Investigations on premixed charge compression ignition type combustion using butanol-diesel blends

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Abstract
Renewable biodegradable butanol blended to diesel fuel was used in an engine that operates on PCCI mode shows excellent combustion characteristics and offer efficient high load performance with minimum exhaust emissions. Its higher octane number prevents engine knock, higher cooling effects have potential to reduce the NOX emissions and well-mixing ability with air substantially reduces the smoke emission. In the present experimentation, n-butanol and diesel blend B10, B20, B30 and B40 were tested on PCCI mode which was mainly accomplished by DI timing 20 degree CA bTDC and injection pressure 400 bar. For high load operation, B40 blend provided 6.9%, 8.1%, 12.9% and 13.7% higher brake thermal efficiency over B30, B20, B10 and neat diesel respectively at the cost of small increment in brake specific fuel consumptions. Smoke and CO emissions reduction were observed. However, NO and HC emissions produced were higher than the B30, B20, B10 and diesel respectively. Considering the benefits in terms of higher high load efficiency and lower emissions, in addition, delayed CA50 (50% burn at crank angle) than all fuel blends, B40 blend was preferred for higher premixing to attain higher performance.

Keywords : Butanol, Compression ignition (CI) engine, Carbon monoxides (CO), Exhaust gas recirculation (EGR), Hydrocarbons (HC), Nitric oxide (NO), Smoke emissions

1. Introduction

CI engines are popular due to their high fuel efficiency, durability and load-carrying capacity. However, the main drawback in their applications is exhaust emissions; in addition, the cost of fuel is also a limiting factor for longer utilization. Use of alternative renewable fuels like bio-diesel or alcoholic fuels provide improved combustion and reduced CO emissions without affecting much the engine efficiency, and being renewable, they can be utilized for longer periods.

The viscosity of alcohol-diesel blend is less, therefore results in easy injection, higher atomization, and better mixing with air. High blend volatility, high oxygen content, light molecular structures, and high hydrogen to carbon (H/C) ratio potentially reduce the smoke and CO emissions. High latent heat of vaporization produces in-cylinder cooling effects at the time of injection near the end of compression stroke that reduces the required work input for compression and improves volumetric efficiency. High laminar flame propagation speed may lead to faster combustion that potentially improves the thermal efficiency (Sayin 2010). A small quantity of alcoholic fuels such as methanol and ethanol, when blended with diesel, do not require modifications in the diesel engine. Regarding the engine operating conditions, a slight difference in BTE is observed. The improved combustion delivers low EGT compared to conventional diesel engine operations. Variations in NOX and HC emissions are mainly dependent on the quality of in-cylinder fuel-air mixture formation and maximum combustion temperature. The low calorific value and high latent heat of vaporization increase BSFC for an increased percentage of alcohol in the alcohol-diesel blends.

Butanol-diesel is another possible alcoholic diesel blends that can be preferred over methanol and ethanol, due to
the following reasons
1. Butanol having higher energy density, and higher cetane number
2. Lower latent heat of vaporization,
3. Lesser water concentration hence less corrosive.
4. Butanol mixes easily with diesel without using any surface agent.
5. Produced by the alcoholic fermentation of ethanol at the comparable cost to ethanol.

Butanol fuels exist in four isomers, namely normal butanol (n-butanol or 1-butanol), secondary butanol (2-butanol), iso-butanol (i-butanol) and tertiary-butanol (t-butanol). Most of the researches have used n-butanol-diesel blends.

