

Common Rail Direct Injection Mode of CI Engine Operation with Different Injection Strategies - A Method to Reduce Smoke and NO_x Emissions Simultaneously

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ABSTRACT

Compression ignition (CI) engines are most efficient and robust prime movers used in transportation, power generation applications but suffer from the problems of higher level of exhaust smoke and NO_x tailpipe emissions with increased use of fossil fuels. Alternative fuel that replaces diesel and at the same time that result in lower smoke and NO_x emissions is presently needed. Therefore the main aim of this experimental study is to lower the smoke and NO_x emissions and to use non edible oils that replace the diesel. For this locally available honge biodiesel (BHO) and cotton seed biodiesel (BCO) were selected as alternative fuels to power CI engine operated in common rail direct injection (CRDI) mode. In the first part, optimum fuel injection timing (IT) and injection pressure (IP) for maximum engine brake thermal efficiency (BTE) was obtained. In the second part, performance, combustion and emission characteristics of the CRDI engine was studied with two different fuel injectors having 6 and 7 holes each having 0.2 mm orifice diameter. The CRDI engine results obtained were compared with the baseline data reported. The combustion chamber (CC) used for the study was toroidal re-entrant (TRCC). The experimental tests showed that BHO and BCO fuelled CRDI engine showed overall improved performance with 7 hole injector when engine was operated at optimized fuel IT of 10° before top dead centre (bTDC) and IP of 900 bar. The smoke emission reduced by 20% to 26% and NO_x reduced by 16% to 20% in diesel and biodiesel powered CRDI engine as compared to conventional CI mode besides replacing diesel by biodiesel fuel (BDF).

Keywords: honge biodiesel (BHO), cotton seed biodiesel (BCO), common rail direct injection (CRDI) engine, number of holes

INTRODUCTION

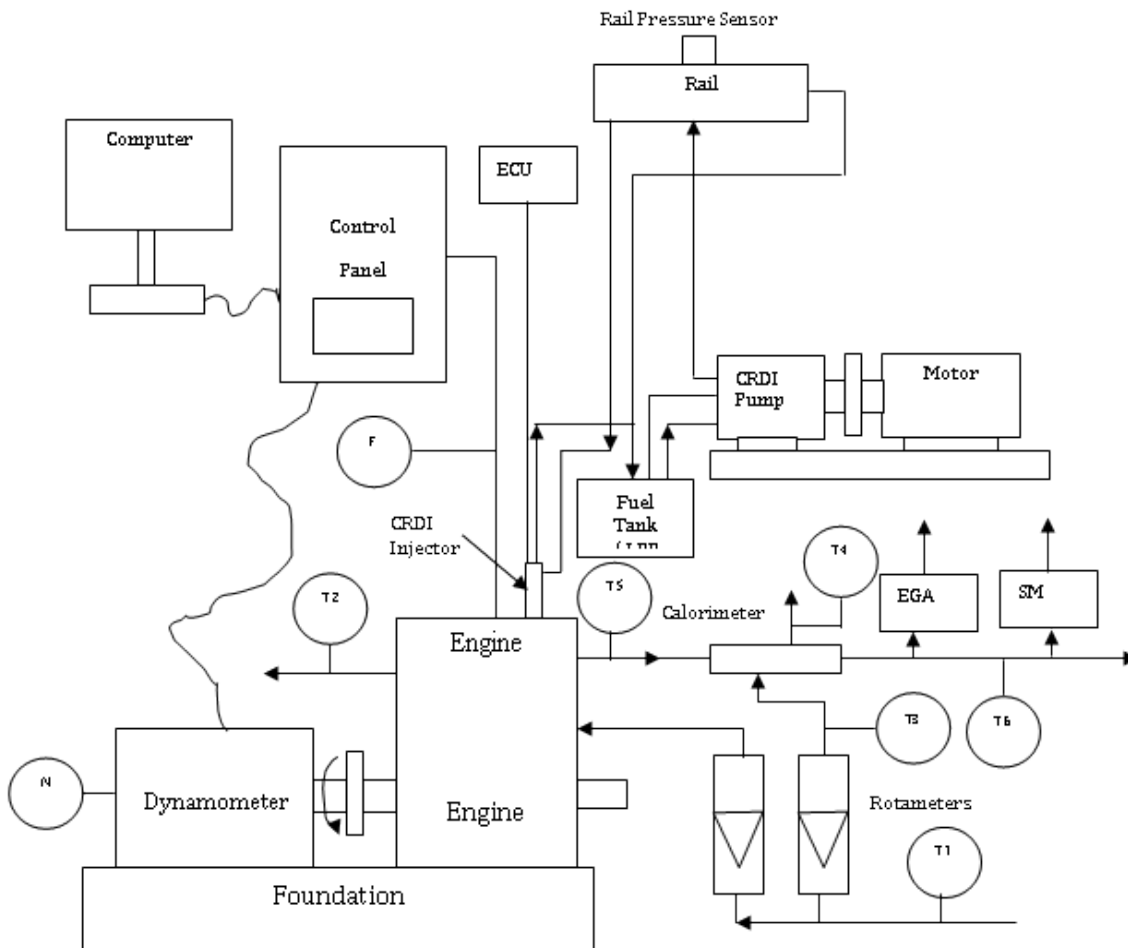
The need for alternative and renewable fuel like BDF in place of depleting diesel fuel demands higher IP due to their higher viscosity. The conventional CI engine has the limitation of fuel IP which can be overcome by employing CRDI facility which injects fuel at any CA, duration and IP. Performance, emission and combustion characteristics study on CRDI engine on rape seed oil methyl ester (ROME) and its blend with diesel showed lower smoke, CO and HC emissions with higher NO_x levels on account of better air fuel mixture prevailed which led to complete combustion (Carlo et al., 2002). Better overall mixing of fuel with air reduced ID with increased fuel droplet velocity and decreased droplet size resulted at higher fuel IP. This led to higher HRR and higher in-cylinder temperature which resulted in increased engine out NO_x emissions (Lee and Park; 2002). The mixing process of fuel with air enhanced at higher fuel IP due to higher charge temperature available at high load condition.

More combustible mixture was formed during ID period because of the higher PP at fuel IP of 900 bar as compared to 800 bar. At the same SOI, variation of fuel IP significantly altered the heat release profile, i.e., higher fuel IP led to earlier SOC and higher as well as narrower peak heat release (Ye and Boehman, 2010; Labecki and Ganippa, 2012). A study on the effect of injection parameters on the combustion and emission characteristics of CRDI engine fuelled with waste cooking oil revealed that HRR of BDF gradually increased with advanced fuel IT irrespective of load condition. At high load, the ISFC was higher and reached minimum when the IT was about 10°aTDC. The HC emissions of the BDF were slightly lower than those of the diesel at the fuel IP of 1600 bar, but NO_x emissions in the engine out gas were found higher. It was also reported that the CO emissions decreased with higher fuel IP at high load condition when fuel IT was retarded (Joonsik et al., 2014). CRDI engine operated at a constant engine speed of 1800 rpm and different loads of 3.2 bar, 5.1 bar IMEP revealed that the control of combustion phasing and EGR of the gasoline/diesel fuels could yield emission lower than CI engine and BTE higher than typical SI engine (Jiakun et al., 2015). The effect of 10%, 20% and 50% Karanja BDF (KOME) blends with diesel were studied on the injection rate, fuel atomization, performance of a CRDI engine at a constant engine speed of 1500 rpm and at different IT. The results showed that with lower blend (20%) HC and CO emissions were reduced in comparison to mineral diesel. It was also reported that up to B10, CRDI engines showed improved BTE and reduced emissions with no change in hardware (Agarwal et al., 2015). The study on CRDI engine confirmed that engine operation was possible with up to 75% swine lard methyl esters blend with diesel but with small decrease in brake fuel conversion efficiency and increase in BSFC (Mikulski et al., 2016). The diesel equivalent levels of soot emission and 22% lower NO_x emission were revealed with 30% RSO at IT of 3°bTDC and IP of 1200 bar (Labecki et al., 2012). Soybean BDF indicated faster ignition, lower pre-mixed spike and lower PP when start of energizing was between 2 to 6°bTDC. For rapeseed BDF with fuel IT varied from 10 to 5°bTDC the combustion process occurred nearer to TDC resulting in higher BTE. More efficient utilization of fuel resulting in better atomization and the advancement or retardation in fuel IT from the optimum value is the reason for deterioration of BTE (Ye and Boehman, 2010; Monyem et al., 2001; Senatore et al., 2008). At higher fuel IP the BTE improved due to efficient utilization of fuel because of better atomization (Bakar et al., 2008). As the fuel IP increases from 800 bar to 900 bar the combustion phasing of BDF advances due to the reduction of the ID through better air entrainment and mixing. When the fuel IP is increased ID decreased due to smaller SMD, shorter break up length, higher dispersion and better atomization of injected linolenic linseed oil methyl ester BDF (Puhan et al., 2009). Retarding the fuel IT decreases the PP due to shorter ID resulting in lower peak cylinder gas temperatures which resulted in lower NO_x concentration (Leung et al., 2006). The smoke emissions of BDF were higher than mineral diesel under the same operating conditions. This could be attributed to the presence of FFA in the BDF leading to poor air-fuel mixture. Smoke emissions of both fuels increased when the IT was retarded due to sluggish and diffusion combustion phase caused by reduced fuel-air mixing rate due to later injection (Sayin et al., 2009). The engine performance with bio fuel-blended diesel fuels (blend range) was similar to that under D100 fuel. Use of BDF–diesel blends lowered the HC and CO emissions but showed higher engine out NO_x. The smoke emissions were reduced by 50% with the use of bio ethanol–diesel blends (Kim and Choi, 2009). The NO_x emissions drastically increased with increase in fuel IP for the BDF and CO, HC, smoke emissions showed decreasing trend with higher fuel IP at higher loads (Hwang et al., 2014; Armas et al., 2006). The amount of fuel delivered during ID could be increased due to enhanced atomization of fuel at the nozzle outlet with high IP which leads to a more distributed vapour phase and better combustion which facilitates increased premixed HRR especially at higher load condition (Ozsezen et al., 2008; Can et al., 2004). It can be noted that at different fuel IT (20°bTDC to 5°aTDC) the PP and the peak HRR of the BDF are slightly lower than those of the diesel. The PP is increased as the fuel IT was advanced due to increased ID. More homogeneous fuel-air mixture is formed which results in a strong premixed combustion phase leading to a higher PP (Narayana et al., 2016).

