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Experimental Investigations on Manifold Injection of Diesel and Biodiesel in a HCCI Engine with Inlet Charge Heating

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ABSTRACT

In the present study, engine tests were conducted on a modified single cylinder four stroke compression ignition (CI) engine operated in homogeneous charge compression ignition (HCCI) mode with injection of diesel and biodiesels through intake manifold. The intake air temperature varied from 50 to 80 °C for diesel and 55 to 85 °C for biodiesel using air pre heater. The coolant temperature varied from 40 to 60 °C for both the diesel and biodiesel operation. For the comparison purpose, CI engine fuelled with diesel was operated at the injection timing of 23°BTDC and an injector pressure of 205 bar. It is seen that HCCI mode of engine operation with diesel and biodiesel resulted into 35-45% lower brake thermal efficiency (BTE) with significant reduction in nitrogen oxide (NO_x) emission by 98% and smoke emissions by 65-75%. On the other hand, HCCI engine operation with diesel and biodiesels showed increased hydrocarbon (HC) emissions by 20-25 times and carbon monoxide (CO) emissions by 30-40%. However, peak pressure (PP) and heat release rate (HRR) decreased by 20-25% when compared to CI mode of engine operation.

Keywords: homogeneous charge compression ignition (HCCI), air preheater, manifold injection, emissions

INTRODUCTION

CI engines are highly efficient in terms of fuel economy but emit more NO_x and smoke emissions in the exhaust. Biodiesels can be used in CI engine as they have similar properties as that of diesel. However, biodiesels demands higher injector opening pressure (IOP) as they have higher viscosity compared to diesel. Use of biodiesel in CI engine addresses the scarcity of fuel but to meet the emission norms led by the authorities, it is necessary to use new concept which meets the legal limits of authority.

HCCI concept is the one to address both of them simultaneously. HCCI concept of combustion started during the late 1970 and this relatively newer concept employment in CI engine in commercial applications is due to its potential to achieve better BTE and lower emissions. In this concept, a homogeneous air-fuel mixture auto-ignites at number of places by compression alone when it reaches the chemical activation energy (Pucher et al., 1996; Furutani et al., 1993).

HCCI engine operation with diesel fuel at different compression ratio (CR) was performed and it was observed that with an inlet air temperature of 90°C, diesel fuel requires about 11:1 compression ratio to ignite it at top dead centre (TDC) (Christensen et al., 1999). HCCI operation has a potential for resulting very low NO_x and particulate matter (PM) emissions with high HC emission, non-optimal combustion phasing, poor BTE of 28% lower than the conventional CI mode (Yao et al., 2009). A gasoline type injector was employed to inject diesel fuel at 50 bar into the intake manifold and found very early combustion phasing at higher CR, sever knocking at higher loads and higher HC, poorer BTE (Suzuki et al., 1998). A study on the diesel HCCI engine using fuel vaporizer technology revealed that NO_x and PM emissions were found lower, while HC and CO emissions found higher with lower indicated thermal efficiency and specific fuel consumption (SFC) (Agarwal et al., 2013). HCCI engine operation in a diesel engine with a fuel injection timing (IT) of 120°CA before top dead centre (BTDC) using a multi hole injector of small diameter was achieved with high levels of HC about 8000 ppm and very early combustion phasing (Nakagome et al., 1997).

To avoid wall wetting (Iwabuchi et al., 1999), developed impinging sprays and found that spray angle of 80° was the most suitable with ITs of 40 to 60° BTDC and revealed that very high smoke opacity and high HC emissions and SFC.

Utilization of the heat energy contained in exhaust gases to vaporize the fuel and adjustment in valve overlap to get high internal EGR techniques were used and this work reported that high negative valve overlap (NVO) increased the combustion stability, low NO_x and smoke at low loads and NO_x increased at high outputs due to uncontrolled knocking combustion (Shi et al., 2005). Use of early in cylinder injection in HCCI engine operation revealed that engine was able to operate at lower brake mean effective pressure (BMEP) (2.1 bar to 4.3 bar) with low NO_x emission and poor BTE but with high HC, CO and smoke emissions (Nathan et al., 2010).

