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Effect of Combustion Chamber Shapes & Injection Strategies on the Performance of Uppage Biodiesel Operated Diesel Engines

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Abstract
In compression ignition engines selection of appropriate combustion chamber shapes integrated with suitable injection strategies play a major role in the combustion processes and emission characteristics. Hence to optimize a combustion chamber shape with appropriate injection strategies, suitable design combinations are essential to meet both emission norms as well as acceptable diesel engine performance. In this context, experimental investigations were carried out on a single cylinder four stroke direct injection diesel engine operated on Uppage oil methyl ester (UOME). Different combustion chamber shapes were designed and fabricated keeping the same compression ratio in the existing diesel engine. Injectors with different number of holes as well as with varying orifice sizes were used to study their effect on the biodiesel fuelled engine. The existing engine was provided with hemispherical combustion chamber (HCC) shape. In order to study the effect of other combustion chamber shapes on the performance of diesel engine, Cylindrical (CCC), Trapezoidal (TrCC), and Toroidal combustion chamber (TCC) shapes were designed and developed. Injectors were used with number of holes varied from 3 to 6 and the size of nozzle orifice varied from 0.18 to 0.3 mm. Engine parameters such as power, torque, fuel consumption, and exhaust temperature, combustion parameters such as heat release rate, ignition delay, combustion duration, and exhaust emissions such as smoke opacity, hydrocarbon, CO, and NOx, were measured. Results obtained with TCC shape and increased number of injector holes (6) with reduced hole size (0.18mm) resulted in overall improved performance with reduced emission levels. Total hydrocarbon emission (THC) and carbon monoxide (CO) were also decreased significantly.

Key Word and Phrases
Biodiesel, Uppage Oil, Emissions, Combustion Chamber Shapes, Injector, Injection Timing, Injector Opening Pressure.

1. Introduction

Diesel engines are widely used for transport and power generation applications because of their high thermal efficiency, and their easy adoption for power generation applications as well. Increased impetus on improving diesel engine performance, with lower noise and vibration levels and lower emissions several techniques involving fuel/engine modifications are essential. Increased energy demand, diminishing fossil fuel reserves in the earth crust and harmful exhaust gases from engine tailpipe have focused major attention on the use of renewable and alternative fuels. To overcome and meet these requirements, use of renewable fuels such as biodiesels and bio-fuels for diesel engines has gained greater momentum. Hence implementing new methods that improve the efficiency of diesel engine for both transport and power generation applications are the need of hour. Renewable energy sources can supply sustainable energy for longer periods of time than their counterparts fossil fuels and have many advantages as well [12]. Liquid biodiesels in particular are more suitable for diesel engine applications as their properties are closer to diesel. A number of vegetable oils have been used for biodiesel production and their respective biodiesels are used as alternative fuels in diesel engines. Biodiesels derived from jatropha, honge (karanja), honne, palm, rubber seed, rape seed, mahua, and neem seed oils were used in diesel engine applications [2]-[5], [9], [19], [27], [29], [32]-[35], [37], [42], [43], [54], [50], Slightly reduced engine performance with increased emissions and poor combustion patterns were reported for biodiesels engine

operation by several researchers [2]-[3], [25], [26], [33], [38]-[39], [44], [47], [49]. Effect of various engine parameters such as compression ratio (CR), injection timing (IT), injection pressure and engine loading on the performance and exhaust emissions of a single cylinder diesel engine operated on biodiesel and their blends with diesel were reported in the literature [8], [10]-[11], [24], [36]. Varying injection timings affect the position of the piston and thereby cylinder pressure and temperature at the injection provided. Retarded injection timings showed significant reduction in diesel NOx and biodiesel NOx [14]. Cylinder pressures and temperatures gradually decreased when injection timings were retarded [41]. Experiments on CI engine using different vegetable oils and their esters at different injection pressures have been reported. Better performance, higher peak cylinder pressure and temperature were reported at increased injection pressures [5], [30], [40], [41]. Kruczynski et al. [18] obtained results of engine tests using camelina sativa oil and reported relatively good engine performance and stressed the need to change the calibration parameters of the engine fuel system that cater to the use of the reported fuel. The high content of Linolenic acid of the oil results in combustion process different to that of diesel. Tompkins et al. [51] highlighted the parameters influencing the gross indicated fuel conversion efficiency of biodiesel derived from palmolein and its B20 blend when compared with diesel oil. Biodiesels inherently shorter combustion durations and inherently lower air-fuel ratios, resulted into lower brake thermal efficiency and this could be linked to its bound oxygen component. Biodiesel with lower heating value requires a longer injector opening to deliver roughly the same amount of energy to produce same torque as obtained with diesel. Varuvel et al. [52] studied feasibility of biodiesel derived from waste fish fat and reported higher NOx emissions for biodiesel. NOx could be reduced by blending biodiesel with diesel and reported lowered brake thermal efficiency and increased particulate matter for the blends studied. Vedaraman et al. [53] used methyl ester of sal oil (SOME) in diesel engine and reported reduced CO, HC and NOx emissions with comparable brake thermal efficiency. They concluded that SOME can be a potential substitute for diesel fuel.

Combustion Chamber

The combustion chamber of an engine plays a major role during the combustion of wide variety of fuels used. In this context, many researchers performed both experimental and simulation studies on the use of various combustion chambers [1], [21]. In re-entrant combustion chamber intensification of swirl and turbulence were reported to be higher when compared to cylindrical chambers which lead to more efficient combustion causing higher NOx emissions and lesser soot and HC emissions [45]. Montajir et al. [20] studied the effect of combustion chamber geometry on fuel spray behavior and found that a re-entrant type combustion chamber with round lip and round bottom corners provides better air and fuel distribution than a simple cylindrical combustion chamber. Experimental study to optimize the combination of injection timing and combustion chamber geometry to achieve higher performance and lower emissions from biodiesel fueled diesel engine has been reported. Toroidal re-entrant combustion chamber and retarded injection timing has been found to improve brake thermal efficiency and reduced brake specific fuel consumption [16]-[17]. Improvement in air entrainment with increased swirl and injection pressure were reported [7], [22]. Prasad et al. [28] studied in-cylinder air motion in a number of combustion chamber geometries and identified a geometry which produced the highest in-cylinder swirl and turbulence kinetic energy around the compression top dead centre (TDC). Three dimensional CFD simulations involving flow and combustion chemistry were used to study effect of swirl induced by re-entrant piston bowl geometries on pollutant emissions of a single cylinder diesel engine fitted with a hemispherical piston bowl and an injector with finite sac volume. The optimal geometry of the re-entrant piston bowl geometry was confirmed by the detailed combustion simulations and emission predictions used. Optimum combustion chamber geometry of the engine showed better performance and emission levels. Suitable combustion geometry of bowl shape helps to increase squish area and proper mixing of gaseous fuel with air [1], [46]. Designing the combustion chamber with narrow and deep and with a shallow reentrance had a low protuberance on the cylinder axis and the spray oriented towards the bowl entrance reduced the NOx emission levels to the maximum extent [15], [21]. Influence of combustion chamber geometry on pongamia oil methyl ester and its blend (B20) fuelled diesel engine were investigated [15] in which toroidal re-

entrant and shallow depth re-entrant combustion chambers were used. Toroidal reentrant combustion chamber resulted into higher brake thermal efficiency, higher NOx and reduced emissions of particulates, CO, UBHC. Lower ignition delay, higher peak pressure with B20 were also obtained when compared to baseline hemispherical and shallow depth reentrant combustion chambers [15].

Injection Strategies

The behavior of fuel once it is injected in the combustion chamber and its interaction with air is important. It is well known that nozzle geometry and cavitations strongly affect evaporation and atomization processes of fuel. Suitable changes in the in-cylinder flow field resulted in differing combustion. The performance and emission characteristics of compression ignition engines are largely governed by fuel atomization and spray processes which in turn are strongly influenced by the flow dynamics inside injector nozzle. Modern diesel engines use micro-orifices having various orifice designs and affect engine performance to a great extent. Effects of dynamic factors on injector flow, spray combustion and emissions have been investigated by various researchers [23], [48]. Experimental studies involving the effects of nozzle orifice geometry on global injection and spray behavior has been reported [31], [6], [13].

From the literature survey it follows that very limited work has been done to investigate the effect of combustion chamber shapes and injector nozzle geometry on the performance, combustion and emission characteristics of diesel engine fuelled with biodiesels. In this context, experimental investigations were carried out on a single cylinder four stroke direct injection diesel engine operated on UOME with different combustion chamber shapes and injectors adopted for this work.

2. Characterization of Uppage oil: Garcinia Cambogia (Uppage) Oil as Biodiesel:

Amongst the many species, which can yield oil as a source of energy in the form of bio-fuel, "Garcinia cambogia" (Uppagi) has been found to be one of the most suitable species in India being grown; it is N_2-fixing trace. It is tolerant to water logging, saline and alkaline soils, and is grown in high rainfall region. Garcinia seeds contain 30 to 40% oil. Garcinia cambogia belongs to the family species. The tree grows in forest and is a preferred species for controlling soil erosion and binding soil to roots because of its dense network of lateral roots. The seeds are largely exploited for oil extraction which is well known for its medicinal properties. So far there is no systematic organized collection of seeds. Mixture seeds consist of 95% kernel and are reported to contain about 27.0 to 40% oil. The yield of oil is reported to be about 35 to 40% if mechanical expellers are used for the recovery of oil from the kernels. The crude oil is brown to creamy in colour, which deepens on standing. It has a bitter taste and disagreeable odour. Fig. 1 shows the Uppage biomass and Fig. 2 shows the biodiesel preparation from Uppage oil.

(a) Uppage Tree (b) Uppage Fruits (c) Uppage Seeds

Fig. 1 Uppage Biomass

(a) 3-Neck conical glass bottle for transesterification
(b) Separation of Glycerin
(c) Washing with hot water
Fig. 2 Biodiesel Preparation

By the present study, Diesel, and Uppage oil methyl ester (UOME) were used as injected fuels. UOME was obtained by transesterification process, where the triglycerides of Uppage oil were transferred to their corresponding monoesters by the reaction of ethanol in the presence of sodium hydroxide catalyst. Table 1 shows the composition of Uppage oil, its fatty acids contribution, chemical formula, structure and their molecular weight with their chemical structure. The properties of UOME were determined experimentally and are summarized in Table 2.

Table 1 Fatty acid contribution of Uppage oil sample and its chemical structure

Sl. No.	Fatty acid	Fatty acid contribution
1	Palmitic	3.7-3.9
2	Stearic	2.4-8.9
3	Lignoceric	----
4	Oleic	44.5-71.5
5	Lignoleic	1.8-18.3
6	Arachidic	2.2-4.7
7	Behenic	----
8	Linolenic	----
9	Eruceic	----

Table 2 Properties of fuels tested

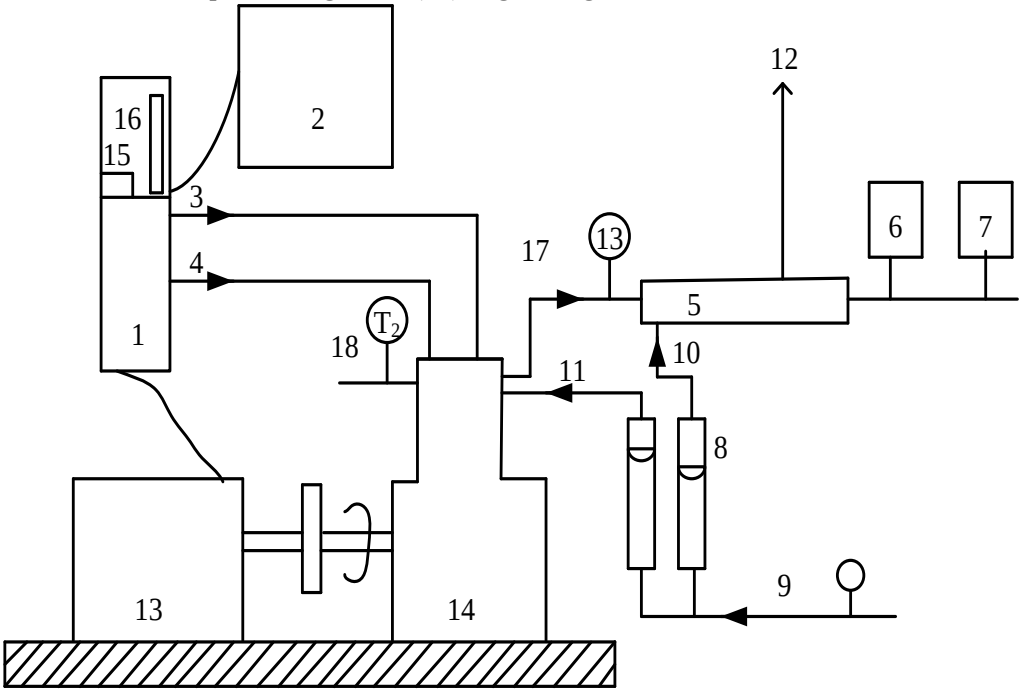
Sl. No.	Properties	Diesel	Uppage oil	UOME
1	Chemical Formula	$C_{13}H_{24}$	----	----
2	Density (kg/m^3)	840	915	860
3	Calorific value (kJ/kg)	43,000	38950	40727
4	Viscosity at 40^{o}C (cSt)	2-5	44.85	5.2
5	Flashpoint (^{o}C)	75	210	178
6	Cetane Number	45-55	40	45

7	Carbon Residue (%)	0.1	0.66	----
8	Cloud point	-2	----	18
9	Pour point	-5	----	21
10	Carbon residue	0.13	0.55	0.01
11	Molecular weight	181		227
12	Auto ignition temperature (^oC)	260		470
13	Ash content % by mass	0.57		0.01
14	Oxidation stability	High	Low	Low
15	Sulphur Content	High	No	No

3. Experimental Setup

Experiments were conducted on a Kirloskar TV1 type, four stroke, single cylinder, water-cooled diesel engine test rig fuelled with UOME. Figure 3 shows the line diagram of the test rig used. Eddy current dynamometer was used for loading the engine. The fuel flow rate was measured on the volumetric basis using a burette and stopwatch. The engine was operated at a rated constant speed of 1500 rev/min. The emission characteristics were measured by using HARTRIDGE smoke meter and five gas analyzer during the steady state operation. Different combustion chamber shapes of cylindrical (CCC), trapezoidal (TrCC), and toroidal (TCC) were used apart from the hemispherical shape provided in the engine. Figures 4 (a), (b), (c) ad (d) shows the different combustion chamber shapes used. To study the effect of number of holes, different injectors with 3, 4, 5 holes each of 0.3 mm and 6 holes of 0.18 mm were selected as shown in Figs. 5 (a), (b). Fig. 5(b) shows the special fixture prepared in-house for fixing the six hole injector. The position of the injector between the inlet and outlet port is also shown in Fig. 5(b).