From the exhaustive literature survey on CRDI engine, it can be noted that utilization of biodiesel, effect of CC shape and nozzle holes on the performance has been scantily reported. Therefore the objective of this experimental study mainly focusses on the use of biodiesels (BHO and BCO) derived from non-edible oils and test their suitability in CRDI engines along with use of injector with varied number of holes and TRCC shape. Initially the fuel IT and IP for best BTE condition was obtained and then keeping optimum IT and IP, experiments were conducted on the engine with injector having varied number of holes. Finally conclusions were drawn from the experimental study.

EXPERIMENTAL TEST RIG AND PROCEDURE ADOPTED

The existing CI engine had HCC. The existing 5.2 kW CI was modified to operate in CRDI mode. In order to facilitate high pressure injection of BDFs in the converted CRDI engine and to facilitate the swirl induced air-fuel mixture the CC was modified to TRCC shape. All the experimental results presented in this study are with TRCC in CRDI mode. The fuel from the tank is pumped to a high pressure CRDI pump through a fuel filter using low



T1, T3 – Inlet Water Temperature, T2 – Outlet Engine Jacket Water Temperature, T4 – Outlet Calorimeter Water Temperature, T5 – Exhaust Gas Temperature before Calorimeter T6 - Exhaust Gas Temperature after Calorimeter, F – Fluid Flow differential pressure Unit, N – Speed Encoder, EGA – Exhaust Gas Analyser, SM – Smoke Meter, LPP-Low pressure pump

Figure 1. Schematic of experimental set up with CRDI facility

pressure pump. The variable speed motor regulates the speed of the high pressure pump which controls the quantity of fuel supplied to a common rail. A pressure relief valve controls the amount of pressure required for fuel injection. In house developed calibrated ECU regulates the fuel SOI (IT) and duration. Modified experimental set up with CRDI facility is shown in **Figure 1**. The experimental tests on CRDI engine were carried out at the engine speed of 1500 RPM for 80 and 100% loads keeping CR 17.5. The fuel injection rate was adjusted to maintain a constant speed of 1500 rpm for both loads and stable engine condition readings were recorded. The temperature of cooling water at exit was maintained at 70°C. The experiments were conducted using diesel, BHO and BCO fuels with two injectors of 6 and 7 holes. The diameter of both the injector orifices was 0.2 mm. Properties of diesel, BHO and BCO are provided in **Table 1**. Specifications of CI engine test rig used for the experimental study are shown in **Table 2**. Pressure sensor and thermocouples provided the cylinder pressure and temperature readings respectively. Exhaust gas emissions were recorded from exhaust gas analyser. The smoke emission was recorded by Hartridge smoke meter. Cooling of the engine was done with the circulating water through the jackets of engine block and cylinder head. HRRs were computed as per the procedure laid down in literature (Hayes et al., 1986; Hohenberg, 1979).

The SOC process was taken as the differentiated cylinder pressure variation time data where a sudden rise in the slope at the point of ignition due to the sudden high premixed heat release. The end of combustion process is the point where 90% of the heat was released (calculated from the cumulative heat release curve). The ID period is the time lag between the SOI and the start of ignition. The SOI was taken based on the IT set in ECU. The start of combustion process was determined from the differentiated cylinder gas pressure variation time data where a sudden rise in the slope at the point of ignition due to the sudden high premixed heat release. The end of combustion process was taken as the point where 90% of the heat release had occurred (calculated from the cumulative heat release curve). Difference between these two is taken as combustion duration (CD).

Table 1. Properties of Diesel, BHO and BCO

| S.No. | Properties | Diesel | BHO | BCO | Standard limits | | ASTM standard |
|-------|--------------------------------------|--------|-------|-------|-----------------|------|---------------|
| | | | | | Min. | Max. | |
| 1 | Viscosity (cSt at 40°C) | 4.59 | 5.6 | 5.0 | 1.9 | 6 | ASTM D445 |
| 2 | Flash point (°C) | 65 | 163 | 167 | 100 – | - | ASTM D93 |
| 3 | Calorific Value (kJ/kg) | 45000 | 36010 | 39648 | - | - | ASTM D5865 |
| 4 | Density (kg/m ³ at 15 °C) | 830 | 890 | 885 | 860 | 900 | ASTM D4052 |
| 5 | Cloud Point (°C) | -10 | -2 | -3 | - | - | ASTM D2500 |
| 6 | Pour Point (°C) | -2 | 1 | 2 | - | - | ASTM D97 |
| 7 | Cetane Number | 50 | 42 | 45 | 47 | - | ASTM D613 |
| 8 | Cold Filter Plugging Point (°C) | 4 | 5 | 6 | - | - | ASTM D6371 |
| 9 | Moisture (%) | 0.02 | 0.02 | 0.02 | - | 0.05 | ASTM D2709 |
| 10 | Carbon Residue (%) | 0.1 | 0.12 | 0.13 | - | 0.05 | ASTM D4530 |

Table 2. CI Engine specifications

| Sl. No. | Parameter | Specification |
|---------------------------|------------------------|---|
| 1 | Type of engine | Kirloskar make Single cylinder four stroke direct injection diesel engine |
| 2 | Rated power | 5.2 KW @1500 RPM |
| 3 | Cylinder diameter | 87.5 mm |
| 4 | Stroke length | 110 mm |
| 5 | Compression ratio | 17.5 : 1 |
| 6 | Software used | Engine soft |
| Air measurement manometer | | |
| 7 | Made | MX 201 |
| 8 | Type | U- Type |
| 9 | Range | 100 – 0 – 100 mm |
| Eddy current dynamometer | | |
| 10 | Model | AG – 10 |
| 11 | Type | Eddy current |
| 12 | Maximum | 7.5 (kW at 1500 - 3000 RPM) |
| 13 | Flow | Water must flow through Dynamometer during the use |
| 14 | Dynamometer arm length | 0.180 m |

Table 3. The accuracies of the measurements and the uncertainties in the calculated parameters

| Measured variable | Accuracy (±) |
|-----------------------|-----------------|
| Load, N | 0.1 |
| Engine speed, rpm | 2 |
| Temperature, °C | 1 |
| Measured variable | Uncertainty (%) |
| Smoke | ±1.1 |
| HC | ±1.2 |
| CO | ±2.5 |
| NO _x | ± 4 |
| Calculated parameters | Uncertainty (%) |
| BTE (%) | ±1.3 |
| HRR (J °CA) | ±0.9 |

Uncertainty analysis of the experimental data

The accuracies of the measurements and the uncertainties in the calculated parameters of the current investigation are provided in the **Table 3**. In order to minimize the errors of measurements six readings were recorded and averaged out results are only presented for plotting the graphs and analysis carried out.

RESULTS AND DISCUSSIONS

The experimental results of CRDI engine are discussed in the following sections. The CI engine was operated at 1500 rpm and a compression ratio of 17.5. The CI engine other operating conditions are provided in **Table 4** for diesel and BDFs. The manufacturer specified engine operating conditions for diesel and optimum engine operating conditions for BDFs for better BTE was used to record the readings. The HCC was employed in CI mode of engine operation. The average of six results are reported in the **Table 4**.

Fuel Injection Timing (IT) Optimization for the Diesel, BDF Powered CRDI Engine

The effect of fuel IT on the performance was studied first. The fuel IT was varied from 25°bTDC to 5°aTDC in a step of 5° CA.

Table 4. Results of CI mode of engine operation with diesel BHO and BCO

| Characteristic | CI Mode of Engine Operation | | |
|-----------------------|---|--|-------|
| | Diesel | BHO | BCO |
| | (IOP: 205 bar, IT: 23° bTDC Injector: 3 hole, 0.3 mm diameter) | (IOP: 240 bar, IT: 19° bTDC, Injector: 3 hole, 0.3 mm diameter) | |
| BTE (%) | 31.25 | 26.0 | 25.6 |
| Smoke (HSU) | 46 | 60 | 62 |
| HC (ppm) | 36 | 49 | 50 |
| CO (% vol.) | 0.13 | 0.154 | 0.156 |
| NO _x (ppm) | 1095 | 1009 | 1000 |
| PP (bar) | 74 | 68 | 67 |
| ID (°CA) | 9.9 | 10.2 | 10.3 |
| CD (°CA) | 38 | 41 | 42 |
| HRR (J/°CA) | 74 | 72 | 71 |

