An Experimental Investigation on Performance and Emission Characteristics of PCCI Engine Using Biodiesel-Ethanol Blends in Dual Fuel Mode

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Abstract. The global automotive industry is faced with the task of reducing global greenhouse gas emissions generated by vehicle exhaust. Heavy machinery like trucks, trailers, off-road vehicles, etc. powered by the diesel engines. These vehicles emit large quantities of NOx and smoke emissions. The simultaneous reduction of both emissions is the key challenge for the automotive industry. Burning a homogeneous charge at relatively low temperature seems to be the means to control both the emissions simultaneously. In this experimental study, the combustion, performance and emissions characteristics of a premixed charge compression Ignition Engine (PCCI) with ethanol injected in the intake manifold of the engine along with biodiesel (Palm oil methyl ester) injected directly into the combustion chamber. The experiments were carried out in a four-stroke, single cylinder vertical water cooled, constant speed diesel engine with the range of 10- 30% premixed ethanol fuel from no load to full load condition is studied. The experiments were conducted with an engine operating in PCCI mode with biodiesel – ethanol blends in dual fuel mode. The experimental results showed that there was a slight increase in Hydrocarbon (HC) and Carbon monoxide (CO) but there was a reduction in Nitrous-oxide (NOX) and Smoke emissions for the blend containing 70% biodiesel and 30% ethanol.

Keywords: Biodiesel;Dual blend;ethanol;PCCI engine;Hydrocarbon;Smoke emission;Ethanol.

1. Introduction
The constant challenges facing day to day are increased emissions and fuel rate consumption which are the environmental as well as energy challenges respectively. To supply the demand for fuel economy, environmentally less dangerous emissions, conservation of energy and, partial Premixed Charge Compression Ignition combustion system idea is ideal. Engine output releases include Unburnt Hydrocarbons, CO, Oxides of Nitrogen, Smoke and matter of particulates, etc; Emissions from the engines remain a condematory situation damaging the layout of design and working of Internal Combustion engines. One favourable way to resolve that kind of problems is to understand Homogeneous Charge Combustion (HCC) at maximum low temperature and with a compression ratio which is higher at that time, which is the last word goal [1]. In HCCI the homogenous mixture and control of combustion are difficult. Hence we have moved to partial premixing of charge which is called Partially Premixed Charge Compression Ignition (PCCI) engine. The aftermath of petroleum fuel which has premixed and correct timid direct injection on half the amount of HCCI were noted and inspected. The answers showed a constant lowering of emission and oxides of nitrogen with a modest CO increase and unburned carbon. To get out of the tough situation, the PCCI idea is taken. Despite the fact that HCCI can give out smoke and benefits of fuel consumption compared to DI-CI Engine it is now still a concept due to its operating difficulties [2]. To come out of these hardships, the PCCI idea is taken for decreasing the emissions of the NOXs and soot. The operation of PCCI involves the pre-mixed charge preparation outside of the cylinder. A little fuel is sent inside with the incoming air at the intake manifold and...
the mixture then goes inside the combustion chamber and the remaining fuel is ejected inside as usual. The diesel injection using port injection is not easy for the surroundings if it is at extreme low temperature for the fuel to dry up. In the CI, the emission and combustion are affected by the atomisation happened and particularly, by the mixture inside the chamber[3]. Several processes are done to achieve correct vaporization which is taking place in the intake manifold. During EGR, air is preheated and premixing chamber was used. Every processing has both success and failures. Employment of hot GCR vaporization of large scale is noted, but it would increase the chamber temperature also, therefore NOx production is increased, and hence the EGR needs a non-manual control mechanism. In the case of EGR is used under various loads conditions[4]. The estimated density, kinematic viscosity, and flashpoint for Palm based oil and biodiesel to mix with the diesel. Results tell the quality of fuel that blends are very high to that of diesel until 30% less than other similarities inside certain restrictions. The experimented data were related as a quantity fraction of oil within the mix. Different comparisons are experimented to note the bio- oil-diesel mix properties supported by our experimental result analysis. The notes relations were calculated by comparing the predictions with experimental data. The equations which were analysed can be used as a frontier for giving diesels engines the best test blending mix. They analysed the total qualities of the C.I. engine charged with the blended diesel. Biodiesel and palm oil methyl ester mix were made in volume percentages of 20 and 100%. The biodiesel mixtures qualities were the same as diesel oil. The experimentation is set on a CI engine and it equals with diesel[5]. UBHC and CO emissions for biodiesel blends have decreased but vice-versa for oil blends comparing it to diesel. NOx emissions have been increased for biodiesel. Blends of diesel to biodiesel up to percentage of biodiesel by volume are recommended [3-6].

