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ABSTRACT

Internal combustion engines in now a days the best available reliable source of power for all domestic, large scale industrial and transportation applications. The major issue arises at the efficiency and exhaust emission of these engines. Every attempt made to improve these engines tends to attain the maximum efficiency and minimum emission. This study is about the effects of air swirl in the cylinder on its performance, combustion and emission. Since the turbulence is necessary for better mixing and the fact that it can be controlled by shape of the combustion chamber, makes this review paper necessary.

Keywords--- Combustion chamber, Performance, Combustion, Emission, Diesel engine, Biodiesel

I. INTRODUCTION

The compression ignition engine is an internal combustion engine that uses the increase in temperature in the compression stroke to ignite a fuel charge (fuel-air mixture). This is also called auto-ignition. These engines are always fuel injected. Air is drawn into the cylinder through the intake manifold and compressed by the piston. The most important function of the CI engine combustion chamber is to provide proper mixing of fuel and air (called carburetion) in a short time to lessen the ignition lag phase. In order to achieve this, an organized air movement called air swirl is provided to produce high relative velocity between fuel droplets and air. When the liquid fuel is injected into the combustion chamber, the spray cone gets disturbed due to the air motion and turbulence inside. The onset of combustion will cause an added turbulence that can be guided by the shape of the combustion chamber. Swirl is defined as organized rotation of charge about the cylinder axis. Swirl is created by bringing an intake flow into the cylinder with an initial angular momentum. Swirl is used in CI engine concepts to promote more rapid mixing between the inducted air charge and the injected fuel. Swirl is also used to speed up the combustion process and in two-stroke engines, it improves scavenging. Squish is the name given to the radially inward or transverse gas motion that occurs towards the end of the compression stroke when a portion of piston face and cylinder head approach each other closely [15].

1.1. Types of combustion chambers

1.1.1. Shallow Depth Combustion Chamber

The depth of the cavity provided in the piston is quite small. This chamber is usually adopted for large engines running at low speeds. Since the cavity diameter is very large, the squish is negligible [16].

Fig-1: Shallow depth combustion chamber [7].

1.1.2. Hemispherical Combustion Chamber

This chamber also gives small squish. However, in this case desired squish can be obtained by varying depth to diameter ratio [16].
1.1.3. Toroidal Combustion Chamber

It provides a powerful squish along with air movement. Due to powerful squish, the mask needed on inlet valve is small and there is better utilization of oxygen. Cone angle of spray for this type of chamber is 150° to 160° [16].

1.1.4. Cylindrical Combustion Chamber

This design was attempted in recent diesel engines. The swirl is produced by masking the cone for nearly 180° of circumference. In this case also, squish can be varied by varying the depth [16].

II. METHODOLOGY

2.1. Transesterification Reaction

It is most commonly used and important method to reduce the viscosity of vegetable oils. In this process triglyceride reacts with three molecules of alcohol in the presence of a catalyst producing a mixture of fatty acids, alkyl ester and glycerol. The process of removal of all the glycerol and the fatty acids from the vegetable oil in the presence of a catalyst is called esterification.

