

Investigation of Fuel Injection Strategies for Direct Injection of Neat n-Butanol in a Compression Ignition Engine

Tadanori Yanai, Christopher Aversa, Shouvik Dev, Graham Reader, and Ming Zheng
University of Windsor

ABSTRACT

In this study, impacts of neat n-butanol fuel injection parameters on direct injection (DI) compression ignition (CI) engine performance were investigated to gain knowledge for understanding the fuel injection strategies for n-butanol. The engine tests were conducted on a four-stroke single-cylinder DI CI engine with a compression ratio of 18.2:1. The effects of fuel injection pressure (40, 60 and 90 MPa) and injection timing in a single injection strategy were investigated. The results showed that an increase in injection pressure significantly reduced nitrogen oxides (NO_x) emissions which is the opposite trend seen in conventional diesel combustion. The parallel use of a higher injection pressure and retarded injection timing was a proposed method to reduce NO_x and cylinder pressure rise rate simultaneously. NO_x was further reduced by using exhaust gas recirculation (EGR) while keeping near zero soot emissions. However, the high load limit was around 6.5 bar indicated mean effective pressure (IMEP) due to high cylinder pressure rise rate. Next, the effect of a split injection, in which the fuel delivery is divided into two injections (first and second injection ratio: 50:50) near top dead center, on the engine performance was investigated to reduce the cylinder pressure rise rate. The split injection could significantly reduce the cylinder pressure rise rate without compromising the indicated thermal efficiency, NO_x and soot emissions. It was also demonstrated that the high load limit could be extended to 11.5 bar IMEP by modifying the split injection ratio while maintaining high thermal efficiency and clean emissions.

CITATION: Yanai, T., Aversa, C., Dev, S., Reader, G. et al., "Investigation of Fuel Injection Strategies for Direct Injection of Neat n-Butanol in a Compression Ignition Engine," *SAE Int. J. Engines* 9(3):2016, doi:10.4271/2016-01-0724.

INTRODUCTION

n-Butanol is a type of alcohol with a molecular formula of C₄H₉OH. It can be made from biomass sources such as starch of corn or wheat, and cellulose, and is then referred to as bio-butanol [1, 2, 3]. In recent years, efficient and economical bio-butanol production techniques have been developed [1, 2, 3]. Hence, the use of n-butanol as a fuel can contribute to addressing a social issue such as the reduction of carbon dioxide (CO₂) emission.

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. Furthermore, like ethanol, n-butanol contains oxygen atom in its molecule. The fuel-borne oxygen (21.6% by weight) can potentially 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, 16, 17]. Numerical simulation studies related to development of reaction mechanism have also been conducted to understand fundamental n-butanol combustion in CI engines [18, 19, 20]. 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 (NO_x) 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 combustion in a DI CI engine fuelled with neat n-butanol have been published [14, 15, 16, 17]. Yanai et al. [14, 15] reported neat

n-butanol combustion characteristics in a direct injection diesel engine. They added a fuel lubricity improver to the test fuel and installed fuel coolers in order to be able to run a common rail type modern DI diesel engine using neat n-butanol. Their results indicated that the ignition delay was around three times longer than that of diesel due to a lower cetane number, and the applicable injection timing range was much narrower than that of diesel. The engine operating range was limited to low load due to a high cylinder pressure rise rate. However, not only near-zero soot emissions, but also very low NO_x emissions were achieved while the thermal efficiency was similar to that of diesel. Jeftić et al. [17] investigated the impact of a post injection on combustion and emissions of a neat n-butanol DI engine. Neat n-butanol combustion produced near-zero smoke emissions throughout the post injection sweep, and enabled the use of earlier post injection timings compared to diesel.