Further to study the effect of orifice size a 4 hole injector with 0.2, 0.25 and 0.3 mm were also selected. Finally the results obtained with biodiesel operation were compared with diesel. The specification of the compression ignition (CI) engine is given in Table 3.



1- Control Panel, 2 - Computer system, 3 - Diesel flow line, 4 - Air flow line, 5 – Calorimeter, 6 - Exhaust gas analyzer, 7 - Smoke meter, 8 - Rota meter, 9, 11- Inlet water temperature, 10 - Calorimeter inlet water temperature,12 - Calorimeter outlet water temperature, 13 – Dynamometer, 14 - CI Engine, 15 - Speed measurement,16 - Burette for fuel measurement, 17 - Exhaust gas outlet, 18 - Outlet water temperature, T1- Inlet water temperature, T2 - Outlet water temperature, T3 - Exhaust gas temperature.

Fig. 3 Experimental Set up

(a) Hemispherical (b) Cylindrical

(c) Toroidal (d) Trapezoidal

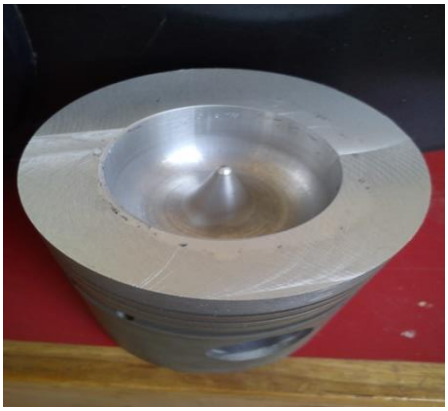
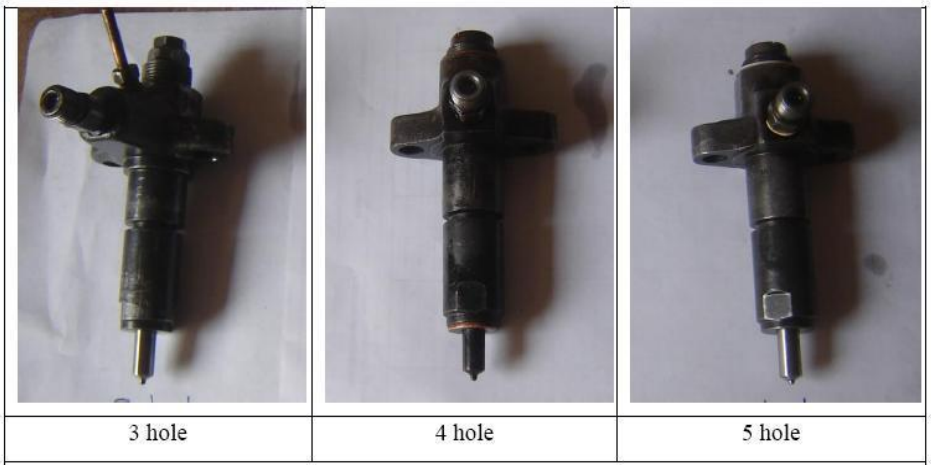
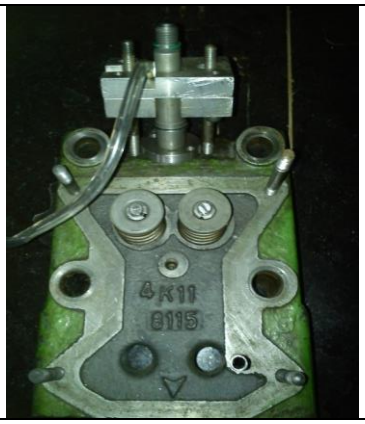


Fig. 4 Combustion chamber shapes



(a)

Side view of six hole injector with fixture



Top view of six hole injector with fixture

Injector position

(b) 6 hole injector fixed attachment

Figs. 5 (a), (b) Injectors with different number of nozzle holes

Table 3 Specifications of the engine

Sl No	Parameters	Specification
1	Type of engine	Kirloskar make Single cylinder four stroke direct injection diesel engine
2	Nozzle opening pressure	200 to 205 bar
3	Rated power	5.2 KW (7 HP) @1500 RPM
4	Cylinder diameter (Bore)	87.5 mm
5	Stroke length	110 mm
6	Compression ratio	17.5 : 1

4. Results and Discussions

In this section effects of injection timing, injection opening pressure (IOP) and nozzle geometry on the performance of diesel engine fuelled with UOME are presented in following sections.

3.1 Effect of Injection Timing

Studies on the biodiesel engine performance were conducted at three injection timings of 19^0, 23^0 and 27^0 bTDC. Compression ratio of 17.5, injector opening pressure of 205 bar (20.5 MPa) was maintained. An injector of three holes each having 0.3 mm diameter orifice was selected for the study.

3.1.1 Brake Thermal Efficiency (BTE)

The effect of injection timing on brake thermal efficiency of CI engine operation with UOME at three injection timings is shown in Fig. 6. For 80% load, highest brake thermal efficiency of 27.57% was obtained with diesel at a static injection timing of 23^0 bTDC. The optimum injection timing for standard diesel was found to be 23^0 bTDC, which also matches with the manufacturer's specification. However brake thermal efficiency with UOME operation at 23^0 BTDC is 21.90 %. Brake thermal efficiencies were lower for UOME as compared to diesel for all the three injection timings tested. The decrease in brake thermal efficiency for UOME might be attributed to lower energy content of the fuel and higher fuel consumption for the same power output. Also higher viscosity of UOME results into poor formation of the mixture and subsequent combustion were poorer than diesel. However by retarding the injection timing by 4 degree crank angle (CA) there was an improvement in brake thermal efficiency. The BTE is 23.27 % injection timing of 19^0 bTDC. Based on the values of brake thermal efficiency the optimum injection timings for diesel, and COME were 23^0 bTDC and 19^0bTDC respectively.

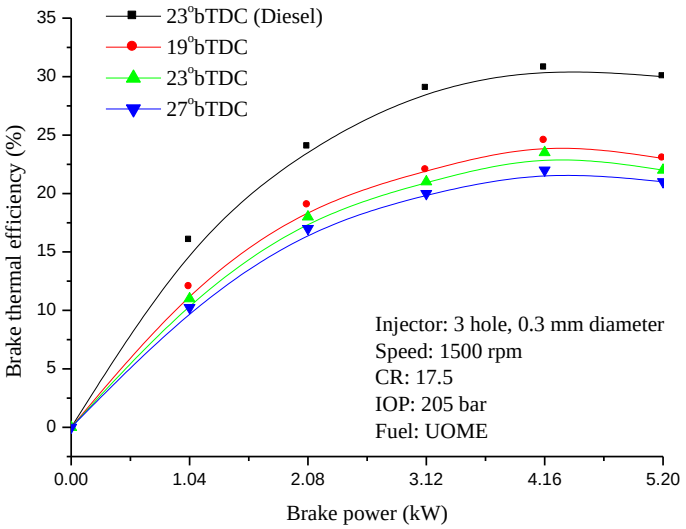


Fig. 6 Effect of injection timing on brake thermal efficiency for UOME

3.1.2 HC and CO Emissions

Fig. 7 and 8 show the effect of injection timing on HC and CO emissions for standard Diesel and UOME. Hydrocarbon emissions in diesel engines are caused due to lean mixture during delay period and under mixing of fuel leaving fuel injector nozzle at lower velocity. The general trend of increased HC and CO emissions for UOME is observed as compared to diesel for all three injection timings tested. This may be attributed to decreased combustion efficiency with UOME as shown in Fig. 6. The poor spray characteristics of UOME oil resulting in poor mixing, and consequently poor combustion may be responsible for this trend. The HC emission at 80% load was observed to be 64 ppm, 76 ppm, 84 ppm for 19^0, 23^0 and 27^0 bTDC injection timings respectively. Lowest HC levels were found at the optimum injection timing of 23^0 bTDC for standard diesel. However, for UOME biodiesel it was lowest at 19^obTDC.

Carbon monoxide is a toxic by-product and is a clear indication of incomplete combustion of the pre-mixed mixture. The amount of CO at 80% load was 0.21%, 0.31% and 0.42% for 19^0, 23^0 and 27^0 BTDC injection timings respectively. Lowest CO levels were found at the optimum injection timing of 23^0 BTDC for standard diesel. It may be concluded that improved combustion, increased cylinder pressure and temperature at 19^0 BTDC results in lower HC and CO emission compared to other injection timings for UOME.

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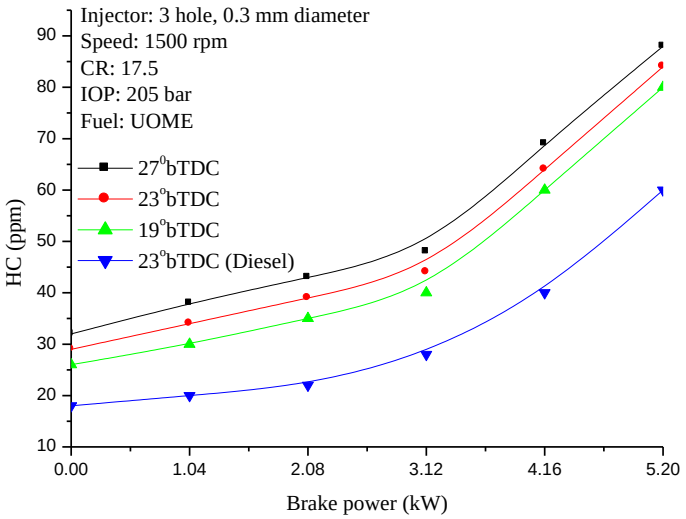

Fig. 7 Effect of injection timing on HC emissions

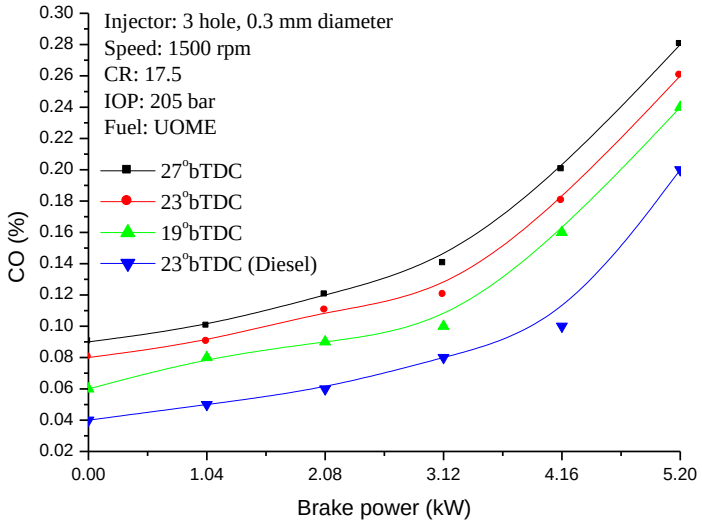

Fig. 8 Effect of injection timing on CO emissions

3.1.3 NO_x Emissions

The effect of injection timing on emission of NO_x with load for diesel, and UOME is shown in Fig. 9. In general retarded injection results in substantial reduction in NOx emissions. As the injection timing is retarded, the combustion process gets retarded. NOx concentration levels were lower as peak temperature is lower. NOx levels are higher at the injection timings of 23^0 and 27^0 BTDC as they lead to a sharp premixed heat release due to higher ignition delay. With UOME fuel NOx were slightly lower compared to diesel fuel. From these results the best injection timing was taken as 19^0 BTDC for UOME.