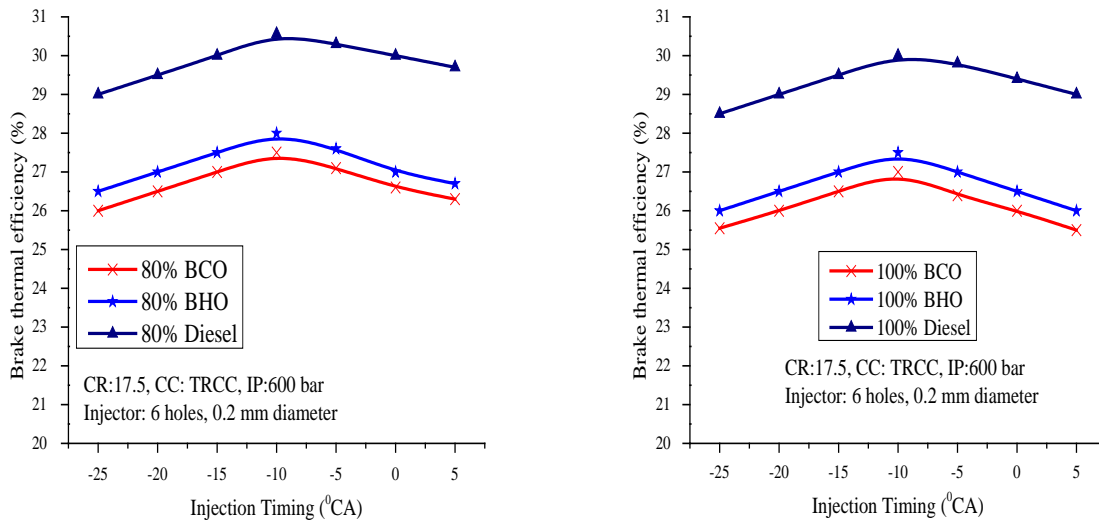


Figure 2. Variation in the BTE at different fuel IT at 80% load and 100% load

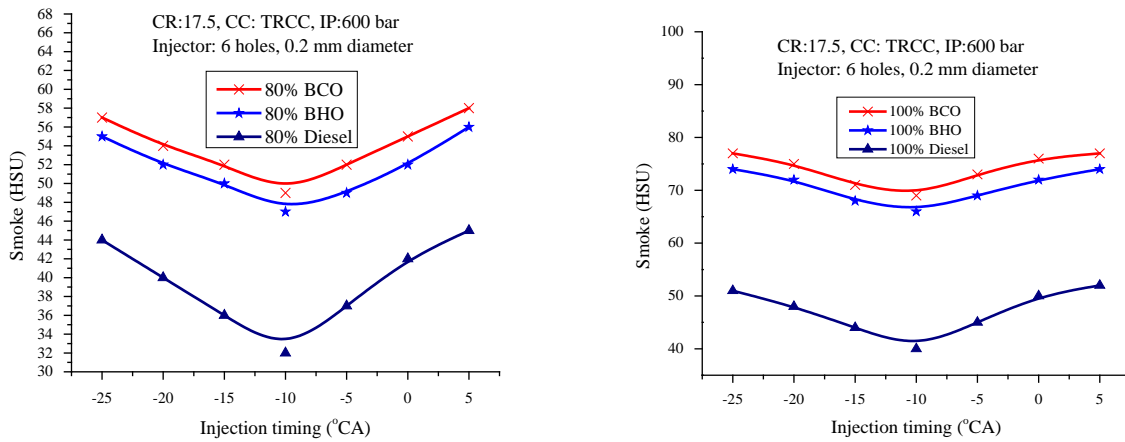


Figure 3. Variation in the smoke emission at different fuel IT at 80% and 100% loads

Performance: Brake thermal efficiency

Figure 2 depicts the variation in fuel IT on the BTE of CRDI engine fuelled with diesel, BHO and BCO at 80% and 100% loads. From Figure 2 it was evidenced that the maximum BTE was resulted at a fuel IT of 10° bTDC for all fuels. This could be due to better atomization of injected fuel at IP of 600 bar, enhanced air-fuel mixing and reduced wall wetting that led to better burning of fuel. The advancement or retardation from this fuel IT deteriorated the BTE of the CRDI engine. The advanced fuel IT led to wall wetting due to higher fuel IP. On the other hand retarded fuel IT increased fuel entering into crevices and reduced the time available for air fuel mixing. Similar findings were reported in the literature (Can et al., 2004; Hwang et al., 2014; Labecki et al., 2012). Maximum BTE was achieved in the CRDI mode with diesel, BHO and BCO and were lower by 2.2%, 10.4% and

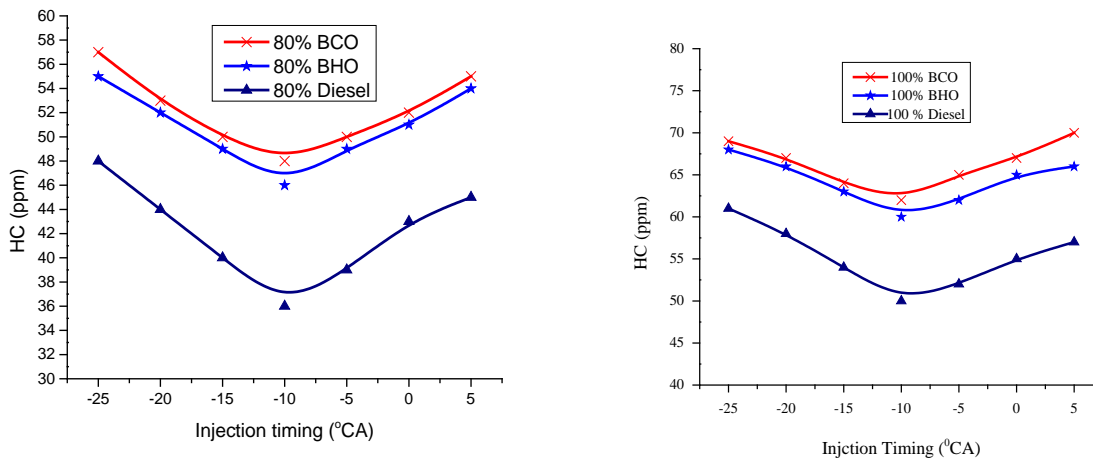


Figure 4. Variation in the HC emission at different fuel IT at 80% and 100% loads

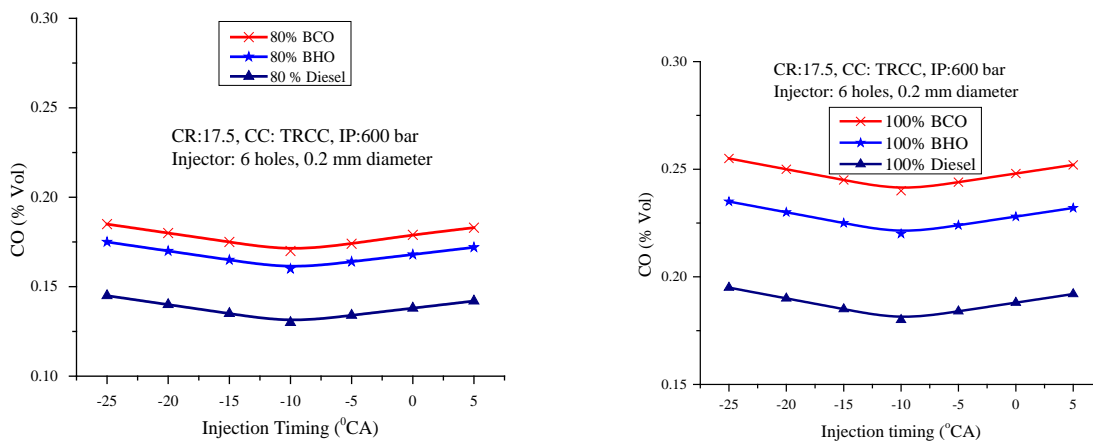


Figure 5. Variation in the CO emission at different fuel IT at 80% and 100% loads

12% respectively at 80% load and fuel IT of 10°bTDC compared to the CI mode. At 100% load slight reduction in BTE about 2% was seen for all fuels. BTE of the engine was lower with the BDF as compared to mineral diesel due to their higher viscosity and lower calorific values and also it was observed that BHO showed better BTE as compared to BCO due to its good combustion qualities.

Emissions: Smoke Emission

Figure 3 shows the effect of change in fuel IT on the engine out smoke emission of CRDI engine fuelled with diesel and BDF used for the study. The smoke emissions in the tail pipe with BDF were higher than the mineral diesel. This could be attributed to the FFA present in the selected BDFs which led to poor air-fuel mixture. At a fuel IP of 600 bar, the smoke emissions of all injected fuels decreased up to an IT of 10°CA bTDC due to reduced wall wetting and better combustion as more time was available for the oxidation process and beyond which increased trends were observed. Beyond 10°CA bTDC sluggish diffusion combustion phase with reduced rate of fuel-air mixing at late injection of fuel was observed. The smoke emission in the exhaust gas of the CRDI engine with BDF were slightly higher by 4% to 7% at fuel IT of 10°bTDC, 80% load and 100% load as compared to CI mode.