The combined manifold and in cylinder injected HCCI operation was employed, where a homogeneous mixture was obtained by injecting maximum quantity of fuel into the intake manifold. The mixture was ignited by a small quantity of fuel injected directly into the cylinder and reported that both NO_x and smoke emissions were better than ordinary diesel engine (Suzuki et al., 1997a; Suzuki et al., 1997b; Suzuki et al., 1998; Odaka et al., 1999). Employment of external mixture preparation device in HCCI engine operation at CR of 18 resulted in lower levels of NO_x and smoke emissions with good BTE (Midlam-Mohler, 2004). Expansion of HCCI engine operating range using charge stratification revealed that the combustion becomes more stable but heat release rate (HRR) decreases and main HRR advances with large stratified charge (Berntsson and Denbratt, 2007). Supplying gasoline through intake manifold to control combustion and diesel fuel direct injection into the cylinder to initiate combustion showed that engine operating load can be increased to 12 bar indicated mean effective pressure(IMEP) with lower NO_x and smoke emissions (Inagaki and Fuyuto, 2006). The characteristics and importance of heat transfer phenomena in combustion chamber walls were reported. Minimum correction coefficient for heat transfer correlations was determined theoretically under the same operating conditions. The study suggest that the proposed model could predict results better than Eichelberg's heat transfer coefficient equation and Burnt's specific heat ratio equation at all ignition timings with all fuels selected (Gürbüz, 2016). The utilization of after treatment devices plays very important role in decreasing emissions. But these can increase and decrease the CO_2 emissions depending on the type of fuel and engine loads. The maximum fuel consumptions was more for biodiesel as compared to diesel. BDF100 was most sustainable at any engine load, with/without after treatment options (Caliskan and Mori, 2017). The effect of cetane number (CN) and ignition delay (ID) on the energy and exergy efficiencies of IC engine was studied. It is reported that a lower CN, a longer ID and a higher level of premixed combustion could increase the exergetic efficiency of a diesel engine (Tat, 2011). The specific fuel consumptions (SFC) decrease for all test fuels with increase in engine load. It was revealed that NO emissions increased with lower CO as amount of biodiesel in the test fuels (Uyumaz et al., 2014). Combustion duration (CD) and the cyclic variation in hydrogen run SI engine could be reduced with optimum swirling flow. The combustion stability in hydrogen run SI engine is mainly dependent on cyclic variations in the flame initiation period and the cyclic variations in this period can be reduced with controlled swirling flow (Gurbuz et al., 2014). Ethanol has higher latent het of evaporation and octane number besides its higher flammability temperature. These characters have better influence on the engine performance and reduce exhaust emissions. Engine torque increase with lower SFC was observed with the ignition timings (Calam et al., 2013).

From the detailed literature review undertaken it was observed that scanty work on low temperature combustion studies using renewable fuels like biodiesel has been reported. Hence the objective of the present study is to evaluate performance of HCCI engine fueled with Ceiba Pentandra Oil Methyl Ester (CPOME) using different coolant and intake charge temperatures.

Therefore, an attempt has been made to evaluate the performance of HCCI engine with different operating strategies.

Figure 1. Separation of Biodiesel and glycerin

Table 1. Properties of Diesel, and CPOME

Sl. No.	Properties	Diesel	CPOME
1	Chemical Formula	C13H24	-
2	Density (kg/m^3)	840	884.4
3	Calorific value (kJ/kg)	43,000	39,790
4	Viscosity at 40°C (cSt)	2-5	4.3
5	Flash point (°C)	75	202.5
6	Cetane Number	45-55	42.4
7	Carbon Residue (%)	0.1	0.06
8	Cloud point	-2	3
9	Pour point	-5	5

PROPERTIES OF FUELS USED

Preparation of Ceiba Pentandra Oil Methyl Ester (CPOME)