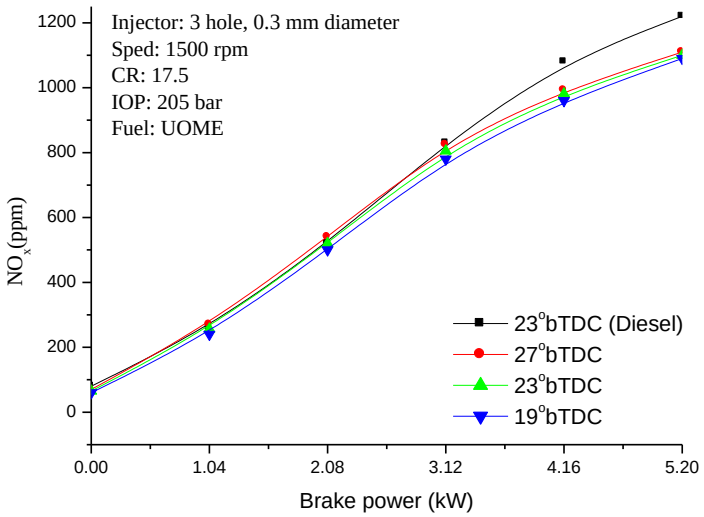

Fig. 9 Effect of injection timing on NO_x emissions

3.1.4 Combustion Analysis

Fig.10 and 11 show the effect of injection timing on the pressure variation and heat release rate at full load engine operation. The delay period increases with increase in injection advance angle as shown in Fig.10. This is because the pressures and temperatures are lower at the beginning of injection. As the injection advance angles are small, the delay period reduces and operation of the engine is smoother. Optimum angle of injection advance would cause peak pressure to occur 10°C to 15°C after top dead centre. As the injection advance increases the ignition delay period also increases and the heat release rate during premixed combustion phase decreases while minor effect on mixing controlled combustion phase is observed. From Fig.11, it is clear that with advanced injection timing of 27°bTDC this effect is more pronounced.

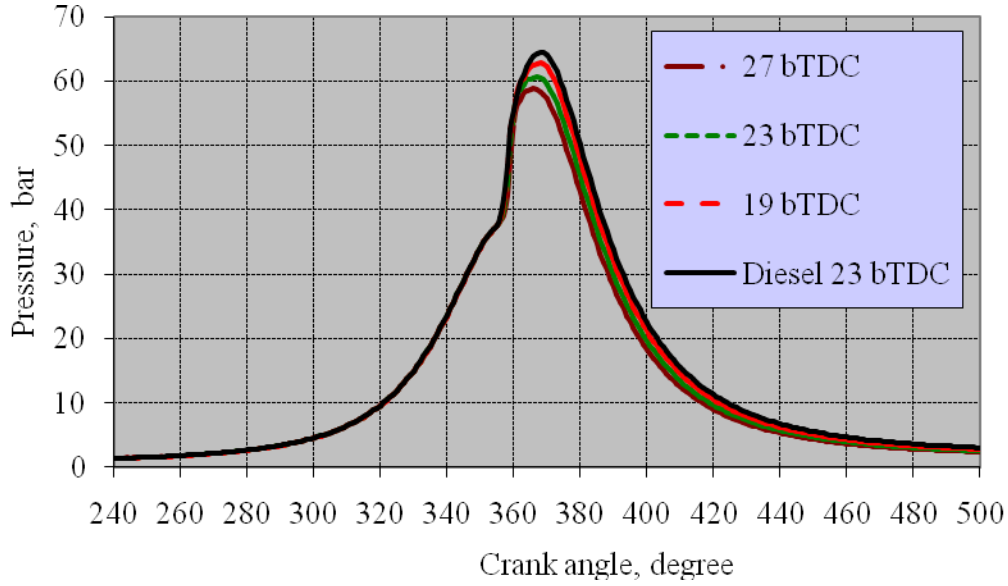

Fig.10 Effect of Injection timing on the pressure variation at full load engine operation

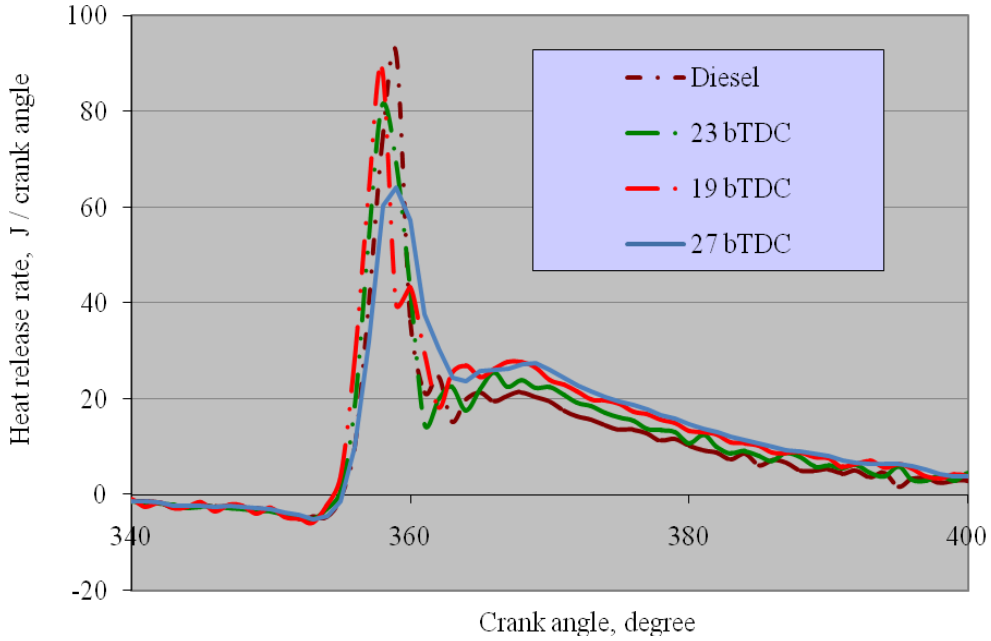

Fig.11 Effect of Injection timing on the heat release rate at full load engine operation

3.2 Effect of Injector Opening Pressure (IOP)

Injector opening pressure of 205 bar for diesel was prescribed by the engine supplier. Effect of injector opening pressure on the UOME performance was undertaken and was subsequently varied from 210 to 240 (21.0-24.0 MPa) insteps of 10 bar. Also, nozzle geometry (3 hole, 4 hole, and 5 hole nozzle) were varied against these pressures. The part load (80% load) and full load tests were conducted at these injection pressures operating the engine at 1500 rev/min. At these loads, air flow rate, UOME flow rates, exhaust gas temperatures, HC, CO, and NOx emissions were recorded. Based on the results, the optimum injection pressures and nozzle geometry were identified and fixed for UOME. Subsequently performance, emission and combustion parameters with the standard diesel were compared.

3.2.1 Standard Diesel Operation

For diesel engine was operated only at manufacturer specified injector opening pressure (IOP) of 205 bar, optimum start of injection (SOI) 23^0 bTDC and maximum compression ratio 17.5. It was observed that maximum brake thermal efficiency at 80% load comply with BTE as specified in standards and was found to be 27.57%.

3.2.2 Operation on UOME

3.2.2.1 Performance Parameters

The effect of injector opening pressure (IOP) and varying nozzle geometry at the static injection timing of 19^0 bTDC and maximum compression ratio 17.5 is presented in the following graphs. The effect of brake power on brake thermal efficiency at different injector opening pressures (IOP) and different nozzle geometry such as 3, 4, 5 and 6 holes are shown in Fig. 12-14. Amongst all the IOPs tested, the highest brake thermal efficiency occurred at 230 bar for a nozzle geometry with 4 hole having an orifice diameter of 0.3 mm each. This is because at higher injection pressures atomization, spray characteristics and mixing with air were better, resulting in improved combustion. Higher IOP (240 bar) will lead to delayed injection negating the gain due to higher IOP. The BTE is found to be 25.25% at 80% load and its maximum value obtained with 4-hole nozzle at an IOP of 230 bar. The BTE reported for 3-hole and 5-hole nozzles were 24.80% and 24.56% at 230 bar respectively. The results revealed that, BTE was found to be more with 4-hole nozzle geometry and IOP of 230 bars amongst all IOP and nozzle geometries studied. The IOP for other two nozzles (3-hole and 5-hole) were also found to be optimum at 230 bar respectively.

Increase in number of holes has not much effect on ignition delay, but the fuel-air mixing rate increases.

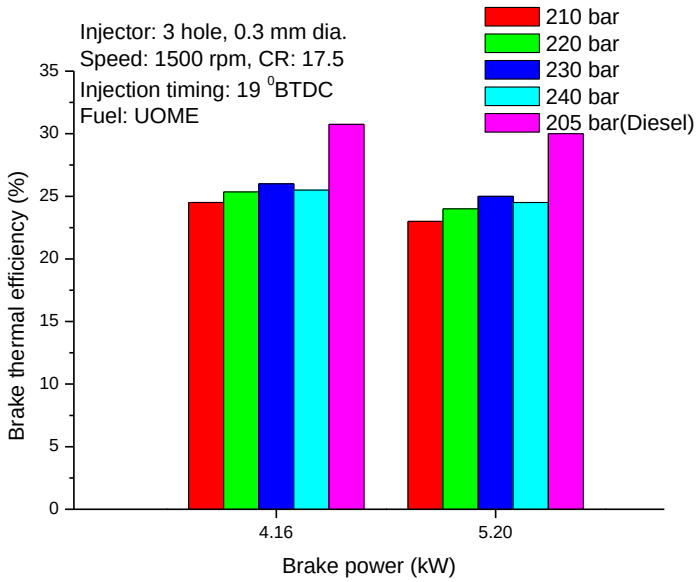

Fig. 12 Effect of 3-hole nozzle and varying injection pressure on BTE

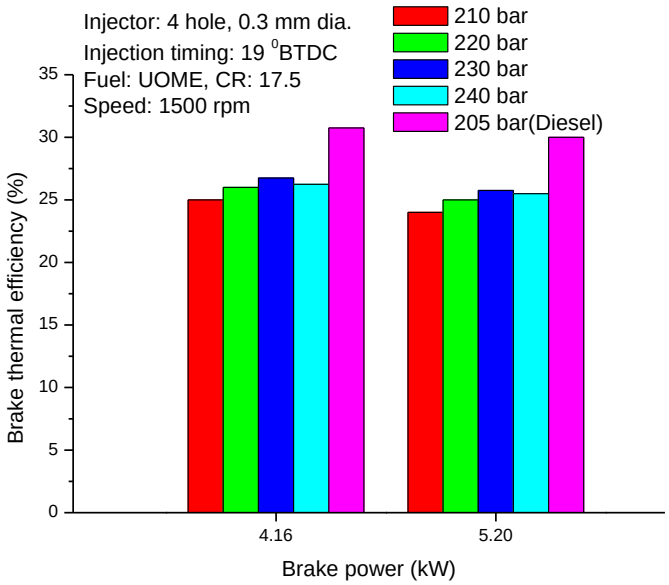

Fig. 13 Effect of 4-hole nozzle and varying injection pressure on BTE

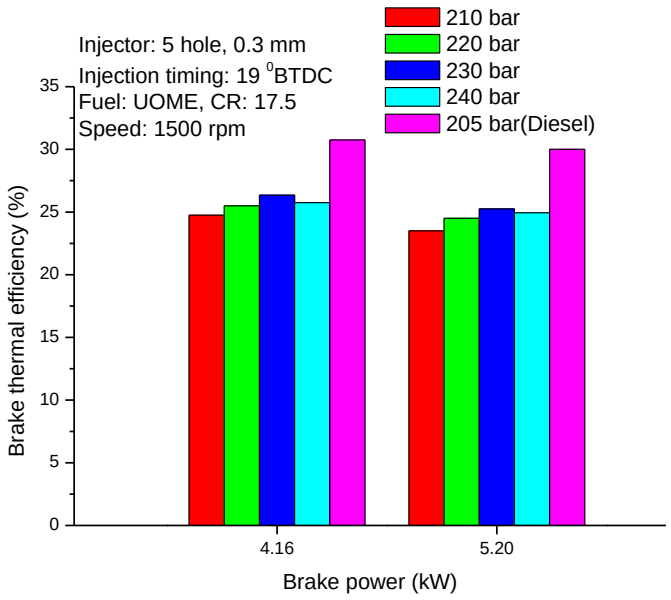

Fig. 14 Effect of 5-hole nozzle and varying injection pressure on BTE

3.2.2.2 Emission Parameters

HC Emission

Figures 15-17 show the effect of brake power on HC emission at different IOP and different nozzle geometry with COME. A significant drop in HC emission is observed at 230 bar IOP with 4-hole nozzle geometry because of better combustion. Enhanced atomization will also lead to a lower ignition delay. This will enhance the performance of the engine with biodiesels, which normally have a higher ignition delay on account of their higher viscosity. An improvement in the spray, will lead to a lower physical delay. HC is reduced from 90 to 80 ppm after increasing the IOP from 210 to 230 bar at full load. The highest IOP of 240 bar leads to an increase in the HC level to 85 ppm probably because it leads to a reduction in the brake thermal efficiency. Also a very high injector opening pressure will lead to a considerable portion of the combustion occurring in the diffusion phase on account of the small ignition delay. Higher IOP (240 bar) will lead to delayed injection negating the gain due to higher IOP.

HC emissions are found to be lower at IOP 230 bar for 4-hole compared to 3 and 5-hole nozzle geometry respectively. The conclusion is that, un-burnt hydrocarbons are less during 4-hole nozzle and IOP of 230 bar operation and is due to improved atomization and proper combustion of COME.