III. RESULTS AND DISCUSSION

3.1. Performance Parameters

3.1.1. Brake Thermal Efficiency (BTE)

C. V. Subba Reddy et al. (2012) found that, the brake thermal efficiency of the diesel engine depends on the combustion efficiency of the engine. That further depends on the formation of homogeneous mixture with turbulence which increases with whirl motion of air in the combustion chamber. The brake thermal efficiency of the base line engine with TGP-2 (tangential grooved piston 2) configuration was increased by 7.6% for 70% load operation compared with base line engine. Similarly the brake thermal efficiency of BLE-5 (base line piston of diesel engine) and BLE-3 were 6.2% and 3.68% more than BLE-1 [1]. S. Jaichandar et al. (2012) found that, BTE of 20% POME was lower compared to that of diesel with standard engine having HCC. Since the engine was operated under constant injection timing and POME had a smaller ignition delay, combustion was initiated much before TDC was reached. That increases compression work and more heat loss and thus reduces BTE of engine. BTE for TCC was higher when compared to SCC and HCC at all loads, may be due to better mixture formation of 20% POME and air, as a result of better air motion in TCC, which leads to better combustion of biodiesel and thus increases BTE. At all loads, BTE for SCC was lower than that of TCC. BTE for HCC lies between those of TCC and SCC at all loads [2]. Chandrashekharapuu Ramachandraiah Rajashekar et al. (2012) found that, an improvement of 2 to 3% brake thermal efficiency had been found with the modified piston due to better combustion. The brake thermal efficiency at 80% load for CR=17.5, IP=175, 200 and 225 bar was 22.15%, 24.52% and 22%. The brake thermal efficiency was improved at 200 bar injection...
pressure compared to 175 and 225 bar injection pressure since more surface area to volume ratio and finer spray which in turn reduce the physical delay period. At 225 bar injection pressure the brake thermal efficiency decreases due to ineffective combustion because of decreased depth of penetration [3]. M. Ravi et al. (2013) found that, the brake thermal efficiency for PSME (palm stearin methyl ester) flat piston at injection pressure of 240 bar had lower value of 23% compared to diesel that was due to lower momentum of very fine droplet penetration in combustion chamber and for ATME (animal tallow methyl ester) hemispherical piston had higher value which was 35.67% compared to diesel that was attributed to lower calorific value, lower viscosity coupled with density of fuel [4]. M. N. Channappagoudra et al. (2013) found that, brake thermal efficiency for 6-grooved piston was found to be higher compared to other two configurations of the piston. That might be due to enhanced mixing rate in case of six grooves carried by turbulence in combustion chamber [5]. M. V. S. Murali Krishna et al. (2013) found that, BTE increased up to 80% of the full load operation (BMEP=4.2 bar) due to conversion of increase of fuel efficiency and beyond that load it decreased due to decrease of air fuel ratios as oxygen was completely used up with both test fuels. CE (conventional engine) with biodiesel showed the compatible performance for entire load range when compared with the pure diesel operation on CE at recommended injection timing. Although carbon accumulations on the nozzle tip might play a partial role for the general trends found, the difference of viscosity between the diesel and biodiesel provided a possible explanation for the compatible performance of CE with biodiesel operation. BTE increased with the advancing of the injection timing in the CE with the biodiesel at all loads, when compared with CE at the recommended injection timing and pressure. That was due to initiation of combustion at earlier period and efficient combustion with increase of air entrainment in fuel spray giving higher BTE. BTE increased at all loads when the injection timing was advanced to 31° bTDC in CE at the normal temperature of biodiesel. Similar trends were found with preheated biodiesel also. Preheating of the biodiesel reduced the viscosity, which improved the spray characteristics of the oil. LHR (low heat rejection) combustion chamber with biodiesel showed the improved performance for the entire load range compared with CE with pure diesel operation. That was because of efficient combustion of biodiesel in the hot environment provided by LHR combustion chamber. The optimum injection timing was found to be 28° bTDC with LHR combustion chamber with normal biodiesel and preheated biodiesel operations. Since the hot combustion chamber of LHR combustion chamber reduced ignition delay and combustion duration and hence the optimum injection timing was obtained earlier with LHR combustion chamber when compared with CE with the biodiesel operation [6]. Ranganatha Swamy L. et al (2014) found that, BTE for diesel was varying with different combustion chambers over the entire load range. Different mixture formation occurring in the individual combustion chamber shapes leads to varying combustion behavior and hence varying BTE was obtained. TCC (toroidal combustion chamber) 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 was found 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 [7]. S. Jaichandar et al. (2014) found that, Brake thermal efficiency for TRCC (toroidal re-entrant combustion chamber) was higher when compared to the other four types of combustion chambers at all loads. That may be due to better mixture formation of POME (pongamia oil methyl ester) and air, caused by better air motion in TRCC. Improved mixture formation, leads to better combustion of the biodiesel and thus increases the brake thermal efficiency. At all loads the BTE for SCC (shallow depth combustion chamber) was lower than that of other combustion chambers. The BTE for SRCC lies between those of TRCC and TCC (toroidal combustion chamber) at all loads. BTE for TRCC (33.09% at full load) was higher, when compared to the other types of combustion chambers at all loads with POME20 [8]. Hariram V. et al. (2014) found that, initially at low loading conditions the BTE of STP (standard toroidal piston) with diesel and B100 was found to be 11.23% and 13.16% respectively whereas STRP (shallow toroidal re-entrant piston) exhibited 12.1% and 14.02% with straight diesel and B100 blends respectively. The HBP (hemispherical bowl piston) showed a very minimal BTE of 9.53% and 11.15% with straight diesel and B100. At part loads the BTE of all the piston geometries showed an increasing trend with the STRP had the highest BTE further it had been found that at full load conditions the BTE of STRP with straight diesel and B100 RBBD as 27.33% and 29.31% respectively whereas STP showed only 27.24% with B100 blend which may be due to better combustion, swirl and squish created in the re-entrant piston geometry. The HBP showed a negative increase in BTE at full load which may be due to poor premixed combustion phase [9]. Vaibhav Bhatt et al. (2014) found that, the brake thermal efficiency for normal engine at full load was 35.81%. The engine with NP-8 (normal piston with 8 grooves) and NP-12 (normal piston with 12 grooves) give thermal efficiencies of 37.08% and 38.43%. With NP-12 configuration had higher thermal efficiency. There was a gain of 7.2% with NP-12 compared to normal engine. That increase in brake thermal efficiency might be due to enhanced mixing rate carried by the turbulence in the combustion chamber [10]. D. N. Basavarajappa et al. (2014) found that, BTE for diesel was higher than UOME (uppage oil methyl ester) operation over the entire load range. That was mainly due to its lower calorific value, lower volatility
and higher viscosity. The improper mixture formation leads to incomplete combustion and hence lower BTE was 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 was found that the TCC had 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 [12].Nimesh A. Patel et al. (2015) found that, the BTE of the engine increased with increasing loads. The thermal efficiency was highest for engine with Bpinst. The brake thermal efficiency of piston B was increased by 1.93% when compare to the base line engine the increased and decreased of pressure give negative effect on brake thermal efficiency [13].Banapurmath N. R. et al. (2015) found that, BTE for diesel fuel mode of operation was higher than both biodiesels of MahOME (mahua oil methyl ester) and NOME (neem oil methyl ester) operation over the entire load range. That was mainly due to lower calorific value of the biodiesels and lower volatility as well. The study with different combustion chamber shapes showed that for biodiesel 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. The TCC had 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 [14].

### 3.1.2. Brake Specific Fuel Consumption (BSFC)

S. Jaichandar et al. (2012) found that, BSFC for 20% POME was slightly higher than that of diesel in standard engine, may be due to lower calorific value of POME than that of conventional diesel. BSFC for SCC and HCC were higher than that for TCC, may be due to poor air fuel mixing, which leads to poor combustion and thus increases BSFC for SCC and HCC [2].Chandrashekarapua Ramachandraiah Rajashekar et al. (2012) found that, at higher loads, the SFC had decreased slightly, compared with standard piston engine. That may be due to better combustion owing to induced turbulence due to squish motion generated with the modified piston. The specific fuel consumption at 80% load for CR=17.5, IP=175, 200 and 225 bar was 0.406, 0.351 and 0.382 kg/kW-hr respectively. The specific fuel consumption was reduced with 200 bar injection pressure compared to 175 bar and 225 bar injection pressures, due to more area coverage of spray formed in the combustion chamber and utilization of air effectively. At 225 bar injection pressure, the SFC was increased due to too rich mixture [3].M. Ravi et al. (2013) found that, the BSFC for ATME (animal tallow methyl ester) hemispherical piston had lower value of 0.245 kg/kW-hr and for PSME (palm stearin methyl ester) flat piston was higher by 49.81% compared to diesel. That phenomenon was due to higher viscosity of fuel [4].M. N. Channappagoudra et al. (2013) found that, BSFC for 6-grooved piston was lower compared to other two configurations of piston. Due to poor combustion there was higher specific fuel consumption for three grooved piston as compared to 6-grooved piston and normal piston [5].S. Jaichandar et al. (2014) found that, BSFC for POME20 was slightly higher than that of diesel at all loads for all combustion chambers. That may be attributed to the lower calorific value of POME than that of conventional diesel. Similarly the BSFC for open combustion chambers such as SCC, HCC and TCC was higher than that for re-entrant combustion chambers such as SRCC and TRCC. The higher specific fuel consumption for open combustion chambers may be attributed to poor air fuel mixing, which leads to poor combustion and thus increases the specific fuel consumption. Compared to HCC, the BSFC for the modified re-entrant combustion chambers viz. SRCC and TRCC was lower by 7% and 13.9% respectively at full load operation with POME20 [8].Vaibhav Bhatt et al. (2014) found that, the BSFC for normal engine at full load was 0.243 kg/kW-hr. The engine with NP-4, NP-8 and NP-12 give brake specific fuel consumption of 0.240 kg/kW-hr, 0.234 kg/kW-hr and 0.226 kg/kW-hr. NP-12 configuration had the lowest brake specific fuel consumption 6.9% when compared to normal engine. That was because of the complete combustion of charge in the combustion chamber by liberating maximum energy due to the inducedness of enhanced air swirl in the combustion chamber [10].Nimesh A. Patel et al. (2015) found that, fuel consumption decreases with increase in load. One possible reason for that reduction was that the brake power increases in higher percentage compare to fuel consumption. Fuel consumption decreases about 5.98% compare to baseline condition [13].