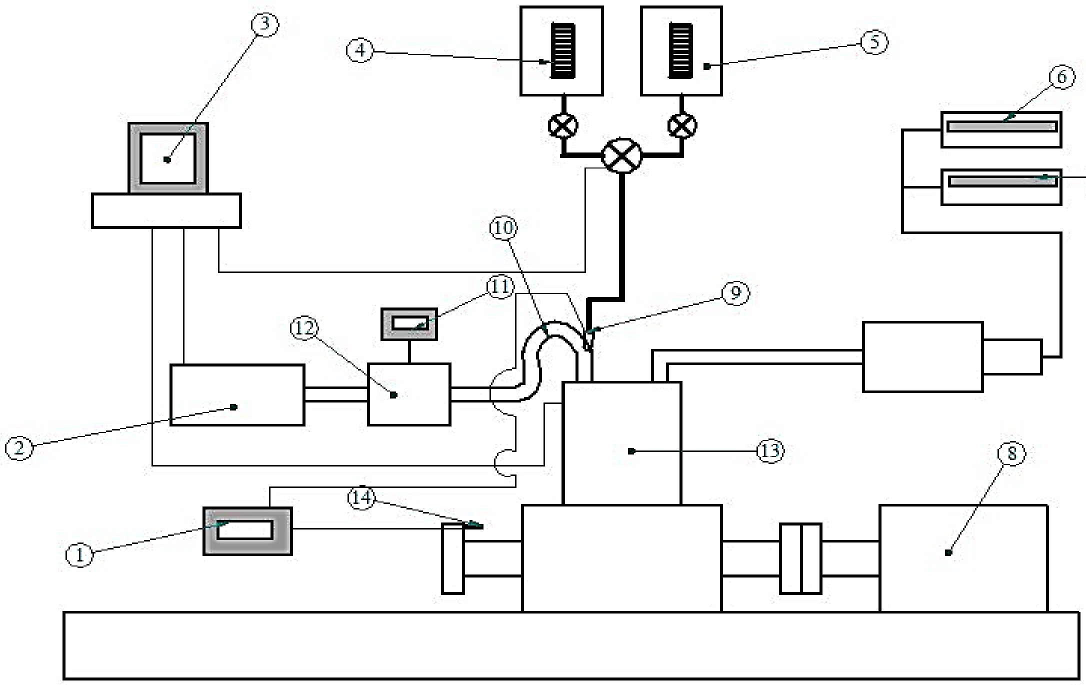
Three stages are involved in transesterification of Ceiba Pentandra Oil (CPO). The acid transesterification and alkali transesterification stages are required for production of CPOME. Following steps explain an optimized method of production of CPOME.

First Stage: In the first stage 2000 ml of raw CPO is heated up to 105°C in a round bottom flask in order to remove the moisture content. Then the oil sample is titrated against NaOH solutions to calculate Free Fatty Acid (FFA) and it was about 8 %.

Second Stage: In the second stage 1000 ml of moisture-free CPO is heated to 60°C and then 150 ml of methanol & 1.5 ml of sulphuric acid are added to the oil. The obtained solution is stirred vigorously using magnetic stirrer at 450 rpm for approximately one hour. Then the solution so obtained is then allowed to settle for about eight hours and the top layer which contains acid is removed. Residue oil is tested for FFA and it was about 5.6%. Since the value was higher than required, the above procedure was repeated until the FFA value was 2.8%.

Third Stage: In the third stage the residue oil is heated to a temperature of 60 °C. 100 ml of methanol and 7.6 grams of NaOH are added to the residue oil. The resulting mixture is maintained at the same temperature and stirred at 450 rpm. After 90 minutes, the mixture is transferred to separating funnel for settling under gravity as shown in **Figure 1**. After eight hours, heavier part glycerin which settled at lower level is separated out to obtain methyl ester of CPO which is further washed with water for 3 or 4 times with a solution containing 10ml of acetic acid and hot water to remove moisture and other sediments to obtain clean CPOME. The biodiesel yield per 1000 ml oil was found to be 890 ml or 89%.

The properties of CPOME was determined in the fuel testing laboratory and summarized in **Table 1**.



01. Injector ECU
02. Air box
03. PC interfaced to engine
04. Diesel tank
05. Biodiesel tank
06. Exhaust gas analyzer
07. Smoke meter
08. Eddy current dynamometer
09. Fuel injector
10. Intake manifold
11. Temperature controlling unit
12. Electric air heater
13. CI engine
14. Crank angle sensor

Figure 2. Experimental set up with Manifold injection facility

EXPERIMENTAL SET UP AND METHODOLOGY ADOPTED

Manifold Injection of Diesel and Biodiesel

The conventional diesel engine was suitably modified to operate in HCCI mode. Air-preheater was used to supply hot air and manifold fuel injection facility developed in house was used to inject diesel and biodiesels with a pintle nozzle at an opening pressure of 100 bar for diesel and 130 bar for the biodiesel respectively at IT of 40°ATDC that is during the suction stroke. A separate inline injection facility that operates with the help of cam shaft was developed in house. This facility provides an opportunity to set IT at any desired value with external mechanical arrangement developed. Actual start of fuel IT was obtained with the help of a needle lift sensor. The engine was always operated at BMEP of 2.5 bar (50% load). The coolant temperature was controlled by varying the water flow rates with the help of a flow control valve. The temperature of cooling water was measured with the help of thermocouples. The experimental set up for HCCI engine operation fuelled with diesel and biodiesels is shown in Figure 2. Table 2 shows the specification of the engine used. Specification of smoke meter is given in Table 3 and the specification of exhaust gas analyzer is given in Table 4. Exhaust gas analyzer and Hartridge smoke meter were used in order to measure HC, NO_x, CO and smoke emissions respectively. The effect of charge temperature on emissions and performance characteristics of HCCI engine was studied.