Rakopoulos et al. (2010), Rakopoulos et al. (2010) and Doğan (2011) have found that the increase in butanol volume by 8%, 16% and 24% in butanol-diesel blends increases BTE, BSFC, and HC while EGT, NO\(_X\), CO and smoke are decreased at 1500 engine rpm. Al-Hasan and Al-Momany (2008) used i-butanol-diesel blend up to 40% with an interval of 10% and found a reduction in thermal efficiency over neat diesel fuel. Algayyim et al. (2017) compared the performance of i-butanol and n-butanol 10%, 20% each in diesel for three different speeds of 1400, 2000 and 2600 rpm. It was observed that the increase in butanol volume in blends provides higher BTE among all of the blends, and n-butanol-diesel blends delivered higher thermal efficiency than i-butanol-diesel blends for all speeds. The addition of straight vegetable oil and biodiesel to n-butanol and diesel blends provide stability to fuel blends, improves the oxygen concentration brings out high oxidation of HC (Atmanli et al. 2013; Atmanli et al. 2015). Atmanli et al. (2013) and Atmanli et al. (2015) have found that, the addition of 20% straight vegetable oil to 70 % diesel and 10% butanol causes a slight decrease in BTE and a bit increase in NO\(_X\). Altun et al. (2011) used in their experiment diesel-biodiesel-butanol fuel blend as 60:20:20 percentage by volume and found reduction in HC, CO, NO\(_X\) and smoke emissions. The BTE was also observed less.

However, the NO\(_X\) emissions still remains a major challenge even using alcohol-diesel blends in conventional CI engine. HCCI and PCCI are the most popular strategies to reduce the NO\(_X\) and deliver better efficiency like conventional diesel engines. HCCI uses a highly premixed air-fuel charge which is prepared by port fuel injection or early direct injection mode. The HCCI operations subjected to several challenges like poor operating load range due to a higher tendency to knock at high load and poor combustion efficiency at low load (Hwang, Dec, and Sjöberg 2007; Olsson, Tunestål, and Johansson 2004). The PCCI combustion strategy in which in-cylinder charge is partially premixed at the start of combustion is also known as partially premixed charge compression ignition (PPCI). The strategy implemented with variable direct injection timing, earlier than conventional diesel and retard than HCCI. In PCCI, the in-cylinder fuel stratification effects offer improved combustion efficiency and provide a high load limit compared to HCCI. Moreover the NO\(_X\)-soot trade-off can be achieved by optimizing injection timing.

Cheng et al. (2014) in their experiment used 30% of butanol-diesel blend with injection timing 33\(^\circ\), 28\(^\circ\), 23\(^\circ\), 18\(^\circ\), 13\(^\circ\), 8\(^\circ\), 2\(^\circ\), and -5\(^\circ\) CA bTDC respectively. It was observed that the higher blend ratio shows higher tendency to knock for DI advance beyond 23\(^\circ\) bTDC for P\(_{\text{BMEP}}\) =1200 bar and cannot be operated at high load (BMEP =11.2 bar) as it delivered low BTE than diesel. However high BTE was observed using higher blend ratios than diesel at low load (BMEP = 4.3 bar). Valentino et al. (2015) in their experiments prepared a partially premixed charge of 20% and 40% butanol diesel blend by using single-stage and two-stage direct injection strategies and used 50% EGR. The NO\(_X\) was observed high for 20% butanol-diesel blend followed by diesel and 40% butanol-diesel blend for all operating load conditions. Advanced DI time was responsible for high NO\(_X\) production and low smoke emissions. Two-stage injection produced low NO\(_X\) over single-stage injection. Leermakers et al. (2013) used in their experimentation high butanol-diesel blends such as 90:10, 80:20, 70:30, 60:40 and 50:50. A high volume of butanol in blends produced high heat release for CA50 and injection pressure 1400 bar. The thermal efficiency starts decreasing as the blend ratios increased beyond 60% butanol in blend. Higher butanol-diesel blend ratios were not effective at low load condition. This increases CO, HC and NO\(_X\) emissions but decreases PM effectively.

The high volume of butanol in butanol-diesel blends when operating engine at advanced DI timing and high injection pressure, deliver poor combustion and thermal efficiency at low load condition. Hence it is required to adjust combustion phase control parameters like intake temperature, injection pressure or to use high cetane fuel. Overly retarded DI time gives much reduction in NO\(_X\) but increases the smoke emissions and also drop the thermal and combustion efficiency.