From the studies mentioned in the previous paragraph, the use of neat n-butanol in DI CI engines has the potential to produce low NO_x and soot emissions. However, the challenges are a low load operating range due to high cylinder pressure rise rate, and understanding the fuel injection strategy suited for the properties of n-butanol to lead to low emissions and high thermal efficiencies simultaneously. Thus, the objective of this study was to gain knowledge for understanding the fuel injection strategies for neat n-butanol DI CI engines to extend the operating load range while maintaining low emissions and high thermal efficiency.

In order to realize this objective, the study first investigated the impact of fuel injection pressure on the engine performance because it may significantly affect the spray development as well as the mixture formation process of neat n-butanol. In addition to this, the high engine load limit was explored using single injection mode. Then, the effect of a split injection technique, in which the fuel delivery is divided into two injections, on cylinder pressure rise rate was studied. Furthermore, the maximum engine load was investigated using the split injection technique with the minimum engine performance targets being 0.60 g/kWh of indicated NO_x, 0.020 g/kWh of indicated soot and 13.0 bar/°CA of maximum cylinder pressure rise rate.

EXPERIMENTAL SETUP

Test Engine and Equipment

Table 2 lists the main specifications of the test engine. A four-stroke water-cooled four-cylinder diesel engine was used for all engine tests. 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. The exhaust pressure was adjusted to be similar to the pressure of the intake by controlling the back pressure valve. This was done to prevent an increase in the engine load due to the external work input of the air compressor. When EGR was used under boosted intake condition, the exhaust pressure was adjusted to be slightly higher than that of the intake to force EGR gas into the intake manifold.

The test n-butanol fuel was only supplied to the first cylinder by an in-house common rail type high pressure injection system. The high pressure pump was from a diesel engine in a commercial vehicle (pickup truck) and it has a fuel-lubricated system which is generally very sensitive to fuel lubricity. The main fuel properties of n-butanol, ethanol and diesel fuel are listed in Table 1. As the viscosity of n-butanol [21] 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. An inline feed pump with a feed pressure of 1.2 bar to 1.4 bar gauge was used upstream of the high pressure pump.

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]	858	780	810
Oxygen Mass [%]	0.0	34.8	21.6
Octane Number	-	108	96
Cetane Number	45.0	2-11 [2, 24]	17-25 [2, 24, 25]
LHV [MJ/L]	36.2	21.1	26.8
Stoic. A/F ratio	14.4	9.0	11.1
Boiling Point [°C]	189.0-345.0	78.4	117.5
$Q_{evaporation}$ [kJ/kg]	316.6	904	582
Kinematic viscosity [mm ² /s]	2.7 @40 °C	1.5 @20 °C	2.3@40 °C [21] 3.6@20 °C [21]

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 °

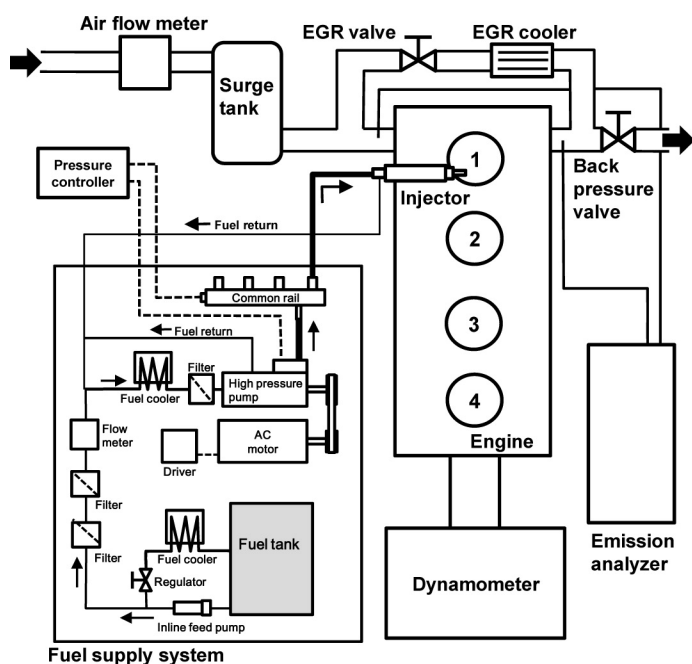


Figure 1. Schematic diagram of the experimental configuration

Analysis Technique

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 first law analysis [22]. 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 fuel injection pressures because there are mechanical losses in driving the high pressure pump.

