

Impact of Fuelling Techniques on Neat n-Butanol Combustion and Emissions in a Compression Ignition Engine

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ABSTRACT

This study investigated neat n-butanol combustion, emissions and thermal efficiency characteristics in a compression ignition (CI) engine by using two fuelling techniques - port fuel injection (PFI) and direct injection (DI). Diesel fuel was used in this research for reference. The engine tests were conducted on a single-cylinder four-stroke DI diesel engine with a compression ratio of 18.2 : 1. An n-Butanol PFI system was installed to study the combustion characteristics of Homogeneous Charge Compression Ignition (HCCI). A common-rail fuel injection system was used to conduct the DI tests with n-butanol and diesel. 90 MPa injection pressure was used for the DI tests. The engine was run at 1500 rpm. The intake boost pressure, engine load, exhaust gas recirculation (EGR) ratio, and DI timing were independently controlled to investigate the engine performance.

The pressure rise rate of the DI could be lowered easily by retarding the injection timing without using EGR, whereas a very high EGR ratio was required for the PFI to reduce the pressure rise rate to the same level of the DI. Soot and nitrogen oxides (NOx) emissions for both fuelling techniques were at a very low level compared to those of base diesel, but the NOx for DI was slightly higher than that for PFI at 0% EGR. However, by using EGR, NOx was reduced to around 20 ppm, similar to the PFI level. Indicated thermal efficiency of the DI was greater than that of the PFI with the same NOx and cylinder pressure rise rate.

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INTRODUCTION

n-Butanol is a type of alcohol with a molecular formula of C_4H_9OH . It can be made from biomass sources, such as starch of corn or wheat, and is then called bio-butanol [1, 2, 3]. In recent years, cellulose based n-butanol production techniques have been developed [1, 2, 3] which are independent of food security concerns. Hence, use of n-butanol as a fuel can contribute to the reduction of carbon dioxide (CO_2) emission, and address energy security concerns.

In addition, n-butanol has more suitable fuel properties (shown in Table 1) for compression ignition (CI) engines in practical use [1, 2, 3] than ethanol which is one of the most commonly used biofuels. Compared to ethanol, the lower heating value (LHV) of n-butanol is higher, potentially resulting in lower fuel consumption. The tendency of water absorption is lower, and thus corrosion problems are less severe. n-Butanol also has a higher kinematic viscosity than ethanol. As a result, better wear characteristics between the moving parts of the fuel injection system may be expected. The fuel-borne oxygen (21.6% by weight) can help suppress soot formation during combustion.

Consequently, n-butanol shows considerable promise as a future biofuel for CI engines, and many researchers have been performing studies related to CI engines fueled with n-butanol [4, 5, 6, 7, 8, 9, 10, 11, 12, 13, 14, 15]. Miers et al. [5] studied the impact of n-butanol

and diesel blend fuels on the emissions of a diesel passenger car using a chassis dynamometer. They reported that an 80% reduction in filter smoke number was observed on a steady-state test when using a 40% n-butanol blend by volume. Rakopoulos et al. [7] investigated the effect of the n-butanol blend ratio on performance and emissions of a high speed direct injection (DI) diesel engine. They reported that smoke density was significantly reduced with increasing percentages of n-butanol in the blend. Nitrogen oxides (NOx) were also slightly reduced with the blended fuel. Valentino et al. [11] evaluated the potential of n-butanol diesel blends under premixed low temperature combustion using a turbocharged DI diesel engine. They reported that a 20% and 40% blend by volume of n-butanol and diesel enabled smoke emissions to be drastically reduced. From the literature review above, increase of n-butanol blend ratio with conventional diesel fuel appears to have the potential to considerably reduce smoke emissions. Therefore, it may be anticipated that smoke emissions can be further reduced if neat n-butanol is used.

Over the last few years, several research reports related to the combustion in a CI engine fuelled with neat n-butanol have been published. Yanai et al. [12, 13] reported neat n-butanol combustion characteristics in a direct injection diesel engine. Their results indicated that very low NOx and soot emissions were achieved while the thermal efficiency was similar to diesel. On the other hand, He et al. [14] investigated neat n-butanol combustion in an HCCI

(Homogenous Charge Compression Ignition) engine with variable valve timing and lift control. They reported that steady combustion and low NO_x were achieved using a negative valve overlap strategy. Maurya et al. [15] investigated performance of neat n-butanol HCCI using a pre-heating technique. Intake air temperature and lambda (λ) were varied, and the engine performance was compared to that of gasoline HCCI.