Some modifications were done on a conventional CI engine to carry out experimentations using n-butanol and
diesel blend. The butanol was used as 10%, 20%, 30% and 40% by volume in neat diesel. The performance and emission characteristics of the modified CI engine using butanol-diesel blends fuel were compared to the baseline CI engine. The fuel injection time was set at 20° CA bTDC and the injection pressure was maintained at 400 bar to achieve the effective fuel stratification that allows the use of a higher blend ratio (40% butanol-diesel blends) for a wide range of load without the requirement of any phase control parameters. This operating condition provided least NO and smoke emissions while limiting the knock and partial burn cycle for diesel and butanol-diesel blends. The enhanced performance and improved combustion were obtained with higher butanol ratios. Whereas, B30 provided the best combustion for low loading conditions because EGT was lower than all fuel blends. B40 delivered the highest high load fuel efficiency and least CO and smoke. Higher charge premixing drops down the smoke emissions by 83% over diesel with the penalty of NO emissions. High is the cetane rating of fuel, higher is the premixing which leads to an elevated pressure rise rate and increases the chances of engine knock particularly at high load. The fuels having autoignition properties lies in between diesel and gasoline fuels, combustion shows strong cool-flame chemistry can deliver high performance at increased load without knock (Mehl et al. 2009; Wada and Senda 2009). B40 blend offers high resistance to autoignition hence preferred for higher premixing by injection time advance to achieve better performance and soot reductions. The high heat of vaporization enhances the thermal stratifications. The fuel stratifications mainly depend on the direct injection timing. The levels of fuel stratification reduce with DI time advance and high injection pressure (Dempsey, Curran, and Wagner 2016; Sjöberg and Dec 2012; Hwang, Dec, and Sjöberg 2007). The combined effects of thermal and fuel stratifications increase the high load limit and provide higher combustion efficiency even at low load in PCCI (Karwade and Thombre 2019).

Direct injection time was advanced from 20° CA bTDC to 35° CA bTDC with an interval of 5° CA. The BTE obtained was 24.7%, 27.4%, 27% and 26.4% for DI at 20°, 25°, 30° and 35° CA bTDC respectively at 4.13 bar BMEP. DI time advance reduces the smoke emissions whereas NO emissions were lowest at DI at 20° CA bTDC, and were highest at DI 25° CA bTDC and start decreasing for DI advanced where cool flame chemistry of blend B40 becomes stronger. NO emissions were 3.7, 7.9, 7.3 and 5.4 g/kWh, and smoke opacity obtained, 4.4, 4.1, 3.4 and 3% for DI at 20°, 25°, 30° and 35° CA bTDC respectively. The B40 blend while operating with advanced DI time (beyond 30° CA bTDC) in engine, delivered poor low-load efficiency. Therefore the DI timing is recommended in the range of 25° - 30° CA bTDC when the engine is operated on low load conditions.

2. Experimental set – up

The experimental set up consists of single-cylinder, four-stroke, water-cooled DI diesel engine. The schematic experimental set-up is shown in figure 1 and the engine specifications are given in table 1. The mechanical fuel injection system of the conventional CI engine is modified to the CRDI system with an electronic pressure sensor and pressure regulating valve retrofitted. The injection system is governed by a solenoid injector driver, calibration cable with programmable ECU and Nira i7r software.

![Fig. 1. Schematics of experimental set-up](image-url)
A pressure transducer, (make PCB, USA) was fitted to engine head for measuring in-cylinder pressure. To measure the engine speed and crank angle positioning, a crank angle sensor range 0 to 5500 rpm and resolution 1° CA was attached to engine crankshaft. The engine was loaded with an eddy current dynamometer to measure the torque in the range of 0 to 90 Nm. The NI data acquisition system and “EngineSoft” software was used for recording the engines performance and combustion analysis over 100 consecutive cycles.

Table 1  Experimentation engine set-up.

|                |          |          |          |          |          |
|----------------|----------|----------|----------|----------|----------|
| Bore           | 87.5 mm  |          |          |          |          |
| Stroke         | 110 mm   |          |          |          |          |
| Compression ratio | 18       |          |          |          |          |
| Rated power    | 3.5 kW @ 1500 RPM |          |          |          |          |
| Piston geometry| Bowl type|          |          |          |          |
| Injector type  | 8 hole solenoid type |          |          |          |          |
| Injection timing °CA BTDC | 20, 25, 30 and 35 |          |          |          |          |
| Injection pressure | 400 bar |          |          |          |          |

The emissions such as HC, CO, NO measurements were carried out with AVL 444 di-gas exhaust gas analyzer and AVL 437c smoke meter provide smoke opacity in % HSU (Hatridge smoke unit).