The emissions were evaluated using gas emission analyzers (California Analytical Instruments). The NO_x emissions were measured using a chemiluminescence detector (CLD) and the total hydrocarbons (THC) by a flame ionization detector (FID). Although an FID generally does not completely detect oxygenated hydrocarbons [23], 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.

Test Conditions

The test conditions are listed in Table 3. Around 6.5 bar indicated mean effective pressure (IMEP) and 1500 rpm were used as base operating conditions because these correspond to the lower mid-level load range and mid-level speed of the test engine, and are dominant operating conditions in practice. Only the engine load was increased to investigate the high load limit. The use of a high injection pressure was of concern due to the issues of wear and vaporization of n-butanol. Hence, the maximum pressure in this study was limited to 90 MPa to prevent these issues, which is a relatively low pressure compared to modern diesel engines. On the other hand, there was a concern that an unstable fuel injection may occur if a very low injection pressure was used. Thus, the injection pressure for neat n-butanol was set to be 40, 60 and 90 MPa. It has been reported that intake heating or high intake boost pressure is required to ignite neat n-butanol in a CI engine because of the low reactivity [13, 14, 16].

Table 3. Test conditions

Test fuels		-	Neat n-butanol, low sulfur diesel
Engine speed		rpm	1500
Engine load (IMEP)		bar	Around 6.5 to 12.5
Intake boost pressure		bar gauge	1.0
Intake temperature		°C	26 ~ 27 (at 0% EGR)
Intake O ₂		%	15.5 ~ 20.9
n-Butanol (single injection)	Injection pressure	MPa gauge	40, 60 and 90
	Injection timing	°ATDC	-30 ~ -10
n-Butanol (split injection)	Injection pressure	MPa gauge	90
	First injection timing	°ATDC	-26 and -17
	Second injection timing	°ATDC	0 and 2
Diesel (single injection)	Injection pressure	MPa gauge	60, 90 and 150 (only for 9.8 bar IMEP)
	Injection timing	°ATDC	-2.0 and -1.5 (only for 150 MPa)
	Split injection ratio	-	50:50 and 40:60
Operating limit	COV _{imep}	%	5.0
	dp/dO _{max}	bar/°CA	25
	p _{max}	bar gauge	180

In this study, in order to obtain both stable ignition and a sufficient amount of air at high engine load, 1.0 bar gauge intake boost pressure was used. The average intake temperature was 26 ~ 27 °C at 0% EGR. When EGR was applied, intake O₂ concentration was controlled from 20.9% (corresponding to no EGR) to 15.5%. Intake temperature increased with increasing EGR ratio. Thus, an EGR cooler was used to suppress the excessive rise in intake temperature.

For the single injection case, the injection timing was adjusted from -30 to -10 °ATDC. For the split injection case, the first injection timing was controlled to -26 and -17 °ATDC. The second injection timing was controlled to 0 and 2 °ATDC. The first and second injection duration ratio was adjusted to 50:50 and 40:60. The reasons for these settings are explained in the Results and Discussion section. The low sulfur diesel fuel was also used in this research as a reference for the neat n-butanol engine performance investigations. For the diesel case, injection pressure was set to 60, 90 and 150 MPa, but 150 MPa was only for high engine load (9.8 bar IMEP) to suppress soot emissions. 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 unstable combustion limit was defined as when COV_{imep} exceeded 5.0%. The operating pressure rise rate limit for safety 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.

