Available online at http://www.journalcra.com

INTERNATIONAL JOURNAL OF CURRENT RESEARCH

International Journal of Current Research
Vol. 8, Issue, 09, pp.37843-37859, September, 2016

ISSN: 0975-833X

RESEARCH ARTICLE

A REVIEW ON EFFECT OF COMPRESSION RATIO ON THE PERFORMANCE, COMBUSTION AND EMISSION CHARACTERISTICS OF COMPRESSION IGNITION ENGINE FUELLED WITH BIODIESEL

***Praveen A. Harari, Ghadge, S. S., Bharat N. Kokku and Vignesh D. Ezhava**

Department of Mechanical Engineering, SGOICOE, Belhe (Pune), Maharashtra, India

ARTICLE INFO

Article History:
Received 23rd June, 2016
Received in revised form
30th July, 2016
Accepted 04th August, 2016
Published online 20th September, 2016

Key words:
Biodiesel,
Compression Ratio,
Performance,
Combustion,
Emission.

ABSTRACT

In the present world it is essential to find an alternate fuel source due to the increased industrialization and depletion in natural resources. The method of obtaining biodiesel from various sources and blending them with diesel is adopted in many economically developed and developing countries around the world. Presently lots of researches are being performed to evaluate biodiesel as alternative fuel for diesel engine due to associated problems with use of conventional diesel such as environmental degradation, petroleum depletion etc. However, the combustion behavior of biodiesel is different from diesel. To compensate the effects of utilization of biodiesel and maximize the performance, input parameters of the engine should be adjusted. Biodiesel is receiving increased attention as an alternative, non-toxic, biodegradable and renewable diesel fuel. It is derived from oils and fats by transesterification with alcohols. The main hurdle to the commercialization of biodiesel is the cost of raw materials. Increasing the consumption of fuel in power and automobile sector, increase the pollution of the environment. Smoke and NOx are main pollutants of emission from diesel engine and it is very difficult to control them simultaneously petroleum based fuels is a finite resource that is rapidly depleting. Consequently, petroleum reserves are not sufficient enough to last many years. Biodiesel is one of the alternative fuel made from vegetable oil, friendly for environment and has no effect on health and can reduce the emission compared with diesel fuel.

Citation: Praveen A. Harari, Ghadge, S. S., Bharat N. Kokku and Vignesh D. Ezhava, 2016. "A review on effect of compression ratio on the performance, combustion and emission characteristics of compression ignition engine fuelled with biodiesel" *International Journal of Current Research*, 8, (09), 37843-37859.

INTRODUCTION

With crude oil reserves estimated to last for few decades, there has been an active search for alternate fuels. The depletion of crude oil would cause a major impact on the transportation sector**.** Among the various alternate fuels under consideration biodiesel derived from vegetable oils, is the most promising alternative fuel to diesel due to the following reasons. Biodiesel can be used in the existing engine without any modifications. Biodiesel is made entirely from vegetable sources also it does not contain any sulfur, aromatic hydrocarbons, metals or crude oil residues. Biodiesel is an oxygenated fuel; emissions of carbon monoxide and soot tend to reduce. The use of biodiesel can extend the life of diesel engines because it is more lubricating than petroleum diesel fuel. Biodiesel is produced from renewable vegetable oils/animal fats and hence improves the fuel or energy security and economy independence.

****Corresponding author: Praveen A. Harari,***
Department of Mechanical Engineering, SGOICOE, Belhe (Pune), Maharashtra, India

A lot of research work has been carried out to use vegetable oil both in its neat form and modified form. Since India is net importer of vegetable oils, edible oils cannot be used for production of biodiesel. India has the potential to be a leading world producer of biodiesel, as biodiesel can be harvested and sourced from nonedible oils like jatropha curcas, pongamia pinnata, neem, mahua, castor, linseed, etc. Some of these oils produced even now are not being properly utilized. Out of these plants, India is focusing on jatropha curcas and pongamia pinnata, which can grow in arid and wastelands. Implementation of biodiesel in India will lead to many advantages like green cover to wasteland, support to agriculture and rural economy and reduction in dependence on imported crude oil and reduction in air pollution (Venkatraman and Devaradjane, 2010). In the modern society having much advancement in technology there is also some issues relating to an alternate source of fuel to sustain the transportation sector for the future generation. However our dependence is on diesel and petroleum for fueling the transportation sector and if this continues then this could threaten our energy resource, affect our economy and even affect our environment so badly that it may even take hundreds of years for a seed to sprout. Thus we are in search of alternate source of fuel to have a sustainable economy. This is possible with the use of Biodiesel which is a renewable source of energy. Though it is not possible to run a CI Engine on 100% biodiesel like jatropha and pongamia without any major modifications in the presently available engine, when blended with diesel in various proportions it would make the world wonder with its Eco-friendly nature. Biodiesel is nothing but long-chain alkyl esters which is obtained from animal fat and plant seeds. They are regarded as carbon sink as they absorb 78.5% of carbon in the atmosphere as they burn and even considered as cleaner than fossil fuels (Balajee *et al*., 2013).

MATERIALS AND METHODS

Transesterification Reaction

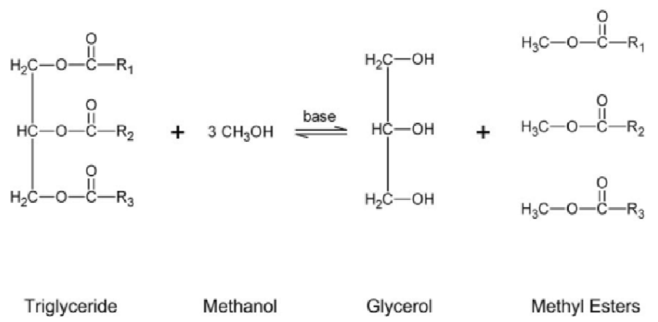

Fig. 1. Transesterification reaction

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.

RESULTS AND DISCUSSION

Performance Characteristics

Brake Thermal Efficiency (BTE)

Anand *et al*. (2009) found that, BTE increased with an increase in load at various compression ratios. That could be attributed to reduction in heat loss and increase in power with increase in load. The maximum BTE obtained was about 27.37%for B5 at 15:1 compression ratio (26.65% diesel), 27.98% for B5 and B10 at 17:1 compression ratio (26.65% diesel) and 29.28% for B20 at 19:1 compression ratio (27.92% diesel) which was quite higher compared to that of diesel. The mixing of biodiesel in diesel oil yields good thermal efficiency curves. Initially the thermal efficiency of the engine was improved by increased concentration of the biodiesel in the blend. The possible reason for that may be additional lubricity provided by biodiesel. The molecules of biodiesel contain some amount of oxygen, which takes part in the combustion process. After certain limit with respect to diesel ester blend, the thermal efficiency trend was reversed and it start decreased as a function of the concentration of blend (Anand *et al*., 2009).

Venkatraman *et al*. (2010) found that, the maximum BTE found was about 30.5% for PME20 and 30.1% for diesel. Increase in thermal efficiency due to percentage of oxygen presence in the biodiesel, the extra oxygen leads to better combustion inside the combustion chamber. The thermal efficiency of the engine was improved by increased the concentration of the biodiesel in the blends and also the additional lubricity provided by biodiesel. The main reason for increased the thermal efficiency with increase in injection pressure may be due to atomization (Venkatraman and Devaradjane, 2010).

Sanjay Patil (2012) found that, BTE for all the test fuels was increased with increase in compression ratio (CR). The BTE at every compression ratio was increased with increase in proportion of biodiesel in the blend. That may be due to the presence of oxygen molecule in the biodiesel which enhanced the combustion phenomenon. The BTE of test fuels was lower at compression ratio of 15.5:1 and 16.5:1 and higher at compression ratio of 17.5:1 (Sanjay Patil, 2012). Sejal Narendra Patel *et al*. (2012) found that, as compression ratio increased brake thermal efficiency was considerably increased for all the blends. Thatmay because as higher compression ratio the combustion was much better compare to lower compression ratio. At CR-18, at every load there was higher efficiency with diesel compare to biodiesel. The brake thermal efficiency of an engine increased significantly with load. At full load the brake thermal efficiency of biodiesel 8.29% lower than diesel and at 50% part load it was 5.28% lesser than diesel. Thathappened because of the lower calorific value of the biodiesel (Sejal Narendra Patel et al., 2012).

Balajee *et al*. (2013) found that, BTE was increased with increased loads for all blends of biodiesels and diesel. It may be due to reduction in heat loss and increase in power. BTE of jatropha B10 and B20 was greater than diesel whereas the BTE of jatropha B30 was similar to diesel. It may be because of the presence of oxygen in biodiesel which enhanced the combustion as compared to diesel and biodiesel was more lubricant than diesel that provides additional lubrication. Jatropha oil biodiesel had higher viscosity, higher density and lower calorific value than diesel. Higher viscosity leads to decreased atomization, fuel vaporization. These may be possible reasons of jatropha B30 had lowest BTE for all loads. The highest value of BTE was obtained at 12 kg load for all the fuels. Similarly, in pongamia oil the BTE was maximum for B10 blend and the BTE of blend B10 and B20 was similar to diesel. That may be due to lower heating or calorific value of the pongamia oil than diesel (Balajee *et al*., 2013). Pavanendra Kumar *et al*. (2013) found that, the thermal efficiency increased with increase in load also it could be found that brake thermal efficiency improved at higher compression ratios. The reason for the improvement of brake thermal efficiency was better combustion and better lubricity of biodiesel. The maximum brake thermal efficiency was found at a compression ratio of 18, due to the superior combustion and better intermixing of the fuel (Pavanendra Kumar *et al*., 2013).

Shelke, (2013) found that, BTE decreased as the proportion of soyabean oil increased in the soyabean oil-diesel oil blends in both the compression ratios. As the compression ratio increased the wall temperature of the combustion chamber increased and due to that the volumetric efficiency decreased. The decrease in volumetric efficiency and the effect of ignition delay were the main reasons for the deterioration of fuel combustion resulted in increase in fuel consumption and subsequent reduction in BTE. In CR-16, the wall temperature was comparatively lower than CR-18 and due to the fact that, the reduction in BTE in CR-16 was slightly lower than in CR-18 (Shelke, 2013). Nageswara Reddy *et al*. (2013) found that, maximum thermal efficiency achieved was about 21.84% at a compression ratio of 14. Minimum thermal efficiency achieved was about 19.31% at a compression ratio of 18. It was found that upto a compression ratio of 14 the brake thermal efficiency showed an increased trend, but reversed its trend and start decreased with further increase in the compression ratio. Reason could be due to the better intermixing of fuel and air along with better combustion at compression ratio of 14 maximum thermal efficiency was achieved. At CR-12, CB10 and CB20 had almost similar brakethermal efficiency as that of diesel. Only CB40 showed value on the lower side. Maximum thermal efficiency achieved was about 21.91% for diesel. Minimum thermal efficiency achieved was about 19.84% for CB40. At CR-14, CB10 had the maximum thermal efficiency whereas CB40 had minimum thermal efficiency at full load conditions (Nageswara Reddy*et al*., 2013).