Table 2. Engine specifications

SI. No	Parameter	Specifications
1	Type	TV1 (Kirlosker make)
2	Software used	Engine soft
3	Nozzle opening pressure	200-225 bar
4	Governor type	Mechanical centrifugal type
5	No. of cylinders	Single cylinder
6	No. of strokes	Four stroke
7	Fuel	H. S. Diesel
8	Rated power	5.2 kW (7 HP at 1500 RPM)
9	Cylinder diameter (Bore)	0.0875 m
10	Stroke length	0.11 m
11	Compression ratio	17:5:1
Air measurement manometer		
12	Made	MX 201
13	Type	U-Type
14	Range	100-0-100 m
Eddy current dynamometer		
15	Model	AG-100
16	Type	Eddy current
17	Maximum	7.5 (kW at 1500 -3000 RPM)
18	Flow	Water must flow through Dynamometer during the use
19	Dynamometer arm length	0.180 m
20	Fuel measuring unit – Range	0-50 ml

Table 3. Specifications of smoke meter

Type	HARTRIDGE SMOKEMETER-4
Object of measurement	Smoke
Measuring range opacity	0-100 %
Accuracy	+/-2 % relative
Resolution	0.1 %
Smoke length	0.43 m
Ambient Temperature Range	-0 oC to +45 oC
Warm up time	10 min. (self controlled) at 20 oC
Speed of response time	Within 15 sec. for 90% response
Sampling	Directly sampled from tail pipe
Power supply	100 to 240 V AC / 50HZ 10-16 V DC @ 15 amps
Size	100 mm x 210 mm x 50 mm

Table 4. The accuracies of the measurements and the uncertainties in the calculated parameter

Type	DELTA 1600S
Object of measurement	Carbon monoxide (CO) and Hydrocarbons (HC)
Range of measurement	HC = 0 to 20.000 ppm as C3H8 (Propane) CO = 0 to 10% NO_x = 0 to 5000 ppm (as Nitric Oxide)
Accuracy	HC = +/- 30 ppm HC CO = +/- 0.2% CO NO_x = +/- 10 ppm NO
Resolution	HC = 1 ppm CO = 0.01% Vol. NO_x = 1 ppm
Warm up time	10 min. (self-controlled) at 20 oC
Speed of response time	Within 15 sec. for 90% response
Sampling	Directly sampled from tail pipe
Power source	100 to 240 V AC / 50Hz
Weight	800 gm
Size	100 mm x 210 mm x 50 mm

Table 5. The accuracies of the measurements and the uncertainties in the calculated parameter

Measured variable	**Accuracy**
Load, N	0.1
Engine speed, rpm	1
Temperature, °C	1
Fuel consumption, g	0.1
Measured variable	**Uncertainty (%)**
HC	±1.5
CO	±2.1
NO_x	±2.4
Calculated parameters	**Uncertainty (%)**
BTE (%)	±0.9

Experimental Uncertainty Analysis

The accuracies of the measurements and the uncertainties in the calculated values of each parameters of the current investigation are shown in the Table 5. In order to minimize the errors of measurements five readings were recorded and averaged out results are only presented for the analysis. All the measurements of physical quantities are subject to uncertainties and were estimated as below:

Uncertainty in the of Mass Flow Rate of Fuel (m_f)

Mass flow rate of fuel, m_f

$$m_f = \frac{\text{Volume flow rate of fuel in cc} \times 10^{-6} \times \text{Density of fuel}}{\text{Time taken in seconds}}$$

i.e,

$$m_f = \frac{V_f \times 10^{-5} \times \rho_f}{t}$$

Volume flow rate of rate of fuel is measured using burettes, so uncertainty in burette reading, $\omega_{v_{ff}} = \pm 1 \text{ cc} = \pm 1 \times 10^{-3}\, m^3$

$$\text{Uncertainty in } m_f = \pm\sqrt{\left(\frac{\partial m_f}{\partial V_f} \times \omega_{V_f}\right)^2 + \left(\frac{\partial m_f}{\partial t} \times \omega_t\right)^2} = \text{in } kg/s$$

Uncertainty in brake power (BP)