The physical-chemical properties of n-butanol and diesel fuels are given in table 2.

Table 2  Test fuel properties.

| Fuels          | n-butanol | Diesel | B10 | B20 | B30 | B40 |
|----------------|-----------|--------|-----|-----|-----|-----|
| Density (kg/m³) | 810       | 830    | 828 | 826 | 824 | 822 |
| Kinematic viscosity at 20° C (mm²/s) | 3.64 | 3.4 | 3.424 | 3.448 | 3.472 | 3.496 |
| Cetane         | ~25       | 45-55  | --- | --- | --- | --- |
| Antiknock Index | 87       | ---    | --- | --- | --- | --- |
| C-atoms        | 4         | ---    | --- | --- | --- | --- |
| H-atoms        | 10        | ---    | --- | --- | --- | --- |
| Molecular weight (g/mol) | 74.12 | 204 | 191.01 | 178.02 | 165.03 | 152.04 |
| A/F Stoichiometric | 11.1 | 14.6 | 14.22 | 13.9 | 13.55 | 13.2 |
| Lower heating value (MJ/kg) | 33.1 | 42 | 41.11 | 40.22 | 39.33 | 38.44 |
| Latent heat of vaporization | 585 | 260 | 292.5 | 325 | 357.5 | 390 |
| Laminar flame velocity (cm/s) | 45 (Doğan 2011) | 33 | --- | --- | --- | --- |

3. Results and discussions

3.1 Comparison of butanol-diesel blends for variable load and fixed injection timing

The experimentations were carried out with the aim of improving engine performance, combustion, and emissions characteristics. The n-butanol-diesel blend fuel B10 (10% butanol + 90% diesel), B20 (20% butanol + 80% diesel), B30 (30% butanol + 70% diesel) and B40 (40% butanol + 60% diesel) used in the experiment. The engine was initially run on diesel fuel. The fuel was injected at 20° CA BTDC and the injection pressure set at 400 bar. Various engine parameters were started measuring when the engine coolant temperature reaches a steady value. For naturally aspirated conditions, the above engine operating condition delivered smooth engine running at zero to high load with lower NO emissions. Same operating conditions were maintained for butanol-diesel blends to study the performance, combustion, and emission parameters.
3.1.1 Engine combustion parameters

Figure 2 and 3 represent pressure versus crank angle (P–θ) diagram and heat release rate versus crank angle (HRR–θ) diagram, are functionally useful to study the combustion behavior of diesel and butanol-diesel blends. The plots cylinder pressure (P–θ) and heat release rate (HRR-θ) are under high load (BMEP = 4.13 bar) condition.

![Cylinder pressure vs crank angle (P–θ).](image1)

![Heat release rate vs crank angle (HRR–θ).](image2)

The start of combustion (SOC) retards with the increasing volume of butanol in blends. The higher butanol-diesels blends get a longer time for premixing hence the combustion phase is faster and therefore high heat release rate (HRR) was observed compared to diesel. The diesel produced the highest peak cylinder pressure followed by B30, B40, and B10 blends as shown in fig. 2. While in-cylinder peak pressure was lowest in case of B20 blends. Figure 3 shows the peak HRR produced by B30 and B40 blends were highest because higher blend volatility causes high rate of premixed combustion. While B40 blend produced bit lower peak HRR than B30 blend mainly due to higher evaporative cooling of butanol cause combustion retard. The Peak HRR was found lower in case of B20 and B10 blends than diesel possibly due to higher evaporative cooling and lower energy density. Increase in butanol blend ratios contribute to the retarded combustion. The CA50 (50% burn at crank angle) was observed 362°, 364°, 365°, and 366° CA for diesel, B10, B20, B30, and B40 respectively. It means higher butanol blends can be efficient for better premixing to achieve greater performance and emission reductions at high load conditions.