RESULTS AND DISCUSSION

Engine Performance Using Single Injection

Impact of Fuel Injection Pressure

Fuel injection pressure is commonly known as one of the most important injection setting parameters because it usually affects the spray development as well as the mixture formation process significantly. However, since n-butanol has a low cetane number [2, 24, 25], low boiling point, high latent heat and fuel-borne oxygen, the impact of fuel injection pressure on combustion and emissions may be different from those of diesel in a CI engine. Thus, this study first investigated the impact of fuel injection pressure. In this test, IMEP was around 6.6–6.9 bar, which was near the base load condition in this study, and 0% EGR was used. Injection timings for both n-butanol and diesel were fixed to -12 and -2 °ATDC respectively to only see the effect of injection pressure. With these injection timings, the CA50 was 10 ± 1 °ATDC at 60 MPa injection pressure for both fuels. Injection pressure was adjusted to 40, 60 and 90 MPa, but the 40 MPa was only for n-butanol.

Figure 2 shows the comparison of the impact of injection pressure on cylinder pressure and rate of heat release between diesel and n-butanol.

In the rate of heat release curve, the diesel case showed a premixed type combustion and a diffusion type combustion. On the other hand, n-butanol showed only a premixed type combustion, and the peak rate of heat release was much higher than that of diesel. When a higher injection pressure was used, the rate of heat release was sharply raised and the maximum cylinder pressure increased for both fuels.

Figure 3 shows the comparison of ignition delay and combustion duration between diesel and n-butanol. n-Butanol showed a similar trend to diesel, even though n-butanol had 2.5 to 3.0 times longer ignition delay and around half of the combustion duration compared

to those of diesel. The reason for shortening of the ignition delay at higher injection pressures could be due to a faster formation speed of a stoichiometric mixture region due to an enhanced fuel-air mixing process. However, in the n-butanol case, ignition delay and combustion duration were slightly less sensitive to injection pressure than that of diesel.