### 3.1.3. Brake Specific Energy Consumption (BSEC)

C. V. Subba Reddy et al. (2012) found that, the BSEC had decreased with increase in load. The BSEC of BLE (base line piston of diesel engine) with tangential grooved piston was lower for 20BD at all load operations. The BSECs for 70% load operation of BLE without grooves and with tangential grooves (TGP-2) were 13.6 and 12.7 MJ/kW-hr respectively. At 70% load operation of BLE with configuration 4 (BLE-4), the reduction in BSEC may be due to improved combustion with better evaporation and mixing of fuel with air was improved by 7.02% compared BLE without tangential grooves. The 20% blended fuel (20BD) showed low BSEC compared other blended fuels of 0BD, 10BD, 20BD, 30BD and 100BD [1].Hariram V. et al. (2014) found that, at low loads the BSEC for HBP (hemispherical bowl piston) was found to be 40 MJ/kW-hr with diesel as fuel whereas B100 showed 38.12 MJ/kW-hr. At part load condition the BSEC for all type of piston and fuel showed a decreasing trend with STRP
(shallow toroidal re-entrant piston) consuming 12.47 MJ/KW-hr which may be due to better air fuel mixing and spray atomization. At full load condition HBP showed an abnormal increase with diesel and B100 which was a result of incomplete combustion. At the same load the STP and STRP with diesel and B100 blend showed a decreasing trend in comparison with HBP geometry. The BSEC of STRP with B100 blend showed 15.08 MJ/KW-hr of energy consumption [9].

3.1.4. Exhaust Gas Temperature (EGT)

C. V. Subba Reddy et al. (2012) found that, the turbulence produced in the combustion chamber depends on swirl motion of air during suction stroke and at the end of compression stroke which increases with number of tangential grooves on the piston crown. From the results, it was concluded that the exhaust gas temperature of base line engine [BLE] was lower than all other base line engines with different configurations over a wide range of operation. The base line engine with tangential grooves [BLE-4] showed maximum exhaust gas temperature due to effective combustion of fuel rated loads. The temperature at 70% load for BLE-4 and BLE-5 with 20% blended fuel (20BD) was 9.2% and 6.7% more than base line engine without grooves (BLE-1) [1].M. Ravi et al. (2013) found that, EGT for PSME hemispherical piston at a pressure of 200 bar had higher value compared to diesel and for ATME hemispherical piston at pressure of 240 bar had lower value compared to diesel therefore it will have higher performance due to reduction in exhaust heat loss [4].Vaibhav Bhatt et al. (2014) found that, the exhaust gas temperature for normal engine at full load was 365°C. The engine with NP-4, NP-8 and NP-12 give exhaust gas temperature of 351°C, 335°C and 327°C. With NP-12 configuration had the lowest exhaust gas temperature 10.4% when compared to normal engine. Lower exhaust gas temperature for NP-12 can be attributed due to low operating temperature in the combustion chamber resulted by the swirl created in the combustion chamber [10].Nimesh A. Patel et al. (2015) found that, exhaust gas temperature increased with engine load for both the combustion chambers used. The exhaust gas temperature of the modify piston B was increase by 4.52% compare to base condition of engine at full load operation. That may be due to more complete combustion as a result of better air fuel mixing [13].

3.1.5. Volumetric Efficiency

C. V. Subba Reddy et al. (2012) found that, the temperature generated in the combustion chamber depends on turbulence created by the tangential grooves on the piston crown. For base line engine with configuration 4 [BLE-4], the drop in volumetric efficiency was more and was about 1.59% as compared to BLE for 70% load operation. The volumetric efficiencies of BLE-2 and BLE-3 lie in between BLE and BLE-5. The fall in volumetric efficiency had an undesirable effect on power output. So the power output of DI diesel engine with configuration 4 (BLE-4) can be compensated by turbo charging [1].

3.1.6. Mechanical Efficiency

Vaibhav Bhatt et al. (2014) found that, as the power increases no doubt the mechanical efficiency also increases. NP-12 configuration had the highest mechanical efficiency 14.8% when compared to normal engine. That may be because of the complete combustion of charge in the combustion chamber by liberating maximum energy due to the inducement of enhanced air swirl in the combustion chamber [10].

