed when shifting the combustion from early ignitions to late ignitions after TDC.

Acknowledgments

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