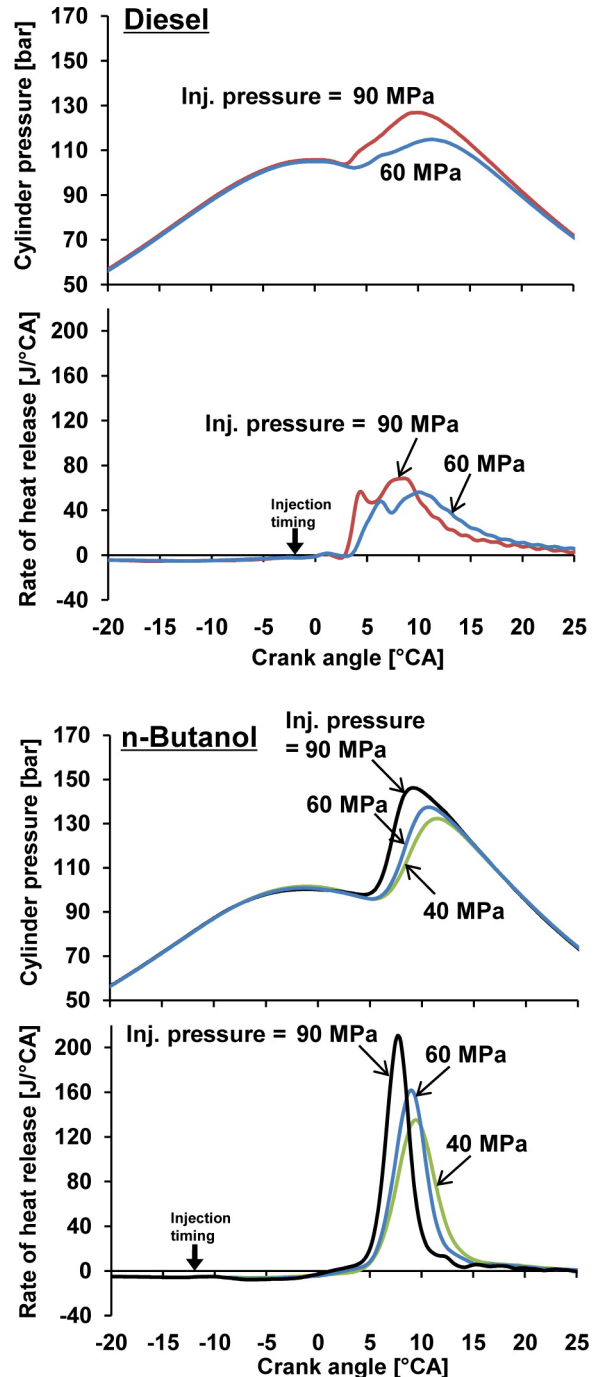


Figure 2. Comparison of impact of injection pressures on cylinder pressure and rate of heat release between diesel and n-butanol (Ne = 1500 rpm, Boost = 1.0 bar (gauge), Diesel: IMEP = 6.6 bar, Int. O_2 = 20.4%, Inj. timing = -2 °ATDC, n-Butanol: IMEP = 6.9 bar, Int. O_2 = 20.5%, Inj. timing = -12 °ATDC).

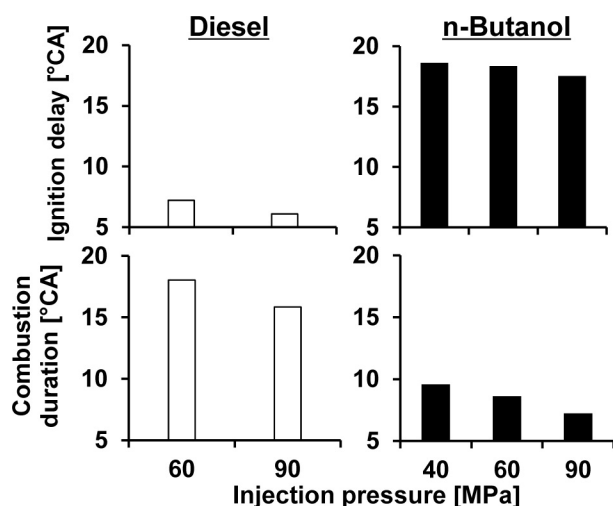


Figure 3. Impact of injection pressure on ignition delay and combustion duration for diesel and n-butanol (Ne = 1500 rpm, Boost = 1.0 bar (gauge), Diesel: IMEP = 6.6 bar, Int. O_2 = 20.4%, Inj. timing = -2° ATDC, n-Butanol: IMEP = 6.9 bar, Int. O_2 = 20.5%, Inj. timing = -12° ATDC).

Figure 4 shows the impact of injection pressure on emissions and indicated thermal efficiency for both fuels. In the n-butanol case, NOx was dramatically reduced while keeping near-zero soot with increasing injection pressure, whereas the diesel showed a general trend of increasing NOx and lowering soot. The reason for the reduction of NOx was attributed to the locally leaner mixture that was obtained due to a longer ignition delay and enhanced spray development, thereby lowering the combustion temperature [26]. Since the NOx was reduced even though the ignition delay was shortened, the spray development of n-butanol was supposed to be significantly changed by increasing injection pressure. Further studies are needed to clarify the impact of injection pressure on neat n-butanol spray development and NOx formation in the future. The reason for the near-zero soot emission was probably due to the formation of a locally leaner mixture, and fuel-borne oxygen in n-butanol. THC of n-butanol was higher than that of diesel. The cause was attributed to entry of portion of fuel spray into the quenching area, and the formation of an over-lean mixture due to longer ignition delay. These tendencies were probably enhanced by an increasing injection pressure, because the THC was slightly increased at a higher injection pressure in the n-butanol case. CO of n-butanol was much higher than that of diesel. This could be due to partial oxidation of the THC, as well as a lower CO oxidation speed due to lower local in-cylinder gas temperature. No clear trend of CO was identified for both fuels as injection pressure was changed. Indicated thermal efficiency for both fuels were similar, and injection pressure had no significant change.