HC and CO Emissions

Figures 4 and 5 shows the effect of change in the fuel IT on the engine out HC and CO emissions of the CRDI engine powered with diesel, BDF at 80% and 100% loads respectively. The HC and CO emissions in the engine out gas were lower for the fuel IT of 10°bTDC. Improved combustion of injected fuel resulted at fuel IP of 600 bar and the higher BTE resulted might be the reason for this result seen. The HC and CO emissions of the engine increased at other fuel ITs. The reason for the result seen could be the advanced IT led to the CC wall wetting and retarded fuel IT increased the fuel entering into the crevices of CC. Similar results were reported in the literature (Agarwal et al., 2015). These emissions level in the exhaust gas with BDF operation were higher compared to mineral diesel operation. The higher viscosity of BDF resulted in to bigger sized fuel droplets at the

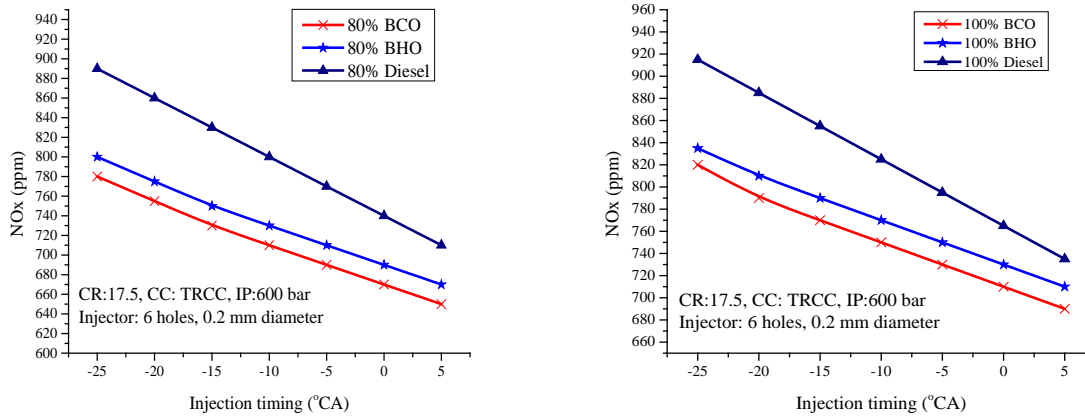


Figure 6. Variation in the NO_x emission at different fuel IT at 80% and 100% loads

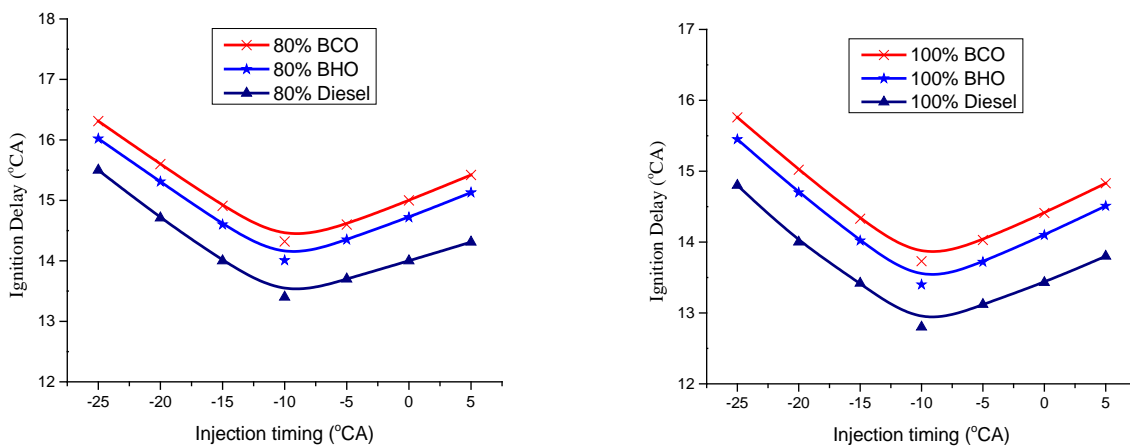


Figure 7. Variation in the ID at different fuel IT at 80% and 100% loads

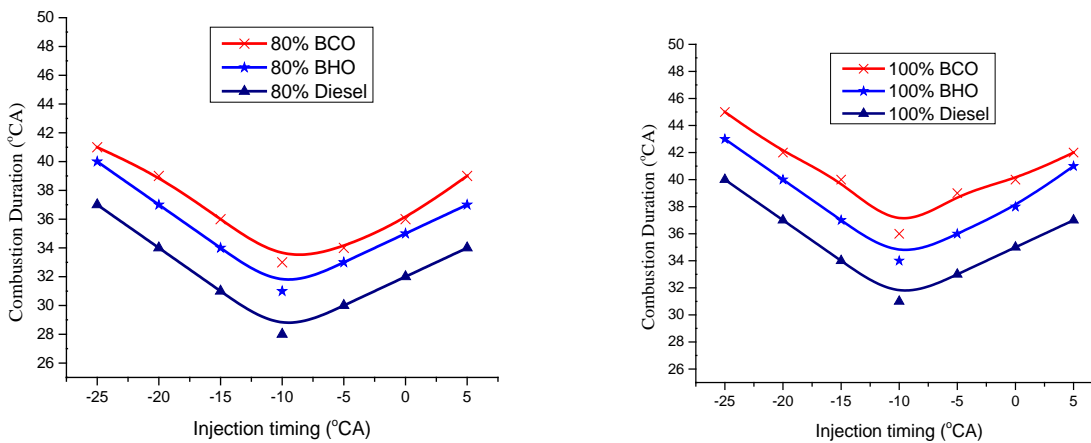


Figure 8. Variation in the CD at different fuel IT at 80% and 100% loads

same IP as compared to diesel. Comparatively lower energy density of biodiesel might lead to lowered combustion temperatures which support the results obtained. HC emissions with BDFs were higher by 27% to 33% and 15% to 20% at 80% and 100% load respectively compared to CI mode. Similarly the CO emissions with BDFs were higher by 23% to 33% at 80% load and 100% load compared to CI mode. BHO yielded slightly lower HC emission as compared to BCO as it burns better.

NO_x Emission

The effect of change in fuel IT on the emissions of NO_x with diesel and BDF at 80% and 100% loads is depicted in Figure 6. From Figure 6, it is noted that the NO_x emission increased with advanced IT. It is well known fact

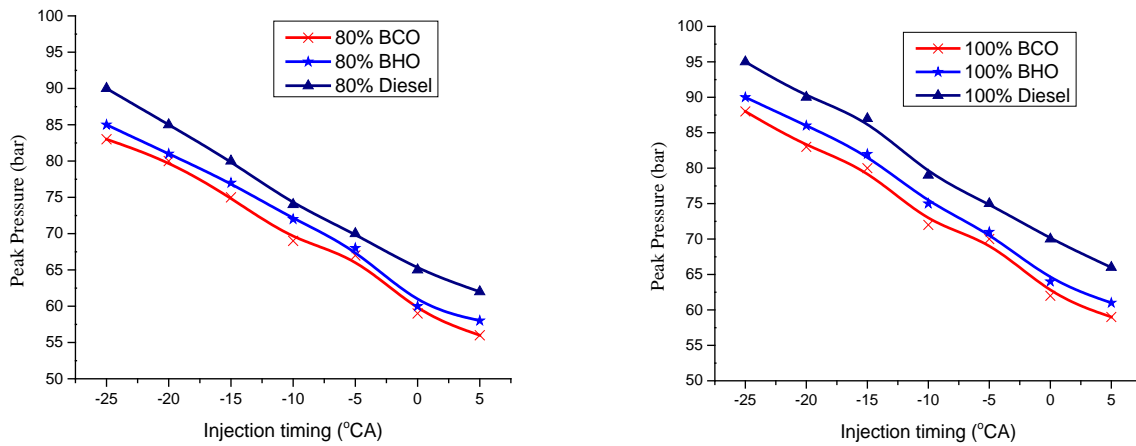


Figure 9. Variation in the PP at different fuel IT at 80% and 100% loads

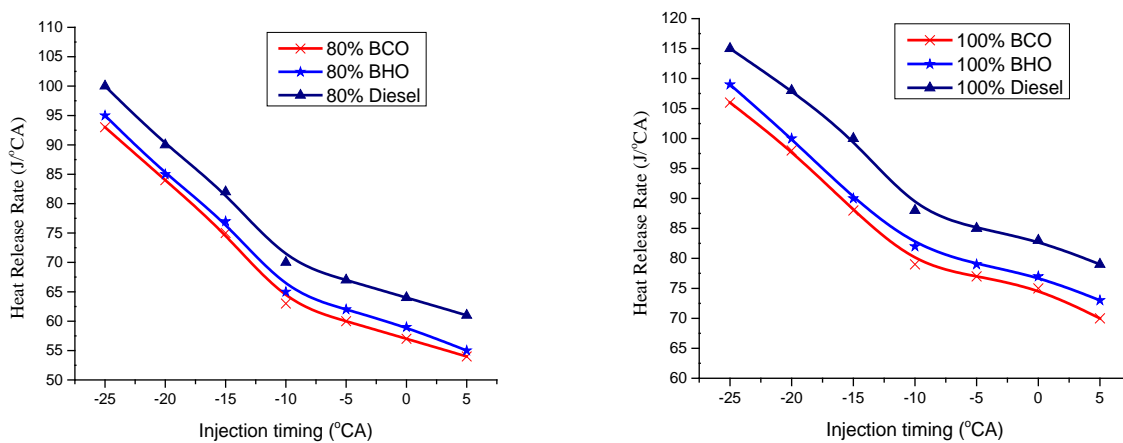


Figure 10. Variation in the HRR at different fuel IT at 80% and 100% loads

that at advanced fuel IT, the PP and HRR increases due to longer ID. This resulted in higher peak cylinder gas temperature which elevated the engine out NO_x emission. Similar results were seen in the literature (Mikulski et al., 2016). NO_x emissions in the engine out gas were lower for BDF operation compared to mineral diesel operation. This is because of lower gas temperature inside the CC and lower CN of selected BDF. The NO_x emission with BDF were lower by about 33% to 38% at fuel IT of 10° CA bTDC for 80% load and 100% load compared to CI mode.

Combustion Parameters

The ID, CD, PP, and HRR variation with BHO and BCO for different fuel IT of 25°CA bTDC to 5°CA aTDC at constant IP of 600 bar is shown in Figures 7 to 10. From Figures 7 and 8, it can be seen that at all fuel IT's (25°bTDC to 5°aTDC) the ID and CD of the BDF were found to be higher than the mineral diesel. Both ID and CD decreased with retarded IT up to 10°bTDC then showed increasing trend. The reason could be the higher gas temperature obtained at retarded IT. From Figures 9 and 10 it can be seen that at all fuel IT (25°bTDC to 5°aTDC) the PP and the peak HRR of the BDF were found to be lower than the mineral diesel. PP and peak HRR of the BDF were lower due to deterioration in preparation of air-fuel mixture observed with such high viscosity and lower volatile fuels. In addition, the lower energy content of the BDF resulted in reduced PP and HRR. The gas PP and HRR were found higher at advanced fuel IT because of increased ID period. Also more fuel available for chemical reaction in the premixed combustion phase could also be responsible for the trend observed. Similar results were reported in the literature (Hayes et al., 1986; Sayin et al., 2009).