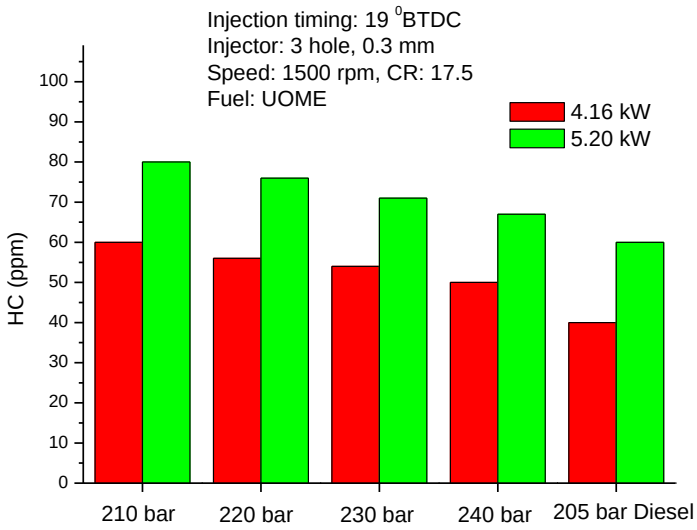

Fig. 15 Effect of brake power on HC at 3-hole nozzle and varying pressure

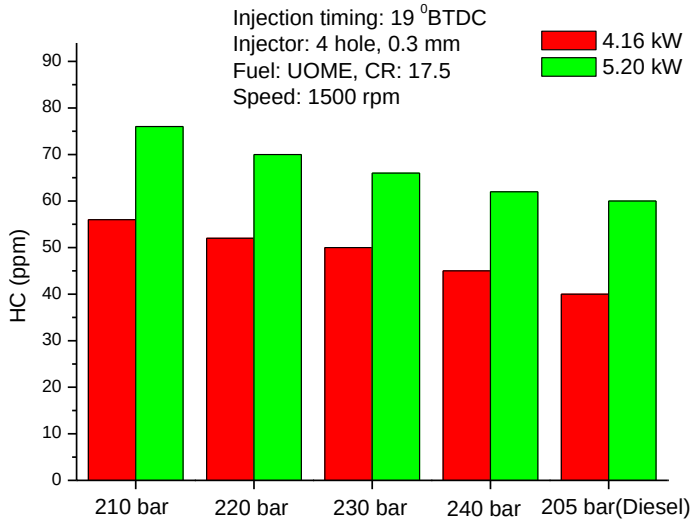

Fig. 16 Effect of brake power on HC at 4-hole nozzle and varying pressure

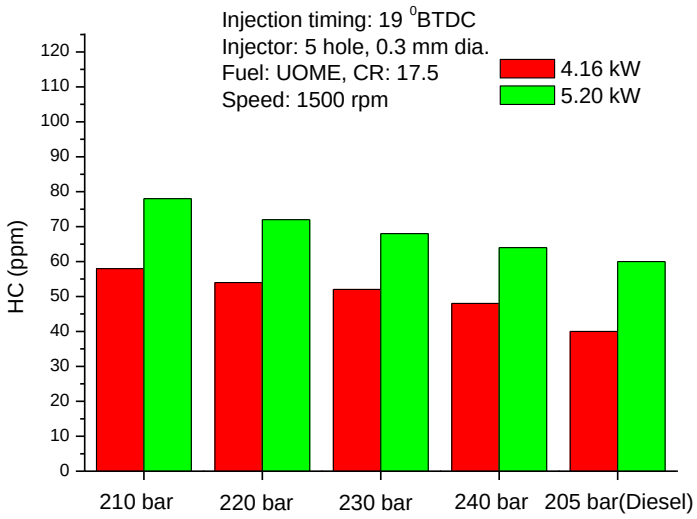

Fig. 17 Effect of brake power on HC at 5-hole nozzle and varying pressure

CO Emission

Fig. 18-20 shows effect of brake power on CO emission. Observed trends for CO emissions were similar to HC emissions, with lower CO emissions occurring at 230 bar injection opening pressure and 4 hole injector.

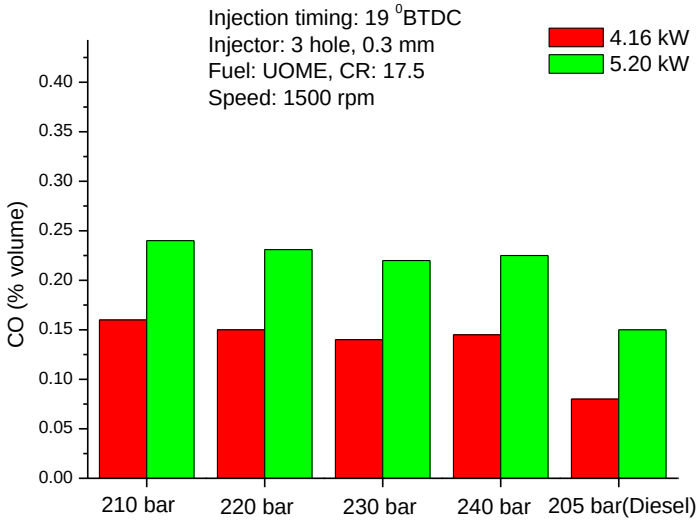

Fig. 18 Effect of brake power on CO at 3-hole nozzle and varying injection pressure

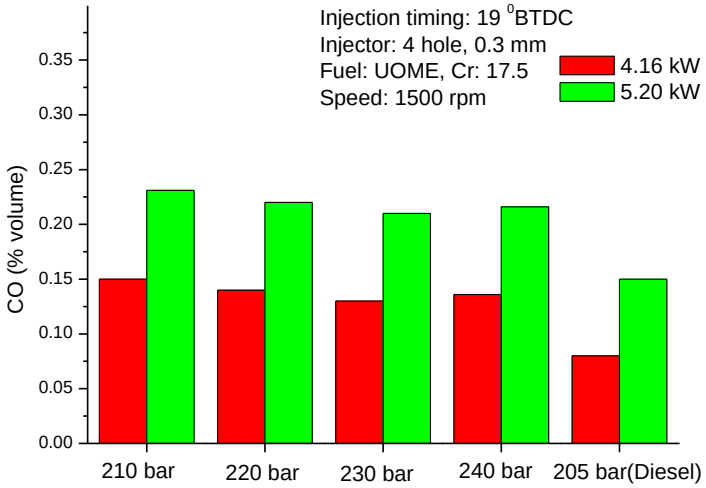

Fig. 19 Effect of brake power on CO at 4-hole nozzle and varying IOP

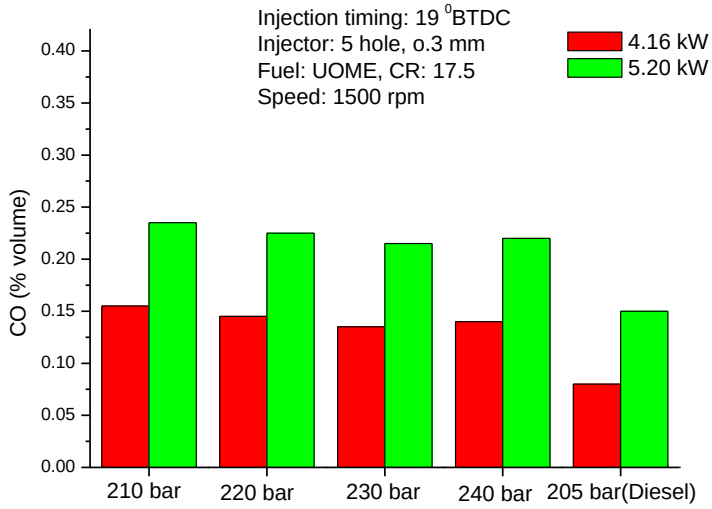

Fig. 20 Effect of brake power on CO at 5-hole nozzle and varying pressure

NO_x Emission

NO_x emissions increases with the increase in IOP due to faster combustion and higher temperatures reached in the cycle as shown in Fig. 21-23.

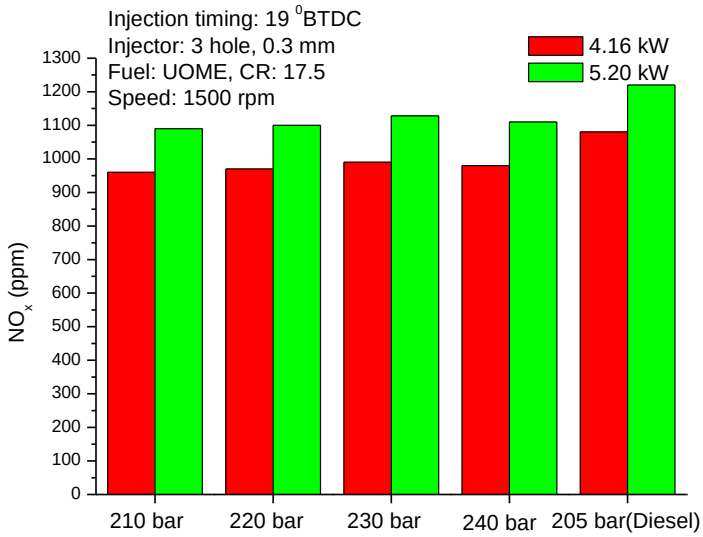

Fig. 21 Effect of brake power on NO_x at 3-hole nozzle and varying injection pressure

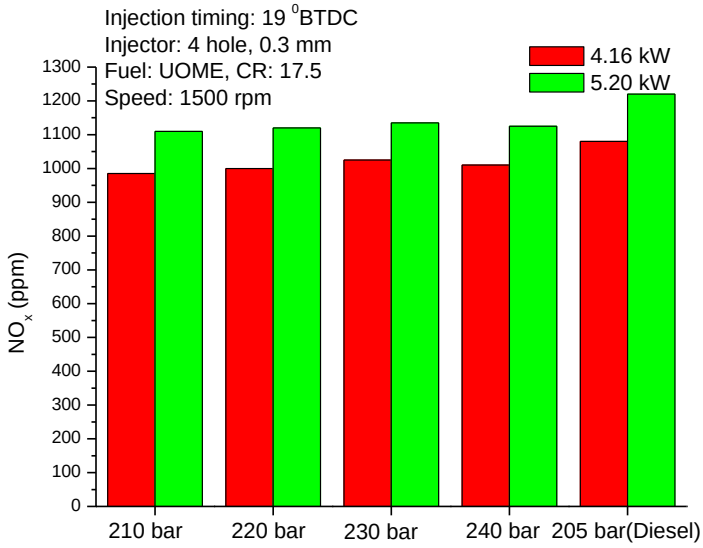

Fig. 22 Effect of brake power on NO_x at 4-hole nozzle and varying IOP

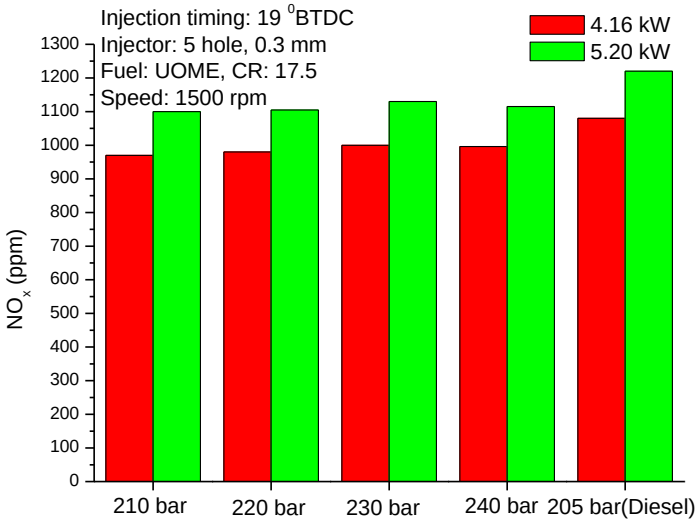

Fig. 23 Effect of brake power on NO_x at 5-hole nozzle and varying pressure

3.2.2.3 Combustion Analysis

Fig. 24 and 25 show the effect of injection pressure on the pressure and heat release rates with crank angle at full load engine operation. Throughout the combustion process the peak pressure of UOME increased with increase in fuel injection pressure. The increase in peak pressure was observed when the injection pressure was varied from 210 bar to 230 bar as shown in Fig. 24. Beyond 230 bar the peak pressure was lowered due to the negation effect.

Fig. 25 shows that heat release rates for UOME operation throughout the combustion process also increased with increased fuel injection rate achieved by increasing the fuel injection pressure when IOP was increased from 210 bar to 230 bar. Heat release rates were lower for injection opening pressures of 210 and 220 bar compared to 230 and 240 bar respectively. Heat release is the highest for an injector opening pressure of 230 bar followed by 240 bar and 220 bar respectively. It is lowest for an injector opening pressure of 210 bar where the brake thermal efficiency was lower compared to other injector opening pressures.

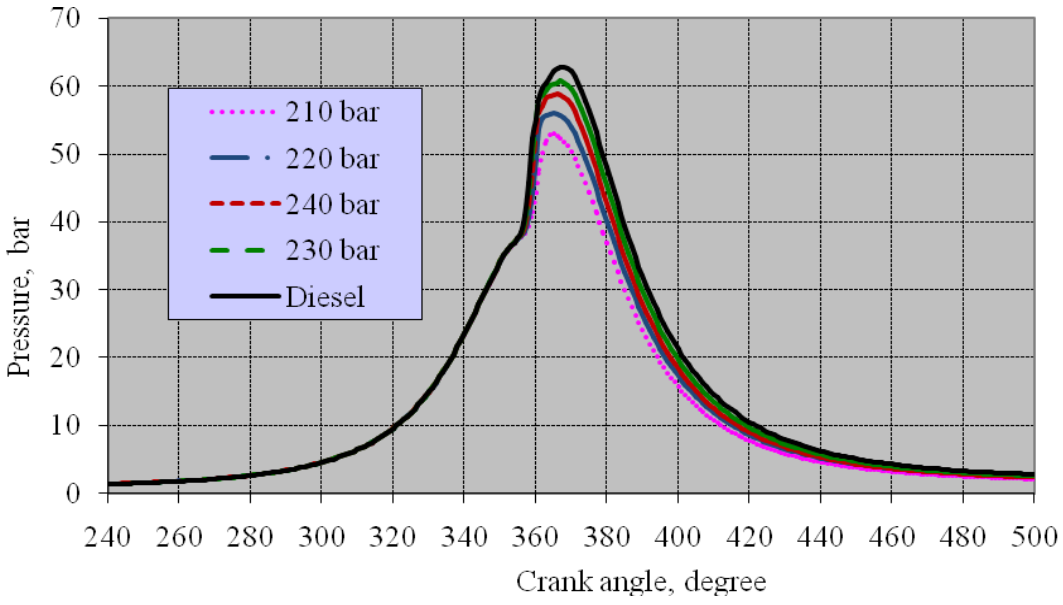

Fig. 24 Effect of injection pressure on the pressure variation with crank angle at full load engine operation.