Therefore, the main impact of injection pressure on engine performance for neat n-butanol combustion was that NOx emissions were noticeably reduced by increasing the injection pressure, which was the opposite trend seen in conventional diesel combustion.

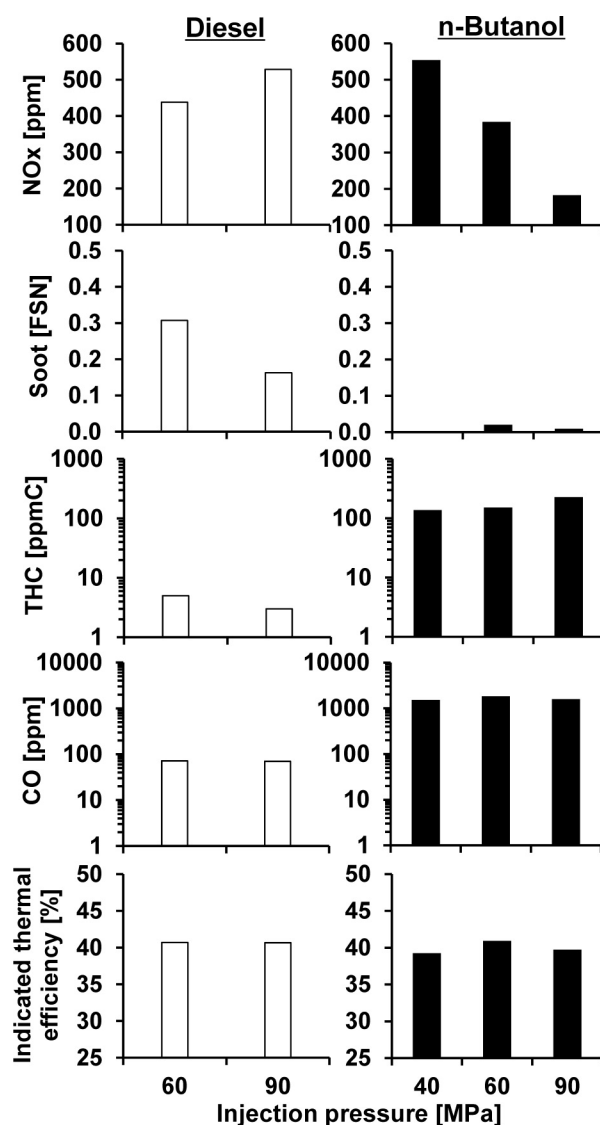


Figure 4. Impact of injection pressure on emissions and indicated thermal efficiency for diesel and n-butanol (Ne = 1500 rpm, Boost = 1.0 bar (gauge), Diesel: IMEP = 6.6 bar, Int. O_2 = 20.4%, Inj. timing = -2° ATDC, n-Butanol: IMEP = 6.9 bar, Int. O_2 = 20.5%, Inj. timing = -12° ATDC).

The use of a high injection pressure may increase cylinder pressure rise rate. Hence, the impact of injection pressure and injection timing on the relationship between emissions and thermal efficiency, and cylinder pressure rise rate was investigated. Figure 5 shows NOx, soot, indicated thermal efficiency and CA50 with respect to cylinder pressure rise rate. The injection timing is indicated in the bracket in the figure. With retarding injection timing, NOx and cylinder pressure rise rate could be reduced simultaneously for all injection pressures. However, the NOx at 90 MPa was much lower than that of 40 MPa injection pressure even at the same cylinder pressure rise rate. Soot emissions and indicated thermal efficiency had no major change at different injection pressures. The practical window of operation for injection timing was limited by the maximum cylinder pressure rise rate limit and the unstable limit. When combustion became unstable, CA50 was around 12° ATDC for all cases. Thus, the combustion phasing limit was deemed to be around 12° ATDC.