Santosh Kumar Kurre *et al*. (2013) found that, BTE increased with the compression ratio that may because at higher compression ratio the rate of combustion of ethanol increase which resulted better thermal efficiency and if compared to neat diesel it was decrease for all blends. The major decrease in BTE was found 20% with higher compression ratio for E10. That may be due to lower heating value of the test fuel (Santosh Kumar Kurre*et al*., 2013). Jalpit B. Prajapati *et al*. (2014) found that,brake thermal efficiency for all compression ratio decreased as the load increased. As compared with tested fuels it increased. That could be due to that as the load increased, suction pressure developed will be higher which might have resulted in efficient combustion (Jalpit *et al*., 2014). Naveen *et al*. (2014) found that, BTE increased with increase in the load for compression ratio selected from 15:1 to 18:1. That may be due to increasing the load on engine increased the brake power output. When the percentage of blending increased then BTE reduced for all the compression ratios selected. That may be due to increased density and viscosity with increase of blending from 20% to 60%. High density of blending increased the mass of fuel injected for same power output. At higher load condition 20% blending of ziziphus jujuba oil (B20) with diesel had higher BTE among the three different blending for all the compression ratios selected except CR-17. At the same time, the percentage blending of ziziphus jujuba oil increased with diesel results that elevated flash point and fire point of the fuel. Increased the compression ratio of the engine produced higher peak pressure and higher temperature at the end of compression. Such increased peak temperature helped to achieve improved combustion quality of blended diesel even it has higher viscosity. At CR-18, BTE of engine fuelled with 20% blending of ziziphus jujuba oil given same result as diesel (Naveen *et al*., 2014). Pavan *et al*. (2014) found that, at CR-13.5 BTE of B25 blend was higher as compared to that of diesel and B15. The BTE of the blends was increased with increase in applied load. It was happened due to reduction in heat loss and increase in power developed with increase in load. The maximum BTE at full load was 38.3% for B25 at CR-13.5 which was 8% higher than that of diesel. At CR-16.5 BTE of B25 blend was higher as compared to that of diesel. The BTE of the blends was increased with increase in applied load and the maximum BTE was 43.1%. The increase in BTE for higher blends may be due to the improved atomization fuel vaporization and frictional losses decrease. At CR-18 BTE of B25 blend was higher as compared to that of diesel. The BTE of the blends was increased with increase in applied load and the maximum BTE was 40.8%. The increase in BTE for higher blends may be due to the combined effect of its higher heating value and decrease in fuel consumption (Pavan *et al*., 2014). Eknath *et al*. (2014) found that, for jatropha fuel BTE was higher at any load compare with any other fuel. For CR-18, compare with diesel jatropha fuel had 37.75% higher efficiency at low load of 2 kg and about 9.29% higher at maximum load of 10 kg. Similarly for karanja fuel compare with diesel efficiency was higher by 15.94% at low load of 2 kg and 6.71% higher at maximum load of 10 kg. For the blend KJD20, at maximum load of 10 kg the thermal efficiency was 4.89% less than diesel fuel and 15.6% less than karanja and jatropha fuel. For KJD40 at maximum load of 10 kg the thermal efficiency was 1.18% less than diesel, 11.54% less than jatropha and 8.5% less than karanja. For K20J40D at maximum load of 10 kg the thermal efficiency was 7.08% less than diesel, 18.05% less than jatropha and 14.8% less than karanja. For K20J60D the thermal efficiency was 2.73% less than diesel 13.26% less than jatropha and 10.1% less than karanja. With increase in content of jatropha for fix content of karanja the thermal efficiency decrease first and then again increases that may because of change in heating value and specific fuel consumption. Decrease in compression ratio had significant change in thermal efficiency for all the fuel blends. For fuel blend K20J40D change was less than 1% (Eknath and Ramchandra, 2014).

Navaneetha Krishnan *et al*. (2015) found that, as the applied load increased the BTE of the fuel blends also increased. That may be due to increase in power developed and reduction in heat loss with increase in load. The maximum BTE at full load was 41.72% for B40. The BTE of B20 and standard diesel were 39.68% and 36.49% respectively and while compared the values fuel blend B40 was 5.2% higher than standard diesel. BTE of fuels increased when the load of the engine increased. Due to lower heating value and increased fuel consumption the BTE for higher blends decreased (Navaneetha Krishnan and Vasudevan, 2015). Silambarasan *et al*. (2015) found that, the BTE gradually increased with increasing compression ratio. At maximum load with the CR-19.5, the BTE for A20 was 31.67% and it was almost equal to the neat diesel fuel (30.47%). That may be due to increase in compression ratio ensured better air-fuel mixing and faster evaporation and leads to complete combustion (Silambarasan *et al*., 2015). Venkateswara Rao (2015) found that, the BTE was increased with increase in compression ratio for all fuels at all loads. For blend fuel, the BTE was always less when compared to diesel because of lower calorific value of biodiesel. The increase in BTE was found as compression ratio increased, due to better mixing of biodiesel at higher temperature causes complete combustion of fuel (Venkateswara Rao, 2015).

Biju Cherian Abraham *et al*. (2015) found that, at lower brake power the BTE of biodiesel blend and biodiesel mixtures were same as that of diesel fuel. The reason may be better combustion and better lubricating property of biodiesel (Biju Cherian Abraham and Chindhu Prasad, 2015). Hariram *et al*. (2015) found that, the BTE increased with increased load conditions which may be due to the fact that power loss would be relatively less when load increased. At standard setting of CR-18 IT-23, the BTE of B20 was higher than diesel at 75% and 100% loads. At full load condition the BTE of B20 blend in comparison with diesel increased by 1% which was about 3.5% higher than diesel 75% load, the increase was around 2.4% which could have been due to the presence of higher amount of oxygen content in B20 accounting for better combustion than diesel. On increased the compression ratio, the BTE showed an overall improvement for both diesel and biodiesel blends owing to reduction in ignition delay. CR-18, exhibited high BTE at all load conditions. When compression ratio was increased from 16 to 18, about 12% and 14% increase in BTE was found for diesel and B20 blend at standard injection timing at full load conditions and similarly 12.5% and 13.5% increase in BTE at advanced timing and 10% and 11.5% increase in BTE at retarded timing was found for diesel and B20 blend. On the advancement of injection timing, the BTE increased for both the diesel and B20 blends. At full load conditions, the BTE slightly increased by about 1.8%, 2.6% and 2.4% at compression ratio of 16, 17 and 18 when the injection timing was advanced from 23° to 25° bTDC for diesel. The BTE increased by 1.5%, 2.6% and 1.8% at compression ratio of 16, 17 and 18 on advanced the injection timing. BTE reduced by 1.5%, 1.8% and 2.8% for diesel and 2%, 3.4% and 3.2% for B20 blend at full load when injection timing was retarded by 3° bTDC from standard setting of 23° bTDC at compression ratios of 16, 17 and 18 respectively (Hariram and G. M. Kumar, 2015).

Brake Specific Fuel Consumption (BSFC)

R. Anand *et al*. (2009) found that, for all fuels tested BSFC decreased with increase in load at all compression ratios. The reason may be that higher percentage of increase in brake power with load compared to fuel consumption. At CR-15:1, the BSFC of 0.305 kg/kW-hr was found for the diesel fuel. In spite of the different calorific values, the fuel blends B5 and B20 maintain about the same BSFC as that diesel fuel. The B10 and B15 blends showed a BSFC of increase by 9.5% and 6.2% respectively at equivalent loads. At higher CR-17:1, the BSFC of the fully loaded engine for diesel fuel was 0.295 kg/kW-hr, whereas that of B15 and B20 showed BSFC increased by 5% and 6.1%. The BSFC of blend B5, which differs itself had the lowest, 0.925% amount of oxygen appeared to be nearly the same as that of blend B10, both of them had higher BSFC by 3.7% relative to diesel fuel. The fuel blend B20 maintain about the same BSFC as that of diesel fuel 0.291 kg/kW-hr whereas blends B5 and B15 maintain their BSFC higher by 7.6% and blend B15 suggested the BSFC was higher by 5.1%. The reason may be due to lower calorific values of biodiesel blends compared to those of neat diesel (Anand *et al*., 2009). Venkatraman *et al*. (2010) found that, the BSFC of 0.273kg/kW-hr was found for diesel and 0.272 kg/kW-hr PME20.It was found that BSFC decreased with the increase in injection pressure, the increase in BSFC with the increase in concentration of PME in diesel (Venkatraman and Devaradjane, 2010). Sejal Narendra Patel *et al*. (2012) found that, the value of brake specific fuel consumption on compression ratio at full load for B20, B30, B40, B60, B80 and B100 was 0.32kg/kW-hr, 0.33 kg/kW-hr, 0.34 kg/kW-hr, 0.36 kg/kW-hr, 0.38 kg/kW-hr and 0.40 kg/kW-hr respectively. Such a high value of BSFC for the blends, especially for B80 and B100 may be attributed to the poor combustion characteristics of those blends (Sejal Narendra Patel and Ravindra Kirar, 2012).

D. Balajee *et al*. (2013) found that, the specific fuel consumption using biodiesel fuel was expected to increase as compared to the consumption of diesel fuel. BSFC decreased sharply with increase in load for all fuel samples at all the compression ratio. The main reason may be that the percent increase of fuel required to operate the engine was less than the percent increase in brake power due to relatively less portion of the heat losses at higher loads. As the BSFC was calculated on weight basis, higher densities resulted in higher values of BSFC. But unexpectedly BSFC of jatropha B10 was slightly lesser than the diesel. The jatropha B30 showed the maximum BSFC followed by pongamia B20 and B30 (Balajee*et al*., 2013). Pavanendra Kumar *et al*. (2013) found that, the BSFC decreased with increase in load due to fact that the ratio of increase in brake power was more as compared to increase in fuel in fuel consumption. It was found the BSFC of B50 more as compared to diesel when experimented at different compression ratios. The reason for that increase in BSFC was low heat value of esters of vegetable oils compared to diesel so more biodieselwas needed to maintain the power output (Pavanendra Kumar*et al*., 2013). Shelke R. E. (2013) found that, at higher compression ratio the temperature and pressure of air at the beginning of injection were higher. 100% diesel oil has the lowest BSFC than other blends and esterified soyabean oil in both the compression ratios. The possible reason behind the result was that, the dissimilar specific gravity and heating value among the blends, esterified soyabean oil and diesel oil were responsible for the increase in fuel flow and BSFC. The increase in BSFC for esterified soyabean oil operation with respect to 100% diesel operation was slightly lower in CR-16 than in CR-18 (Shelke, 2013).