Thus, in a CI engine, there are two possible fuelling techniques of neat n-butanol: port fuel injection that generally induces the HCCI type of combustion [16, 17, 18] (referred to as PFI in this context), and direct fuel injection (DI). For the PFI case, a port fuel injection system needs to be installed on an engine, although very low NO_x and soot emissions can be expected because of the lean and homogeneous charge resulting in a low combustion temperature [16, 17, 18]. On the other hand, for the DI case, no major engine modification is required [12, 13]. The differences in the engine performance between PFI and DI are yet to be investigated in detail, and need to be identified in order to understand the suitable combustion mode for neat n-butanol in a CI engine.

Therefore, the objective of this study was to understand the difference between the two fuelling techniques (PFI and DI) of neat n-butanol in a CI engine in terms of combustion, emissions and indicated thermal efficiency.

EXPERIMENTAL SETUP

Table 2 lists the main specifications of the test engine. A four-stroke water-cooled four-cylinder diesel engine was used. Figure 1 shows the experimental configuration. As a single cylinder test setup, the first cylinder was used for research, and the remaining three cylinders were used to motor the first cylinder. The fuel injection, intake and exhaust system of the first cylinder were separated from the rest of the engine to gain independent control of injection duration, injection timing, intake boost pressure, and exhaust gas recirculation (EGR) ratio. The intake boost pressure was controlled by an external air compressor.

Two identical gasoline port fuel injectors were installed near the intake port for the n-butanol PFI tests. The test fuel was pressurized by using an in-line feed pump designed for gasoline use. The port fuel injection timing and duration were controlled by an independent injection control system. An in-house common-rail direct injection system was used to conduct the DI n-butanol combustion studies. The high pressure pump was from a diesel engine of a commercial vehicle and it had a fuel-lubricated system which is generally very sensitive to fuel lubricity. As the viscosity of n-butanol (shown in Table 1) is lower than that of diesel fuel, the wear rate of the moving parts such as the plunger in the high pressure pump was a concern. To minimize this wear, a fuel lubricity improver for low sulfur diesel (Innospec OLI-9070x) was added to the n-butanol fuel (concentration of 500 ppm by volume). The boiling point of n-butanol is 117.5 °C, which is much lower than that of diesel fuel. To avoid vaporization of the n-butanol fuel in the high pressure pump, two fuel coolers were installed to reduce the fuel temperature.

Table 1. Fuel properties

	Diesel Fuel	Ethanol	n-Butanol
Formula	$C_nH_{1.8n}$	C_2H_5OH	$n-C_4H_9OH$
Density [kg/m ³]	846	780	810
Oxygen Mass [%]	0.0	34.8	21.6
Octane Number	25	110-115	87
Cetane Number	46.5	2-11	17-25 [2, 19, 20]
LHV [MJ/L]	36.8	21.1	26.8
Stoic. A/F ratio	14.6	9.0	11.1
Boiling Point [°C]	288.0-339.0	78.0	117.5
$Q_{evaporation}$ [kJ/kg]	316.6	728.2	595.0
Kinematic viscosity [mm ² /s]	3.5 @30°C	1.5 @20°C	2.5@30°C 3.6@20°C

Table 2. Main specifications of the test engine

Engine model	-	4 stroke 4 cylinder direct injection diesel, Ford Duratorq
Displacement	cm ³	1998
Bore × stroke	mm	86 × 86
Compression ratio	-	18.2 : 1
Peak cylinder pressure	bar gauge	180
Injection system	-	Common-rail
Direct injection Injector	-	Solenoid type, 6 holes, nozzle hole diameter 0.13 mm, umbrella angle 155 °