Injection Pressure (IP) Optimization for the Diesel, BDF Powered CRDI Engine

Experiments were conducted on a CRDI engine powered with diesel and two selected BDFs to study the influence of fuel IP on its performance. The fuel IP was varied keeping the fuel IT constant at 10°CA bTDC and the engine speed was maintained constant at 1500 rpm. A CRDI injector of 6 holes each of 0.2 mm orifice size was adopted in the study. For the study only two loads were selected (80% and 100%).

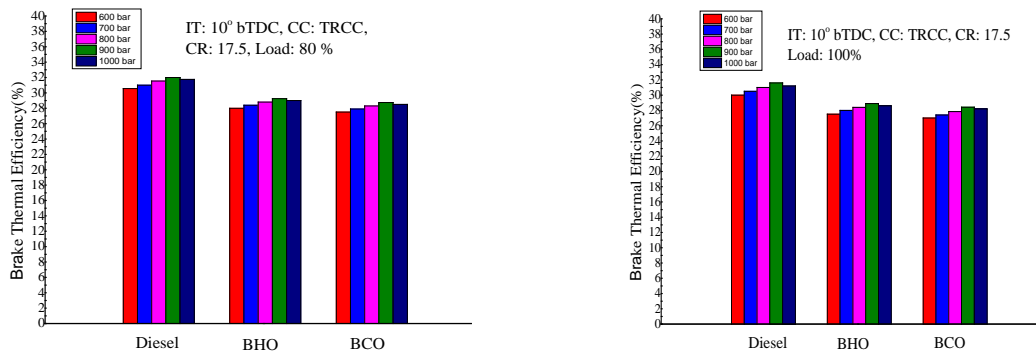


Figure 11. Variation in the BTE with the IP at 80% and 100% loads

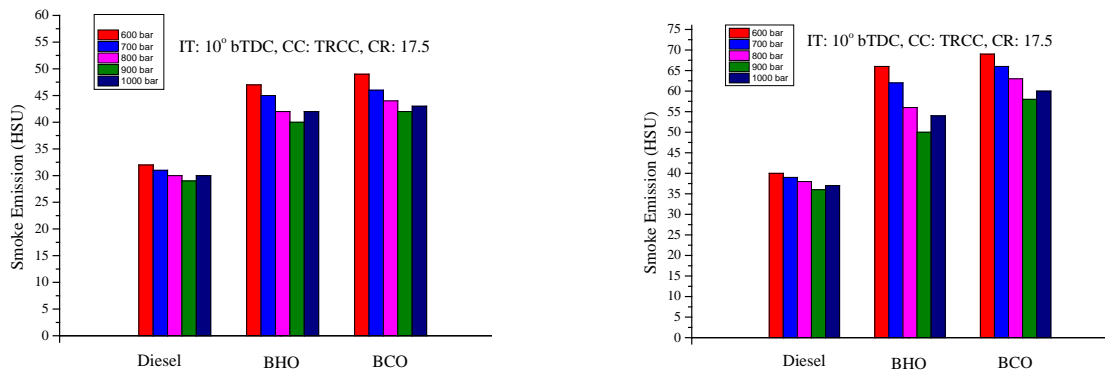


Figure 12. Variation in the smoke emission with the IP at 80% and 100% loads

Performance: Brake Thermal Efficiency

Figure 11 shows the effect of variation in fuel IP on the BTE at different fuel IP (600, 700, 800, 900 and 1000 bar). As the fuel IP was increased, the BTE of the engine increased up to IP of 900 bar then it showed decreasing trend as shown in Figure 9. Improved atomization resulted in homogeneous mixture and reduced ID at higher pressure up to 900 bar. However at fuel IP of 1000 bar BTE decreased slightly due to increased wall wetting. The higher fuel dispersion and penetration at higher fuel IP could also be the reason for higher BTE. Similar results were seen in the literature (Lee and Park, 2002; Leung et al., 2006). Amongst all the fuel IP used for the study, the maximum BTE was obtained at IP of 900 bar. The BTE were lower for both BDF compared to mineral diesel. Lower CN, higher FFA and lower volatility of the BDF injected resulted in later SOC. The peak HRR for BDF were lower compared to that of mineral diesel under the same engine operating conditions. Maximum BTE was achieved with BHO and BCO and were lower by 6.4% and 8% respectively at 80% load and IP of 900 bar compared to the CI mode. At 100% load slight reduction in BTE by 1.5% was seen for all fuels.

Emissions: Smoke Emission

Figure 12 depicts the effect of variation in fuel IP on the smoke emission at 80% and 100% loads. The smoke emission were lower at higher fuel IP of 900 bar and it might be due to enhanced atomization resulting in complete combustion. Biodiesels have heavier molecular structure due to their higher viscosity. The improper air fuel mixture formed resulted into higher smoke emissions. The lowest smoke level achieved at a fuel IP of 900 bar. The smoke emissions of the CRDI mode with BDF were higher by 26-32% at 80% and 100% load compared to CI mode.

HC and CO emissions

Figures 13 and 14 shows the effect of change in fuel IP on the engine out HC and CO emissions at 80% and 100% loads. At higher fuel IP the decreasing trend of engine out HC and CO emissions for all fuels resulted and might be due to complete combustion of the injected fuels on account of better atomization and mixing with air. Good ignition qualities and higher oxygen content of both BDF produced lower engine out HC and CO emissions but these were slightly higher than mineral diesel. The HC emission achieved with BDF were higher by 8% to 14% at 80% and 100% load compared to CI mode. Similarly the CO emission with BDF were higher by 11% to 23%

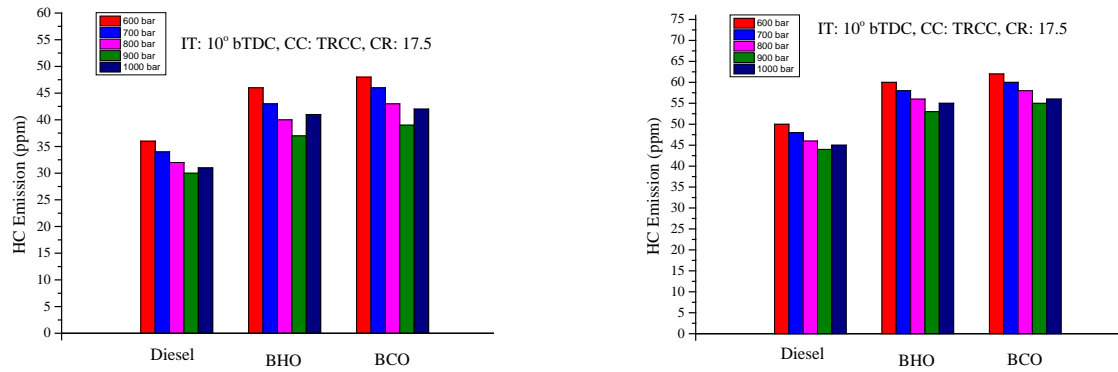


Figure 13. Variation in the HC emission with the IP at 80% and 100% loads

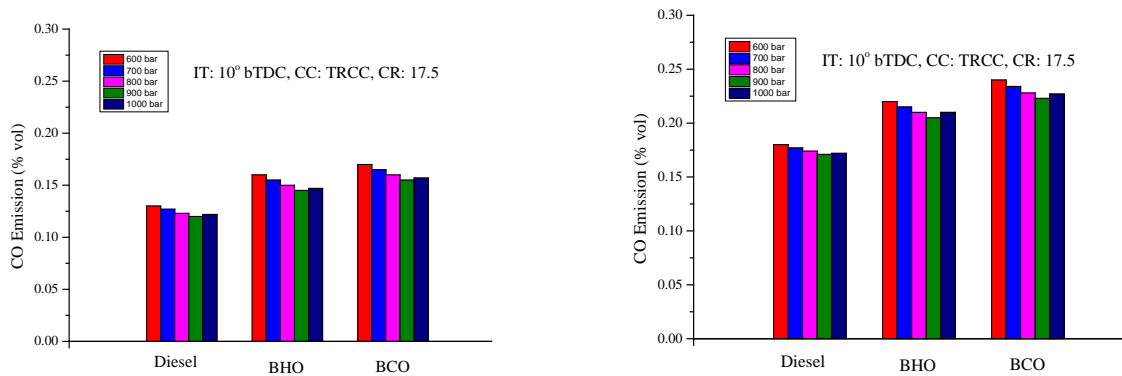


Figure 14. Variation in the CO emission with the IP at 80% and 100% loads

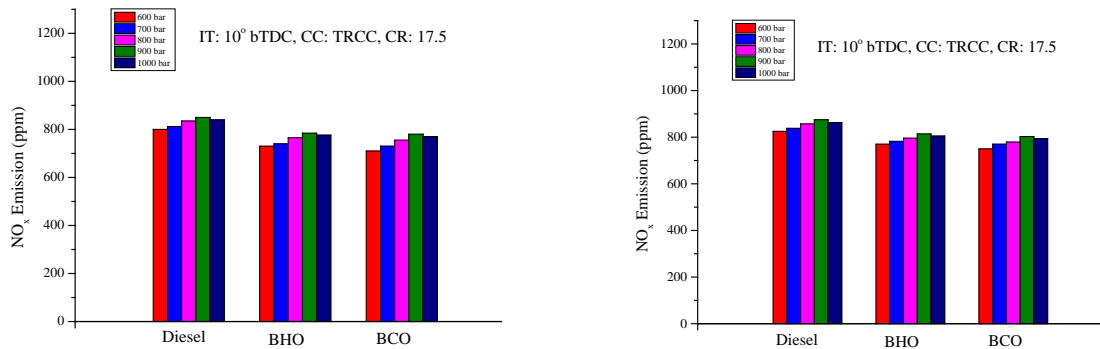


Figure 15. Variation in the NO_x emission with the IP at 80% and 100% loads

at 80% load and at 100% load compared to CI mode. Also the lower values of BTE with BDF were responsible for these trend observed. A fuel IP of 1000 bar resulted in slightly higher engine out HC and CO emissions due to increased wall wetting. At higher IP the fuels spray reaches CC wall quickly as it penetrates through longer distance.