V. Nageswara Reddy *et al*. (2013) found that, at full load condition the lowest BSFC found was about 390 g/kW-hr at CR-14. Highest BSFC found was 440 g/kW-hr at CR-16, 17 and 18. That could be contribution to charge dilution. BSFC was almost same at CR-12 and CR-13 due to incomplete combustion of the fuel at these compression ratios. At CR-12, at full load condition the lowest BSFC found was about 390 g/kW-hr for diesel and highest found was 430 g/kW-hr for CB40. BSFC was same for CB10 and CB20 i.e. 400 g/kW-hr. At CR-14, at full load condition the lowest BSFC consumption was about 390 g/kW-hr for CB10 and highest found was 440 g/kW-hr for CB40. BSFC was nearly same for CB10 and diesel (Nageswara Reddy *et al*., 2013). Santosh Kumar Kurre *et al*. (2013) found that, BSFC decreased with the compression ratio increased. E20 showed decrease of 20% for lower compression ratio and 16% for higher compression ratio. That may be due to the fact that, at higher compression ratio the pressure and temperature increased because of that the rate of combustion increased which gave better thermal efficiency thereby decreased BSFC (Santosh Kumar Kurre*et al*., 2013). Jalpit B. Prajapati *et al*. (2014) found that, BSFC decreased with increased in load. From all compression ratios, CR-18 had better performance as compared to CR-16. Also B30 had better performance as compared to B0. Because of lower calorific value of biodiesel and its blends compared to diesel, more amount of fuel was required to produce the same power output (Jalpit *et al*., 2014).

Naveen *et al*. (2014) found that, specific fuel consumption was reduced with increased the load for all compression ratios. The main reason for that brake power developed was higher than fuel consumption in higher load. Specific fuel consumption increased with increased the percentage of blend from 20% to 60%. At higher load condition B20 registered lesser fuel consumption when compared to other blending (B40 and B60) for all the compression ratios selected except the compression ratio of 17:1. B60 registered higher fuel consumption for the entire compression ratio. That may be due to combined effect of higher density and lower calorific value with respect to increasing the blend and also varying chemical structure. High density of ziziphus jojoba oil caused higher mass injection of fuel for the same volume at same injection pressure. Lower calorific value caused higher fuel consumption for the same power development. Also, different chemical structure of oil have poor combustion quality increased the fuel consumption. Specific fuel consumption, decreased with increased compression ratio in consequence of higher temperature produced at the end of compression (Naveen *et al*., 2014).

Eknath *et al*. (2014) found that, with increase in load on engine specific fuel consumption decreased for all the fuels. At maximum load of 10 kg and CR-18, karanja fuel had 10% more fuel consumption than diesel and 6.6% more than jatropha. For KJD20 fuel consumption was 10% higher compare with diesel, 6.66% compare with jatropha and same compare with karanja. For KJD40 fuel consumption was higher by 6.89% compare with diesel, 3.45% higher compare with jatropha and 3.4% lower compare with karanja. For K20J40D fuel consumption was higher by 12.9% compare with diesel, 9.67% higher compare with jatropha and 3.23% higher compare with karanja. That indicated for fixed content of karanja, increase in content of jatropha resulted in higher fuel consumption. Density of blends of the fuel was 2 to 5% more than petro-diesel. Since density was higher than diesel fuel specific fuel consumption was higher for blends compare with diesel (Eknath and Ramchandra, 2014).

Navaneetha Krishnan *et al*. (2015) found that, when load increased the specific fuel consumption of the fuel will maintain a gradual decrease. The specific fuel consumption values were 0.2234 kg/kW-hr, 0.2268 kg/kW-hr and 0.2201 kg/kW-hr for the fuel blends B20, B40 and standard diesel respectively. The specific fuel consumption values increased only for the higher percentage of blends. It may be due to heating value, density and viscosity of the fuels. Fuel blend B40 had lower energy content than standard diesel but higher energy content than blends B10, B20 and B60 (Navaneetha Krishnan and Vasudevan, 2015). Silambarasan *et al*. (2015) found that, the BSFC gradually decreased with increased compression ratio. At maximum load with the compression ratio of 19.5 the BSFC for A20 was 0.301 kg/kW-hr and 3% reduction of BSFC was achieved when compared with the neat diesel fuel. That may be due to increase in compression ratio leads to reduction in dilution of charge by residual gases which resulted in better BTE and lower BSFC (Silambarasan *et al*., 2015).

Venkateswara Rao (2015) found that, BSFC decreased with increased of brake power and compression ratio and the BSFC for blend fuel was 2.4% higher as compared to diesel at maximum load for compression ratio of 20:1. The BSFC was 18.23% less with increased compression ratio from 14:1 to 20:1 for the blend fuel at full load operation. Maximum BTE of 24.5% was found at minimum BSFC of 0.38 for blend fuel at compression ratio of 20:1 (Venkateswara Rao *et al*., 2015). Biju Cherian Abraham *et al*. (2015) found that, the specific fuel consumption of the diesel as well as the blends were decreased with increased load upto 80% load and then decreased. Due to lower calorific value of biodiesel for the same power output B20 needed more fuel flow than diesel. At maximum power, the fuel flow rate of J20 was about 2% higher than diesel (Biju Cherian Abraham and Chindhu Prasad, 2015). Babu *et al*. (2015) found that, at CR-16.5 the trend of BSFC of the engine with PSME were in line with petro-diesel operation. The BSFC of the engine were on the higher side with biodiesel operation when compared to baseline performance. As the fuel injection pressure was increased, the BSFC values were decreased. But the values with 210 bar were close to baseline values. It was better to adopt higher injection pressures when a fuel with higher viscosity was used compared to petro-diesel. That may be due to the fact that as the fuel injection pressure was increased smaller and smaller size droplets should have produced and which in turn mixed thoroughly with available air and let near complete combustion. Further increase in pressure beyond 210 bar had not helped. Therefore, for the engine configuration 210 bar could be an optimum injection pressure. At CR-19, the BSFC values for fuel injection pressure of 190 bar were close to baseline data. Higher compression ratio had helped a vital role in obtaining better performance with neat biodiesel (Babu *et al*., 2015).

Brake Specific Energy Consumption (BSEC)

Venkatraman M. *et al*. (2010) found that, BSEC increased with the increase in concentration of PME in diesel and decreased with increased in injection pressure. The found value of BSEC for PME20 of 11796kJ/kW-hr was lower compared to diesel 12200kJ/kW-hr (Venkatraman and Devaradjane, 2010). V. Hariram *et al*. (2015) found that, BSEC decreased with increase in load which was found at all the compression ratios tested for all the blends at varying injection timing and that may be due to the lesser percentage intake in fuel consumption with load compared to brake power and relatively lower heat loss at higher loads. At standard setting conditions the BSEC at full load and 75% loads were lower by 8% and 10% for B20 blend compared to diesel which could be due to the higher amount of oxygen content in the blends and the possibility of better complete combustion than diesel. BSEC reduced with the increase in compression ratio and at full load conditions BSEC decreased by 8% and 20% when compression ratio was increased from 16 to 17 and from 16 to 18 for diesel and for B20 blend that reduction found was 20% and 25% for above increase in compression ratio. At 75% load, BSEC reduced by 17% and 25% for diesel and B20 blend when compression ratio was increased from 16 to 18. Higher reduction in BSEC was found for B20 blend than diesel on increased the compression ratio. The better performance of biodiesel blend at higher compression ratio might have been due to the lesser volatility and higher viscosity. At CR-18, when injection timing was advanced from 23° to 25° bTDC, BSEC reduced by almost 10% and 14% for diesel and B20 blend and the reduction was 15%, 16% and 14%, 16% at CR-17 and CR-16 for diesel and B20 blend respectively at 100% load. On retarded the injection timing by 3° bTDC, BSEC increased by 22% and 23% at full load condition at CR-18 for diesel and B20 blend and the increase was about 23.5%, 28% and 24%, 30% at CR-17 and CR-16 for diesel and B20 blend (Hariram and Kumar, 2015).

Exhaust Gas Temperature (EGT)

Shelke R. E. (2013) found that, as the proportion of diesel oil decreased in the blends the EGT increased in both the compression ratios. Esterified soyabean oil had higher cetane number which helped to burn the fuel comparatively better than the higher soyabean oil-diesel blends. In CR-18, EGT was slightly higher than in CR-16 (Shelke, 2013). Nageswara Reddy *et al*. (2013) found that at maximum compression ratio the exhaust gas temperature was maximum. Exhaust gas temperature was increased with increase in compression ratio. Exhaust gas temperature was found highest compression ratios of 16, 17 and 18 (Nageswara Reddy *et al*., 2013). Santosh Kumar Kurre *et al*. (2013) found that, EGT decreased with compression ratio for all blends. That may be due to the fact that at higher compression ratio the pressure and temperature increased because of that the rate of combustion increases more complete combustion resulted in lower losses during exhaust. At higher compression ratio it was nearer to neat diesel for all the blends (Santosh Kumar Kurre*et al*., 2013).