3.2. Combustion Parameters

3.2.1. Peak Pressure Rise

C. V. Subba Reddy et al. (2012) found that, the maximum rate of pressure rise was with BLE-4 and minimum for base line engine [BLE-1]. Due to high turbulence, complete combustion occurs and the combustion efficiency increases, consequently higher rate of pressure increases. At 70% load operation the rate of pressure rise was 10.4% and 9.76% for BLE-4 and BLE-5 compared to BLE-1 [1].M. V. S. Murali Krishna et al. (2013) found that, peak pressure for normal biodiesel was slightly higher than that of diesel fuel; even though biodiesel was having lower value of lower calorific value. Biodiesel advanced the peak pressure position as compared to fossil diesel because of its higher bulk modulus and cetane number. That shift was mainly due to advancement of injection due to higher density and earlier combustion due to shorter ignition delay caused by higher cetane number of biodiesel. When, a high density (or high bulk modulus) fuel was injected, the pressure wave travels faster from pump end to nozzle end, through a high pressure in-line tube. That causes early lift of needle in the nozzle, causing advanced injection. Hence, the combustion takes place very close to TDC (lower value of time of occurrence of peak pressure) and the peak pressure slightly high to existence of smaller cylinder volume near TDC. However, the peak pressures of preheated methyl ester were less than that of normal biodiesel. When the engine was running on preheated biodiesel the fuel injection was slightly delayed, due to decrease in bulk modulus of biodiesel with the increase in fuel temperature. The reasons for lower peak pressures of preheated biodiesel was also attributed to earlier combustion caused by short ignition delay (due to faster evaporation of the fuel) at their preheated temperatures. Peak pressures increased with the increase of injector opening pressure and with the advancing of the injection timing with the test fuels. Peak pressure increased as injector opening increased. That may be due to smaller sauter mean diameter shorter breakup length, better dispersion, and better spray and atomization characteristics. That improves combustion rate in the premixed combustion phase. Maximum rate of pressure rise (MRPR) was highest for normal diesel followed by the biodiesel. With biodiesel, as injector opening pressure increased, spray characteristic improved and in turn burned fuel increased again and in turn combustion rate increased in the premixed combustion phase. When the engine was operated under the full load condition, the mechanical loading was at the maximum level. The differences in maximum rate of pressure rise, the peak cylinder pressure and the
occurrence of peak cylinder pressure during the maximum mechanical loading may cause performance loses. Both peak cylinder pressure and maximum rate of pressure rise of biodiesel was lower and occurrence of peak cylinder pressure slightly deviated away when compared to normal diesel. However it decreased as injector opening pressure of biodiesel increased. The value of time of occurrence of peak pressure (TOPP) decreased with the advancing of the injection timing and with increase of injector opening pressure at different operating conditions of the test fuels. Preheating of the biodiesel showed lower TOPP, compared with biodiesel at normal temperature. That once again confirmed by observing the lower TOPP, the performance of the engine improved with the preheated biodiesel compared with the normal biodiesel. That trend of increase of maximum rate of pressure rise indicated improved and faster energy substitution and utilization by biodiesel in engine, which could replace 100% diesel fuel. That too, all these combustion characters were within the limits hence the biodiesel can be effectively substituted for diesel fuel [6].

D. N. Basavarajappa et al. (2014) found that, 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. Higher in-cylinder pressure for biodiesel operation with TCC compared to other combustion chamber shapes was found. It could be due to the combined effect of longer ignition delay, lower adiabatic flame temperature and slow burning nature of the biodiesel operation. That 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 [12].

3.2.2. In-cylinder Pressure

S. Jaichandar et al. (2012) found that, pressure variations of three types of open combustion chambers follow similar pattern of pressure of pressure rise as that of diesel at all loads. When compared to diesel oil, pressure data of 20% POME were lower for all types of combustion chambers, may be due to variations of viscosity and heating value with POME % in fuel. Cylinder pressure trend of TCC with 20% POME was found closer to that of standard engine operated with diesel fuel and well above for SCC and HCC with 20% POME, may be attributed to better combustion due to better air fuel mixing [2]. M. Ravi et al. (2013) found that, the maximum rise of cylinder pressure during combustion near to TDC i.e. 325°-450° crank angle. ATME hemispherical piston at 220 bar pressure was having higher in-cylinder pressure compared to all other fuels at different pressures. The main cause for higher peak in-cylinder pressure in the CI engine running with biodiesel was because of the advanced combustion process initiated by easy flow ability of biodiesel due to physical properties of biodiesel [4]. Lava K. R. et al. (2014) found that, the standard piston produces higher cylinder pressure compared to threaded piston. That trend may be attributed due larger delay period with the standard piston in which more amount of fuel was accumulated in the combustion chamber [11]. Banapurmath N. R. et al. (2015) found that, biodiesel with TCC resulted in higher peak pressure. The pressure for MhOME biodiesel operation with TCC was higher compared to NOME biodiesel tested. It could be due to the combined effect of longer ignition delay, lower adiabatic flame temperature and slow burning nature of the biodiesel operation. That 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 [14].

3.2.3. Heat Release Rate (HRR)

Lava K. R. et al. (2014) found that, the premixed combustion region was rather higher for threaded piston indicating that higher of delay period due greater mixing of fuel with air because of swirl generation [11]. D. N. Basavarajappa et al. (2014) found that, biodiesel operation for HCC, CCC and TrCC resulted into lower heat release rate compared to the operation with TCC. That was due to the result of higher second peak obtained with HCC, CCC and TrCC in the diffusion combustion phase compared to the TCC operation [12]. Banapurmath N. R. et al. (2015) found that, biodiesel operation for TCC resulted into higher heat release rate compared to the operation with other combustion chambers. Combustion chamber being common NOME had higher second peak in the diffusion combustion phase compared to the MhOME operation with TCC operation [14].

3.2.4. Mass Fraction Burnt

Lava K. R. et al. (2014) found that, the mass fraction burnt was high with threaded piston. That could explain higher NOx, lower CO and HC emissions with threaded piston [11].

3.2.5. Ignition Delay Period

C. V. Subba Reddy et al. (2012) found that, the general phenomenon was that with increasing the load the ignition delays were reduced. The ignition delay of diesel engine of piston crown with different number of tangential grooves was lowered as compared with base line engine (BLE-1) due to higher swirl flow in the cavity of piston. The effect of that was to reduce the time lag for initiating combustion, bringing down delay periods. The diesel engine with TGP-2 (BLE-4) showed the lowest ignition delay among all the configurations tested and base line engine without grooves (BLE-1) showed marginally higher ignition delay. The reduction in the ignition delay of BLE-4 and BLE-5 are 5.2% and 4.95% for 70% load operation compared to BLE-1. The ignition delay of BLE-2 and BLE-3 lies between BLE-1
and BLE-5 [1]. D. N. Basavarajappa et al. (2014) found that, the ignition delay was calculated based on the static injection timing. 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. That 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 found 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 [12]. Banapurmath N. R. et al. (2015) found that, ignition delay decreased with an increase in brake power for almost all combustion chamber shapes. With an increase in brake power, the amount of fuel being burnt inside the cylinder gets increased and subsequently the temperature of in-cylinder gases gets increased. That 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. However, lower ignition delays were found 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 [14].