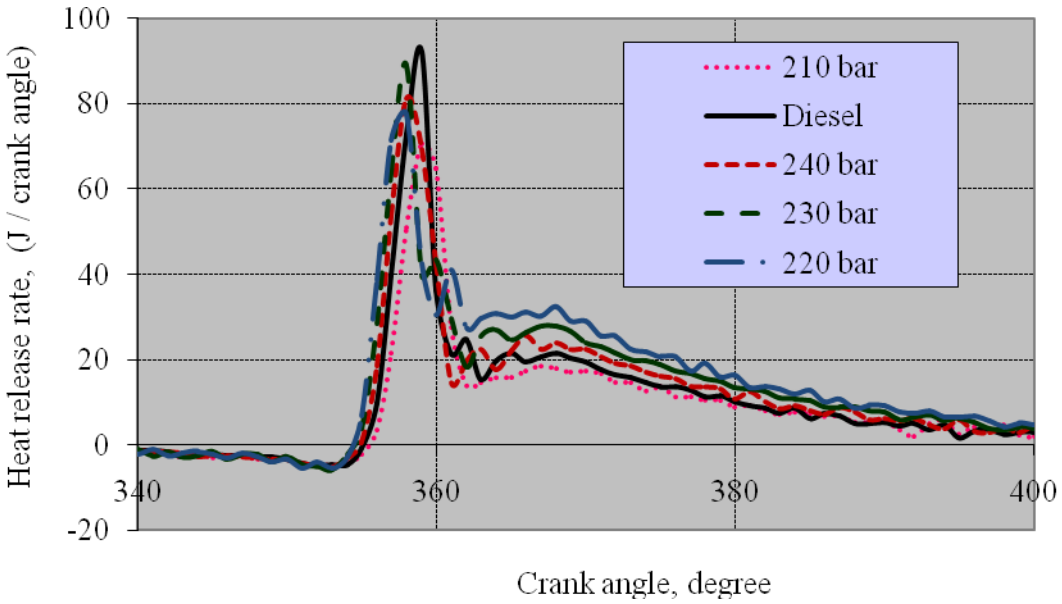

Fig. 25 Effect of injection pressure on the heat release rate with crank angle at full load engine operation.

3.3 Effect of Size of Injector Nozzle Holes

This section deals with the performance of UOME fuelled engine using a 4-hole injector with varying nozzle hole sizes. In order to study the effect of nozzle orifice on the biodiesel engine performance the orifice diameter was varied from 0.2-0.3 mm in a 4-hole injector. In order to further substantiate the simultaneous effect of nozzle orifice size and number of holes an injector having 6 hole with a nozzle orifice of 0.180 mm was selected.

3.3.1 Brake Thermal Efficiency

Fig. 26 shows variation of BTE for the UOME injected with a 4-hole injector for orifice size decreased from 0.3 to 0.2 mm. Decreasing the size of holes from 0.3 to 0.2 mm, ensures better mixing of air and biodiesel inside the combustion chamber and further leads to better combustion and hence increased BTE is found. However with less than 0.2 mm the above effect is nullified as fuel droplets move faster than air associated with poor mixing further resulting in inferior engine performance. BTE further increased with a 6 hole injector having a nozzle orifice of 0.180 mm and this could be mainly attributed to improved fuel-air mixing.

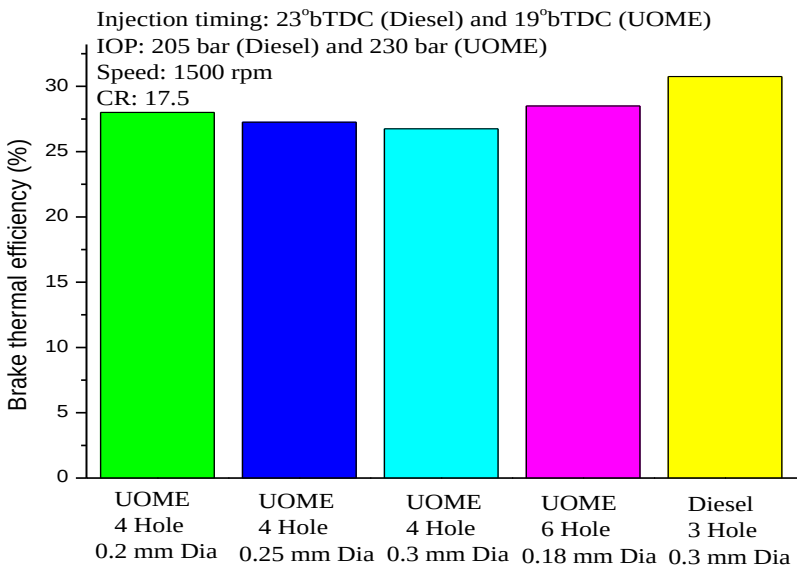

Fig. 26 Variation of brake thermal efficiency for UOME injected with different size orifice injectors

3.3.2 Emissions

This section explains the effect of injector nozzle orifices or holes and their size on the variation of smoke opacity, HC, CO, NO_x emissions for UOME operation.

HC Emission

Variation of HC and CO for the UOME with fixed 4-hole injector having varying orifice size from 0.3 to 0.2 mm is shown in Figs. 27, 28. It may be noted that higher HC and CO from exhaust are the direct result of incomplete combustion. HC and CO emissions were found to be lower for decreased injector hole sizes as the wall impingement with UOME is less compared to that with larger hole size. However higher hole size injectors leads to deposition of fuel on the combustion chamber walls. Hence higher HC and CO emissions were found to be more with 0.3 mm hole injector. HC and CO emissions further decreased with a 6 hole injector having a nozzle orifice of 0.180 mm. The reason could be attributed to increased BTE obtained with improved fuel-air mixing leading to reduced wall wetting in this injector.

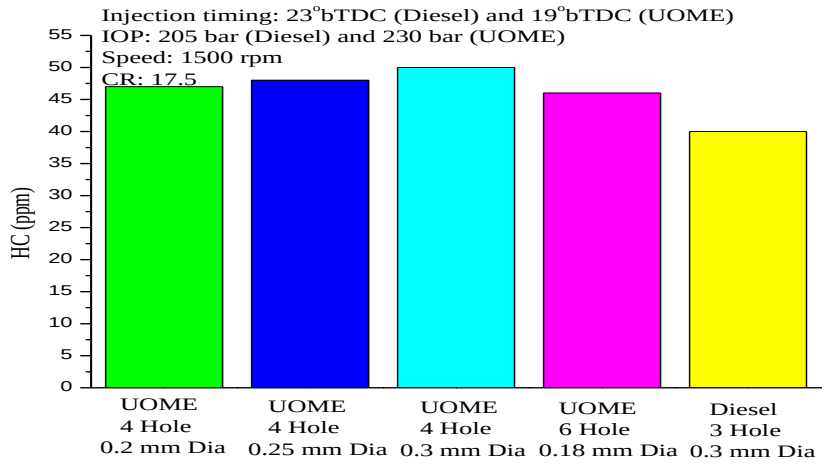

Fig. 27 Variation of HC for UOME injected with different size orifice injectors

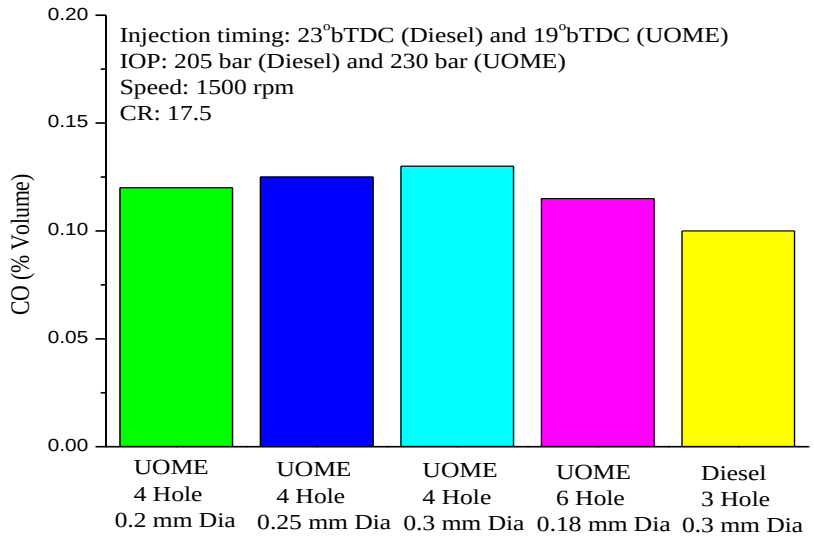

Fig. 28 Variation of CO for UOME injected with different size orifice injectors

NO_x Emission

Fig. 27 shows the variation of NO_x with 0.2, 0.25 and 0.3mm size of a 4 hole injector for UOME biodiesel. The variations in NO_x follow changes in adiabatic flame temperature. The reason for decreased NO_x with increased size of holes could be due to better combustion prevailing inside the engine cylinder and more heat released during premixed combustion. These effects also vary with spray pattern of liquid fuels, suggesting that reaction zone stoichiometry and post combustion mixing are also influenced by fuel composition. Reducing the injector orifice size further lowers NOx emission with a 6 hole injector.

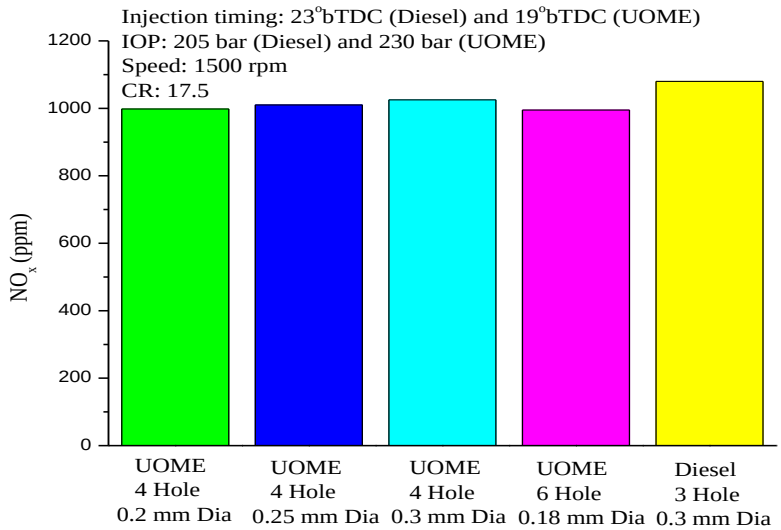

Fig. 29 Variation of NOx for for UOME injected with different size orifice injectors

3.3.2 Combustion Analysis

The cylinder pressure crank angle history was obtained for 100 cycles for UOME operation and the average pressure variation with crank angle at 80 % load using 4-hole injector of different orifice sizes is shown in Fig. 30. The UOME operation with a 4 hole injector having an orifice size of 0.2 mm showed higher peak pressure compared to other orifice sizes.

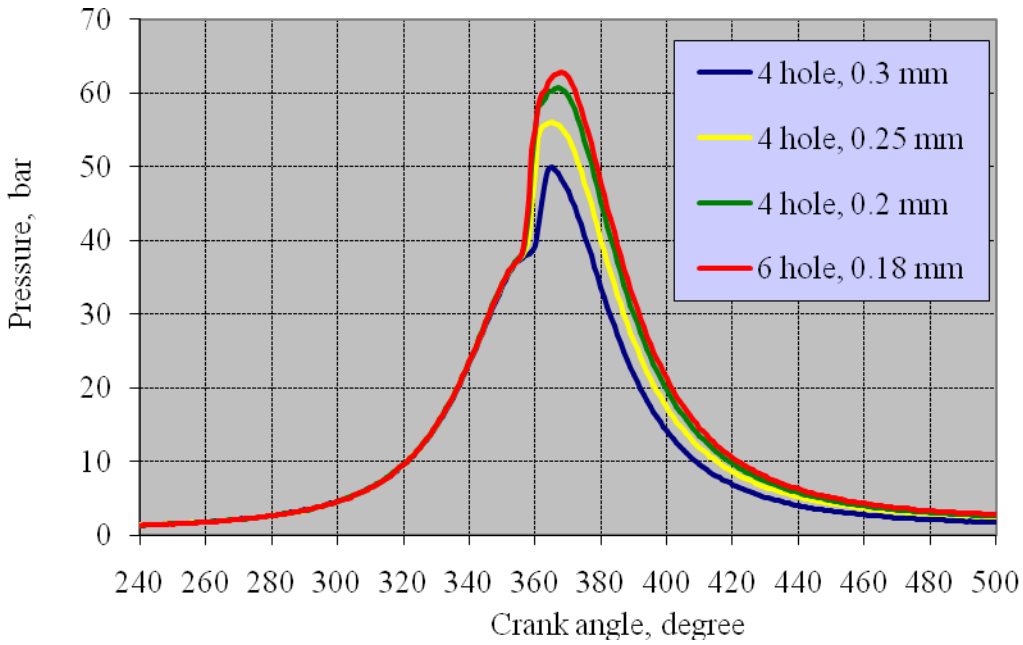

Fig. 30 Variation of pressure with crank angle for 4-hole injector with different size orifices

Figure 31 shows the variation of heat release rate with biodiesel injected with 4-hole injector having different orifice sizes. The premixed burning phase associated with a high heat release rate is significant with diesel operation which is responsible for higher peak pressure and higher rates of pressure rise. This is the reason for the higher thermal efficiency of diesel. The diffusion-burning phase indicated under the second peak is greater for HOME respectively when compared to diesel. The significantly higher combustion rates during the later stages with these biodiesel leads to high exhaust temperatures and lower thermal efficiency. UOME operation using 4 hole injector the heat release rate increased with reduced nozzle hole size from 0.3 to 0.20 mm. Fine spray of the biodiesel obtained with reduced hole size of the 4 hole injector ensures better mixing of the air and fuel and the associated combustion leading to higher heat release rates.