Naveen *et al*. (2014) found that, with increased the load on the engine EGT were increased due to higher heat loss from the combustion. EGT decreased with increased the blending percentage. B20 registered the highest EGT among the B20, B40 and B60 at full load condition. Lowest EGT registered by B60 among the three different blending and diesel. Lesser EGT of biodiesel blends could be due to lower calorific value and higher viscosity led to poor atomization rate. The lower EGT suggested that the engine was not thermally overloaded but more fuel was required to maintain the same power output. EGT decreased significantly for all the blending when raising the compression ratio from 15:1 to 18:1. Combustion quality increased with increased compression ratio because the negative effects of higher viscosity overcome by improved combustion temperature. Raising the compression ratio and blending of ziziphus jujuba make the positive effect on EGT aspect (Naveen *et al*., 2014). Pavan *et al*. (2014) found that, at CR-13.5 EGT for diesel was found to increase with increase in load. Among the blends B15 and B25 showed relatively less exhaust temperature than diesel. The maximum exhaust temperature was found for diesel which was 445°C which was high as compared to 407°C of B15. The increase in temperature was may be due to longer ignition delay. At CR-16.5, the EGT for diesel was found to increase with increase in load. Among the blends B15 and B25 showed relatively less exhaust temperature than diesel. The maximum exhaust temperature was found for diesel which was 479°C which was high as compared to 458°C of B25. The relatively high boiling point constituents were not adequately evaporated during the main combustion phase and continued to burn in the late combustion phase. At CR-18, the EGT for diesel was found to increase with increase in load. Among the blends B15 and B25 showed relatively less EGT than diesel. The maximum EGT was found for diesel which was 459°C which was high as compared to 435°C of B25. That increase in temperature may be due to unburnt fuel was coming out as oxides of nitrogen, sulphur and carbon (Pavan *et al*., 2014).

Navaneetha Krishnan *et al*. (2015) found that, the EGT decreased for different blends when compared to that of diesel. Standard diesel had highest EGT and it was 323.97°C, whereas the blends B20 and B40 had lower temperature 306.23°C and 312.93°C respectively. Blended fuels had lower calorific value that leads to reduction in EGT as compared to the standard diesel and lesser temperature at the end of compression. Higher performance was due to lower exhaust loss (Navaneetha Krishnan and Vasudevan, 2015). V. Hariram *et al*. (2015) found that, the EGT increased steadily with increase in load for all blends at all blends at all compression ratios tested due to the increased amount of fuel at higher loads. The EGT range was found from 127°C to 280°C at all blends and variables tested. The EGT increased with B20 compared to diesel at all compression ratio and injection timing which may be due to higher heat loss at increased load conditions. The mean EGT increased by an overall 8% with B20 blend compared to diesel at standard setting. Increased the compression ratio from 16 to 18, the mean EGT dropped by around 5% for diesel and 6.5% for B20 blend and the drop was around 4% and 5% for diesel and B20 blend when compression ratio was increased from 16 to 17 which could have been due to the complete combustion of fuel. On advancing the injection timing the mean EGT reduced by 6% and 7% for diesel and B20 blend at CR-18, 5.5% and 8% for CR-17 and 6.5% and 7.5% at CR-16 for diesel and B20 blend. Retarding the injection timing increased the EGT where 7.5% and 9% increase in EGT was found at CR-18 for diesel and B20 and at CR-17 and CR-16, the EGT increase was 6.7%, 8% and 8%, 9.5% for diesel and B20 blends (Hariram and Kumar, 2015).

Volumetric Efficiency

Eknath *et al*. (2014) found that, for compression ratio of 18 and 16 no any significant variation in volumetric efficiency was found for jatropha and karanja compared with diesel fuel. Similarly for other blends of fuel changes found was less than 1% only (Eknath and Ramchandra, 2014). Venkateswara Rao P. (2015) found that, the volumetric efficiency of blend fuel was less by 0.9% compared to that of diesel at maximum load with increase in compression ratio from 14:1 to 20:1 the volumetric efficiency decreased by 3.1% for blend fuel at maximum load. The residual gas left in the clearance volume at high pressure and temperature caused to decrease volumetric efficiency (Venkateswara Rao, 2015).

Biju Cherian Abraham *et al*. (2015) found that, for jatropha biodiesel blend the volumetric efficiency was very low. The volumetric efficiency of the diesel engine mainly depends upon the combustion chamber temperature. The increase in the chamber temperature increased the intake air temperature and consequently reduced the mass of air drawn in each cycle thereby decreased the volumetric efficiency. Biodiesel mixture has higher volumetric efficiency when compared to biodiesel (Biju Cherian Abraham and Chindhu Prasad, 2015).

Mechanical efficiency

Balajee *et al*. (2013) found that, mechanical efficiency increased with increase in load for all type of fuel blends. That may be due to increase in the brake power. Mechanical efficiency of the biodiesel was greater than the diesel at all compression ratios. Pongamia B10 and jatropha B10 had the maximum mechanical efficiency 62.77% and 62.20% respectively. Diesel had the minimum mechanical efficiency of 55.49%. The increase in efficiency for all biodiesel blends may be due to improved quality of spray, high reaction activity in the fuel-rich zone and decrease in heat loss due to lower flame temperature of blends than that of diesel (Balajee*et al*., 2013). Pavan *et al*. (2014) found that, at CR-13.5 in the beginning with increasing load of the engine the mechanical efficiency of various concentrations of blends and pure diesel were increased. The maximum mechanical efficiency of the engine was 63% for hibiscus oilB25 at full load where it was slightly less for diesel which was 56.5%. That may be due to improved atomization fuel vaporization, better spray characteristics and improved combustion through mixture. At CR-16.5, the mechanical efficiency for diesel was found to increase with increase in load. Among the blends B15 and B25 showed the maximum mechanical efficiency was achieved at full load for B15 that was 64.8% than other blends and pure diesel.

The minimum mechanical efficiency was found for diesel at full load condition. The increase in efficiency may be decrease in heat loss due to low flame temperature of blends than that of diesel. At CR-18, mechanical efficiency for diesel was found to increase with increase in load. Among the blends B15 and B25 showed the maximum mechanical efficiency was achieved at full load for B25 that was 65.8% than other blends pure diesel. The minimum mechanical efficiency was found for diesel at full load condition. That may be due to better combustion and an increase in the energy content of the blend (Pavan *et al*., 2014). Eknath *et al*. (2014) found that, at CR-18 for jatropha and karanja biodiesel at part load operation mechanical efficiency was close to diesel fuel. With increase in load on engine for both these fuels efficiency decreased by about 15 to 17%. For other blends similar effects were found. At load range of 6 to 10 kg loss in mechanical efficiency was about 20 to 22% compare with diesel fuel. Decrease in compression ratio to 16 it was found that all the fuel blends had the efficiency was more or less same compared with diesel fuel (Eknath and Ramchandra, 2014).

Navaneetha Krishnan *et al*. (2015) found that, as the load increased mechanical efficiency for all the blends were also increased in a steady state. Blends B40 and B10 at full load hadmaximum mechanical efficiency and it was 93.77% and 87.62% respectively. Due to difference in fuel properties such as viscosity and density negligible difference among the curve may be accounted. The mechanical efficiency of the fuel blends was in general very close to that of diesel. While compared to diesel fuel blends had improved quality of spray, high reaction activity in the fuel rich zone and decreased in heat loss due to lower flame temperature will be the cause for efficiency increases. Mechanical efficiency increased with increasing load for all the blends (Navaneetha Krishnan and Vasudevan, 2015). Biju Cherian Abraham *et al*. (2015) found that, the mechanical efficiency of the blends was less which may be due to difference in fuel properties such as viscosity and density. The mechanical efficiency of the fuel blendswas in general very close to that of diesel (Biju Cherian Abraham and Chindhu Prasad, 2015).

Brake Power (BP)

Sejal Narendra Patel *et al*. (2012) found that,the values of brake power at CR-18 full load for B0, B10, B20, B30, B40, B60, B80 and B100 were 3.35kW, 3.30kW, 3.43kW, 3.37kW, 3.18kW, 3.24kW, 3.14kW and 3.08kW respectively. The brake power was higher for B20 and beyond that as percentage of biodiesel increased the value of biodiesel decreases. The nature of brake power increased with the increase in load. At CR-18, the brake power of an engine increased significantly with load. At full load the brake power of biodiesel was comparatively 8.05% and at part load (50% load) it was 8.27% lesser than diesel because of the higher viscosity and density of the biodiesel (Sejal Narendra Patel and Ravindra Kirar, 2012). Pavanendra Kumar *et al*. (2013) found that, theBP increased with increase in load. For higher compression ratio B50 produced more brake power especially at higher load. Due to shorter ignition delay the combustion started earlier for biodiesel. That effect diminished with higher CR, as the temperature and pressure of the cylinder increased. Further, at higher load the inbuilt oxygen of biodiesel assisted in complete combustion. The friction power of B50 was less than diesel. That factor also contributed in increased BP for B50 (Pavanendra Kumar *et al*., 2013).

Jalpit Prajapati *et al*. (2014) found that, the brake power decreased with decreased in compression ratio. Thatmay be due to the fact that when the compression ratio decreased, cylinder head slightly tilted. Also clearance volume increased in combustion chamber. Because of that peak cylinder pressure decreased. It also affected by diesel-biodiesel blends. The higher density and kinematic viscosity of fuels resulted in the injection of a larger mass of fuel to the engine for the same fuel volume. The greater the viscosity of the blend, the less the leakage in the fuel pump (Jalpit *et al*., 2014). Eknath *et al*. (2014) found that, there were no significant differences in break power of the engine between forbiodiesel, its blends and diesel fuel for both compression ratios 18 and 16. At maximum load for any compression ratio difference between petroleum diesel and pure biodiesel was less than 1.5% only. That was mainly due to higher SFC at higher load at any compression ratio, lower heating value and higher oxygen content of biodiesel. Since density of the biodiesel fuel was more than petro diesel, biodiesel supplied to the engine was more than diesel fuel, which compensate for the loss of heating value

(Eknath and Ramchandra, 2014). Navaneetha Krishnan *et al.* (2015) found that, the fuels blendsB10, B20, B40 and B60 have decreased brake power while compared with standard diesel. At higher compression ratio the conversion of chemical energy into mechanical energy leads to decrease in brake power. In addition to that uneven combustion and lower heating value of the fuel blends leads to decreased brake power. The maximum brake power obtained for B40 and standard diesel while the other blends had a reduction in brake power with load due to lower heating value. The brake power values were 3.7149 kW and 3.6973 kW for B40 and standard diesel respectively (Navaneetha Krishnan and Vasudevan, 2015).