The inspection ratios of emulsified fuels on all basic properties of single cylinder 4 stroke Kirloskar engine under various loads[6]. For this particular test Benzalkonium chloride is mixed to the diesel blend as an emulsifier to resist formation of layers and to be made as a blend of homogenous mixture. At top most BP, the contrast between the diesels and emulsified fuels shows improved properties in brake power efficiency in results with depletion in smoke and SFC. The CO2, HC, cylinder force, and warmth rate of release for D50 E40 emulsions are greater than diesel[7-8]. The partial Premixed Charge Compression Ignition-Direct Injection (PCCI-DI) Engine with premixed fuels such as ethanol and gasohol is injected with mixed diesel and all the basic properties such as performance, emission etc has been analysed and investigated. The experiments were done on all load conditions in a naturally aspirated air-cooled 4 stroke engine. The premixed fuel additive strengthens the strength of fuel-air mixture and the combustion duration is made lesser in bi-fuel injection. The above study, it had been noted that 70% reduction in smoke from mixed gasoline and 67% from ethanol fuel in comparison to pure diesel. The emission of nitrogen oxides were also decreased to 30% and 24% for ethanol fuel and premixed gasoline. In Particular smoke, NOx emissions are reduced in premixed gasoline slightly equal to the amount of ethanol, brake thermal efficiency was increased gradually and noted in 20% and ethanol and premixed gasohol in bi-fuel mode, in comparison to pure diesel experiment [9].

They checked the power output and emitted pollutant properties of a CI engine filled with ethanol and palm-oil methyl ester. The performance and exhaust emission characteristics of Palm oil methyl ester biodiesel (PB10, PB20, PB30) adding ethanol (E5% and E10%) in an sudden Injection – Compression Ignition engine over the changing loading conditions on the engine and found lesser CO emissions. The corrections can be done through a smaller amount of Carbon Monoxide emissions and Exhaust Gas centigrade reduced inside the raises of doping of ethyl alcohol content in mixture of Methyl Ester and its mixture [10]. They investigated the causes of double bio-fuel (Jatropha Methyl esters and purified Turpentine oil) on a cylinder normally aspirated diesel (Kirloskar) unaccompanied exhaust gas recirculation.
Jatropha Methyl esters and purified turpentine oil have a more and less viscosity fuel addition with comparative boiling values to that of diesel; so this will offer use in a compression-ignition engine. The further conclusion of BT50 in all load conditions, depletion for 2.9%, 4.56%, 4.72%, 29.16%, 42.5% in brake energy efficiency, smoke, CO, NOx and HC even though the CO2 emissions raises 10.7% [11]. This review suggests that fuel injection is a suitable way for HCCI mode, so its dividend to the compression-ignition engine requires be analysing and studying. Inside a diesel engine compression process takes place based fuel injection into the combustion chamber. The necessary properties are timing and period of the ejection of fuel working, the giving out of the fuel and atomization in the combustor, the ignition timing, the injected fuel are proportional to the rotation of the crankshaft, and the complete fuel injected in connection to powerhouse(engine) capacity. The arrangement idea has developed and introduced to the matter more emission levels of conventional direct injection compression-ignition engine systems. During early stages the premixed charge compression ignition technique is used in injections for the burning process when a higher amount of fuel is burned, which disturbs combustion noise. The combustion noise reduced by using a tool of pilot injection usage. The most important motive of this article is to analyse by experimenting the combustion noise, emissions, and diesel performance in operation below premixed charge compression ignition in this operation of pilot injection. Besides, a special process supported the atomization of the in-cylinder force signal, combustion noise analysis was experimented.