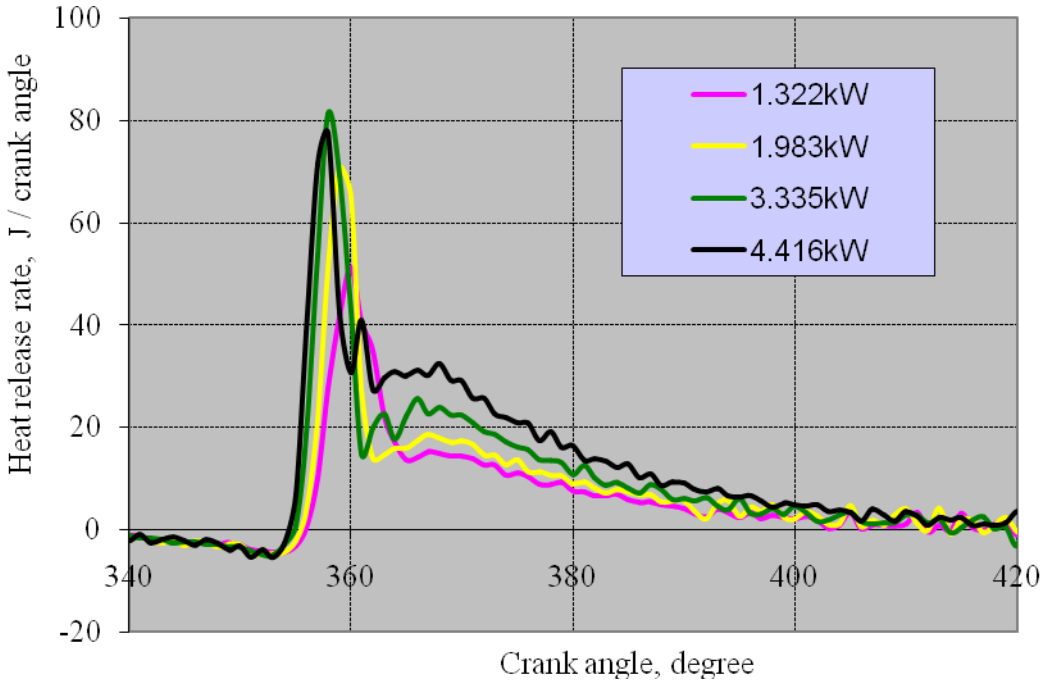

Fig. 31 Variation of heat release rate with crank angle for 4-hole injector with different size orifices

4.0 Effect of Combustion Chamber Shapes on the UOME Operation

In the present work, diesel engine was operated on diesel, and UOME with different combustion chamber shapes namely cylindrical (CCC), trapezoidal (TrCC), and toroidal combustion chamber (TCC) shapes. The results obtained are presented in the subsequent paragraphs. The injector with 6 hole each having a diameter of 0.18 mm was used for all the combustion chamber shapes used. The biodiesel is injected at 230 bar pressure.

4.1 Performance and Emission Characteristics

Figure 32 shows the variation of brake thermal efficiency (BTE) with brake power. BTE for diesel was higher than UOME operation over the entire load range. This is mainly due to its lower calorific value, lower volatility and higher viscosity. The improper mixture formation leads to incomplete combustion and hence lower BTE is obtained with UOME. UOME operation with TCC resulted in better performance compared to other combustion chambers. It may be due to the fact that, the TCC prevents the flame from spreading over to the squish region resulting in better mixture formation of biodiesel-air combinations, as a result of better air motion and lowers exhaust soot by increasing swirl and tumble. Based on the results, it is observed that the TCC has an ability to direct the flow field inside the sub volume at all engine loads and therefore substantial differences in the mixing process may not be present.

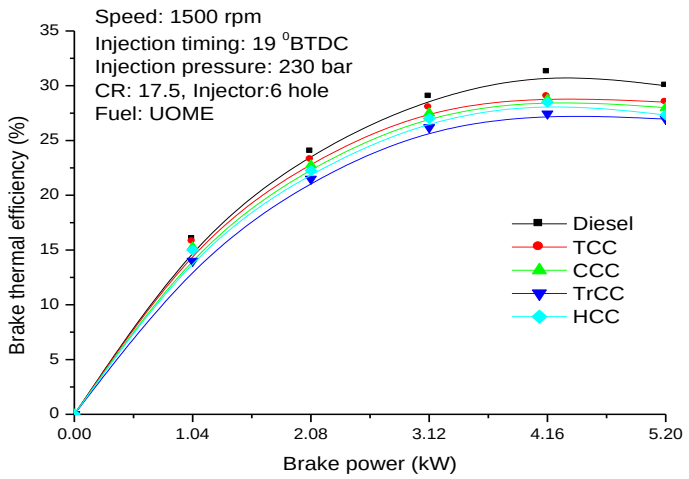

Fig. 32 Variation BTE with BP

Figure 33 shows variation of smoke opacity with brake power. Smoke opacity for diesel was lower than UOME over the entire load range. This may be attributed to improper fuel-air mixing due to higher viscosity and higher free fatty acid content of biodiesel considered. However, TCC gives lower smoke emission levels compared to other combustion chambers. It may be due to the fact that, the air-fuel mixing prevailing inside combustion chamber and higher turbulence resulted in better combustion and oxidation of the soot particles which further leads to reduction in the smoke emission levels.

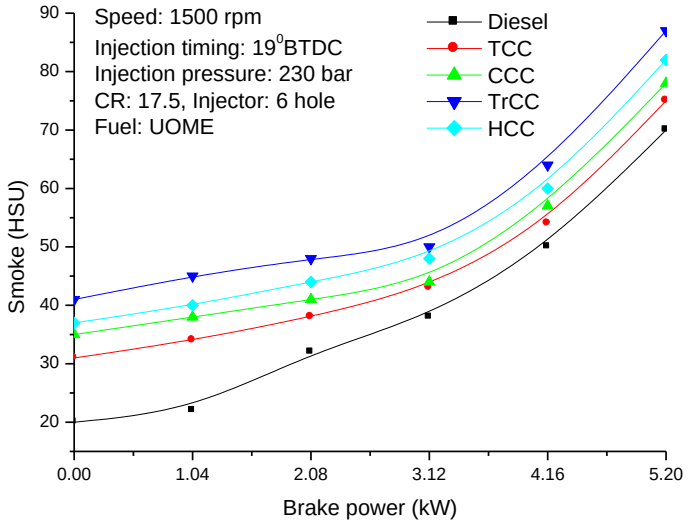

Fig. 33 Variation of Smoke opacity with BP

Figure 34 and 35 shows the variation of hydrocarbon (HC) and carbon monoxide (CO) emission levels for diesel, and UOME at all loads. Both HC and CO emission levels were higher for and UOME compared to diesel operation. Incomplete combustion and lower BTE of UOME is responsible for this observed trend. It could be due to the lower calorific value, lower adiabatic flame temperature and higher viscosity and lower mean effective pressures obtained with UOME.

However, TCC resulted in lower HC and CO emission levels compared to other combustion chamber shapes. It could be due to higher turbulence and comparatively higher temperature prevailing inside the combustion chamber that resulted into minimum heat losses and better oxidation of HC and CO and hence reduced emission levels. However, other combustion chambers may marginally contribute to the proper mixing of fuel combinations.

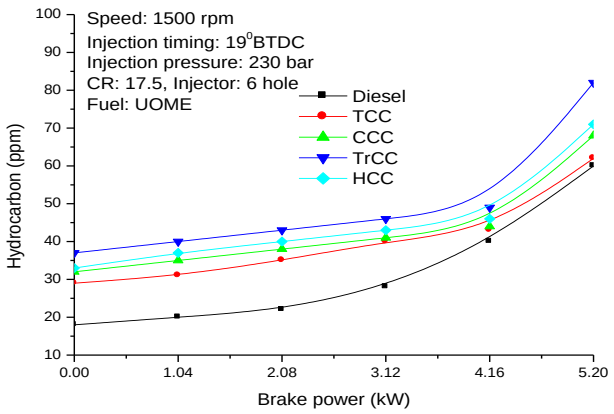

Fig. 34 Variation of HC with BP

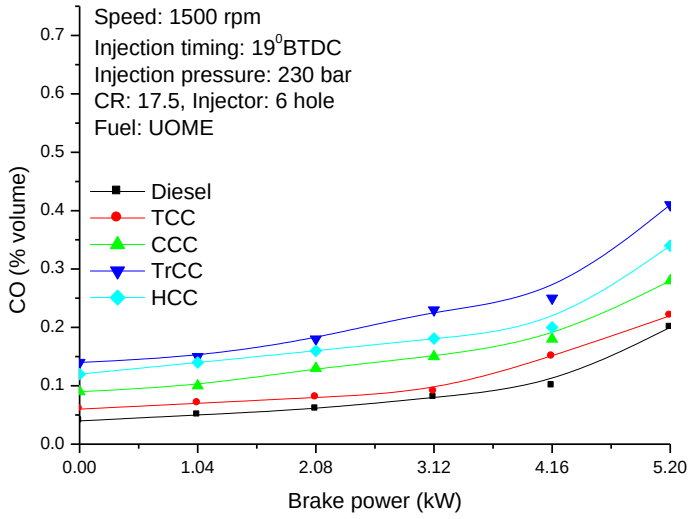

Fig. 35 Variation of CO with BP

The NO_x emission levels were found to be higher for diesel compared to biodiesel over the entire load range (Fig. 36). Higher heat release rates during premixed combustion phase observed with diesel compared to biodiesel operation is responsible for this trend. Slightly higher NO_x were resulted with TCC compared to other combustion chamber shapes tested. This could be due to slightly better combustion occurring due to more homogeneous mixing and larger part of combustion occurs just before top dead center. Presence of oxygen in the biodiesel is also responsible for this trend. Therefore it is resulted in higher peak cycle temperature.

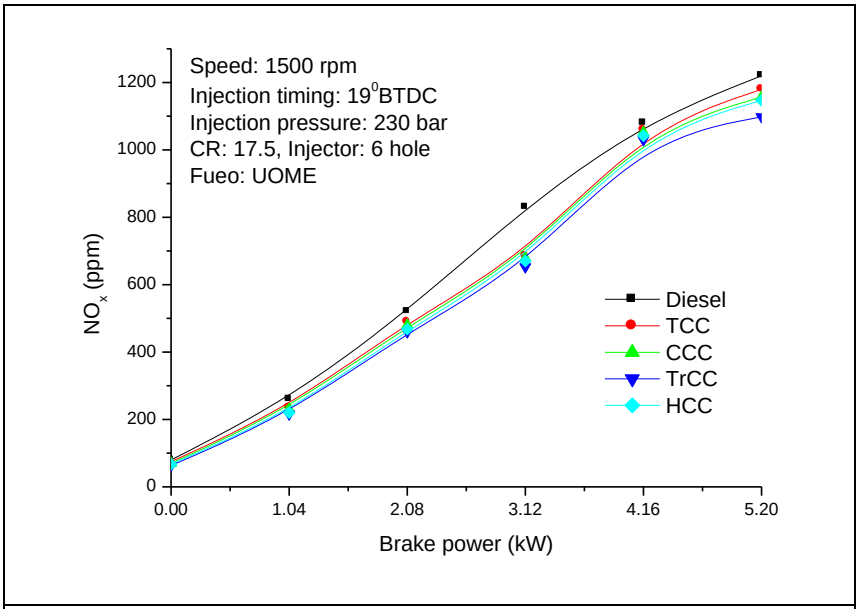

Fig. 36 Variation of NOx with BP

4.2 Combustion Characteristics:

In this section combustion characteristics of diesel engine fuelled with UOME biodiesel has been presented.

Ignition Delay

The variation of ignition delay with brake power for different combustion chamber shapes are shown in Fig. 37. The ignition delay is calculated based on the static injection timing. It is observed that ignition delay decreased with increased brake power for all combustion chamber shapes. With increased brake power, the amount of fuel being burnt inside the cylinder gets increased and subsequently the temperature of in- cylinder gases gets increased. This leads to reduced ignition delay with all combustion chamber shapes. However, the ignition delay for diesel was lower compared to biodiesel operation with all combustion chamber shapes used. However, lower ignition delays were observed for biodiesel operation with TCC compared to the operation with HCC, CCC and TrCC. It could be attributed to better air-fuel mixing and increased combustion temperature.