Combustion Characteristics

Cylinder Peak Pressure

Anand *et al.* (2009) found that, overall combustion characteristics for all blends were found quite similar to diesel. Peak pressure increased with increase in compression ratio. Since cetane number for biodiesel was 49 which was higher than neat diesel i.e. 45 hence ignition delay was shorter for all biodiesel blends than neat diesel. Peak pressure increased with increase in loads and biodiesel blends at all compression ratios. Blends had more peak pressure than neat diesel at higher compression ratio (17:1 and 19:1). The reason for thatmay be the presence of oxygen in the biodiesel resulted complete combustion of fuel leading to increase in peak temperature and peak pressure. Blend B20 had maximum peak pressure and it decreased with decrease in biodiesel percentages. While at compression ratio of 15:1, blend B15 and B20 had highest peak pressure at full load. At 19:1 CR, the maximum peak pressure for B20 was about 90.66 bar, B15 was about 90.61bar, B10 was 90.14bar, B5 was 89.65 bar and for diesel was 81.23 bar. In a compression ignition engine, peak pressure depends on the combustion rate in initial stages, which in turn was influenced by the amount of fuel taking part in the uncontrolled combustion phase. The premixed or uncontrolled combustion phase was generated by the ignition delay period and by the mixture preparation during the delay period. Thus higher viscosity and lower volatility of the COME which lead to poor atomization and mixture preparation with air and fast burning nature of COME blends during the ignition delay period were the reasons for this trend of peak pressure. Peak pressure was highest for neat diesel and all the fuel blends have nearly same peak pressures and ignition delay at compression ratio of 15:1. But in after burning, period rate of decrease in pressure was highest for diesel and least for B20. Peak pressure was highest for B20 at 75% load and rate of pressure rise was highest for B5. That might be due to presence of oxygen in the biodiesel that made complete combustion of fuel possible, thereby producing more $CO_2$and hence released more heat from the gases. Thus, both the peak temperature and pressure of biodieselfuelled engine was higher than that of diesel fueled engine (Anand*et al*., 2009).

Venkatraman *et al.* (2010) found that, the cylinder pressure PME20 (82.3bar) anddiesel (81bar) at CR-19:1. The pressure increased with increase in injection pressure 240bar and advanced injection timing 27° bTDC (Venkatraman and Devaradjane, 2010). Sanjay Patil (2012) found that, with increase in compression ratio, the peak pressure was increased for all test fuels. At every compression ratio, the peak pressure decreased with increase in proportion of biodiesel in the blend and also found that the peak pressures of all test fuels were less in comparison with that of diesel. Increase in compression ratio enhanced the pressure and temperature of air-fuel mixture in compression stroke resulted in increased peak pressure. Increase in proportion of biodiesel in blend burned more fuel during diffusion phase of combustion and lower calorific value of blend caused in decrease of peak pressure. Increase in load increased the peak pressure and brake thermal efficiency. Same trend had found with all test fuels. Predicted peak pressure at full load when engine was fuelled with B0, B20, B60 and B100 were compared with experimental results were found in closer approximation (Sanjay Patil *et al*., 2012). Balajee *et al.* (2013) found that, at CR 17.5 themaximum pressure attained during idling was around 47 bar by diesel whereas at maximum load maximum pressure (60.8 bar) was attained by jatropha B30. At compression ratio 16 the maximum pressure (49 bar) in idling condition was attained by diesel whereas at maximum load condition the maximum pressure (62 bar) was attained by jatropha 30. At compression ratio 15, the maximum pressure (31.19 bar) in idling condition was attained by jatropha B10whereas at maximum load condition the maximum pressure (62 bar) was attained by jatropha B30 (5).

Nageswara Reddy *et al.* (2013) found that, in both the load conditions maximumpressure attained was at a compression ratio of 12 and minimum at a compression ratio of 18. While moving from no load to full load condition the change in the cylinder pressure along with the decrease in the ignition delay period was maximum at a compression ratio of 14. At CR-12, maximum cylinder pressure attained was for CB10 at both no load and full load condition. Lowest cylinder pressure attained was for CB40 at both no load and full load condition. Longer ignition delay foundfor CB40 whereas CB10 had the shortest ignition delay. At CR-14, maximum cylinder pressure attained was for CB10 at no load condition but with a delay of 2 degree in crank angle as compared to diesel. At full load condition maximum cylinder pressure attained was for diesel followed by CB10,CB20 and lowest for CB40. Shorter ignition delay was observed at this compression ratio as compared to compression ratio 12 (Nageswara Reddy *et al*., 2013). Eknath *et al.* (2014) found that, for CR-18, the peak pressure for jatropha fuelwas maximum by 6% compared with pure diesel, whereas for karanja it was 4%, for KJD20 and KJD40 it was more by 1% compared with pure biodiesel. However for the blends of K20J40D peak pressure was low by about 12% compared with pure diesel fuel. For K20J60D fuel peak pressure was low by 2% compared with pure diesel. For a compression ratio of 16, peak pressure for all fuel was also decreased and it was found that the peak pressure was low for each blends compare with diesel fuel. That was mainly because of increase in ignition delay. Higher pressure rise was found for fuel due to longer delay. Pressure reached during the second stage of combustion depends on the duration of delay period. Long delay period resulted in high pressure rise, since more fuel was present in the cylinder before the rate of burning comes under control. Same trend was found compared with jatropha fuel since, jatropha has 6% more pressure rise compared with diesel fuel. For the same proportion of karanja and jatropha pressure rise found to be same. However, with increase in jatropha content for the fix value of karanja content pressure was low compare with diesel fuel. That was due to higher density of biodiesel fuel. It was found that for all blends peak pressure was found to be after TDC only that ensured the smooth running of engine (Eknath and Ramchandra, 2014). Navaneetha Krishnan *et al*. (2015) found that, the tamanu oil blend B40 gave higher combustion pressuredue to longer ignition delay of tamanu oil. Longer ignition delay occurred due to the fuel absorbed more heat from the cylinder immediately after injection. The peak pressure value for standard diesel and tamanu oil blends B10, B20, B40, B60 were 72.89 bar, 73.07 bar, 70.76 bar, 71.30 bar and 69.16 bar respectively at full load. The higher combustion pressure obtained due to the rapid and complete combustion of fuel inside the combustion chamber the peak cylinder pressure decreased at the start of combustion and increases further. The pressure rise was due to the combustion rate in initial stages, which was influenced by the fuel intake component in the uncontrolled heat release phase. The peak pressure recorded for standard diesel, B10, B20, B40 and B60 were 64.12 bar, 66.03 bar, 67.03 bar, 65.52 bar, 67.68 bar respectively. Due to the high viscosity and low volatility of biodiesel, the cylinder peak pressure was lower that of standard diesel (Navaneetha Krishnan and Vasudevan, 2015).

Biju Cherian Abraham *et al*. (2015) found that, the maximum cylinder gas pressure was found to be 53.48 bar, 43.40 bar, 45.03 bar at 0 kg and 64.68 bar, 58.87 bar, 57.86 bar at 9 kg for D100, J20 and JR20 respectively. At all engine loads, combustion started earlier for biodiesel than for diesel. That may be due to a short ignition delay and advanced injection timing for biodiesel (because of a higher bulk modulus and higher density of biodiesel). In spite of the slightly higher viscosity and lower volatility of biodiesel, the ignition delay seems to be lower for biodiesel than for diesel. The reason may be that a complex and rapid pre-flame chemical reaction takes place at high temperatures. As a result of the high cylinder temperature existed during fuel injection, biodiesel may undergo thermal cracking and lighter compounds were produced, which might have ignited earlier to result in a shorter ignition delay. At lower engine speed the variation was low as compared to higher loads. As the engine load decreased, the residual gas temperature and wall temperature decreased, which resulted in lower charge temperature at injection time and lengthened the ignition delay (Biju Cherian Abraham and Chindhu Prasad, 2015). Babu *et al*. (2015) found that, the trend of P-θ lines with PSME was found similar to the baseline of petro-diesel. The P-θ line was seen lower at injection pressure of 210 bar and at a Compression ratio of 16.5, whereas in P-θline was also found higher for compression ratio of 19 at the rated injection pressure of 190 bar with compared the baseline data of CR-16.5 and 190 bar. Compression ratio had played a dominant role in achieving better performance (Babu *et al*., 2015).

Heat Release Rate (HRR)

Anand *et al*. (2009) found that, the premixed burning phase associated with a high heat release rate was significant with biodiesel blends operation. Slightly higher peak HRR of 60 J/deg (17:1 CR) and 65 J/deg (19:1 CR) were found for biodiesel under full load. It was found to be 50 J/deg (17:1 CR) and 48 J/deg (19:1 CR) for diesel under similar conditions. Increase in HRR was an indication of better premixed combustion and was probably the reason for increased NO emissions. Higher HRR for biodiesel blends was probably due to excess oxygen present in its structure and dynamic injection advance apart from static injection advance. Higher boiling point of biodiesel could also result in higher HRR. However for 15:1 CR there was no much variation in peak HRR in compared with diesel at full load. Increase in diffusion combustion without increase in premixed combustion was indicative of poor combustion and was probably the reason for decreased NO emissions at full load (Anand *et al*., 2009). Venkatraman *et al*. (2010) fond that, the value of heat release rate reduced with increase in PME blends and the value of heat release rate was 102J/CA for diesel and heat release rate was 81.6J/CA for PME20 (Venkatraman and Devaradjane, 2010).

Sanjay Patil (2012) found that, decrease in compression ratio increases heat release inpremixed phase, however occurrence of maximum heat release moved away from TDC. That may bedue todecrease in compression ratio increased the ignition delay period, which caused more fuel to burn late in the expansion stroke. Same trend was found for all the test fuels. Increase in proportion of biodiesel increased the cetane number of blend, decreased the delay period. Decrease in delay period burned less amount of fuel in premixed phase, hence decrease in net heat release rate was found at every compression ratio (3). P. Navaneetha Krishnan *et al*. (2015) found that,the maximum heat release rate for fuel blends B10, B20, B40, B60 and standard diesel measured to be17.23, 17.61, 17.45, 16.10, 17.92 J/°CA. The heat release rate decreased at the start of combustion and increased further. That variation was due to the air entrainment combined with lower air-fuelmixing rate and effect of viscosity of the fuel blends. The heat release rate for B20 blends quite similar to that of standard diesel, whereas other blends deviated more from that of standard diesel. The heat release rate of tamanu oil blends decreased compared to that of diesel at full load. The heat release rate of standard diesel was higher than oil blends due to its reduced viscosity and better spray formation (Navaneetha Krishnan and Vasudevan, 2015).

Cumulative Heat Release Rate

Venkatraman M. *et al*. (2010) found that, the cumulative heat release rate was increased for PME20 (1398J) compared to diesel (1458J). That may be due to higher exhaust gas temperature and NOx emission (Venkatraman and Devaradjane 2010).

Combustion Duration

Navaneetha Krishnan *et al*. (2015) found that, the time duration measured from the beginning of the heat release to the end of heat release. While compared with standard diesel, the total duration of combustion was shorter for biodiesel and diesel blends. Due to lower calorific value, a higher quantity of fuel was required to keep the engine speed stable at different loads for biodiesel blends. The combustion duration for the fuel blends B10, B20, B40, B60 and diesel at full load condition were 36.37, 35.38, 34.17, 29.85 and 38.94°CA respectively. Due to the efficient combustion of the injected fuel, the combustion duration for fuels gets decreased (Navaneetha Krishnan and Vasudevan, 2015).