Brake power of engine,

$$BP = \frac{2\pi NT}{60 \times 1000} = \frac{2\pi N \times W \times R}{60 \times 1000} = kW$$

where, W is Load applied in N;
Uncertainty in BP, ω_{BP},

$$\omega_{BP} = \pm\sqrt{\left(\frac{\partial BP}{\partial W} \times \omega_W\right)^2 + \left(\frac{\partial BP}{\partial N} \times \omega_N\right)^2} = \pm \text{ in } kW$$

Uncertainty in the brake thermal efficiency (η_{th})

Brake thermal efficiency is given by

$$\eta_{th} = \frac{\text{Brake power in } kW}{m_f \times CV_1} \times 100 = \%$$

Uncertainty in Brake thermal efficiency,

$$\omega_{\eta_{th}} = \pm\sqrt{\left[\frac{\partial \eta_{th}}{\partial BP} \times \omega_{BP}\right]^2 + \left[\frac{\partial \eta_{th}}{\partial m_f} \times \omega_{m_f}\right]^2} = \pm \%$$

Table 4 provides the uncertainties calculated for the parameters on performance, emission and combustion. Uncertainty values for all the parameters are within 3% and are acceptable for the analysis presented.

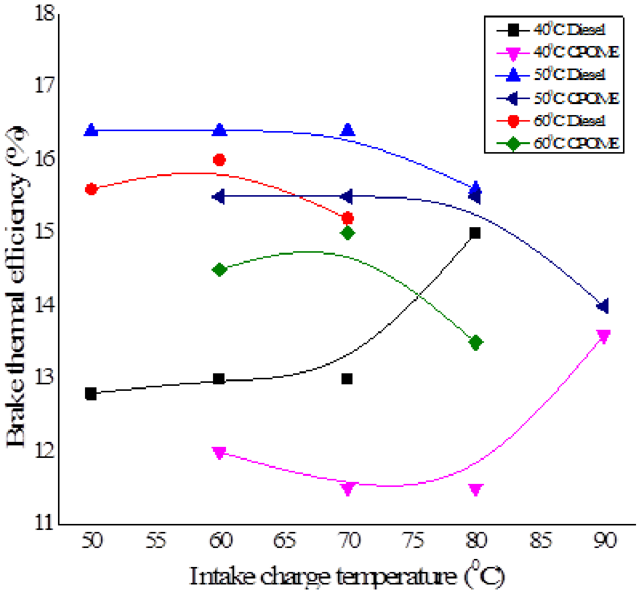


Figure 3. Variation of brake thermal efficiency with intake charge temperature

RESULTS AND DISCUSSIONS

This section discusses the results of the experiments carried out using diesel and biodiesel in manifold assisted HCCI mode of engine operation.

Brake Thermal Efficiency

Figure 3 shows the effect of charge and coolant temperatures on BTE in case of manifold injection of diesel, and CPOME. The BTE is found to be lowest with all the injected fuels for coolant temperature of 40°C and this is observed up to the charge temperature of 70°C. However for coolant temperature of 50 and 60°C, the BTE is much higher compared to 40°C but drop in BTE beyond 70°C is observed. Lower HRR available at lower coolant/charge temperatures could be responsible for these trends obtained. Lower coolant temperatures demands higher charge temperatures to achieve better BTE. However the injected biodiesel have shown poorer performance than the diesel and the reasons could be attributed to their higher viscosity (nearly twice), lower volatility and calorific value and also they require 5 to 10°C higher intake charge temperature compared to its counterpart diesel. When the coolant temperature increased is from 40°C to 60°C, advancement in the main heat release for diesel and biodiesels was observed with increase in the fuel injection rate. For 40°C coolant temperature, when the charge temperature was increased to a value higher than 80°C the HRR goes up that resulted in increased BTE. The highest BTE was obtained at a coolant temperature of 50°C which as seen from Figure 3 and after this coolant temperature, the BTE dropped due to very early heat release. The engine became erratic due to knock at high charge temperatures of about 80°C for diesel and 85°C for biodiesels and the high coolant temperature about 60°C though the HRR was high, but it occurs at too early crank angles.