![Load vs exhaust gas temperature](image3)

Longer burn duration leads to higher exhaust gas temperature whereas low exhaust gas temperature (EGT) represents improved combustion that means high heat release at shorter duration near the TDC during expansion. As shown in fig. 4, EGT increases with increase in load for fuel blends. However, at low load (BMEP = 1.03 bar) the lowest EGT was found for B30 blend. The burn duration was retarded in case of B40 over B30 blend at low load condition due to its larger cooling effects. The improved combustion for higher blend ratios give reduced EGT at high engine load operation.
3.1.2 Engine performance parameters

Figure 5 shows higher BTE for all load with increase in butanol blend ratios, except B40 blend at low load. The higher in-cylinder cooling effects of B40 blend cause much retarded combustion at low load (BMEP = 1.03 bar). The BTE was observed increased for higher blend ratios. The possible reasons can be high volatility of blends, and increased ignition delay that causes high rate of air-fuel premixing before the start of combustion. Furthermore a high flame travel velocity of butanol cause higher heat release at shorter burn durations (Cheng et al. 2014; Doğan 2011; Algayyim et al. 2017). The increased rate of premixed combustion for higher blend ratios decreases the rate of diffusion combustion in the overall combustion phase (Yamamoto et al. 2013) hence the combustion is less gradual as compared to diesel which also provides the benefit in soot reduction as local equivalence zones becomes leaner.

![Fig. 5 Load vs brake thermal efficiency](image1)

![Fig. 6 Load vs brake specific fuel consumption](image2)

The high heat of vaporization of butanol produces higher evaporative cooling which results in low combustion temperature hence low brake power and torque (Algayyim et al. 2017; Ileri, Atmanli, and Yilmaz 2016). This increases the required fuel supply. For high load (BMEP = 4.13 bar), B40 blend could provide 6.9%, 8.1%, 12.9%, and 13.7% higher BTE over B30, B20, B10, and diesel respectively due to the improved combustion. On the other hand, BSFC shows slight increment as 379, 394, 397, 401, and 402 g/kWh for diesel, B10, B20, B30, and B40 respectively at high load conditions as shown in fig. 6. For low load conditions (BMEP = 1.03 bar), the BSFC for B40 blend was much higher than that of all fuel blends.

3.1.3 Engine emissions parameters

There are various possible sources for unburned hydrocarbon formation in CI engines like fuel trapped in the nozzle, crevice areas and cylinder piston interface that accounts for a majority of HC emission. Other factors include incomplete evaporation of fuel in a mixture, local over rich/over lean mixture and liquid wall film interface with more spray impingement (Heywood 1988).

The HC emissions decrease with the increased load as shown in fig. 7. Overly retarded combustion of B40 blends at low load responsible for poor combustion efficiency which results in high HC emissions. HC emissions were still higher at increased load condition for B40 blend because, butanol-diesel blends produce cooling effects that cause slower evaporation of diesel and poor fuel-air mixing of the blends. Increased spray penetration causing unwanted fuel impingement on the combustion chamber walls (so flame quenching) and cushioning in the ring land areas, also the so-called 'lean outer flame zone' increases with blend ratios where the flame is unable to exist (Rakopoulos, et al. 2010). The level of HC emissions was lowest for B10 blend for all load conditions followed by B20, B30 blends. The fuel impingement due to higher spray penetrations is lesser for lower butanol blend ratio moreover, the oxygenated fuel causes oxidation of HC at high temperatures.

![Fig. 7 Load vs unburned hydrocarbon emissions](image3)
The oxidation of CO is encountered similar to oxidation of HC but the generation of CO emissions mainly depends on the equivalence ratio and combustion temperature. Figure 8 shows, a low load condition where equivalence ratio is too lean leads to poor combustion efficiency and cannot produce adequately high combustion temperature for CO oxidation. While operating engine at high load required high equivalence ratio which decreases the in-cylinder oxygen concentration hence lesser conversion of CO into CO$_2$ takes place even at high combustion temperature. The oxygenated butanol-diesel blend brings out the appropriate oxidation of CO to CO$_2$, moreover its lighter molecular chain and high H/C ratio provide much reduction in CO emissions. Under low load conditions, high butanol blend ratio provides a significant reduction in CO emissions and CO emissions were comparatively lower at high load than diesel.