Mass Fraction Burnt

Eknath *et al*. (2014) found that, at compression ratio of 18 it was found that start of combustion occurred at earlier stage for K20J40D fuel, 21° bTDC, i.e.33.33% earlier than diesel, 38.1% earlier than jatropha and 47.62% earlier than karanja. It was found that 90% of the mass was burn at about 14.44° a TDC whereas for other fuel same mass was burn at 18° aTDC and more. That indicated early burning property of the blend. As compression ratio decreased to 16 for the fuel K20J40D fuel combustion started at 18° bTDC. Thatwas mainly due to decrease in temperature of the charge (Eknath and Ramchandra, 2014). Navaneetha Krishnan *et al*. (2015) found that, at full load conditions the mass fraction burnt of blends was slightly higher than that of standard diesel. Due to the oxygen content of the fuel blends the combustion was sustained in the diffusive combustion phase. At crank angle 340°-360°, the mass fraction burnt for the fuel blend B40 was higher than the standard diesel. But at crank angle 360°-390°, the mass fraction burnt was slightly closer to each other (Navaneetha Krishnan and Vasudevan, 2015).

Indicated Mean Effective Pressure (IMEP)

Navaneetha Krishnan *et al*. (2015) found that, at lesser load condition the indicated mean effective pressure for blend B40 increased but lower at higher load while compared with standard diesel. At 50% and 75% loading conditions the blend B40 closely followed standard diesel values. The indicated mean effective pressure for blend B40 and standard diesel at 50% load was 1.908 bar, 1.919 bar and for 75% load was 2.834 bar and 2.841 bar respectively (Navaneetha Krishnan and Vasudevan, 2015).

Ignition Delay

P. Navaneetha Krishnan *et al*. (2015) found that, the ignition delay decreased with biodiesel in the diesel blends with increase in load. The ignition delay period for B10, B20, B40 and B60 at full load condition were 5.34, 1.61, 3.68and 3.39/°CA respectively that were higher than diesel. It was mainly due to maximum cylinder pressure higher temperature and higher cetane number (Navaneetha Krishnan and Vasudevan, 2015).

Mean Gas Temperature

Eknath *et al*. (2014) found that, at a compression ratio 18, mean gas temperature was low forK20J40D. That temperature was 6.01% less than diesel, 0.8% less than jatropha, 41.82% less than karanja and 47.83% less than KJD20 and 6.37% less than K20J60D. That low temperature resulted in to low NOx emission and even combustion efficiency was improved due to complete combustion. As the compression ratio decreased to 16 jatropha had lower mean gas temperature compared with other fuels. K20J40D fuel temperature was 766.46°Cwhich was 48-49% less than any other fuel except jatropha. It was found that for both compression ratios for fixed content of karanja (20%) with increase in content of jatropha mean gas temperature was decreased first and again increased, it was found to be minimum when jatropha content was 40% by volume. That indicated that the fuel K20J40D could be low emission environmental friendly fuel for Indian rural requirements where karanja and jatropha could easily available (Eknath and Ramchandra, 2014).

Emission Characteristics

CO Emissions

Anand *et al*. (2009) found that, reduced CO emissions at higher loads when running on biodiesel blends at increased compression ratios from 15:1 to 19:1. That was typical with all internal combustion engines since air/fuel (A/F) ratio decreased with increased load. Increase in CO emissions as the load was increased up to 50% when running on biodiesel blends and then decreased at the BMEP of 3.3 bar and 4.4 bar. The diesel fuel operation produced the highest CO emissions at high load levels and when the engine load reached a certain BMEP (4.4 bar), the CO emissions start to increase more rapidly for both diesel and biodiesel blends.At high compression ratio of 19:1, the highest CO emission of 0.69% by volume was measured for diesel fuel, and the lowest of 0.13% was found for blend B10. The CO emissions for blends B5, B15, and B20 were 0.27%, 0.18% and 0.37% by volume, respectively. Reduced CO emissions were maintained, probably, owing to the oxygen inherently present in biofuels (Anand *et al*., 2009).

Venkatraman M. *et al*. (2010) found that, CO emission of 0.49%vol for diesel and 0.42%vol for PME20 which was maintained due to presence of oxygen in the biofuels. CO emissions decreased with increase in PME in the blends had sufficient time for combustion process because of advanced injection timing (Venkatraman and Devaradjane, 2010). Sejal Narendra Patel *et al*. (2012) found that, at entire load range of including 100% biodiesel had lesser emission of carbon monoxide than diesel. That decrease may be because of higher oxygen content in biodiesel which caused the complete combustion (Sejal Narendra Patel and Ravindra Kirar, 2012). Pavanendra Kumar *et al*. (2013) found that, the presence of CO in the exhaust of an engine was representation of the chemical energy of the fuel that was not fully utilized. Carbon monoxide is a colorless and odorless but a poisonous gas. Generally, the CO emission was affected by the equivalence ratio, fuel type, combustion chamber design and atomization rate, start of injection timing, engine load and speed. The most important among these parameters was the equivalence ratio. The CO emission decreased as the load on an engine increased. That was a typical result for internal combustion engines because the combustion temperature increased with the engine load and CO emission reduced. CO emission decreased as load increased and it had been found that CO emission decreased as the compression ratio increased. That could be explained with the help of HC emission at higher compression ratio i.e. higher the HC emission, lower the CO emission. The temperature reached inside the cylinder was low at low compression ratio, so more CO was exhausted from the engine (Pavanendra Kumar *et al*., 2013). V. Nageswara Reddy *et al*. (2013) found that, at CR-12 at full load condition lowest carbon monoxide of about 336 ppm was found for CB10 followed by CB20 (489 ppm), diesel (510 ppm) and highest of about 961 ppm for CB40. That could be attributed to the longer ignition delay found for CB40. Longer ignition delay along with increased brake specific fuel consumption decreased the air-fuel ratio inside the cylinder leaving less amount of air for complete combustion which in turn gives rise to higher CO emissions. At CR-14, CB10 and CB20 had carbon monoxide emission on the lower side whereas diesel and CB40 had emissions on the higher side (Nageswara Reddy*et al*., 2013).

Santosh Kumar Kurre *et al*. (2013) found that, there was no effect of CR on the emission of CO. while compared to neat diesel it was slightly higher for E5 and E15. When ethanol was added with diesel it causes reduction in gas temperature which restrains the oxidation of CO, so CO emission slightly increased (Santosh Kumar Kurre*et al*., 2013). Jalpit B. Prajapati *et al*. (2014) found that, CO was usually formed when there was no sufficient O_2 to oxidize the fuel. CO increased with decreased in compression ratio. But CO decreased with increased in loads. From the different blends, B30 had better performance as compared to B0. Lower CO emissions could be attributed to the combined effect of their oxygen content and higher cetane number. A higher cetane number exhibited a shorter ignition delay and allowed longer combustion duration. Then the oxygen content of biodiesel comes into play, which enhanced the combustion process (Jalpit *et al*., 2014). Pavan *et al*. (2014) found that, at CR-13.5 the CO emissionof the blend B25 was less than the standard diesel at full load condition and it was found to be higher for the blend B15. Emission for the blend B25 and diesel came exactly the same at 3/4th load.

The percentage of CO increased due to rising temperature in the combustion chamber, physical and chemical properties of the fuel, air-fuel ratio, shortage of oxygen at high speed. At CR-16.5, The CO emission of the blend B15 and B25 was less than the standard diesel at CR-16.5 for full load condition and at half load emission had been slightly increased for blend B15 the percentage of CO increased due to less amount of time available for complete combustion. At CR-18, The CO emission of the blend B15 and B25 was less than the standard diesel at CR-18 for full load condition and at half load emission had been slightly increased for blend B15 the percentage of CO increased may be due to poor vaporization and atomization (Pavan *et al*., 2014). R. D. Eknath *et al*. (2014) found that, for a compression ratio of 18 it was found that at low load 2 kg CO emission was higher, with increase in load emission reduced. That may because of increase in load increased the temperature and hence combustion was complete. At a low load of 2 kg for karanja fuel CO emission reduces by 50% compare with diesel fuel and for jatropha fuel it reduced by 12.5%. For the blend KJD20 had same emission as that of diesel fuel. For a blend KJD40 and K20J60D about 75% emission was reduced compare with diesel and jatropha fuel. Compare with karanja reduction was found to be 50%. For K20J40D fuel CO emission was found to be zero for the entire load range. Higher oxygen content of and lower carbon to hydrogen ratio of a biodiesel may tend to reduce CO emission. Also it was found that reduction in compression ratio increased CO emission for the fuels. That was mainly because of decrease in mean gas temperature (Eknath and Ramchandra, 2014). P. Navaneetha Krishnan *et al*. (2015) found that, the carbon monoxide emission of the blend B40 was found to be higher for light and medium loads and closer to that of standard diesel. Due to rising temperature in the combustion chamber, air fuel ratio, lack of oxygen at high speed, physical and chemical properties of fuel and smaller amount of time available for complete combustion, the proportion of carbon monoxide emission increases. The carbon monoxide emission increased for vegetable oil fuels due to the effect of fuel viscosity on the fuel spray quality (Navaneetha Krishnan and Vasudevan, 2015). R. Silambarasan *et al*. (2015) found that, the CO gradually decreased with increased compression ratio. At maximum load with the compression ratio of 19.5, the CO for A20 was 0.03% and 40% reduction of CO was achieved when compared with the neat diesel fuel. That may be due to complete combustion, less dilution of charge by residual gases accelerated the carbon oxidation to form carbon dioxide (Silambarasan *et al*., 2015). Venkateswara Rao P. (2015) found that, CO emission decreased with increased both CR and brake power. The blend fuel CO emission was 32.6% less compared to diesel at CR-20:1 for blend fuel at full load. At higher CR, combustion rate of fuel increased due to higher temperature and adequate turbulence was created in the combustion chamber to complete combustion, hence the emission of CO decreased (Venkateswara Rao, 2015).

Biju Cherian Abraham *et al*. (2015) found that,generally CO emissions were affected by start of injection timing, injection pressure, physical and chemical properties of the fuel used, engine load, speed and improper air-fuel mixing. J20 and JR20 produced lower CO than diesel. Biodiesel contained 10-11% oxygen content which promoted more complete combustion resulting lower CO emissions. Effects of biodiesel on CO emissions vary significantly among vehicles, engine technology, test cycle and the feed stocks used to produce the biodiesel (Biju Cherian Abraham and Chindhu Prasad, 2015).