3.2.6. Combustion Duration

D. N. Basavarajappa et al. (2014) found that, the combustion duration increases with increase in the power output with all combustion chamber shapes adopted. That was due to the amount of fuel being burnt inside the cylinder gets increased. Combustion chamber being same, higher combustion duration was found 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 chamber shapes tested. That 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 thermal efficiency. However, biodiesel operation with TCC showed improvement in heat release rate compared to other combustion chamber shapes [14].

3.3. Emission Parameters

3.3.1. CO Emissions

C. V. Subba Reddy et al. (2012) found that, in diesel engine, swirl motion of air played a major role in the increase of fuel evaporation and air-fuel mixing, thereby resulting in reduction of the combustion time and increasing of the combustion efficiency. With the higher turbulence in the combustion chamber, the oxidation of carbon monoxide was improved which reduces the CO emissions. The lowest carbon monoxide emission was with BLE-4 configuration compared to BLE-1 configuration and about 9.75% by volume at 70% load operation. For the other configurations the values varies in between these two extremes [1]. S. Jaichandar et al. (2012) found that, at low and medium loads CO emissions for different combustion chambers were not much different from those of standard diesel. At full load, CO emissions of the blend decreased significantly when compared with those of standard diesel. CO emissions decrease with TCC than SCC and HCC. Higher air movement in TCC and presence of oxygen in POME, lead to better combustion of fuel, resulting in decrease in CO emissions. Increase in the proportions of oxygen in POME promotes further oxidation of CO during engine exhaust process [2]. Chandrashekharapu Ramachandraiah Rajashekhar et al. (2012) found that, the exhaust emissions of carbon monoxide was lower for J20 blend with modified piston at CR=17.5, IP=200 bar, compared to standard piston engine. Lower concentration of CO in exhaust was a clear indication of complete combustion of fuel. The CO levels with standard piston are high due to combustion inefficiencies. The CO emission at 80% load for CR=17.5, IP=175, 200 and 225 bar was 0.33%, 0.286% and 0.32% The CO emission was decreased at 200 bar injection pressure compared to 175 and 225 bar injection pressure. With increase in injection pressure from 175 to 200 bar the CO emission reduces due to effective combustion of air fuel mixture and reduced viscosity [3]. M. Ravi et al. (2013) found that, CO emissions for ATME flat piston at pressure of 200 bar had lower value and for ATME flat piston at a pressure of 240 bar was higher by 85.99% compared to diesel. That was due to incomplete combustion of fuel [4]. M. N. Channappagoudra et al. (2013) found that, generally CI engines operate with lean mixtures and hence the CO emission would be low. With higher turbulence and temperature in combustion chamber the oxidation of carbon monoxide was improved and which reduces CO emissions. With 6-grooved piston configuration emission of CO was low as compared to other two piston configurations [5]. Ranganatha Swamy L. et al (2014)
found that, CO emission levels were found with the configurations of combustion chambers used. Incomplete combustion and associated lower BTE with respective configuration was responsible for that trend. However, TCC resulted in lower CO emission levels compared to other combustion chamber shapes. Marginally comparable performance was found with SDCC as well. 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 [7].

S. Jaichandar et al. (2014) found that, carbon monoxide emissions were greatly reduced with the addition of POME to diesel. That showed CO emissions were greatly reduced with the addition of POME to diesel. In addition it was found that CO emissions further decreased with the re-entrant combustion chambers than with the open combustion chambers. Higher air movement in TRCC lead to better combustion of fuel resulting in the decrease in CO emissions. Secondly increase in the proportion of oxygen in POME promotes further oxidation of CO during the engine exhaust process. Reduction in CO emissions was a strong advantage in favor of POME. There was a reduction of 44.5% CO emissions for the TRCC compared to baseline HCC, when tests were carried out with POME20 [8]. Hariram V. et al. (2014) found that, CO emissions were mainly formed due to incomplete combustion and improper oxidation of carbon atoms to carbon dioxide. The STRP exhibited 0.09 to 0.04% of CO emissions at part load conditions with diesel and RBBD B100. At similar load the HB and STRP showed 0.11 to 0.08% and 0.08 to 0.03% of CO for straight diesel and RBBD respectively. The STRP emits lesser quantity of CO for both diesel and biodiesel blends which may be due to better oxidative stability and enhanced swirl squish motion in the combustion chamber. Generally HB and showed a negative increase of 5 to 7% on comparison with STP [9]. Vaibhav Bhatt et al. (2014) found that, the CO emissions for NP-8 and NP-12 were 0.78% and 0.74% by volume respectively, whereas for normal engine it was 1% by volume. The CO emissions were lower by 10% for NP-12 when compared to normal engine at full load. Generally, CI engines operate with lean mixtures and hence the CO emission would be low. With the higher turbulence and temperatures in the combustion chamber, the oxidation of carbon monoxide was improved and which reduces the CO emissions [10]. Lava K. R. et al. (2014) found that, generally CI engines operate with lean mixtures, hence the CO emission would be low. With increase in turbulence due to swirl motion in threaded piston the oxidation of carbon monoxide was improved, which results in reduction of CO emissions. The CO levels with standard piston were high at full load conditions due to combustion inefficiencies [11].

D. N. Basavarajappa et al. (2014) found that, CO emission levels were higher for and UOME compared to diesel operation. Incomplete combustion and lower BTE of UOME was responsible for that 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 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 CO and hence reduced emission levels. However, other combustion chambers may marginally contribute to the proper mixing of fuel combinations [12]. Nimesh A. Patel et al. (2015) found that, CO emission decreases with increase in load for any condition and also different values at even same load. The CO emission was 0.08% at baseline and 0.07% at modified piston condition [13]. Banapurmath N. R. et al. (2015) found that, CO emission levels were higher for MhOME, and NOME compared to diesel operation. Incomplete combustion of the MhOME and NOME biodiesels was responsible for that trend. It could be due to their lower calorific value, lower adiabatic flame temperature and higher viscosity and lower mean effective pressures. However, TCC resulted in lower CO emission levels compared to other combustion chamber shapes. It could be due to higher turbulence and comparatively higher temperature prevailing in the combustion chamber that resulted into minimum heat losses and better oxidation of CO and hence reduced emission levels. However, other combustion chambers may not contribute to the proper mixing fuel combinations [14].