The NO emission formation mainly depends on in-cylinder maximum combustion temperature. In case of butanol-diesel blends, the in-cylinder cooling effects and lower calorific value keep maximum combustion temperature lower. At low load condition B40 produced lowest NO emissions because of low in-cylinder temperature generates due to higher cooling effects and retarded combustion. At high load conditions, NO emitted by B10 blend was observed lower than diesel due to its lower calorific value and higher cooling effects (as shown in fig. 9). Still, the major conflict observed was the NO emissions were higher for higher blends. The possible reason can be higher premixing before the start of combustion that causes a higher rate of heat release at shorter burn duration as compared to diesel this leads to producing high combustion temperature. Moreover high oxygen content of butanol also contributes more NO emissions formation compared to diesel.

Smoke emissions mainly contain soot and particulate matters (PM). Soot is produced by oxygen-deficient thermal cracking of long-chain molecules (Heywood 1988). Lighter molecular structure and high H/C ratio make butanol smoke free fuel. Furthermore, a higher rate of premix combustion causes a considerable reduction in smoke emissions.
for higher butanol blend ratio than that of diesel (as shown in fig. 10.)

3.2 Direct injection time advance using B40 blend.

Engine operating on B40 blend shows high performance, excellent combustion characteristics, and considerable reduction in emissions over diesel and B10, B20, B30 blends. Its low cetane number can prevent early ignition in case of high premixing of charge. Besides, cool flame chemistry of B40 blend can potentially deliver higher performance with lower emissions for injection time advance. The rate of charge premixing for B40 blend was increased by DI time advance from 20° to 35° CA bTDC with an interval of 5° CA for injection pressure 400 bar.

3.2.1 Engine combustion

Figure 11 and 12 show the combustion behavior of B40 blend at high load (BMEP = 4.13 bar) for DI time advance from 20° to 35° CA bTDC. The highest peak pressure and maximum heat release rate were obtained at 35° CA bTDC (as shown in fig. 11 and 12). For advanced DI time, a small fraction of fuel burn as a diffusion flame in the early phase of combustion. The combined effects of thermal and fuel stratification exhibit the cool-flame chemistry and delay the start of main combustion event. The advanced DI time shows stronger cool-flame chemistry which provides higher premixing before the start of main combustion phase and results in high HRR and peak cylinder pressure.

![Cylinder pressure vs crank angle (P–θ)](image1)

![Heat release rate vs crank angle (HRR–θ)](image2)

Fig. 11 Cylinder pressure vs crank angle (P–θ). Fig. 12 Heat release rate vs crank angle (HRR–θ)

The CA50 was between 360°- 370° CA for all DI timings suggests smooth and efficient combustion without knock and or misfires. The maximum pressure rise rate was observed 10.2° bar/ CA at TDC for DI time 35° CA bTDC, hence further DI time advance can increase the knocking tendency.

3.2.2 Engine performance

As shown in figure 13, the highest BTE was observed at 25° CA bTDC where the condition was most favorable for in-cylinder mixture formation which offers efficient and improved combustion at all load conditions. The higher BTE at injection time advance may be due to the presence of local rich fuel-air equivalence zone results in high-temperature combustion and causes a shorter burn duration of the main combustion event (Yusri et al. 2019).

Further advanced DI causes early combustion, this increases the work transfer by in-cylinder gases to the piston hence a little reduction in BTE was observed at BMEP = 4.13 bar. Advanced DI time decreases the BTE and under low load operation, and BTE becomes poor (at DI = 35° CA bTDC) because of the significant influence of cool-flame chemistry of B40 increases the required fuel supply. The significant improvement in BTE was occurred for increased load due to the shorter burn duration for injection time advance.
Figure 14, load and BSFC relationship shows that B40 delivers the highest performance at high load, where DI at 25° CA bTDC gives highest performance for all loading conditions as the specific fuel consumption was lowest. Increase in loading improves BTE for advanced DI time and reduces the marginal difference between BTE and BSFC.