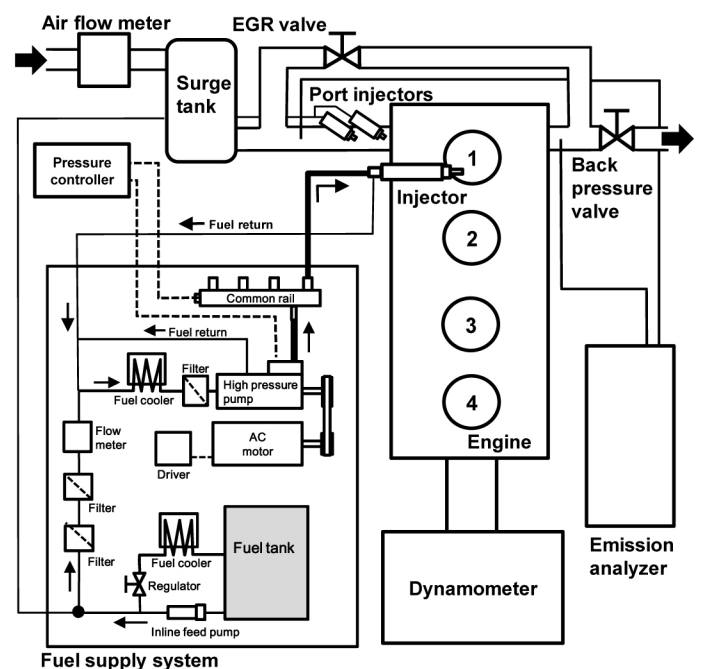


Figure 1. Schematic diagram of the experimental configuration

A piezoelectric pressure transducer (AVL GU13P) was installed in the first cylinder to acquire the cylinder pressure for the combustion analysis. The cyclic cylinder pressure was sampled for 200 consecutive engine cycles, and each cylinder pressure curve was filtered by using fast Fourier transform (FFT) analysis. The average of the 200 cycles was used in this paper to present the cylinder pressure curve. The rate of heat release was calculated from the cylinder pressure using 1st law analysis [19]. The rate of heat release discussed in this paper is the net rate of heat release, which includes heat transfer. The ratio of specific heat (γ) used for the calculation was fixed to 1.37. The mass burned fraction was calculated from the accumulated rate of heat release. The start of combustion was defined as 5% of mass burned fraction (CA5), and the end of combustion was defined as 95% of mass burned fraction (CA95). The combustion duration was the period between CA5 and CA95. The ignition delay was defined as the period between the start of injection command and the start of combustion. The crank angle at 50% of mass burned fraction was denoted by CA50. In this study, only the indicated thermal efficiency was evaluated to understand the basic trends of thermal efficiency, although the brake thermal efficiency should also be considered in the comparison of fueling techniques between PFI and DI for practical use.

The emissions were evaluated by using gas emission analyzers (California Analytical Instruments). The NO_x emissions were measured using a chemiluminescence detector (CLD) and the total hydro-carbons (THC) by a flame ionization detector (FID). Although an FID generally does not completely detect oxygenated hydrocarbons [20], the THC values in this paper were not adjusted to compensate the FID response. The carbon monoxide (CO) and CO₂ were measured by a non-dispersive infrared (NDIR) detector. The oxygen (O₂) was measured by a paramagnetic analyzer (PMA) and the soot was measured by an AVL 415S smoke meter. Aldehyde emissions can be expected from n-butanol combustion since oxygenated hydro-carbon fuels generally tend to form aldehydes during the combustion process, although it was not measured in this study. The aldehyde emission trend should be investigated in the future.

The test conditions are listed in Table 3. The engine was run at 1500 rpm. The engine load was changed from idle to the load limit. The intake boost pressure was increased from naturally aspirated to 1.0 bar gauge. Intake temperature was in the range of 24–30 °C at 0% EGR (no EGR). An EGR cooler was not used, so the intake temperature increased with increasing EGR ratio. The EGR ratio was varied from 0% to 75%. For the PFI test, the injection pressure was 7.0 bar gauge, which allowed sufficient energy quantity to be supplied even under high boost conditions. The injection timing was fixed to −350 °ATDC (beginning of the intake stroke), and the duration was determined according to the load condition of the test engine. For the DI test, the injection pressure was 90 MPa gauge, which is a relatively low pressure compared to modern diesel engines, but this was chosen to prevent the wear of the moving parts

and the evaporation of n-butanol inside the high pressure pump. The injection timing was adjusted from −17 °ATDC to −12 °ATDC. The engine load, intake boost pressure, EGR ratio, and direct injection timing were adjusted to investigate the engine performance. The low sulfur diesel fuel was also used in this research as the reference for the neat n-butanol engine performance investigations.