V. Hariram *et al*. (2015) found that, CO emissions increase with increased load due to the presence of rich fuel air mixture and this trend was found at all CR. CO emissions reduced with biodiesel blend due to the presence of oxygen content where carbon monoxide gets oxidized. The CO emissions reduced by around 16% with B20 blend at standard setting. On increasing the CR from 16 to 18, CO emissions reduced by 10% and 8% diesel and B20 blend at IT 23° bTDC and reduction of about 10% and 16% for diesel and B20 was found for same IT on increasing CR from 16 to 17. An overall increase of 20% was found when CR was increased from 16 to 18 for various IT and blends. CO emissions reduced by 15.5%, 14% and 12% on advancing the IT at CR 18, 17 and 16 for diesel and for B20 the reduction was 18%, 17% and 14% respectivily which may be due to reduced ignition delay of the combustion process. Increase in CO emissions by 14.1%, 13.5% and 15% was found on retarding the IT at CR 18, 17 and 16 for diesel and for B20 the increase was 14.5%, 11% and 16.5% (Hariram and Kumar, 2015).

HC Emissions

R. Anand *et al*. (2009) found that, the emission of unburnt hydrocarbons (HC) for all fuels and compression ratios was small, 15-88 ppm, increased slightly with load and proportion of fuel injected. It was quite difficult to determine any reliable dependencies; however, the HC emissions for biofuels proceed a bit lower levels than that of diesel fuel. The COME operation showed very low (about a factor of 2) HC emissions throughout the load range. HC emissions peaked at a particular A/F ratio and then dropped with further increase in load. Increase in HC emissions as the quantity of diesel fuel in the blend increased. A very significant difference was found at low and high load conditions at the compression ratio 19:1. The increase in diesel/COME blend showed highest HC concentration in the exhaust when compared with the results of increase in COME/diesel blend. However at light load, the COME diesel blends did not show any marked difference in HC emissions irrespective of compression ratios. For minimum load the reduction was practically unaffected by the addition of biodiesel in the diesel fuel at all compression ratios. The most beneficial reduction appeared at intermediate loads (Anand *et al*., 2009).

Venkatraman *et al*. (2010) found that, diesel had the maximum rate of hydrocarbon 36ppm amongthe tested fuels. It was also found that the hydrocarbon of 29ppm for PME20 decreased with increase in concentration of the biodiesel blends. That may be due to improved combustion because of increased in injection pressure and advanced injection timing (Venkatraman and Devaradjane, 2010). Sejal Narendra Patel *et al*. (2012) found that,the emission of biodiesel was considerably less compare to diesel. At compression ratio 18 the emission of hydrocarbon of diesel and biodiesel were 38 ppm and 19 ppm respectively. The higher cetane number of biodiesel and oxygen availability of fuel was responsible for that decrease (Sejal Narendra Patel and Ravindra Kirar, 2012). Pavanendra Kumar *et al*. (2013) found that, the unburned fuel component present in exhaust of an engine consists hydrocarbon component was termed as HC emission. These hydrocarbons consist of small non-equilibrium molecules, which were formed when large fuel molecules breaks up by thermal cracking during combustion reactions. The major cause of HC emission was non-homogeneity of fuel-air mixture. Due to that non-homogeneity some local zones in combustion chamber will be too lean to combust properly and other zones may be too rich with not enough oxygen to burn all the fuel. HC emission decreased as the load increased. Generally HC emission from exhaust was measure of unburnt fuel in the exhaust of an engine. HC emission using biodiesel was lower than diesel. The test was performed at compression ratios 18, 16, and 14 and it was found that HC emission of B50 was minimum at CR-18. That was due to the fact that the cetane number of ester based fuel was higher than diesel and so results in better combustion leading to lower emission (Pavanendra Kumar *et al*., 2013).

V. Nageswara Reddy *et al*. (2013) found that, at CR-12 diesel and CB20 showed same hydrocarbon emissions from no load to full load condition. That was due to their nearly same brake specific fuel consumption. Lowest hydrocarbon emission of about 10 ppm was foundfor CB10 at part load condition. Maximum hydrocarbon emission of about 50 ppm was found for CB40 at full load condition. Higher hydrocarbon emission for CB40 was due toits higher brake specific fuel consumption. At CR-14, diesel and CB20 showed same hydrocarbon emissions from no load to full load condition. Maximum hydrocarbon emission of about 50 ppm wasfound for CB40 at full load condition (Nageswara Reddy *et al*., 2013).

Santosh Kumar Kurre *et al*. (2013) found that, HC decreased with the compression ratio increased. When the percentage of ethanol increased in the blend HC also increased. E10 showed 44.2%, increase of HC for CR-17, 62.2% higher HC than the pure diesel for CR-18. E20 showed increase in HC on ranges 56-66% than the diesel fuel for increased CR. High latent heat of vaporization of ethanol tends to produce slow vaporization and mixing of fuel and air. In homogeneity of mixture may be other reason of increasing HC (Santosh Kumar Kurre *et al*., 2013). Jalpit B. Prajapati *et al*. (2014) found that,the concentration of HC in diesel exhaust varies from few ppm to several thousand ppm depending upon the load on the engine. The hydrocarbons in diesel exhaust were composed of mixture of many individual hydrocarbons in the fuel supplied to the engine as well as partly burned hydrocarbons produced during combustion process. HC emission increased with decreased in compression ratio. HC emission was decreased as compared to diesel. The increased gas temperature and higher cetane number of biodiesel and their blends were responsible for that decrease (Jalpit *et al*., 2014).

Navaneetha Krishnan *et al*. (2015) found that,at higher load condition the hydrocarbon emission of various blends were higher except the blend B20. In vegetable oil fuels, the effect of fuel viscosity and the fuel spray quality had been expected to produce some increase in hydrocarbon content in emission. Blend B40 increase in load increase the hydrocarbon emission. The other blends B10, B20 and B60 produce lesser hydrocarbon emission at 50% and 75% load while compared with standard diesel. Higher hydrocarbon emission was due to the longer ignition delay that leads to the accumulation of fuel in the combustion chamber (Navaneetha Krishnan and Vasudevan, 2015). Silambarasan *et al*. (2015) found that,the HC gradually decreased with increased compression ratio. At maximum load with the compression ratio of 19.5, the HC for A20 was 23ppm and 23.33% reduction of HC was achieved when compared with the neat diesel fuel. That was due to increase the air temperature at the end of compression stroke, enhancement in combustion temperature and reduction in charge dilution leads to complete combustion and reduction in hydrocarbon emissions (Silambarasan *et al*., 2015).

Venkateswara Rao (2015) found that,the emission decreased with increase in CR for all loads. The blend fuel emission was 26.7% less compared to that of diesel at maximum load for 20:1 CR. The HC emission of blend fuel at maximum load was 54.3% decreased with increase in CR, that may bedue to the presence of oxygen molecules present in the blend fuel improves the combustion (Venkateswara Rao, 2015). Biju Cherian Abraham *et al*. (2015) found that,the fuel viscosity, fuel spray quality and characteristics also the incomplete combustion affects the HC emission. Unburnt hydrocarbon emission increased with that of load for all prepared test fuels. Biodiesel produced less HC emission in comparison to that of diesel because of better combustion of the test fuel and its blend due to presence of oxygen (Biju Cherian Abraham and Chindhu Prasad, 2015). V. Hariram *et al*. (2015) found that, UBHC decreased with increased loads which could have been due to the adequate amount of oxygen in the fuel mixture enhancing reduced hydrocarbon emissions at higher loads. UBHC emissions reduced with biodiesel blend compared to diesel which was found at all CR. UBHC reduced by 11.5%, 10.8% and 15% at CR 18, 17 and 16 with B20 blend. On increasing the CR from 16 to 18, UBHC reduced by 36% and 33% for diesel and B20 blend and the reduction was around 30% and 28% when CR was increased from 17 to 18 at all loads. The reduction of UBHC steadily at full load conditions with increased CR. On advancing the IT the UBHC reduced by 25%, 19% and 11% for diesel at CR 18, 17 and 16 and the reduction was around 18%, 16% and 12% for B20 blend at CR 18, 17 and 16 respectivily due to the better and complete combustion. Retarding the injection timing increased the UBHC emissions by 18.5%, 17.3% and 13.4% for diesel and by 19%, 21.3% and 20.1% for B20 blend at CR 18, 17 and 16 (Hariram and Kumar, 2015).

CO_2 Emissions

Pavanendra Kumar *et al*. (2013) found that, the higher CO_2 emission in the exhaust of internal combustion engine was indication of better combustion of fuel. The CO_2emission increases as the load on an engine increased. That may be due to the fact that at higher loads the combustion temperature increased which helped in complete combustion of the fuel.CO_2 emission increased as the load increased. Amount of CO_2 in the exhaust was an indication of the combustion of the fuel inside the cylinder. More amount of CO_2 in exhaust means better combustion. CO_2 emission increased with increase in compression ratio. That may be due to better combustion and intermixing of fuel and air at higher compression ratio (Pavanendra Kumar *et al*., 2013). Nageswara Reddy *et al*. (2013) found that, at CR-12 from no load condition to full load condition CB10 ha lower value of carbon dioxide. CB40 showed higher value of carbon dioxide emission at all load condition. At full load condition lowest value found was about 25% for CB10 and highest of about 37% for both CB20 and CB40. At CR-14, from no load condition to full load condition CB10 and CB20 had the same type of variation. Only difference was that carbon dioxide emissions for CB10 were on the higher side of the graph, whereas CB20 had on the lower side. Carbon dioxide emissions for diesel were found higher than both CB20 and CB40 (Nageswara Reddy *et al*., 2013).