Hydrocarbon (HC) Emissions

Figure 4 shows the trend of HC emissions with varied coolant and charge temperatures. The biodiesel showed higher HC emissions compared to diesel and the reasons could be their lower BTE obtained associated with incomplete combustion that prevails inside the combustion chamber due to improper fuel-air mixture and subsequent wall wetting problem observed with these biodiesels. The trend of HC emissions was not uniform and was different for different coolant temperatures. As the coolant temperature increased the cylinder wall temperature goes up increasing the vaporization of the fuel from the wall surface. This could be a reason for increased HC emissions with increase in coolant temperature. Similar trends were reported in the literature. With a coolant temperature of 40°C, HC emission decreased when the charge temperature was increased to 70°C then it increased. Decreasing trend of HC emission for the injected liquid fuels was observed at a coolant temperature of 50°Cwith increase in intake charge temperature whereas at a coolant temperature of 60°C increasing trend was observed with increase in intake charge temperature.

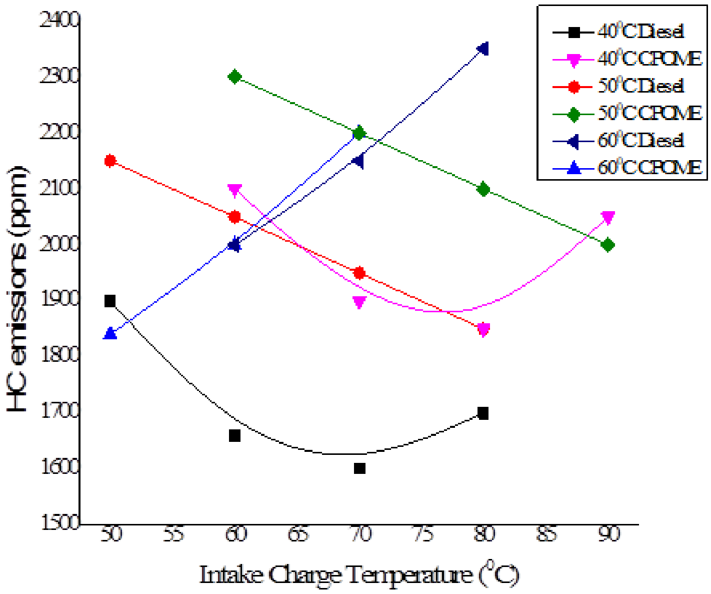


Figure 4. Variation of HC emissions with intake charge temperature

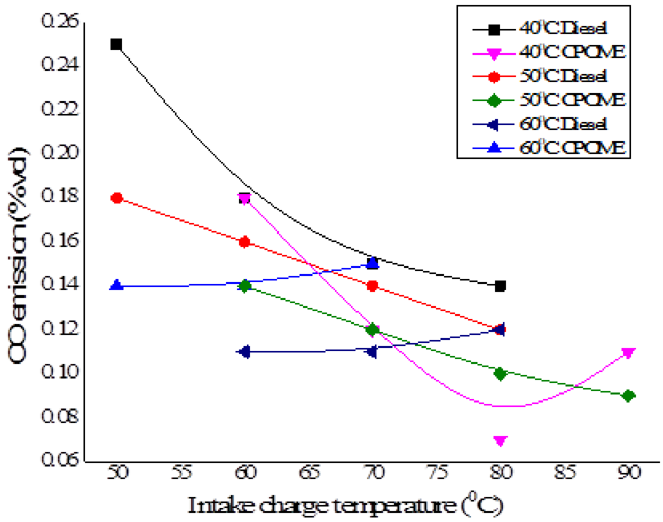


Figure 5. Variation of CO emissions with intake charge temperature

Carbon Monoxide (CO) Emissions

Figure 5 shows the effect of intake charge temperature and coolant temperature on CO emissions. CO emission showed similar trends as compared to HC emissions. The biodiesel showed higher CO emissions compared to diesel and the reasons could be the incomplete combustion that occurs due to improper fuel and air mixtures. With a coolant temperature of 40°C, CO emission decreased when the charge temperature was increased up to 70°C than it increased. Decreasing trend of CO emission was observed at a coolant temperature of 50°C with increase in intake charge temperature whereas at a coolant temperature of 60°C increasing trend was observed with increase in intake charge temperature. Similar results were reported in the literature.

Exhaust Gas Temperature

The variation in exhaust gas temperature with coolant and intake charge temperature is shown in Figure 6. It is clearly observed from the Figure 6 that as the intake charge temperature increases the exhaust gas temperature (EGT) increased. Also with the increase in coolant temperature there was increase in EGT. EGT refers to the excess heat supplied to the engine cylinder and in case of biodiesel; the lower BTE has resulted into higher EGT.