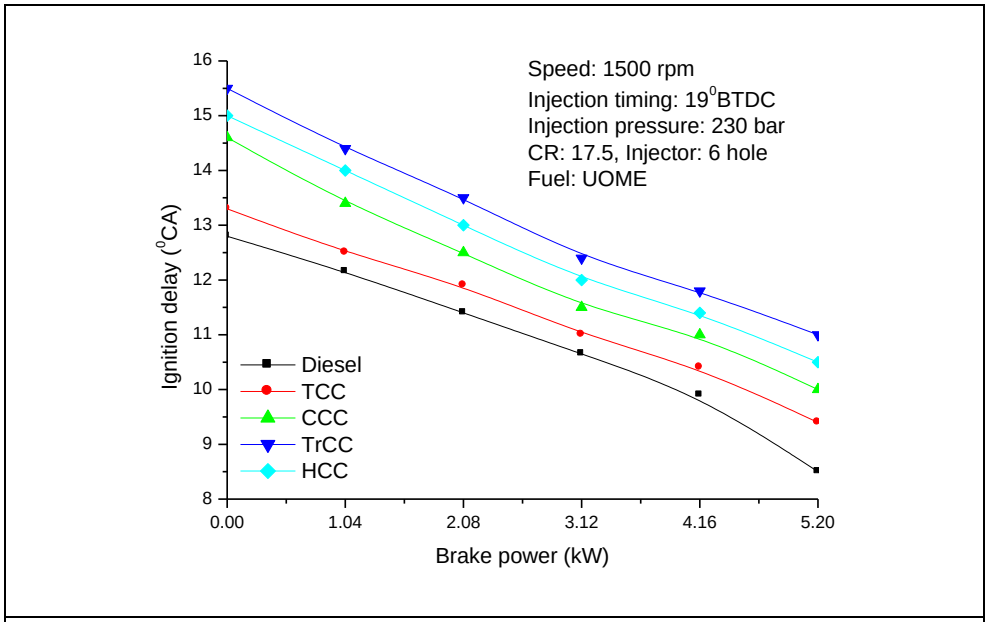

Fig. 37 Variation of ignition delay with BP

Combustion Duration

The combustion duration shown in Figure 38 was calculated based on the duration between the start of combustion and 90% cumulative heat release. The combustion duration increases with increase in the power output with all combustion chamber shapes adopted. This is due to the amount of fuel being burnt inside the cylinder gets increased. Combustion chamber being same, higher combustion duration was observed with biodiesel compared to diesel operation. It could be due to higher viscosity of biodiesel leading to improper air–fuel mixing, and needs longer time for mixing and hence resulting in incomplete combustion with longer diffusion combustion phase. However combustion duration was reduced with TCC compared to other combustion chambers tested. This could be attributed to improvement in mixing of fuel combinations due to better squish. Significantly lower combustion rates with biodiesel operation leads to higher exhaust temperatures and lower brake thermal efficiency. However, biodiesel operation with TCC showed improvement in heat release rate compared to other combustion chamber shapes.

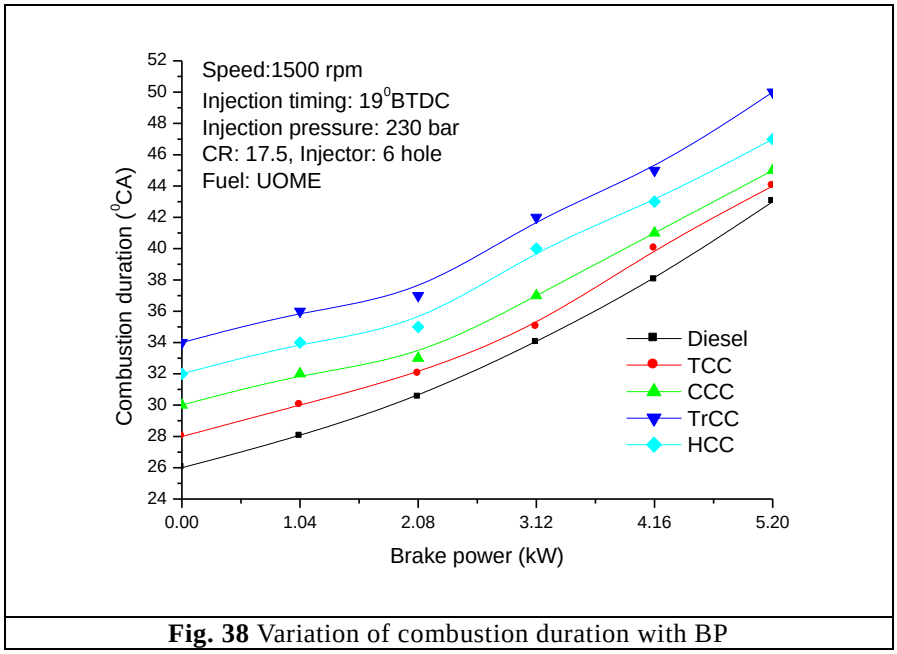

Fig. 38 Variation of combustion duration with BP

Cylinder Pressure

Figure 39 shows the effect of combustion chamber shapes on peak pressure for UOME with different combustion chambers used. The peak pressure depends on the combustion rate and amount of fuel consumed during rapid combustion period. Mixture preparation and slow burning nature of biodiesel during the ignition delay period were responsible for lower peak pressure and maximum rate of pressure rise. Biodiesel with TCC resulted in higher in-cylinder pressure as is evident from Figure 40. Higher in-cylinder pressure for biodiesel operation with TCC compared to other combustion chamber shapes was observed. It could be due to the combined effect of longer ignition delay, lower adiabatic flame temperature and slow burning nature of the biodiesel operation. This could be attributed to incomplete combustion due to improper mixing of fuel combinations, reduction of air entrainment, and higher viscosity of biodiesel. The sharp increase in combustion acceleration showed increased cylinder pressure during the piston's descent and that the combustion energy as efficiently converted into work.

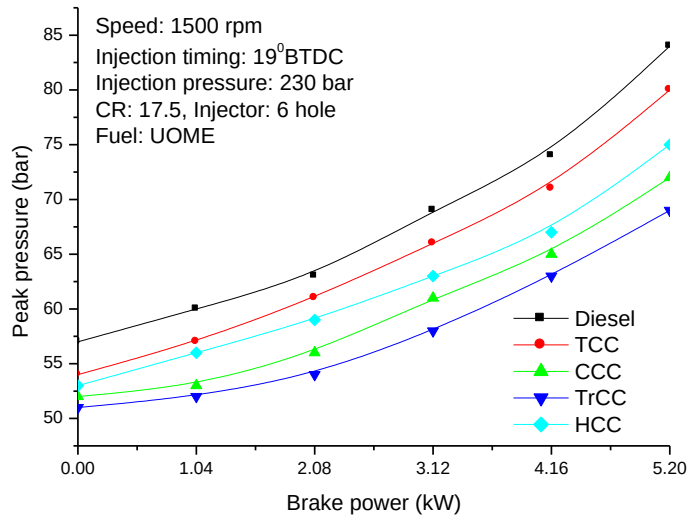

Fig. 39 Peak pressure variation for UOME

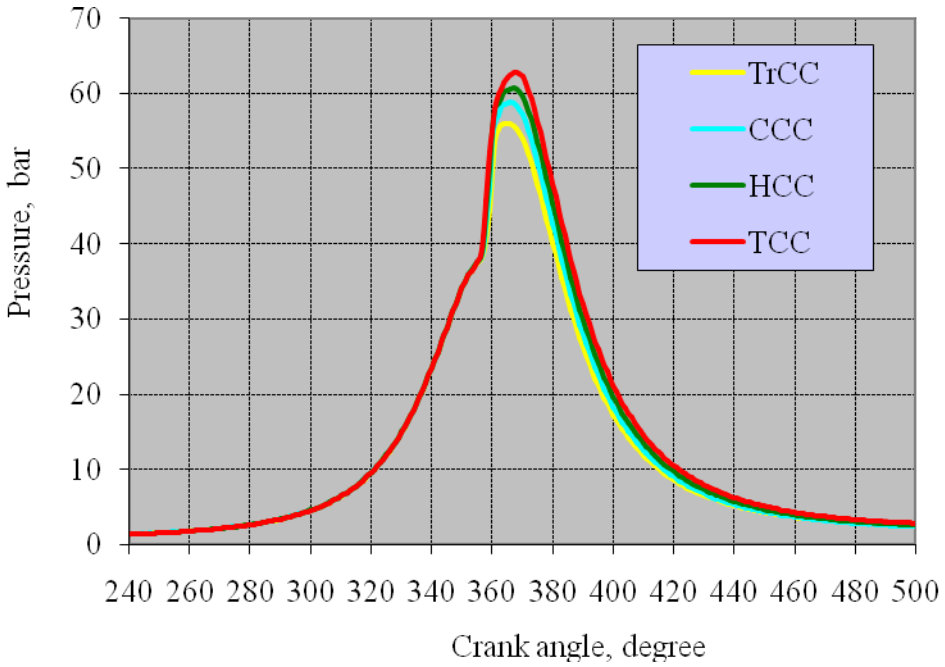

Fig. 40 In-cylinder pressure versus crank angle for different combustion chamber shapes for UOME at 80 % load

Heat Release Rate

Figure 41 shows rate of heat release versus crank angle for different biodiesel combinations with different combustion chamber shapes used. Biodiesel operation for HCC, CCC and TrCC resulted into lower heat release rate compared to the operation with TCC. This is due to the result of higher second peak obtained with HCC, CCC and TrCC in the diffusion combustion phase compared to the TCC operation.

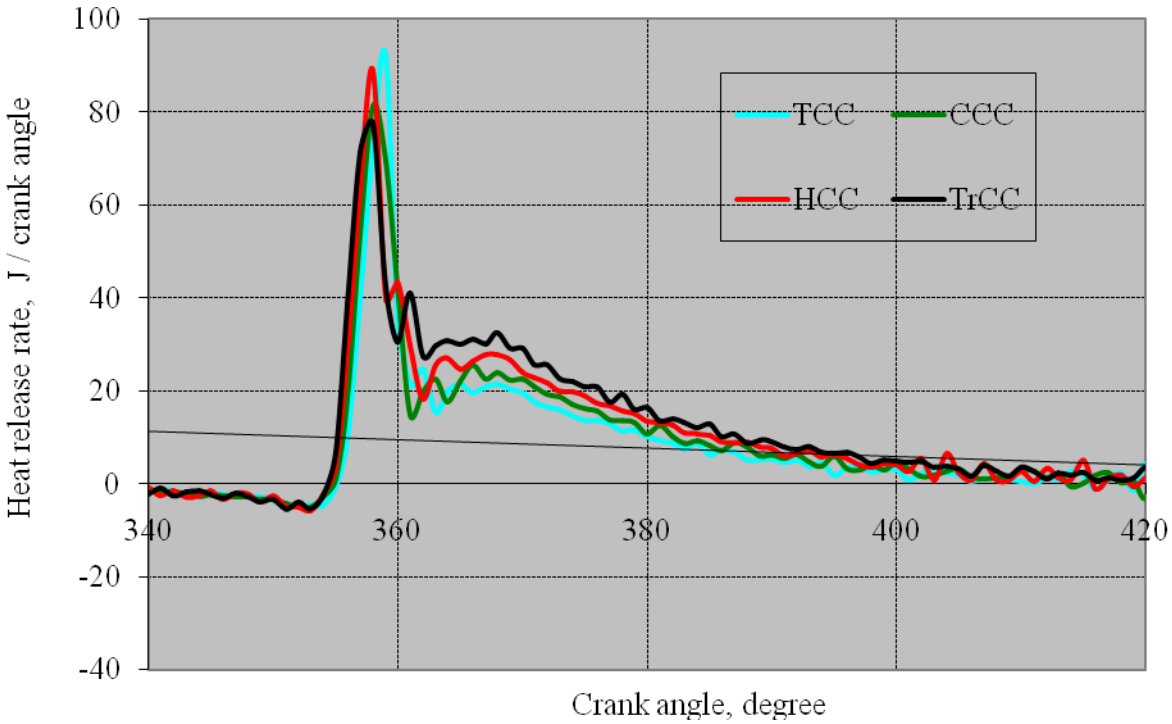

Fig. 41 Rate of heat release versus crank angle for different combustion chamber shapes for HOME at 80 % load.

Conclusions

From this exhaustive study on the feasibility of UOME in diesel engine application it is found that the performance was inferior compared to diesel. By suitable adjustments in the engine parameters such as injection timing, injector opening pressure, nozzle geometry in terms of number of holes and hole size and combustion chamber shape adopted it is found that performance could be improved. Hence UOME can be alternatively used as fuel for diesel engines.

Retarding injection timing from 23 to 19^{o} bTDC showed encouraging results, for UOME. Increasing injection pressure from 210 bar to 230 bar for UOME favored the engine performance with reduced emissions. Increasing the number of nozzle holes in the fuel injector from 3 to 6 improved the performance of the engine with reduced emissions for UOME operation. Keeping number of nozzle holes fixed and reducing their size remarkably affected engine performance. The performance was further improved with TCC combustion chamber provision. Combustion chamber optimization coupled with optimized injector position further improved the biodiesel engine performance. The use of mechanical injection systems (in the present study) has limitation on the injector opening pressure of 300 bar and can be overcome by CRDI and Unit injector injection systems. Such advanced injection systems are more suitable for viscous biodiesels applications and they can be injected to a pressure of 1500 to 3000bar.