Engine operating limits for this test were judged by unstable combustion, cylinder pressure rise rate ($dP/d\theta$) limit and maximum cylinder pressure (P_{max}) limit. The definition of unstable combustion was based on the engine load. At 5.0 bar IMEP and below, combustion was deemed unstable when the coefficient of variation of IMEP (COV_{imep}) exceeded approximately 10%. Above 5 bar IMEP, when the COV_{imep} exceeded around 5%, the combustion was regarded as unstable. The pressure rise rate limit was defined as approximately 25 bar/°CA. This limit is greater than that of typical diesel engine operation since the objective was to conduct the feasibility research for understanding the neat n-butanol combustion in a CI engine. The P_{max} limit was set to 180 bar gauge.

Table 3. Test conditions

Test fuels		-	Neat n-butanol, low sulfur diesel
Engine speed		rpm	1500
Engine load (IMEP)		bar	Idle to around 7.5
Intake boost pressure		bar gauge	0.2 ~ 1.0
Intake temperature		°C	24 ~ 30 (at 0% EGR)
EGR ratio		%	0 ~ 75%
Port injection	Injection pressure	bar gauge	7.0
	Injection timing	°ATDC	-350
Direct injection	Injection pressure	bar gauge	900
	Injection timing	°ATDC	-17.0 ~ -12.0, 0.0 (for diesel only)
Operating limit	COV_{imep}	%	5.0 (above 5.0 bar IMEP) 10.0 (at 5.0 bar IMEP and below)
	$dP/d\theta_{max}$	bar/°CA	25
	P_{max}	bar gauge	180

RESULTS AND DISCUSSION

Port Fuel Injection Combustion Characteristics

Ignition Characteristics

Since n-butanol has a low cetane number in the range of 17–25 [2, 21, 22], there is a possibility that the ignition may not occur under normal diesel operating conditions. Therefore, the study initially focused on the conditions to establish the stable ignition of neat n-butanol. In order to enhance the ignition reaction, increasing intake temperature is one of the possible ways, but the air density becomes lower,

resulting in formation of a richer mixture at the same quantity of supplied fuel. Thus, a high engine load may be difficult to achieve in this way because of the richer mixture. On the other hand, the chemical reaction rate is generally pressure dependent [23]. The ignition delay of n-butanol has been reported using a shock tube [24, 25], and the ignition delay decreased with the increase in the ambient pressure. From this knowledge, the effect of the intake boost pressure on the ignition of the PFI n-butanol was investigated.

Figure 2 shows the boost effect on cylinder pressure and rate of heat release for PFI. The engine load was 5.0 bar IMEP and 0% EGR was used. With increasing intake boost pressure, the cylinder pressure increased and the combustion phasing advanced. No visible heat release from low temperature oxidation reactions [18, 26, 27] was observed at any of the test conditions. This observation has also been noted by Zhang et al. [28]. Figure 3 illustrates the ignition timing (CA5), equivalence ratio (ϕ), and the COV_{imep} under the same conditions in Figure 2. At 0.2 bar gauge intake boost, misfire occurred because the ignition timing was too late. When the intake boost was increased from 0.3 bar gauge to 1.0 bar gauge, the ignition timing advanced by 7.0 °CA (from 6 to -1 °ATDC) even though the mixture became leaner (decreasing ϕ), and COV_{imep} reduced. The explanation for the enhanced ignition by higher boost pressures could be that the low temperature reaction during the ignition process was promoted [29].

Therefore, it was found that the boosted intake pressure can enhance the ignition process of n-butanol under the PFI condition, and that at least 0.3 bar intake boost was required to achieve stable combustion at 5.0 bar IMEP at normal intake temperature.