2. Experimental setup

Internal combustion liquid cooled which has vertically fixed engine type which rotates at 1500rpm. The addition of loads is done through an electrical dynamometer. In an unit time the fuel quantity can be measured by a burette which is linked to the tank. The exhaust gas was measured by AVL437C smoke meter. A test rig is connected with a data acquisition system for obtaining different curves and results during experimental investigation. A five gas analyzer aligned with a stationary engine to provide composition details of emission in exhaust gas are CO, CO2, UBHC, NOx, and other non-useful oxygen. In this method, the cable at one end is connected to the exhaust gas outlet while the other is at the end of the inlet of the analyzer. Figure 1 and 2 shows the experiment setup. Tabulation 1 and 2 shows Engine Specification and properties of fuel.

![Fig. 1 Experimental Setup](image-url)
1- Engine  
2- Flywheel  
3- Dynamometer  
4- Air Surge Tank  
5- Auxiliary Fuel Injector  
6- Main Fuel Injector  
7- Exhaust Gas Analyzer  
8- Flow Meter  
9- Main Fuel Tank  
10- Port Fuel Tank  
11- Flow Meter  
12- Data Acquisition  
13- Relay Controller  
14- Charge Amplifier  
15- Fuel Control Valve  
16- Pressure Transducer  
17- Crank Angle Encoder  
18- Fuel Injection Pump  

Fig.2 Experimental Setup Table

1 Technical specification of the test engine

| Engine type                  | Kirloskar Oil Engine TV1 |
|------------------------------|--------------------------|
| Bore                         | 87.5 mm                  |
| Stroke                       | 110 mm                   |
| Rated power output           | 5.20 kW                  |
| Rated speed                  | 1500 rpm                 |
| No. of cylinder              | 1                        |
| Cooling system               | Water-cooled             |
| Injection pressure           | 200 bar                  |
| Injection timing             | 23 deg bTDC              |
| Compression ratio            | 17.5 : 1                 |
| Connecting rod length        | 234.00(mm)               |
Table 2 Comparison of Fuel properties

| Properties             | Units | Diesel | Palm Oil Methyl Ester | Ethanol |
|------------------------|-------|--------|-----------------------|---------|
| Auto Ignition Temperature | °C    | 210    | 343                   | 363     |
| Octane Number          | Nil   | -      | -                     | 102     |
| Cetane number          | Nil   | 51     | 47                    | -       |
| Density at 15°C         | kg/m³ | 873    | 904                   | 786     |
| Flash Point            | °C    | 65     | 196                   | 26      |
| Fire Point             | °C    | 75     | 210                   | 34      |
| Calorific Value        | kJ/kg K | 45000 | 37500                 | 29700   |

The below equation is used to calculate the premixed fuel ratio ($m_p$);

$$ r_p = \frac{m_p h_p}{m_d (h_p + m_d) h_d} $$

Where $m_p$ be the premixed fuel mass consumed and $m_d$ be the directly injected conventional fuel correspondingly, where $h_p$, $h_d$ are calorific values of premixed fuel ratio and diesel. The experimental tests were taken out for a premixed fuel ratio of 0.1, 0.2 & 0.3 respectively. The duration of fuel injection was calculated within milliseconds and position in an electronic control unit. The premixed fuel ratio is decreased by using electronic control unit (ECU). The ratio for the experiment is the diesel – 100%, Biodiesel (Palm oil methyl ester) 100%, Biodiesel 90%, and 10% Ethanol, Biodiesel 80%, Ethyl alcohol 20%, Methyl esters 70% and Ethanol 30%.

3. Results and Discussion

In the paperwork converse the burning characteristics, amount of heat release for the crank angle manifold injected premixed ethanol, and diesel oil was compared with palm oil. In the same way, the performance emission characteristics of curb energy efficiency and curb particular fuel expending and discharge attribute such as curb particular CO, NOx, curb particular HC and pollution emission was compared with diesel oil. In this paper the result shows the full experimentation of alternative-fuel to advanced want of HCCI ignition.