References

1. *Arturo de Risi, Teresa Donateo, Domenico Laforga*, 'Optimisation of combustion chamber of direct injection diesel engines', *Society of Automotive Engineers, Paper No.: **2003-01-1064**, (2003) USA.*
2. *Banapurmath N.R., and Tewari P.G.,* 'Combustion and emission characteristics of a Direct Injection CI engine when operated on Honge oil, Honge oil methyl ester (HOME) and blends of Honge oil methyl ester (HOME) and diesel', *International Journal of Sustainable Engineering, **1** (2008), 80–93.*
3. *Banapurmath N.R. and Tewari P.G.,* 'Effect of biodiesel derived from Honge oil and its blends with diesel when directly injected at different injection pressures and injection timings in single cylinder

water cooled compression ignition engine', *Proceedings of Institute of Mechanical Engineers, Part A: Journal of Power and Energy, Professional Engineering Publications* ***223(1)*** *(2009), 31-40.*

4. *Bari, S., T.H. Lim, and C.W. Yu,* 'Effect of Preheating of Crude Palm Oil (CPO) on Injection System, Performance and Emission of a Diesel Engine', *Renewable Energy,* ***27*** *(2002), 339-351.*
5. *Bari, S., C.W. Yu, and T.H. Lim,* 'Effect of fuel injection timing with waste cooking oil as a fuel in direct injection diesel engine', *Proceedings of the Institution of Mechanical Engineers, Part D:Journal of Automobile Engineering,* ***218*** *(2004), 93-104.*
6. *Benajes J., Pastor J.V., Pyari R., and Plazas A.H.* 'Analysis of the influence of diesel nozzle geometry in the injection rate characteristics', *Journal of Fluids Engineering,* ***126*** *(2004), 63-71.*
7. *Bharathi Pratiba V.V., and G. Prasanthi,*.'Influence of in cylinder air swirl on diesel engine performance and emission', *International Journal of Applied Engineering and Technology,* ***1*** *(2011), 113-118.*
8. *Cenk Sayin, Metin Gumus,* 'Impact of compression ratio and injection parameters on the performance and emissions of a DI diesel engine fueled with biodiesel-blended diesel fuel', *Applied Thermal Engineering,,* ***31*** *(2011), 3182-3188.*
9. *Gajendra Babu, M. K., Chandan Kumar and L.M. Das,* 'Experimental Investigations on a Karanja Oil Methyl Ester Fuelled DI Diesel Engine, *Society of Automotive Engineers, Paper No.:* ***2006-01-0238****, (2006) USA.*
10. *Gajendra Babu M. K.,* 'Studies on performance and exhaust emissions of a CI engine operating on diesel and diesel biodiesel blends at different injection pressures and injection timings', *Society of Automotive Engineers, Paper No.:* ***2007-01-0613****, (2007) USA.*
11. *Godiganur, S., Murthy, C. Reddy, R.P.*. '6BTA 5.9 G2-1 Cummins engine performance and emission tests using methyl ester mahua (Madhuca indica) oil/diesel blends', *Renewable Energy,* ***34*** *(2009), 2172–2177.*
12. *Goldemberg J., Coelhobn S.T.,* 'Renewable energy- traditional biomass vs. modern biomass', *Energy Policy,* ***32*** *(2004), 711-714.*
13. *Hans J.S., Lu P.H., Xie X.B., Lai M.C., Henein N.A.*, Investigation of diesel spray primary breakup and development for different nozzle geometries. *Society of Automotive Engineers, Paper No.:* ***2002-01-2775*** *(2002) USA.*
14. *Hountalas, D.T., D.A. Kouremenos, K.B.Binder, A. Raab, and M.H. Schnabel,* 'Using advanced Injection timing and EGR to Improve DI engine efficiency at Acceptable NO and Soot levels' *Society of Automotive Engineers, Paper No.:* ***2001-01-0199,*** *(2001) USA.*
15. *Jaichander, S., Annamalai, K.,* 'Performance and exhaust emission analysis on pongamia biodiesel with different open combustion chambers in a DI diesel engine', *Journal of Scientific Industrial Research,* ***71*** *(2012), 487-491.*
16. *Jaichander, S., Annamalai, K.,.'*Effects of open combustion chamber geometries on the performance of pongamia bodiesel in a DI diesel engine', *Fuel* ***98*** *(2012), 272-279.*
17. *Jaichander, S., Annamalai, K.,* '*Combined impact of injection pressure and combustion chamber geometry on the performance of a biodiesel fueled diesel engine*', *Energy* ***47*** *(2013), 388-394.*
18. *Kruczynski Stanislaw W.,* 'Performance and emission of CI engine fuelled with camelina sativa oil', *Energy Conversion and Management,* ***65*** *(2013), 1-6.*
19. *Karnwal A., Naveen Kumar, M.M., Hasan, Rajeev Chaudhary, Arshad Noor Siddiquee, and Z. A. Khan,* 'Production of Biodiesel from Thumba Oil: Optimization of Process Parameters, Iranica', *Journal of Energy & Environment,* ***1*** *(2010), 352-358.*
20. *Montajir R., Tsunemoto H., Minami T.,* 'Fuel spray behavior in a small DI diesel engine: effect of combustion chamber geometry', *Society of Automotive Engineers, Paper No.:* ***2000-01-0946*** *(2000) USA.*
21. *Matsumoto, K., Inoue, T., Nakanishi, K., and Okumura, T.,* 'The effects of combustion chamber design and compression ratio on emissions, fuel economy and octain number requirement', *Society of Automotive Engineers Paper No.:* ***770193****,* (1977) doi:10.4271/770193.
22. *McCracken M. E., and J. Abraham,* 'Swirl-Spray Interactions in a Diesel Engine', *Society of Automotive Engineers Paper No.:* ***2001-01-0996*** *(2001) USA.*
23. *Mulemane A., Hans J.S., Lu P.H., Yoon S.J., and Lai M.C.,* 'Modelling dynamic behavior of diesel fuel injection systems' *Society of Automotive Engineers Paper No.:* ***2004-01-0536 (****2004) USA.*
24. *Muralidharan K., Vasudevan D.,* 'Performance, emission and combustion characteristics of a variable compression ratio engine using methyl esters of waste cooking oil and diesel blends', *Applied Energy,* ***88*** *(2011), 3959-3968.*
25. *Nwafor O.M.I.,* "Effect of advanced injection timing on the performance of rapeseed oil in diesel engines", *Renewable Energy,* ***21*** *(2000), 433–44.*

26. *Nwafor O.M.I.,* "The effect of elevated fuel inlet temperature on performance of diesel engine running on neat vegetable oil at constant speed conditions", *Renewable Energy, **28** (2003), 171–81.*
27. *Onga, H.C., T.M.I. Mahlia, H.H. Masjukia, and R.S.Norhasyimab,* 'Comparison of palm oil, Jatropha curcas and Calophyllum inophyllum for Biodiesel', *Renewable and Sustainable Energy Reviews, **15** (2011), 3501–3515.*
28. *Prasad B.V.V.S.U., Sharma C.S., Anand T.N.C., and Ravikrishna R.V.,* 'High swirl-inducing piston bowls in small diesel engines for emission reduction', *Applied Thermal Energy, **88** (2011), 2355-2367.*
29. *Puhan, S., N. Vedaraman, B.V.B. Ram, G. Sankaranarayanan, K. Jeychandran,.* 'Mahua oil (Madhuca Indica seed oil) methyl ester as biodiesel-preparation and emission characteristics', *Biomass Bioenergy, **28** (2005), 87–93.*
30. *Puhan Sukumar Jegan, R, K. Balasubbramanian and G. Nagarajan*. 'Effect of injection pressure on performance, emission and combustion characetrstics of high linolenic linseed oil methyl ester in a DI Diesel engine', *Renewable Energy, **34** (2009), 1227-1233.*
31. *Pyari R., Salvador F.J., Gimeno J., Zapata L.D.,* 'Diesel nozzle geometry influence on spray liquid-phase fuel penetration in evaporative conditions, *Fuel **87** (2008), 1165-76.*
32. *Ragit, S. S., S. K. Mohapatra, K. Kundu.* 'Performance and emission evaluation of a diesel engine fuelled with methyl ester of neem oil and filtered neem oil*', Journal of Scientific and Industrial Research, **69** (2010), 62–66.*
33. *Ramadhas, A.S., C. Muraleedharan, and S. Jayaraj,* "Performance and emission evaluation of a diesel engine fueled with methyl esters of rubber seed oil.", *Renewable Energy, **30** (2005), 1789–1800.*
34. *Ramadhas, A.S., S. Jayaraj, and C. Muraleedharan.* 'Characterization and effect of using rubber seed oil as fuel in the compression ignition engines', *Renewable Energy, **30 (**2005), 795–803.*
35. *Raheman, H., Ghadge, S.V.,* 'Performance of compression ignition engine with mahua (Madhuca indica) biodiesel', *Fuel, **86** (2007), 2568-2573.*
36. *Raheman, H., Ghadge, S.V.,* 'Performance of diesel engine with biodiesel at varying compression ratio and ignition timing' , *Fuel, **87** (2008), 2659–2666.*
37. *Raheman, H. and A.G. Phadatare,*. 'Diesel engine emissions and performance from blends of karanja methyl ester and diesel', *Biomass and Bioenergy, **27** (2004), 393-397.*
38. *Rao, T. V. G. P. Rao, K.H.C. Reddy*. 'Experimental investigation of Pongamia, jatropha and neem methyl esters as biodiesel on C.I. engine', *Jordan Journal of Mechanical Industrial Engineering, **2** (2008), 117–22.*
39. *Ravindranath, N.H., Balachandra,P.,* 'Sustainable bioenergy for India: Technical, economic and policy analysis', *Energy, **34** (2009), 1003–1013.*
40. *Rosli, Abu Bakar and Abdul Rahim Ismail Semin,* 'Fuel injection pressure effect on performance of direct injection diesel engines based on experiment', *American Journal of Applied Sciences, **5** (2008), 197- 202.*
41. *Roy, M.M.,* 'Effect of fuel injection timing and injection pressure on combustion and odorous emissions in DI diesel engine', *Journal of Energy Resources Technology, ASME Transactions, **131** (2009), 1-8.*
42. *Sahoo, P.K., L.M. Das, M.K.J. Babu, and S.N. Naik,* 'Biodiesel development from high acid value polanga seed oil and performance evaluation in a CI engine', *Fuel **86** (2007), 448-454.*
43. *Sahoo, P.K., L.M. Das, M.K.G. Babu, P. Arora, V.P. Singh, N.R. Kumar, T.S. Varyani*, 'Comparative evaluation of performance and emission characteristics of Jatropha, karanja and polanga based biodiesel as fuel in a tractor engine', *Fuel **88** (2009), 1698-1707.*
44. *Scholl, K. W., Sorenson, S.C.,* 'Combustion analysis of soyabean oil methyl ester in a direct injection diesel engine', *Society of Automotive Engineers Paper No.: **930 – 9341993** (1993) USA.*
45. *Saito, T., Daisho, U.N., Ikeya N.* 'Effects of combustion chamber geometry on diesel combustion', *Society of Automotive Engineers, Paper No.: **1986 861186** (1986) USA.*
46. *Shinde Tanaji Balawant*. 'Experimental investigation on effect of combustion chamber geometry and port fuel injection system for CNG engine', *IOSR Journal of Engineering (IOSRJEN), **2** (2012), 49-54.*
47. *Singh, R. N., Singh, S. P., Pathak, B. S.,* 'Investigations on operation of CI engine using producer gas and rice bran oil in mixed fuel mode', *Renewable Energy, **32** (2007), 1565–1580.*
48. *Som S., Aggarwal S.K., El-Hannouny E.M., Longman D.E.,* 'Investigation of nozzle flow and cavitation characteristics in a diesel injector', *Journal of Engineering and Gas Turbine Power, **132** (2010), 1-12.*
49. *Sridhar G., Sridhar H. V., Basavaraj, M.S, Sudarshan H. I., Somsekhar, Dasappa S., Paul P. J.,*. 'Case study on distributed decentralized generation based biomass gasifier system' *The Energy & Resources Institute (TERI), New Delhi, (2008) 1- 15.*
50. *Sundraapandian, S., Devaradjane. G.,* 'Theoretical and experimental investigation of the performance of vegetable-oil-operated CI engine', *Society of Automotive Engineers, Paper No.: **2007-32-0067 (**2007) USA.*

51. *Tompkins B.T., Song H., Bittle J.A., Jacobs T.J.* 2012. 'Efficiency considerations for the use of blended biofuel in diesel engines', *Applied Energy, **98** (2012), 209-218*.
52. *Varuvel Edwin Geo, Mrad Nadia, Tazerout Mohand, Aloui Fethi*. 'Experimental analysis of biofuel as an alternative fuel for diesel engines', *Applied Energy, **94** (2012), 224-231*.
53. *Vedaraman N., Puhan Sukumar, Nagarajan G., Ramabrahman B.V., Velappan K.C.,* 'Methyl ester of Sal oil (Shorea robusta) as a substitute to diesel fuel-A study on its preparation, performance and emissions in direct injection diesel engine', *Industrial Crops and Products, **36** (2012), 282-288.*
54. *Venkanna, B. K. and C. Venkataramana Reddy,* 'Performance, emission and combustion characteristics of direct injection diesel engine running on calophyllum inophyllum linn oil (honne oil)', *International Journal of Agricultural & Bioogical Engineering, **4** (2011).*