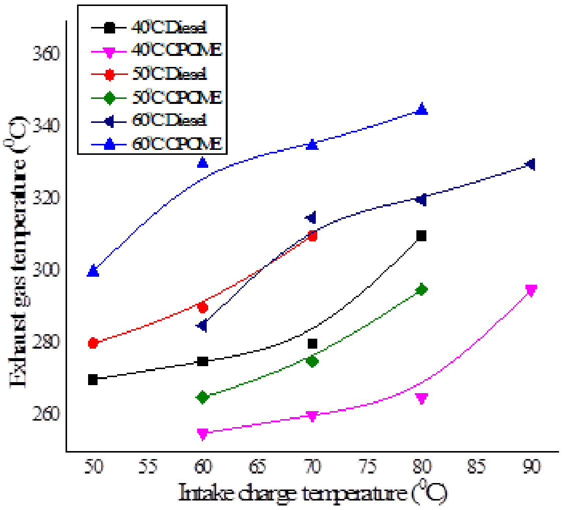


Figure 6. Variation of exhaust gas temperature with intake charge temperature

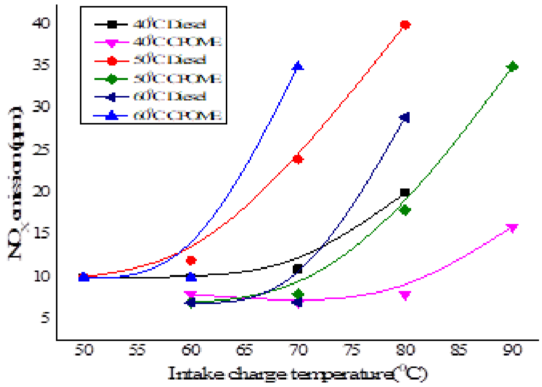


Figure 7. Variation of NO_x emissions with intake charge temperature

Nitrogen Oxides (NO_x) Emissions

Figure 7 shows the variation of NO_x emission with intake charge temperature at different coolant temperatures. Higher combustion temperature inside the cylinder and nitrogen present in atmospheric air are the causes for NO_x formations. The biodiesel showed lower NO_x emissions compared to diesel. The reasons for this could be attributed to lower premixed combustion, lower peak pressure and combustion temperature occurring in the CC. The NO_x emission were extremely lower compared to conventional diesel engine mode and this is mainly due to homogeneous mixture of fuel and air before combustion starts and lower in-cylinder temperature achieved. In the HCCI mode of engine operation the intake charge temperature has a significant influence on NO_x emissions level. The HCCI operational ways resulted in lesser than 30 ppm of NO_x emissions but the NO_x level was about 300 ppm with the base diesel engine.

Smoke Emissions

The variation of smoke emissions with intake charge temperature at different coolant temperature is shown in Figure 8. It can be observed from figure that the smoke level depends on both the charge and coolant temperatures. Smoke level was highest at the coolant temperature of 50°C, where the BTE was reported to be highest for both diesel and biodiesels. However CPOME showed comparatively higher smoke due to their heavier molecular structure that resulted in improper fuel-air mixture compared to diesel. At the lowest coolant temperature used smoke observed was lower, probably because the temperatures were very low and in HCCI operation homogeneous charge formed led to complete combustion. At the intermediate temperature, where the BTE was highest, the smoke level was also highest. An increase in charge temperature increased the local temperature that increased the smoke level. The smoke level was quite high with the HCCI mode of operation.

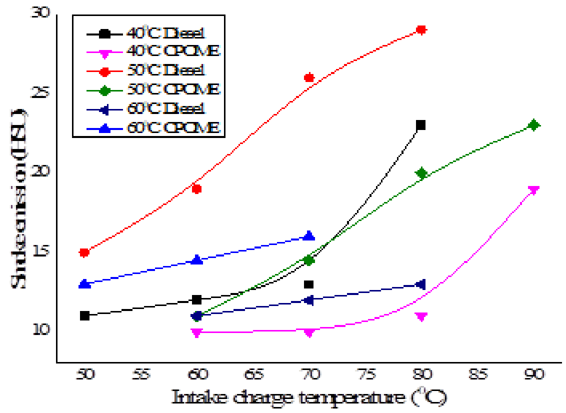


Figure 8. Variation of Smoke emissions with intake charge temperature

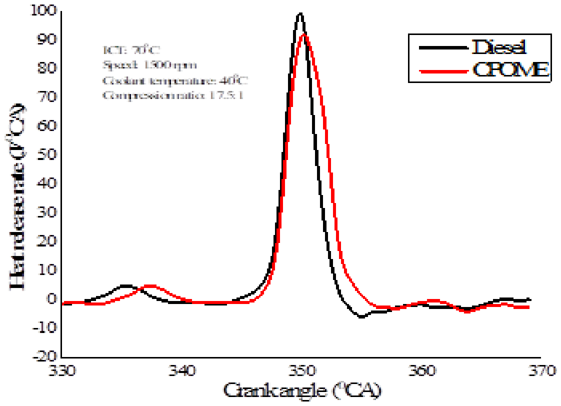


Figure 9. Variation of HRR with Crank angle

Heat Release Rate (HRR)