NO_x Emissions

The change in engine out NO_x emission in the exhaust gas of the engine with fuel IP is shown in Figure 15 at 80% and 100% loads. The enhanced combustion rate due to better fuel atomization and dispersion increased cylinder gas temperature in the cycle at higher fuel IP. Diesel yielded higher engine out NO_x emission compared to BD fuels as these have higher viscosity and lower energy density. The increased fuel dispersion and smaller droplet size obtained due to higher fuel IP led to better air fuel mixing which decreased ID. This increased the HRR and resulted into higher in-cylinder gas temperature which increased the engine out NO_x emissions. The BDFs showed reduced premixed combustion and hence slightly lower engine out NO_x emissions as compared to mineral diesel. The lower adiabatic flame temperature and CN of BDF resulted into lower level of engine out NO_x emissions. The NO_x emission achieved in the CRDI mode with BDF were lower by 26% to 32% at 80% and

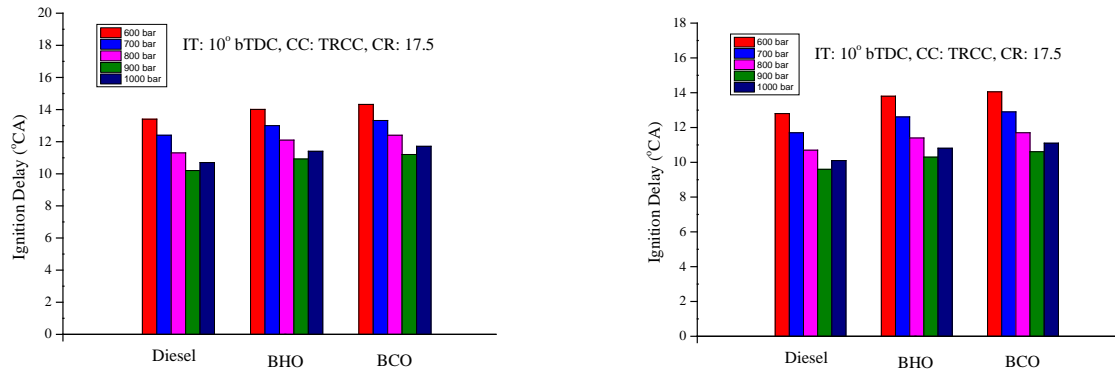


Figure 16. Variation in the ID with the IP at 80% and 100% loads

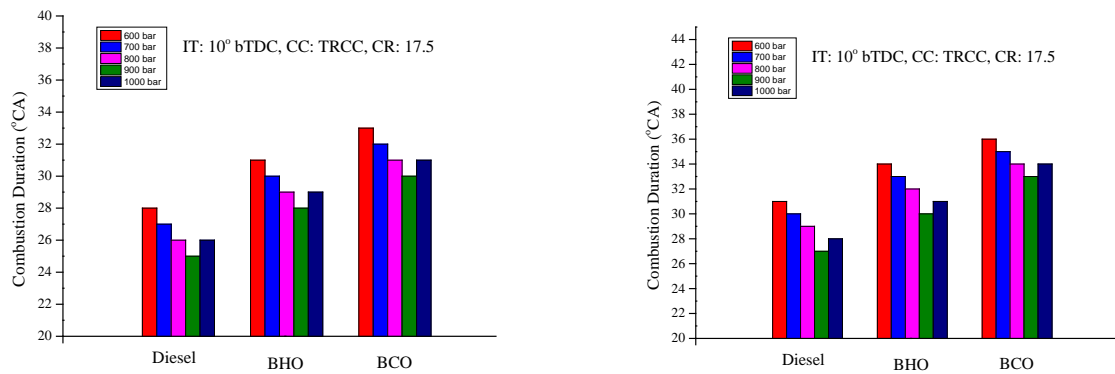


Figure 17. Variation in the CD with the IP at 80% and 100% loads

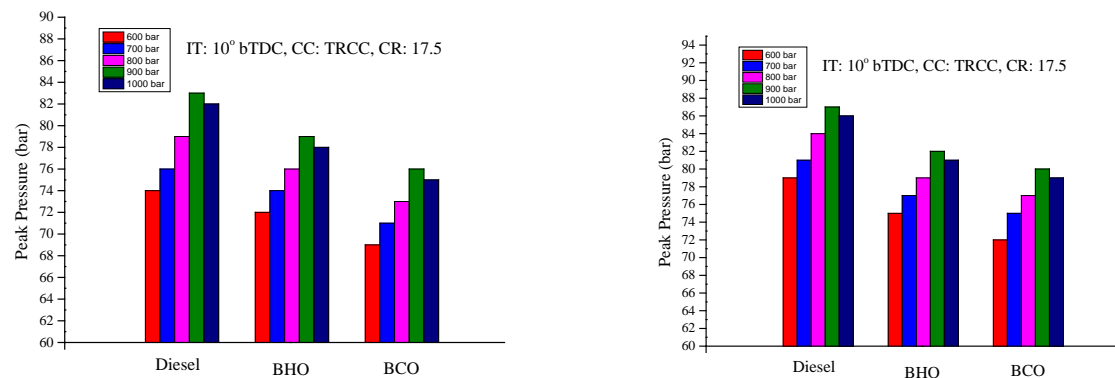


Figure 18. Variation in the PP with the IP at 80% and 100% loads

100% load compared to CI mode. NO_x emission of BHO was slightly higher as compared to BCO due to higher in cylinder temperature that prevailed and HRR.

Combustion Parameters:

The ID, CD, PP and peak HRR variation for CRDI engine powered with diesel and BDF at different fuel IP and at a constant fuel IT of 10°CA bTDC is shown in Figure 16 and Figure 19. PP increased with increase in fuel IP till 900 bar beyond which it reduced. Similar trends could be seen for peak HRR as shown in Figure 16. Better fuel atomization at higher IP and reduced ID resulted in enhanced burning of injected fuels there by the increase in PP and HRR was observed. Similar results could be seen in the literature (Grimaldi et al., 2002; Leung et al., 2006; Ozsezen et al., 2008). Slight decrease in PP and HRR at 1000 bar IP could be due to fuel entering in to the crevices escaping the main combustion and also could be due to delayed injection negating the performance. The BDFs is more viscous which creates higher friction around the injector needle there by indicating slow needle-lift

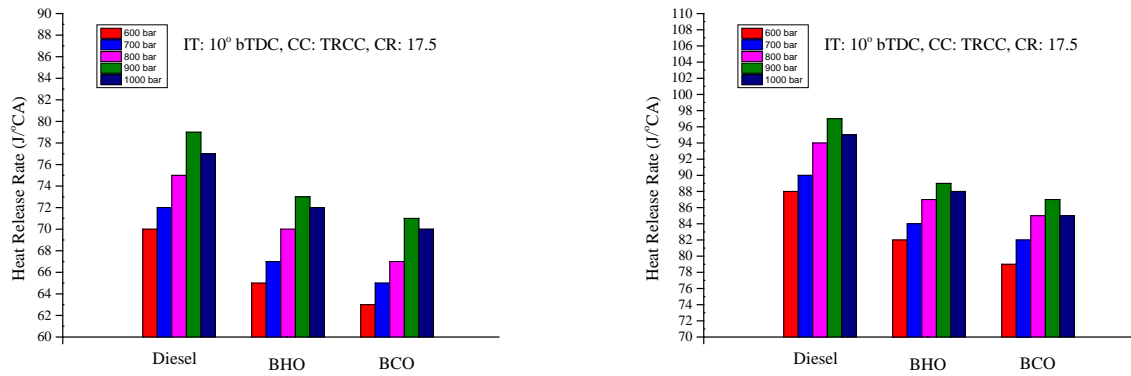


Figure 19. Variation in the HRR with the IP at 80% and 100% loads

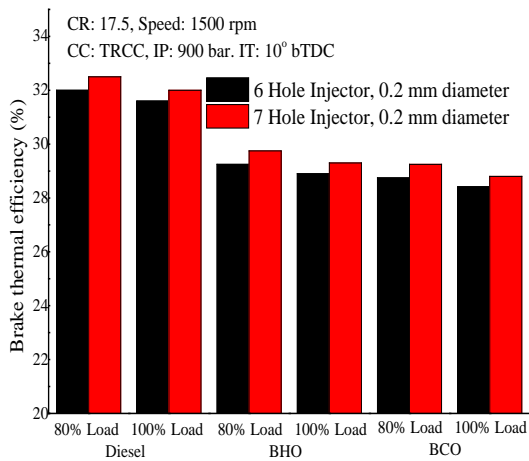


Figure 20. Comparison of the BTE of the CRDI engine using 6 and 7 hole injectors

movement with longer injection delay. At the same SOI of fuel, increased fuel IP changed the heat release profile that is higher fuel IP led to SOC quite early in the cycle and also found higher with narrowed heat release peak.