3.3.2. HC Emissions

C. V. Subba Reddy et al. (2012) found that, the main sources of hydrocarbon emissions in diesel engine were wall quenching, lean mixing and the burning of lubricating oil. The hydrocarbon formation in the higher swirl motion engine was less compared to base line engine and that higher swirl motion in the combustion chamber was produced by the tangential grooves on the piston crown. At 70% load operation the BLE-4 gives a maximum reduction of hydrocarbon emission level was found and was about 10.5% compared to BLE-1. With BLE-2, BLE-3 and BLE-5, the reduction in hydrocarbon levels were about 3.2%, 5.86% and 7.2% compared to BLE-1 [1]. S. Jaichandar et al. (2012) found that, UHBC emissions were reduced on all loads for TCC and HCC, may be due to better burning of POME with better mixture formation of air and POME due to improved swirl motion of air [2]. Chandrashekarapu Ramachandraiah Rajashekar et al. (2012) found that, a substantial reduction in HC emissions due better mixing and complete combustion of fuel. The UBHC at 80%load for CR=17.5, IP=175, 200 and 225 bar was 12, 8 and 11 ppm. The UBHC emission was reduced at 200 bar injection pressure compared to 175 bar and 225 bar injection pressures. As the IP increased from 175 bar to 200 bar unburnt hydrocarbon emission reduced due to the fine spray formed during injection and improved atomization. With further increase in injection pressure
an increase in the unburnt hydrocarbon emission was found due to finer fuel spray which reduces momentum of droplets which might probably result in incomplete combustion [3]. M. Ravi et al. (2013) found that, HC increases at 180 bar for diesel, PSME, ATME for both hemispherical and flat pistons and at 200, 220, 240 bar pressure the HC decreases. So out of these pressures at 220 bar PSME hemisphericalpiston have fewer emissions. PSME hemispherical piston at 180 bar was lower by 7.67% compared to diesel. That was dueto better vaporization and proper atomization [4]. M. N. Channappagoudra et al. (2013) found that, unburnt hydrocarbon emission was the result of incomplete combustion but there was decrease in hydro-carbon emission level with 6-grooved piston as compared to other two piston configurations and normal piston it was attributed to complete combustion because of proper mixing of air and fuel in 6-grooved piston. But there HC emissions were greater in 3-groove and normal piston which was due to improper air-fuel mixing rate [5]. Ranganatha Swamy L. et al (2014) found that, HC emission levels were found with the configurations of combustion chambers used. Incomplete combustion and associated lower BTE with respective configuration was responsible for that trend. However, TCC resulted in lower HC emission levels compared to other combustion chamber shapes. Marginally comparable performance was found with SDCC as well. 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 hence reduced emission levels. However, other combustion chambers may marginally contribute to the proper mixing of fuel combinations [7]. S. Jaichandar et al. (2014) found that, UBHC emissions were reduced over the entire range of loads for all types of combustion chambers fuelled with POME20 when compared to diesel operation. Moreover, the UBHC emissions for re-entrant combustion chambers were compared with the open combustion chambers. UBHC emissions were reduced over the entire range of loads for re-entrant combustion chambers such as TRCC and SRCC. That may be due to better combustion of POME caused by better mixture formation of air and POME due to improved swirl motion of air. There was a reduction of 20.7% UBHC emissions for the TRCC compared to baseline HCC, when tests were carried out with POME20 [8]. Hariram V. et al (2014) found that, the emissions of UBHC showed an increasing trend irrespective of piston geometries with STRP B100 as 3 ppm and HBP B100 as 11 ppm at low loads the UBHC emission was found to be varying between 3 ppm to 17ppm across all piston geometries. At part load and high load HBP exhibit higher emissions of HC which may be due to incomplete combustion and effect of cylinder wall quenching. STRP with diesel and B100 RBBD were found to emit 17 ppm and 11 ppm of UBHC during part load condition 25 ppm and 19 ppm during full load condition which was 13 to 15% lower than HBP. Vaibhav Bhatt et al. (2014) found that, the HC emissions for NP-4, NP-8 and NP-12 were 94 ppm, 93 ppm and 92 ppm respectively, whereas for normal engine it was 95 ppm. The HC emissions were lower by 3.1% for NP-12 when compared to normal engine at full load. The un-burnt hydrocarbon emission was the direct result of incomplete combustion. It was apparent that the hydrocarbon emission was decreasing with the increase in the turbulence, which results in complete combustion [10]. Lava K. R. et al. (2014) found that, the HC emission was the direct result of incomplete combustion. The HC emission was decreasing with the increase in turbulence in threaded piston, which results in complete combustion of fuel [11]. D. N. Basavarajappa et al. (2014) found that, HC emission levels were higher for and UOME compared to diesel operation. Incomplete combustion and lower BTE of UOME was responsible for that 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 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 hence reduced emission levels. However, other combustion chambers may marginally contribute to the proper mixing of fuel combinations [12]. Nimesh A. Patel et al. (2015) found that, the HC emission of modifying piston-B was reduced by 2.12% at same injection pressure when it was compare to baseline engine. It was apparent that the hydrocarbon emission was decreasing with the increase in the turbulence, which results incomplete combustion [13]. Banapurmath N. R. et al. (2015) found that, HC emission levels were higher for MhOME, and NOME compared to diesel operation. Incomplete combustion of the MhOME and NOME biodiesels was responsible for that trend. It could be due to their lower calorific value, lower adiabatic flame temperature and higher viscosity and lower mean effective pressures. However, TCC resulted in lower HC emission levels compared to other combustion chamber shapes. It could be due to higher turbulence and comparatively higher temperature prevailing in the combustion chamber that resulted into minimum heat losses and better oxidation of HC and hence reduced both emission levels. However, other combustion chambers may not contribute to the proper mixing fuel combinations [14].