Combustion Analysis

Load condition at 50%

The crankshaft angle for pure diesel is compared with cylinder pressure figure 3 is given below, biodiesel, biodiesel-ethanol mixture at 50% load condition. It is observed that the neat diesel produced more cylinder pressure compared to B100 and biodiesel-ethanol blends. The lowest cylinder pressure was produced by the B70+E30 blend. As mentioned earlier, the cooling effect of alcohol combined with higher octane number has increased the delay period which led to the attainment of peak pressure after TDC.
Load condition at 100%

In the above graph 4 shows that the crankshaft angle for pure diesel is compared with cylinder pressure, Methyl esters, biodiesel-ethanol mixture at filled load state. This result showing that more cylinder pressure is produced by B70+E30 than pure diesel and other Methyl ester-ethanol mixture. The B100 has lower cylinder pressure. Under full load, higher combustion temperature overtakes the cooling effect of ethanol, and B70+E30 will produce higher combustion pressure thereby leading to better combustion.
Performance Characteristics
Brake Thermal Efficiency

Figure 5 shows differences in the Brake thermal efficiency with various loads for pure diesel, methyl ester, and methyl ester-ethanol mixture. The capability of combustion system is indicated by curb energy efficiency and how proficient the fuel energy is transformed to mechanical energy. From the above graph fig 5, it is seen that the maximum curb energy efficiency was produced by the pure diesel compared to biodiesel and other biodiesel-ethanol blends. However, the B70+E30 blend produced brake thermal efficiency which is closer to neat biodiesel at full load. Also, this blend performed better than other biodiesel-ethanol mixture for the whole operating load of the engine.

![Fig. 5 Variation of Brake thermal efficiency with the load](image)

Brake Specific Energy Consumption

Fig 6 indicates difference in curb energy efficiency consumption to various loads for neat diesel, biodiesel, biodiesel-ethanol blends.

\[
\text{BSEC} = \text{BSFC} \times \text{CV}
\]

From the above graph figure 6, it is seen that the particular energy consumption is found to be more when the engine was operating with B90+E10 blend and lowest with neat diesel. The B70+E30 blend produced lower specific energy consumption compared to other biodiesel-ethanol blends. This may be due to improvement in fuel vaporization characteristics leading to better combustion efficiency. Since the calorific value of methyl ester, methyl ester-ethanol mixture are found to be lower than diesel, the engine will consume more fuel to produce the same power output, leading to the increased specific fuel consumption for biodiesel and methyl ester-ethanol mixture. Also, the fact that during dual fuel mode, more energy is required for injecting both the fuels separately. This is the reason for higher energy consumptions by biodiesel-ethanol blends. The SEC is highly dependent on SFC and Calorific value.
Figure 6 Variation of Brake specific energy consumption with the load.

Emissions Characteristics
Brake Specific Unburnt Hydrocarbon (BSUBHC)

Figure 7 indicates the difference in the Hydrocarbon emissions to different loads for neat diesel, biodiesel, biodiesel-ethanol blends. The neat diesel and biodiesel have lower HC emissions than the Biodiesel-Ethanol blend. The B70+E30 have higher HC emissions than neat diesel, biodiesel, and other methyl ester-ethanol mixture. The increase in HC emissions along with premixed ratio increment because a reduction in the temperature cylinder. The cooling effect of ethanol can lead to flame quenching which is the most significant reason for higher HC emissions. However, B70+E30 produced slightly higher HC emission at full load compared to other blends, because of the increased cooling of induced alcohol. The escape of premixed fuel through the crevice volume is the reason for increased hydrocarbon emissions at the low and part loads. The lower latent heat of ethanol reduces the combustion temperature and affects the burning of biodiesel by forming a quenching layer. At full load, the temperature inside the combustion chamber is necessary to burn all the fuels. This may be since some hydrocarbon components are composed on the surface of the carbon particles which contribute of the engine.
3.2.3 LOAD VS CARBON MONOXIDE

The conventional diesel fuel produces low amount of CO emission for biodiesel and its blends. There is a higher amount of CO emission in light load situation for B70+E30, conversely there promising of decreases in Co emission at maximum engine load at same blends. This is due to rich air fuel production in cylinder. The usage of ethanol blends along with this fuel may reduce the peak cycle temperature leads to decrease carbon dioxide conversion that may left of larger amount of carbon monoxide in the exhaust at minimum load. This inference that HCCI the conversion of Carbon dioxide from carbon monoxide depends upon the peak to peak cycle combustion temperature increase above leads to 1500 K for conversion.
Oxide of Nitrogen