Effect of Number of Injector Holes on the Performance, Emission and Combustion Characteristics of CRDI Engine Powered with Diesel and Biodiesels

The experimental tests were conducted on CRDI engine fuelled with diesel, BHO and BCO at 80 and 100% load to investigate the effect of number of injector holes on its performance. The results of 6 and 7 hole injectors of single injection CRDI operation are discussed in the following section. Both 6 and 7 hole injectors had the orifice diameter of 0.2 mm. The injection of fuel was carried at an IT of 10°bTDC and IP of 900 bar.

Performance: Brake Thermal Efficiency

Figure 20 shows the comparison of the BTE of the CRDI engine with all fuels for 6 and 7 hole injectors. Improved BTE with 7 hole injector compared to 6 hole for all the fuels and injection strategies are shown in Figure 20. Lower penetration of fuels injected with former reduced the wall impingement as mass low rate per hole gets reduced when number of holes increased from 6 to 7. More quantity of fuel participated in combustion and higher percentage vaporization was resulted with 7 hole nozzle and could be the other reasons for higher BTE. Lower CN, higher viscosity, lower volatility of biodiesels led to poor atomization compared to diesel which resulted in lowered BTE compared to diesel.

Emissions: Smoke Emission

Figure 21 depicts the comparison of smoke emissions of CRDI engine operation with 6 and 7 hole injectors. Lower smoke emissions with 7 hole injector compared to 6 hole for all the fuels and injection strategies were obtained. Reduced wall impingement and complete combustion of fuel might be the reasons for lowered smoke emissions with 7 hole nozzle. Higher percentage vaporization with 7 hole nozzle could be another reason for decreased smoke emission. BDF showed higher smoke emissions compared to mineral diesel fuel due to their

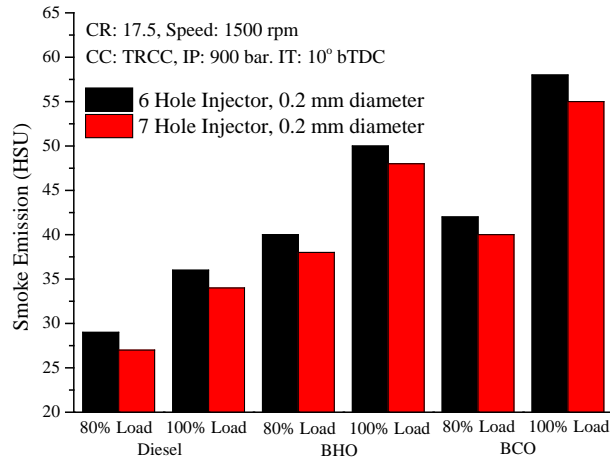


Figure 21. Comparison of the smoke emission of the CRDI engine using 6 and 7 hole injectors

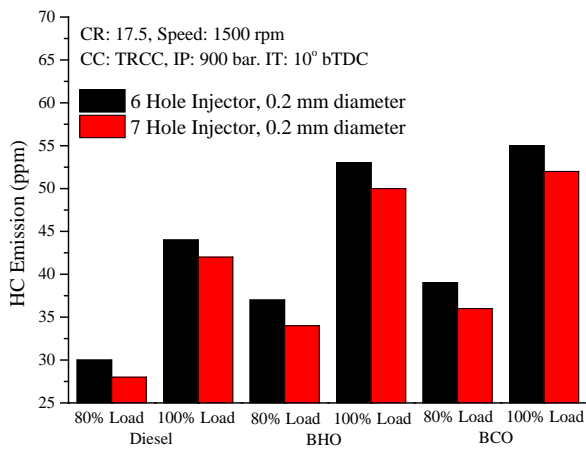


Figure 22. Comparison of the HC emission of the CRDI engine using 6 and 7 hole injectors

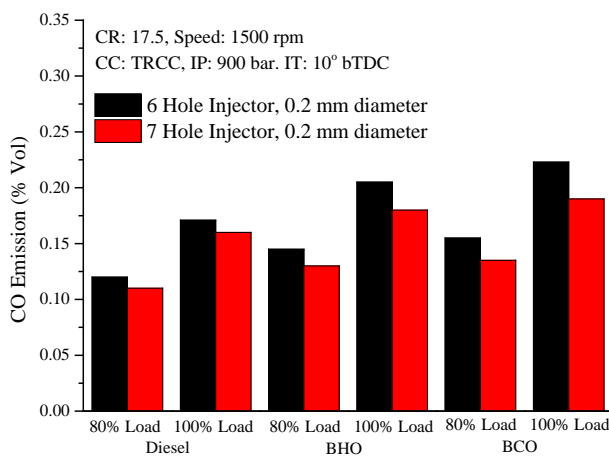


Figure 23. Comparison of the CO emission of the CRDI engine using 6 and 7 hole injectors

lower CN, higher viscosity with lower volatility character that led to poor atomization the IP being same for all fuels. The smoke emission reduced by 20% to 26% in diesel and biodiesel powered CRDI engine as compared to CI mode. BCO showed slightly higher smoke emission as compared to BHO due to its lower burning rate.

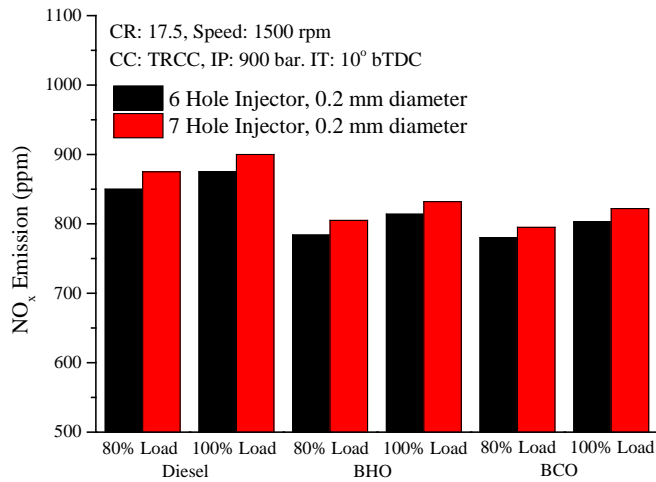


Figure 24. Comparison of the NO_x emission of the CRDI engine using 6 and 7 hole injectors

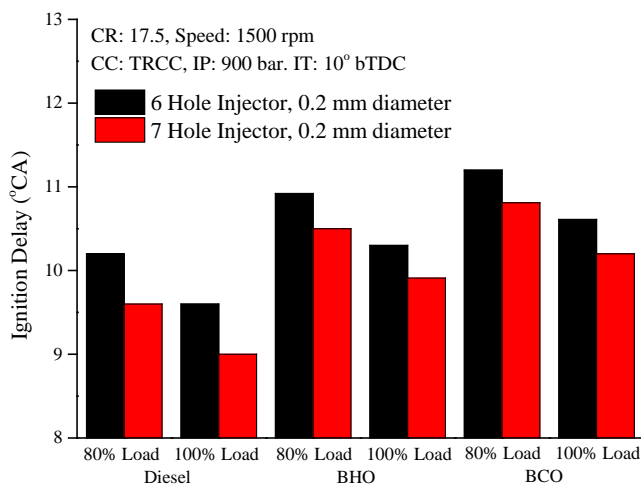


Figure 25. Comparison of the ID of the CRDI engine using 6 and 7 hole injectors

HC and CO Emissions

Figures 22 and 23 indicates the comparison of the engine out HC and CO emissions of CRDI engine operation with 6 and 7 hole injectors. The CRDI operation resulted in lower HC and CO emissions with 7 hole injector compared to 6 hole for all the fuels and injection strategies. Lower fuel penetration distance reduces the wall impingement as mass flow rate per hole gets reduced. A larger quantity of fuel participated in combustion and higher percentage vaporization that resulted with 7 hole nozzle and could be other reasons for lower trend of HC and CO emission. Biodiesels showed higher HC and CO emissions compared to diesel fuel due to their lower CN, higher viscosity with lower volatility character that led to poor atomization compared to diesel though the IP was same for all fuels.

NO_x Emissions

Figure 24 illustrates the comparison of engine out NO_x emissions of CRDI engine operation with injector of 6 and 7 holes. The CRDI operation resulted in higher NO_x emissions with 7 hole injector compared to 6 hole for all the fuels and injection strategies adopted. Lower penetration distance obtained with the former further reduces the wall impingement as mass flow rate per hole gets reduced and more fuel participate in combustion process and this could increase the in cylinder gas temperature, PP and HRR and hence the NO_x emission increased. Higher percentage vaporization with 7 hole nozzle could be another reason for higher NO_x emission. BDF showed lower NO_x emissions compared to mineral diesel due to their lower CN, higher viscosity and lower volatility which led