3.3.3. NOx Emissions

S. Jaichandar et al. (2012) found that, NOx emissions were higher for TCC than the base engine, may be due to higher combustion temperatures arising from improved combustion due to better mixture formation and availability of oxygen in POME. Another reason for increased NOx emissions may be due to larger part of combustion was compared before top dead center for 20% POME compared to diesel due to their lower ignition delay. So, it was highly possible that higher peak cycle temperatures were reached for 20% POME compared to diesel. NOx can be controlled by adopting
exhaust gas recirculation and by employing suitable catalytic converters. At full load with 20% POME for TCC, NOx emission was 738 ppm compared to 712 ppm for standard HCC. There was an increase of NOx emissions (3.5%) for TCC [2]. M. Ravi et al. (2013) found that, PSME flat piston for 220 bar had lower emissions compared to other fuels at different pressures and for PSME hemispherical piston at rated load was higher by 11.59% compared to diesel. That was attributable to higher peak combustion temperature in the combustion chamber influences that factor. NOx increases for biodiesel compared to diesel that was due to more oxygen content in the combustion chamber [4]. M. V. S. Murali Krishna et al. (2013) found that, NOx were the precursor pollutants which can combine to form photochemical smog. Those irritate the eyes and throat, reduces the ability of blood to carry oxygen to the brain and can cause headaches, and pass deep into the lungs causing respiratory problems for the human beings. Long-term exposure had been linked with leukemia. Therefore, the major challenge for the existing and future diesel engines was meeting the very tough emission targets at affordable cost, while improving fuel economy. Temperature and availability of oxygen were two favorable conditions to form NOx levels. At peak load, NOx levels increased with test fuels at recommended injection timing due to higher peak pressures, temperatures as larger regions of gas burned at close-to-stoichiometric ratios. The rice bran oil based biodiesel having long carbon chain (C20-C32) recorded more NOx than that of fossil diesel having both medium (C8-C14) as well as long chain (C16-C28). The increase in NOx emission might be an inherent characteristic of biodiesel due to the presence of 54.9% of mono-unsaturated fatty acids (MUFA) and 18% of poly-unsaturated fatty acids (PUFA). That means, the long chain unsaturated fatty acids (MUFA and PUFA) such as oleic C18:1 and linoleic C18:2 fatty acids were mainly responsible for higher levels of NOx emission. Another reason for higher NOx levels was the oxygen (10%) present in the methyl ester. The presence of oxygen in normal biodiesel leads to improvement in oxidation of the nitrogen available during combustion. That will raise the combustion bulk temperature responsible for thermal NOx formation. Oxygen and nitrogen content of biodiesel can cause an increase in NOx emissions. The production of higher NOx with biodiesel fueling was also attributable to an inadvertent advance of fuel injection timing due to higher bulk modulus of compressibility, with the in-line fuel injection system. Residence time and availability of oxygen had increased, when the injection timing was advanced with test fuels, which caused higher NOx levels. NOx slightly increased with test fuels as injector opening pressure increased. The increase in peak brake thermal efficiency was proportional to increase in injector opening pressure. Normally, improved combustion causes higher peak brake thermal efficiency due to higher combustion chamber pressure and temperature and leads to higher NOx formation. That was an evident proof of enhanced spray characteristics, thus improving fuel air mixture preparation and evaporation process. NOx levels decreased with preheating of the biodiesel. The fuel spray properties may be altered due to differences in viscosity and surface tension. The spray properties affected may include droplet size, droplet momentum, and degree of mixing, penetration and evaporation. The change in any of those properties may lead to different relative duration of premixed and diffusive combustion regimes. Since the two burning processes (premixed and diffused) have different emission formation characteristics, the change in spray properties due to preheating of the biodiesel were lead to reduction in NOx formation. As fuel temperature increased, there was an improvement in the ignition quality, which will cause shortening of ignition delay. A short ignition delay period lowers the peak combustion temperature which suppresses NOx formation. Lower levels of NOx was also attributed to retarded injection, improved evaporation, and well mixing of preheated biodiesel due to their viscosity at preheated temperatures. Biodiesel had higher value of NOx emissions followed by diesel. That was because of inherent nature of biodiesel as it had oxygen molecule in its composition [6]. Ranganatha Swamy L. et al. (2014) found that, higher NOx emission levels were with diesel for the configurations adopted over the entire load range. Higher heat release rates during premixed combustion phase found with TCC and SDCC led to higher NOx formations compared to other combustion chamber shapes used. That could be due to slightly better combustion prevailing in combustion chamber due to more homogeneous mixing and larger part of combustion occurring just before top dead center [7]. S. Jaichandar et al. (2014) found that, the NOx emissions were higher for re-entrant combustion chambers i.e. for TRCC and SRCC than the open combustion chambers such as SCC, HCC and TCC. The reason for the increase in NOx could be the possibility of higher combustion temperatures arising from improved combustion due to better mixture formation and availability of oxygen in POME. However NOx can be controlled by adopting exhaust gas recirculation and by employing suitable catalytic converters. At full load, for the TRCC using POME20, the level of NOx emission was 784 ppm compared to 712 ppm for standard engine having HCC. There was an increase of about 9.2% of NOx emissions for TRCC compared to the baseline engine when fuelled with POME20 and 20.98% with standard diesel [8]. Hariram V. et al. (2014) found that, the STP exhibited 154 ppm, 242 ppm, 428 ppm at low load, part load and full load respectively which was a decrease in 3 to 4% with RBBD B100 as the fuel. HBP exhibited 86 ppm at part load and 274 ppm at full load for RBBD blend and 118 ppm at part load and 304 ppm for straight diesel. That piston was subjected to 20% more than the full load condition which showed a sharp increase in the NOx emission up to 450 ppm which was 2% to 4% lesser than STP. The STRP was found optimum at low load, part load and full load conditions where the NOx emissions were found to be 178 ppm and
210 ppm, 272 ppm and 304 ppm, 453 ppm and 485 ppm respectively for RBBD blend and straight diesel which may be due to lack of oxygen for oxidation, atomization of fuel and fuel viscosity at higher loads. Generally HBP was not suitable at part load and full load operation on comparison with STP and STRP [9]. Vaibhav Bhatt et al. (2014) found that, NOx emission increases with increase in turbulence in the cylinder because of high temperature. The NOx emissions for NP-4, NP-8 and NP-12 were 563 ppm, 559 ppm and 556 ppm respectively, whereas for normal engine it was 565 ppm. The NOx emissions were lower of 1.5 % for NP-12 when compared to normal engine at full load. That may be due to decrease in combustion duration; the residence time for the gas in the combustion chamber was too short to form the NOx at normal level [10]. Lava K. R. et al. (2014) found that, the NOx emissions were slightly increased for S20 blend with threaded piston in comparison with the standard piston. That may be due to the higher temperature in the combustion chamber because of complete combustion of fuel with swirl generated threaded piston [11]. D. N. Basavarajappa et al. (2014) found that, the NOx emission levels were found to be higher for diesel compared to biodiesel over the entire load range. Higher heat release rates during premixed combustion phase found with diesel compared to biodiesel operation was responsible for that trend. Slightly higher NOx were resulted with TCC compared to other combustion chamber shapes tested. That 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 was also responsible for that trend. Therefore it was resulted in higher peak cycle temperature [12]. Nimesh A. Patel et al. (2015) found that, emission of NOx of modify piston-B was increased by 4.22% at 185 bar injection pressure when compare to baseline engine at full load condition. Thereason for the increase in NOx may be attributed to higher combustion temperatures arising from improved combustion due to better mixture formation in piston-B [13]. Banapurmath N. R. et al. (2015) found that, the NOx emission levels were found to be higher for diesel fuel operation compared to biodiesel over the entire load range. That may because of higher heat release rate during premixed combustion phase found with diesel compared to biodiesel operation. Slightly higher NOx resulted with TCC compared to other combustion chamber tested. That 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 was also responsible for that trend. Therefore it was resulted in higher peak cycle temperature [14].