Figure 9 indicates the variation of Nitrous Oxide emissions to various loads for neat diesel, biodiesel, and biodiesel-ethanol blends. The neat diesel has higher NOX than neat biodiesel and other blends. From the experimental observation that NOx is reduced when there is an increase in the amount of premixed ratio when related to neat diesel and biodiesel. The concept of increasing ethanol in the intake manifold has reduced temperature and this pre-cooling might have reduced the in-cylinder combustion temperature. The values of NOx at no load to full load for B70+E30% are lower than neat diesel and biodiesel. The cooling effect of higher ethanol content in the B70+E30 blend combined with a reduction in cetane number led to increase in ignition delay which caused the peak pressure to occur slightly after top dead center, this is the main reason for a decrease in NOx emission with the higher premixed ratio of ethanol. But, there is an increase in NOx emission at the full load. This is due to increased cylinder pressure which causes a high peak combustion temperature. The increase in cylinder temperature causes increased oxides of nitrogen emission.

![Figure 9 variation for oxides of nitrogen emissions with respect to load.](image)

Smoke Emission

Figure 10 shows the difference in smoke emissions of various loads for neat diesel, biodiesel and biodiesel-ethanol mixture. The neat biodiesel has more smoke than pure diesel and methyl ester ethanol mixture. A smoke emission for B70+E30 is reduced due to more amount of oxygen and a lower amount of carbon content in the mixture. The higher viscosity and density of methyl ester is the reason for increasing smoke in pure biodiesel. The indication of maximum energy release and peak pressure for the B70+E30 mixture enhanced the oxidation of carbonaceous soot produced during the mixing controlled combustion which led to lower smoke emission than others mixtures at higher engine power.
4. Conclusion

From the experimental study the following conclusions can be determined:

- The cylinder pressure was found to be higher in full load for B70+E30 (75.8 bar at 6°aTDC). The B100 has lower cylinder pressure (71.25 bar at 9°aTDC). Whereas in the case of part load, the cylinder pressure was found to be higher for neat diesel (64.95 bar at 5° aTDC) and lower for B70+E30 (61.46 bar at 8° aTDC).
- The brake thermal efficiency was found to be higher for neat diesel (33.75%) at full load. And lower brake thermal efficiency was found to be B90+E10 (21.95%) at full load. However, the B70+E30 (30.22%) blend produced brake thermal efficiency which is closer to neat biodiesel at full load.
- The specific energy consumption was found to be higher for B90+E10 (16513.75 kJ/hr) at full load. And lower specific fuel consumptions were found to be neat diesel (10893 kJ/hr) at full load. The B70+E30 blend (12028.22 kJ/hr) produced lower specific fuel consumption compared to other biodiesel-ethanol blends. This may be due to improvement in fuel vaporization characteristics leading to better combustion efficiency.
- The HC emissions were found to be higher for B70+E30 (158 PPM) at full load. And lower was to be found for neat biodiesel (52 PPM) at full load. The cooling effect of ethanol can lead to flame quenching which is the most significant reason for higher HC emissions.
- The CO emissions were found to be higher for neat diesel (0.156 %) at full load. And lower was found to be for B80+E20 (0.099 %) at full load. The main reason for higher CO emission is because of fuel-rich zones and insufficient oxygen availability inside the combustion chamber.
- The NOx emissions were found to be higher for neat diesel (1745 PPM) at full load. And lower emissions were found to be B70+E30 (1225 PPM) at full load. The cooling effect of higher ethanol content in the B70+E30 blend combined with a reduction in cetane number led to an increase in ignition delay and reduction in in-cylinder temperature.
The smoke emissions were found to be higher for neat diesel (90 %) at full load. And lower emissions was found to be B70+E30 (65 %) at full load. Due to maximum heat release and peak pressure for the B70+E30 blend enhanced the oxidation of carbonaceous soot produced during the mixing controlled combustion which led to lower smoke emission compared to other blends at higher engine load.

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