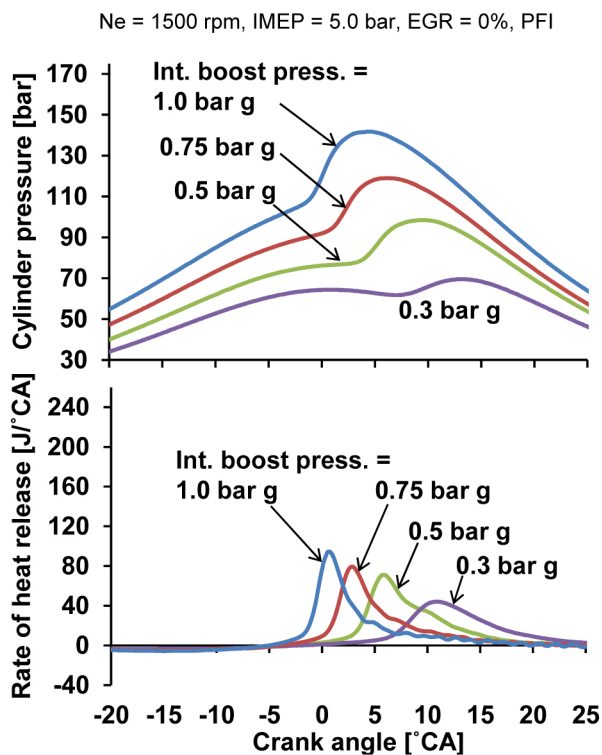


Figure 2. Effect of intake boost pressure on PFI combustion phasing

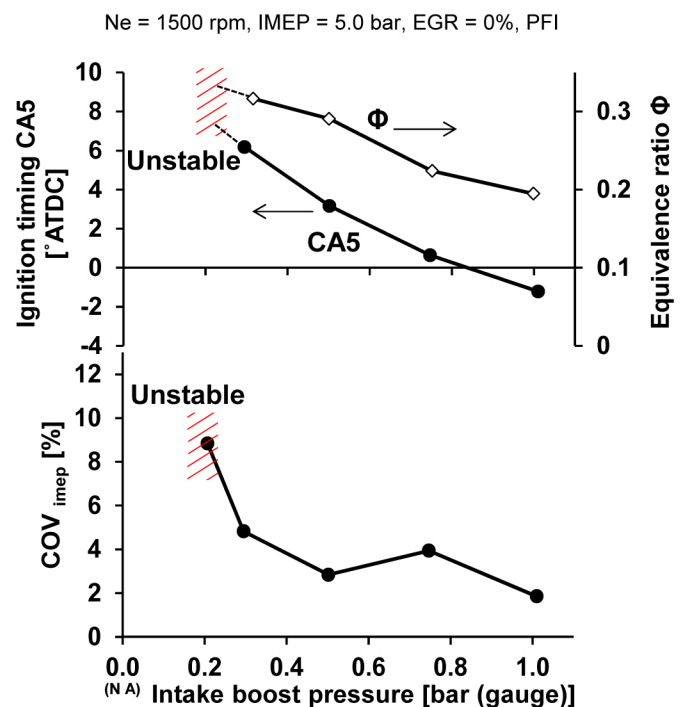


Figure 3. Effect of intake boost pressure on ignition timing, equivalence ratio and COV_{imep} in PFI

Applicable Engine Load Ranges

The combustion may become unstable when the engine load is reduced as the in-cylinder gas temperature decreases. When the engine load is increased, the load may be limited by the pressure rise rate. Therefore, the applicable engine load range was investigated.

Figure 4 shows that the variation of COV_{imep} and the pressure rise rate for the PFI case with respect to the IMEP. The intake boost pressure was set from 0.3 to 1.0 bar gauge based on the previous results. In the 0.3 bar intake boost pressure case, when the IMEP was reduced below 5.0 bar, the COV_{imep} increased sharply, and the combustion became unstable at 3.0 bar IMEP. On the other hand, when the IMEP increased, COV_{imep} was lowered while the pressure rise rate increased. When the IMEP exceeded 6.5 bar, the pressure rise rate increased very sharply until the load limit of around 7.5 bar IMEP was reached. However, when 1.0 bar gauge intake boost pressure was used, the combustion was stable even at the low load of 1.0 bar IMEP. The pressure rise rate was higher than that of the 0.3 bar boost in the range of 2.0 to 7.0 bar IMEP. The load limit was around 7.5 bar which was the same as that under 0.3 bar boost. The results of the 0.5 and 0.75 bar gauge boost pressure cases lay in-between the 1.0 and 0.3 bar gauge boost pressures cases. Hence, the high load limit was not affected by the intake boost pressure.

The impact of engine load on the combustion phasing was investigated to understand the reasons that intake boost pressure had no effect on the high load limits. Figure 5 illustrates the combustion phasing with engine load change at intake boost pressure 1.0 bar and 0.3 bar gauge. In the 1.0 bar gauge case, with increasing engine load, the peak of heat release rate and P_{max} increased gradually. When IMEP reached to 7.4 bar, the cylinder pressure rise was very steep which was near the $dP/d\theta$ limit. The P_{max} was also very high.