3.3.4. Smoke Opacity

C. V. Subba Reddy et al. (2012) found that, the higher turbulence in the combustion chamber results better combustion and oxidation of the soot particles which further reduces the smoke emissions. Due to the complete combustion of fuel with swirl motion of excess air the smoke emissions were marginal. At 70% load operation, the smoke emissions of BLE-4 were reduced by about 26.9% compared to base line engine [BLE-1]. The reduction in the smoke emissions of BLE-5 was 26.54% and also diesel engine with configuration-3 and 2 reduce emissions by 14.97% and 8.2% compared to BLE-1 [1]. S. Jaichandar et al. (2012) found that, at all loads, smoke emissions for the blend decreased significantly than those of standard diesel, may be due to presence of oxygen in biodiesel blend. Oxygenated 20% POME fuel leads to an improvement in diffusive combustion. Smoke emissions were found lower for TCC than HCC and SCC, may be due to more complete combustion due to better air fuel mixing and presence of oxygen in POME [2]. M. V. S. Murali Krishna et al. (2013) found that, drastic increase of smoke levels at peak load operation with biodiesel operation was found compared with pure diesel operation. That was due to the higher value of ratio of C/H (C= number of carbon atoms and H= number of hydrogen atoms in fuel composition (C10H14N2), higher the value of this ratio means, number of carbon atoms were higher leading to produce more carbon dioxide and more carbon monoxide and hence higher smoke levels) of crude vegetable oil (0.7) when compared with pure diesel (0.45). The increase of smoke levels was also due to decrease of air-fuel ratios and volumetric efficiency with biodiesel compared with pure diesel operation. Smoke levels were related to the density of the fuel. Since biodiesel had higher density compared to diesel fuel, smoke levels were higher with biodiesel. Smoke levels decreased at the respective optimum injection timing with test fuels. The inherent oxygen of biodiesel might have provided some useful interactions between air and fuel, particularly in the fuel-rich region. Certainly, it was evident proof of the oxygen content of biodiesels enhanced the oxidation of hydrocarbon reactions thus reducing smoke levels. Smoke levels were lower with pure diesel operation followed by biodiesel. That was due to lower value of C/H ratio, density, air fuel ratio of the diesel fuel. A decrease in smoke levels with increase of injector opening pressure, with different operating conditions of the biodiesel. That was due to improvement in the fuel spray characteristics at higher Injector opening pressure and increase of air entrainment, at the advanced injection timings, causing lower smoke levels. Even though viscosity of biodiesel was higher than diesel, high injector opening pressure improves spray characteristics, hence leading to a shorter physical delay period. The improved spray also leads to better mixing of fuel and air resulting in turn in fast combustion. That will enhance the performance. Preheating of the biodiesel reduced smoke levels, when compared with normal temperature of the biodiesel. That was due to i) the reduction of density of the biodiesel, as density was directly related to smoke levels ii) the reduction of the diffusion combustion proportion with the preheated biodiesel iii) the reduction of the viscosity of the biodiesel, with which the fuel spray does not impinge on the combustion chamber walls of lower
temperatures rather than directed into the combustion chamber. By the esterification process, the viscosity of the biodiesel was brought down many times. Since there was drop in the viscosity, naturally the density of the esterified oil was also dropped at the room temperature. Volatility of the biodiesel also increased with the esterification process. Smoke levels were lower with diesel operation followed by biodiesel. That was due to the nature of the composition of the diesel fuel. Diesel fuel had lower value of C/H, lower viscous and lower theoretical air fuel ratio [6]. Ranganatha Swamy L. et al (2014) found that, diesel being common fuel used type of combustion chamber configuration affects the smoke opacity over the entire load range. That may be attributed to improper fuel-air mixing occurring inside the individual combustion chamber shapes adopted. 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 [7]. S. Jaichandar et al. (2014) found that, at all loads and for all combustion chambers, smoke emissions for the blend decreased significantly when compared with those of diesel. The reduction in smoke emission may be due to the presence of oxygen in diesel blend. Further, it was also found that the smoke emissions were lower for re-entrant combustion chambers than open combustion chambers. That was due to more complete combustion due to better air fuel mixing and the presence of oxygen in the POME. There was a reduction of 28.2% smoke emissions for the TRCC compared to baseline HCC, when tests were carried out with POME20 [8]. D. N. Basavarajappa et al. (2014) found that, smoke opacity for diesel was lower than UOME over the entire load range. That 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 [12]. Banapurmath N. R. et al. (2015) found that, the smoke opacity for diesel fuel operation was lowerthan MhOME and NOME biodiesels 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 biodiesels 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 reduction the smoke emission levels [14].

**3.3.5. CO\textsubscript{2} Emissions**

M. Ravi et al. (2013) found that, PSME flat piston for 220 bar pressure had lower CO\textsubscript{2} emissions compared to other fuels and for PSME hemispherical piston at a pressure of 220 bar was higher by 38.21% compared to diesel. That may because of excess supply of oxygen was the influencing criterion.

**IV. CONCLUSIONS**

This review paper gives an insight into the importance and effects of good combustion chamber design. There is a strong necessity of research and innovation in combustion chamber design as with advent of new technologies in engine and fuel type innovations, this is indispensable. Moreover, whatever is the type of fuel, technology or engine used in present or in future, combustion will be always there as through combustion of fuel only, power is generated. Hence study of combustion chamber is of prime importance and that too in CI engines because their applications are varied and widespread.

**REFERENCES**