### 3.2.3 Engine emissions

Figure 15 shows HC emissions decrease as load increases. The higher combustion temperature brings out the oxidation of HC emissions. Regardless of this HC emissions were higher for DI time advance this is mainly because of increased flame quenching due to spray penetration in less dense air into the engine cylinder. This suggests the use of higher intake air-swirl or higher intake pressure can resolve this issue for advanced injection timing (Sjöberg and Dec 2005).

Fuel-air premixing for advanced DI timing decreases the CO emissions (as shown in fig. 16) because the emitted CO follows the same behavior as the emitted soot by the engine, a fact collectively attributed to the same physical and chemical mechanisms affecting almost in the same way, at least qualitatively, the net formation of these emissions (Rakopoulos et al. 2010).

The in-cylinder cooling effects due to high volume of butanol in blend results in low combustion temperature which is beneficial in NO reduction. High charge premixing provide least NO and was observed at 20° CA bTDC among all injection timings (as shown in fig. 17). NO emissions were highest for DI at 25° CA bTDC where the combustion of overall lean mixture produced high combustion temperature. Further DI advance cause early burning of a small fraction of fuel reduces the in-cylinder oxygen concentration (by producing some amount of CO and HC)
before the start of main combustion event.

Smoke emission increases with increase in load. The in-cylinder local rich equivalence zones are the sources of smoke production. Higher premixing by means of DI timing advance leads to local equivalence ratio zones leaner (Huang et al. 2019) which provide effective reduction in smoke emissions for all load range as shown in fig. 18. The DI timing 35° CA bTDC has shown significant reduction in smoke emissions. It was observed 30% reduction in smoke over DI timing 20° CA bTDC and 85% reduction when compared to diesel for DI timing 20° CA bTDC. The reason is the physical-chemical property of butanol and the leaner local equivalence zones.

4. Conclusions

Renewable, biodegradable butanol blended to diesel increase the volatility of blends and provide higher premixing results in homogeneous combustion that considerably reduce the smoke emissions. Meanwhile the light molecular structure and oxygen content of blends reduced CO emissions and high cooling effects lower the NO emissions.

The n-butanol-diesel blend shows slightly high ignition delay and shorter burn duration because of its high rate of premixing which results in improved thermal efficiency.

Increased volume of butanol in butanol-diesel blend ratio gives high peak cylinder pressure, heat release rate, and pressure rise rate compared to neat diesel. Hence the high butanol blend can be preferred for higher premixing mainly governed by direct injection time advance in order to reduce engine emissions and to achieve high performance.

High butanol blends give better thermal efficiency at part load to high load conditions. Using B40 blend at DI 20° CA bTDC, NO emissions were observed low, and also the brake thermal efficiency. Advancing DI to 25° CA bTDC significantly increases the BTE with the penalty of increase NO emissions. A further advance in DI timing decreases BTE marginally and significantly reduce NO emissions.

B40 blend delivered poor low load efficiency for DI at 35° CA bTDC due to larger in-cylinder cooling effects of butanol may lead to partial burn cycles. This causes the production of high level of HC and CO emissions. HC emissions were found increased with DI time advance for all load range for higher butanol blend ratio. Light molecular chain, oxygen concentration of butanol fuel offer larger reduction in smoke and CO emissions. The high butanol blends up to 40% is suitable for high load engine operations using premixed combustion mode.

Nomenclatures

|  |  |
|---|---|
| γ | : Specific heat ratio |
| BSFC | : Brake specific fuel consumption |
| °CA bTDC | : Crank angle before top dead center |
| CAS0 | : 50% burn at crank angle |
| CO | : Carbon monoxides |
| DI | : Direct injection |
| EGT | : Exhaust gas temperature |
| HC | : Hydrocarbons |
| HCCI | : Homogeneous charge compression ignition |
| HRR | : Heat release rate |
| NO | : Nitric oxide |
| NOx | : Oxides of nitrogen |
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