The account of energy supplied to charge heating is not considered in the calculations of the heat release. **Figure 9** shows the variation of heat release rate with crank angle for diesel, CPOME fuelled HCCI engine for optimized conditions of intake charge temperature of 70°C and a coolant temperature of 50°C. From **Figure 9** it follows that the properties of the injected fuels resulted in two HRRs that is cool flame HRR and main combustion. The cool flame HRR occurs between at 332-340°CA and a small amount of heat energy released was found for a crank angle about 10°. Main combustion occurred between 348-355°CA. Diesel fuel has higher calorific value compared to its counterpart to CPOME and hence higher premixed combustions were obtained for these fuels. The biodiesel resulted in increased diffusion combustion phase as the BTE associated with these fuel combinations were lower.

Peak Pressure

Figure 10 shows the variation of pressure with crank angle for diesel, CPOME operated HCCI engine for optimized conditions of intake charge temperature of 70°C and a coolant temperature of 50°C. Diesel has higher calorific value compared to its counterpart biodiesel and hence higher peak pressure is obtained. The lower calorific value and higher viscosity of biodiesels resulted into lower peak pressures.

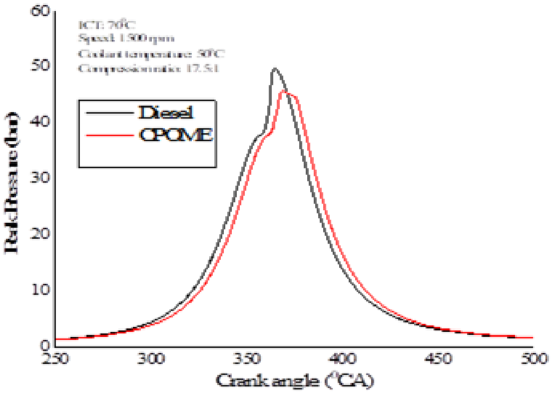


Figure 10. Variation of Peak pressure with Crank angle

CONCLUSIONS

The experiments were performed on a modified single cylinder four stroke CI engine operated in HCCI mode with injection of diesel and biodiesel through intake manifold at 50% load. The air preheater was employed to heat the intake air and the temperature varied from 50 to 80°C for diesel and 55 to 85°C for biodiesel. The coolant temperature was varied from 40 to 60°C for all the fuels. For the comparison purpose, CI engine fuelled with diesel was operated at the IT of 23 °BTDC, an injector opening pressure of 205 bar, compression ratio of 17.5 and engine speed of 1500 rpm. From the experimental studies on manifold injection assisted HCCI engine fuelled with diesel and CPOME, the following conclusions are drawn under the above operating conditions:

- HCCI engine operation with diesel/biodiesel showed 35-45% lower BTE compared to neat diesel CI engine operation.
- HCCI engine operation with diesel/biodiesel resulted in very low NO_x emission about 98% lower compared to neat diesel CI engine operation.
- HCCI engine operation with diesel/biodiesel resulted in very low Smoke emission about 65-75% lower compared to neat diesel CI engine operation.
- HCCI engine operation with diesel/biodiesel resulted in lower EGT about 30-40% lower compared to neat diesel CI engine operation.
- HCCI engine operation with diesel/biodiesel resulted in very high HC emission about 20- 25 times higher compared to neat diesel CI engine operation.
- HCCI engine operation with diesel/biodiesel resulted in very high CO emission about 30-40% higher compared to neat diesel CI engine operation.
- HCCI engine operation with diesel/biodiesel resulted in lower HRR about 20-25% lower compared to neat diesel CI engine operation.
- HCCI engine operation with diesel/biodiesel resulted in lower PP about 20-25% lower compared to neat diesel CI engine operation.

On the whole HCCI engine operation with diesel and biodiesel injection into manifold was successful but the main problem faced was the entry of fuels into crank case through piston–cylinder clearance. In HCCI engine operation with biodiesel intake charge temperature required was 10°C higher than diesel. A coolant temperature of 50°C and Charge temperature of 70°C provided overall better results.

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