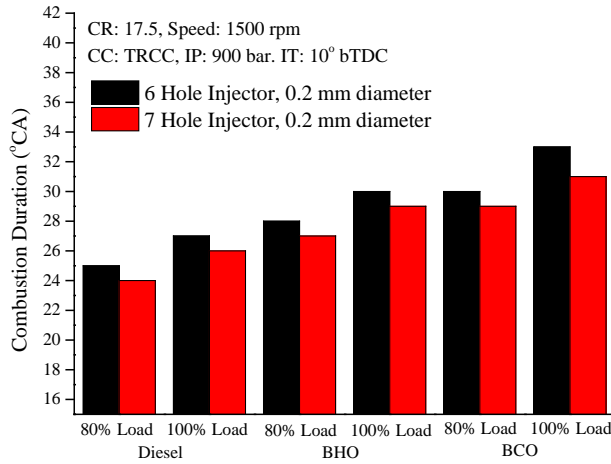


Figure 26. Comparison of the CD of the CRDI engine using 6 and 7 hole injectors

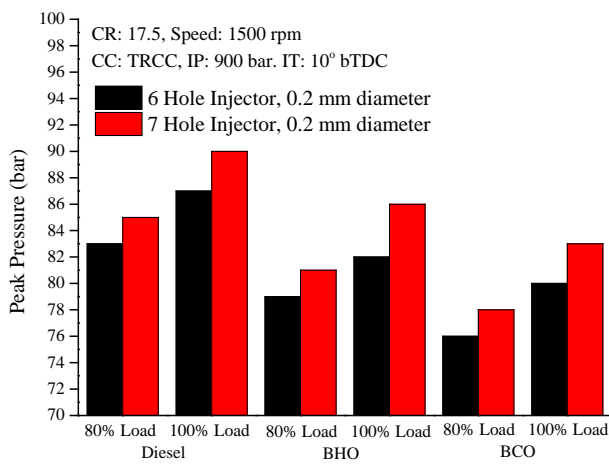


Figure 27. Comparison of the PP of the CRDI engine using 6 and 7 hole injectors

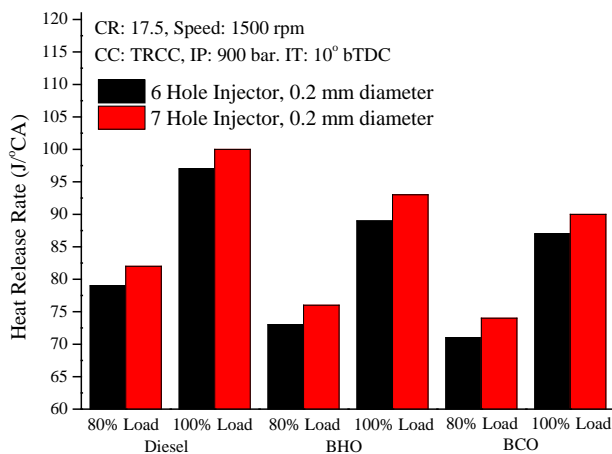


Figure 28. HRR of the CRDI engine using 6 and 7 hole injectors

to poor atomization. NO_x reduced by 16% to 20% in diesel and biodiesel powered CRDI engine as compared to CI mode.

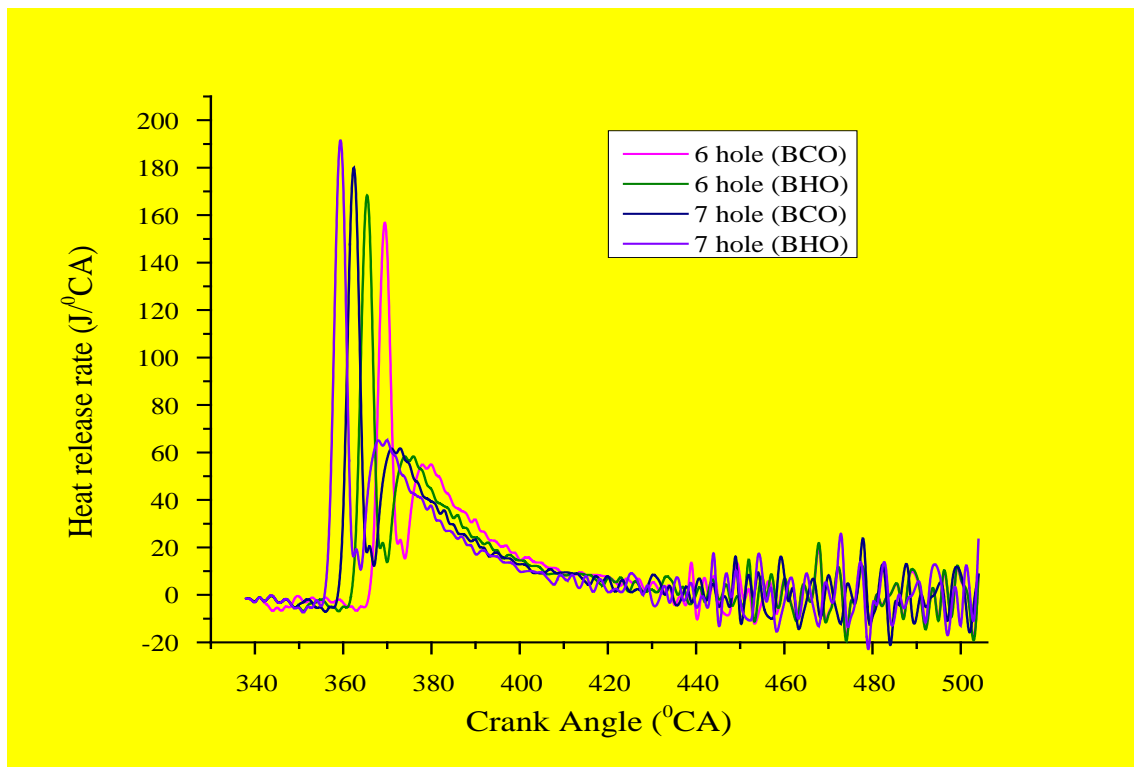


Figure 29. Variation of HRR with crank angle of CRDI engine for 6 and 7 hole injectors using BCO and BHO

Combustion Parameters

Comparison of ID, CD, PP and Peak HRR of CRDI engine operation with 6 and 7 hole injectors is provided in [Figures](#) from 25 to 29 respectively. From the [Figures](#) 25 and 26, it can be seen that the ID and CD of the BDF were higher than the diesel at both loads. But the ID found lower at 100% load compared to 80% and on the other hand the CD was higher at 100% load compared to 80%. Also 7-hole injector resulted in to lower ID and CD as compared to 6 hole injector for all fuels and could be due to higher gas temperature leading to more homogeneous mixture formation. The diesel and BDF powered CRDI engine with injector of 7 holes yielded 6% to 11% lower ID and 15% to 19% lower CD as compared to CI mode.

From the [Figures](#) 27 and 28, it can be seen that the PP and the peak HRR of the BDF were found lower as compared to diesel. The PP and peak HRR of the BDF were lower due to deterioration in preparation of air-fuel mixture as a result of high viscosity and lower volatility of fuels. In addition, the lower energy content of the BDF resulted in reduced PP and HRR. A 7 hole injector resulted in to higher PP and HRR as compared to 6 hole injector for all fuels due to better air fuel mixing. Also lower ID allowed more fuel available for chemical reaction in the premixed combustion phase which resulted in higher PP and HRR. The diesel and BDF powered CRDI engine with injector of 7 holes yielded 10% to 14% higher PP and HRR as compared to CI mode. The ID and CD of BCO was higher compared to BHO but PP and peak HRR were lower which reflects the poor combustion qualities of BCO. Variation of HRR with crank angle is shown in [Figure](#) 29. HRR was found to higher for 7 hole injector compared 6 hole injector due to better premixed combustion phase is likely to be more prominent with increased number of holes and this will affect HRR. For the same hole injector, BHO results in higher HRR compared to BCO due to differences in the fuel properties.

CONCLUSIONS

CI engine was suitably modified to operate with CRDI facility at 1500 rpm using TRCC and injector of 6 and 7 holes with holes of orifice 0.2 mm diameter each. From the detailed experimental study on CRDI engine powered with diesel and BDF, the following conclusions were drawn:

- The optimum fuel IT and IP for the CRDI engine were 10°bTDC and 900 bar respectively for all fuels tested.
- The diesel and BDF powered CRDI engine with an injector having 7 holes yielded 5% to 6% higher BTE as compared to CI mode.

- Lowered HC, and CO emissions with 7 holes injector respectively by 15% to 18%, and 11% to 16% as compared to CI mode.
- Smoke emission reduced by 20% to 26% while NO_x reduced by 16% to 20% in diesel and biodiesel powered CRDI engine as compared to CI mode.
- The diesel and BDF powered CRDI engine with an injector of 7 holes yielded 10% to 14% higher PP and HRR as compared to CI mode.
- The diesel and BDF powered CRDI engine with injector of 7 holes yielded 6% to 11% lower ID and 15% to 19% lower CD as compared to CI mode.
- BHO showed better performance in terms of higher BTE and lower emissions except NO_x as compared to BCO due its better combustion qualities. Also combustion characteristics of BHO were better as compared to BCO.

Overall it can be concluded that the modified CI engine operated in CRDI mode worked satisfactorily with selected BDFs at optimum fuel IT and IP of 10°bTDC and 900 bar respectively. CRDI engine operation with 7 holes injector yield improved BTE and resulted in a drastic reduction in smoke and NO_x emissions as compared to CI mode. Both BDFs were well suited for CRDI engine operation which can reduce the burden on oil import bill.

NOMENCLATURE

aTDC – After top dead centre
 BDF – Biodiesel fuel
 BHO - Honge oil biodiesel
 BSFC - Brake specific fuel consumption
 bTDC – Before top dead centre
 BTE – Brake thermal efficiency
 CA – Crank angle
 CC – Combustion chamber
 CD – Combustion duration
 CI – Compression ignition
 CN – Cetane number
 CO – Carbon monoxide
 CRDI - Common rail direct injection
 ECU – Electronic control unit
 EGR - Exhaust gas recirculation
 FFA – Free fatty acid
 HC – Hydrocarbon
 HCC – Hemispherical combustion chamber
 HRR - Heat release rate
 ID – Ignition delay
 IMEP - Indicated mean effective pressure
 IP – Fuel injection pressure
 ISFC - Indicated specific fuel consumption
 IT – Fuel injection timing
 NO_x - Oxides of nitrogen
 PP - Peak pressure
 SI – Spark ignition
 SMD - Sauter mean diameter
 SOC - Start of combustion
 SOI - Start of fuel injection
 TDC - Top dead centre
 TRCC - Toriodal re-entrant combustion chamber

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