Santosh Kumar Kurre *et al*. (2013) found that, carbon dioxide decreased as the compression ratio increased. There was 35.4% reduction for E5 and E15while 30% reduction of E20 if compared with neat diesel for CR-17. Ethanol had higher oxygen contain and provide more oxygen for combustion may lead to reduction of CO_2 (Santosh Kumar Kurre *et al*., 2013). G. Pavan *et al*. (2014) found that, at CR-13.5 theCO_2 emission for B15 and B25 had been increased than that of diesel and at full load the emissions of diesel were close to that of blends. The reason for increase in CO_2 may be due to efficient combustion inside the combustion chamber. At CR-16.5, The CO_2emission for B15 and B25 had been increased than that of diesel and at full load the emissions of diesel were close to that of blends. The reason for the increase in CO_2 may be due to increase in compression ratio from 13.5 to 16.5 the CO Emission reacted with extra oxygen and thus resulted in high CO_2. At CR-18, Initially B25 Emission was less than that of diesel and by increased the loads the CO_2increased gradually and thus resulted in higher emissions than the diesel. The reason for the increase in emission at CR-18 may be due to abundant availability of oxygen in the air (Pavan *et al*., 2014). Eknath *et al*. (2014) found that, compared with diesel fuel CO_2emission for jatropha and karanja was found to be lower by about 10 to 15%.Thatwas mainly due to the fact that biodiesel had low carbon to hydrogen ratio compare with diesel fuel. For the blend K20J40D the emission was by about 25 to 30% with increase in load compare with diesel, jatropha and karanja fuel. That may be due to improved combustion characteristics (Eknath and Ramchandra, 2014).

Navaneetha Krishnan *et al*. (2015) found that, the complete combustion of fuel in the combustion chamber leads to increased emission of carbon dioxide. The carbon dioxide emission changed with exhaust gas temperature. Due to incomplete combustion and inadequate supply of oxygen carbon dioxide emission of the fuel blends B40 at full load decreased. The increased emission of carbon dioxide in the atmosphere leads to several environmental problems like global warming and ozone layer depletion. The carbon dioxide emission from the combustion of biofuel blends could be intake by the plants and so the carbon dioxide level was kept constant in the atmosphere (Navaneetha Krishnan and Vasudevan, 2015). Venkateswara Rao, (2015) found that, CO_2 emission increased with increased CR for all the loads. CO_2 emission increased with increase in brake power. At 20:1 CR the CO_2emission of blend fuel was 3.2% higher compared to that of diesel at maximum load. At higher compression ratio CO_2emission was 28.6% higher for blend fuel at maximum load due to improved combustion rate of bend fuel (Venkateswara Rao, 2015). Biju Cherian Abraham *et al*. (2015) found that, due to lower calorific value of biodiesel more fuel was needed to produce the same power and also due to fuel-borne oxygen in biodiesel, more carbon was oxidized to CO_2 than partial oxidation of carbon to CO. JR20 had the highest level of CO_2 emission (Biju Cherian Abraham and Chindhu Prasad, 2015).

NO Emissions

Anand *et al*. (2009) found that, the emission of nitric oxide (NO) increased gradually with load, reached the maximum values of 10-205 ppm for COME-diesel blend and 16-153 ppm for diesel fuel at high loads. When operating at the low CR of 15:1, the minimum NO emissions at adequate loads were maintained by the B20 blend. That blend suggested NO emissions lower by 6.12% to 37.5% relative to diesel fuel. Biofuels with higher oxygen contents, 1.45% to 4.075% and neat biodiesel produce higher or a bit lower NO emissions, depending on the load. The reason for the decrease in NO was the cetane number of the biodiesel was higher than that for diesel fuel and that was associated with lower NO emissions. Increased cetane number reduced the size of the premixed combustion by reduced the ignition delay. That resulted in lower NO formation rates since the combustion pressure rises more slowly, giving more time for cooling through heat transfer and dilution and leading to localized gas temperatures. By reducing aromatics the flame temperature will drop, leading to a lower NO production rate. As a result, the aromatics have high carbon-hydrogen ratios and thus fuels with lower aromatics will lead to a smaller amount of $CO_2$and larger amount of H_2O being formed compared to high aromatic fuels. Since H_2O has a lower tendency to dissociate at high temperatures (compared to CO_2), that will lead to low aromatic fuels having lower concentrations of O radical and O_2(from radical-radical recombination) which will further reduce the kinetic production of NO. Furthermore, if the final yield of NO was determined by the high-temperature equilibrium between oxygen, nitrogen versus NO, the lower high temperature O_2 concentrations will lead to lower equilibrium concentration of NO (Anand *et al*., 2009). Nageswara Reddy *et al*. (2013) found that, at CR-12 from no load condition to full load condition CB10 had lower value of nitrogen oxide as NO. At full load condition highest value obtained was about 278 ppm for CB20 followed by CB40, diesel and lowest of about 213 ppm for CB10. At CR-14, from no load condition to full load condition CB40 had lower value of nitrogen oxide as NO. At full load condition highest value obtained was about 310 ppm for CB10 followed by CB20, diesel and lowest of about 148 ppm for CB40 (Nageswara Reddy *et al*., 2013).

NO_2 Emissions

V. Nageswara Reddy *et al*. (2013) found that, at CR-12 from no load condition tofull load condition diesel had lower value of nitrogen oxide as NO_2. CB10 also had values nearly same as that of diesel. At full load condition lowest value found was about 29 ppm for both CB40 and diesel. Lowest value of about 75 ppm was found for CB20. At CR-14, higher amount of NO_2emissions were found for CB10 and diesel. Lowest value of NO_2 emissions were found for CB40. CB20 showed intermediate NO_2 emissions (Nageswara Reddy *et al*., 2013).

NOx Emissions

Venkatraman *et al*. (2010) found that, NOx emissions increases for PME20 (1238ppm) compared to diesel (1096ppm).Due to the advancement of injection timing and pressure all the injected fuel burnt as a result higher combustion temperature was attained. The higher temperature promotes NOx formation (Venkatraman and Devaradjane, 2010). Sejal Narendra Patel *et al*. (2012) found that, the emission of nitrogen oxide for diesel and biodiesel 1093ppm and 960ppm.The emission of nitrogen oxide decreased for biodiesel at full load condition but slightly increase at part load condition. During combustion chemical reaction which made nitrogen oxide were takes place at higher temperature. The exhaust gas temperature remained almost same for all the blends (Sejal Narendra Patel and Ravindra Kirar, 2012). V. Nageswara Reddy *et al*. (2013) found that,at CR-12 at full load condition highest value found was about 353 ppm for CB20 followed by CB40, diesel and lowest of about 245 ppm for CB10. Both CB20 and CB40 showed higher nitrogen oxide emissions than diesel. At CR-14, at full load condition highest value found was about 385 ppm for CB10 followed by CB20, diesel and lowest of about 166 ppm for CB10 (Nageswara Reddy *et al*., 2013). Santosh Kumar Kurre *et al*. (2013) found that,as the compression ratio increased the NOx increased for neat diesel. E10 showed 33% reduction in the NOx emission compared between low compression ratio and higher compression ratio E15 showed 35.8% increase in NOx than diesel fuel for lower compression ratio. The NOx emission was decrease with the addition of ethanol. As the ethanol was an oxygenated fuel and it had higher heat of evaporation because of that it reduced the combustion temperature, caused NOx reduction (Santosh Kumar Kurre *et al*., 2013).

Jalpit B. Prajapati *et al*. (2014) found that, as the compression ratio decreased NOx emission increased. NOx emission of all blends increased as compared to diesel (Jalpit *et al*., 2014). Pavan *et al*. (2014) found that, at CR-13.5 nitrogen oxideemission increase with increase of load. The minimum nitrogen oxide value was 101ppm for diesel and 195ppm for B25. That was a result of low availability of oxygen during combustion. At CR-16.5, the nitrogen oxide emissions were higher for blends of hibiscus oil compared with diesel. The lowest value of NOxwas 120ppm for diesel at full load as compared to B25 with176ppm. That result depends on oxygen quantity and fuel viscosity, in turn atomization. NOxemissions were mainly governed by the magnitude of peak cylinder temperature and crank angle at which it occurred. At CR-18, the nitrogen oxide emissions were lower for all the blends of hibiscus oil compared with diesel. The lowest value of NOxwas 98ppm for diesel at full load as compared to B25 with 180ppm. The reason for increase in NOxemissions was high temperature and better availability of oxygen. The oxygen contain in the biodiesel facilitate in the oxidation of the nitrogen present in the air resulted in the formation of the NOx (Pavan *et al*., 2014).

R. D. Eknath *et al*. (2014) found that,increase in load increased the combustion temperature that increased NOx. It was found that for jatropha and diesel fuel NOx emission was same. For karanja fuel for a load range, NOx emission was low about 20 to 22% compared with diesel fuel. For blends K20J40D at maximum load of 10 kg NOx emission was 70% lower than diesel and jatropha fuel. That may bedue to the fact that the mean gas temperature for this fuel was 807°C whereas the mean gas temperature of diesel fuel was 858°C. Due to low combustion temperature NOx emission was low at compression ratio 18. Decreased the compression ratio to 16 NOx emissions was 64% less than diesel fuel. Improved combustion and low mean gas temperature may be the cause of reduction in NOx (Eknath and Ramchandra, 2014). Navaneetha Krishnan *et al*. (2015) found that, the nitrogen oxide emission for standard diesel was lower than that of biodiesel and its blends except B40 at lower loads. Usually vegetable based fuel contains a small amount of nitrogen. That leads to the nitrogen oxide production. For 50% load, nitrogen oxide emission from the tamanu oil blend B40 was slightly lower than that of standard diesel. But in case of 100% load condition the nitrogen oxide emission from the B40 blend was higher than that of standard diesel, while the other blends close follows the standard diesel. Due to the higher peak temperature higher nitrogen oxide emission for fuel blends will occur. The nitrogen oxide emission for fuel blend B40 and standard diesel for 50% load was 18 ppm and 22 ppm respectively (Navaneetha Krishnan and Vasudevan, 2015). Silambarasan *et al.* (2015) found that, the NOx gradually increase with increased compression ratio. At maximum load with the compression ratio of 19.5, the NOx for A20 was 740ppm and it was almost equal to the neat diesel fuel 734ppm. That may be due to increase in compression ratio increased the combustion pressure and temperature which accelerated the oxidation of nitrogen to form oxides of nitrogen (Silambarasan *et al*., 2015). Venkateswara Rao (2015) found that, the NOx emission increased with increase in CR for all loads. The blend fuel NOx emission was 9.8% less compared to that of diesel at maximum load for 20:1 CR. NOx emission of 63.2% increased for blend fuel at maximum load and increased in CR. Production of NOx depends upon the maximum temperature in the cylinder and concentration of oxygen. It was found that the oxygen concentration in exhaust gas at higher CR was less, hence lower the NOx formation (Venkateswara Rao, 2015). Biju Cherian Abraham *et al*. (2015) found that,the production of NOxwas significantly influenced by the in-cylinder gas temperature, availability of oxygen and residence time. NOx emission increased almost linearly with increase in engine load which may bedue to higher cylinder pressure and temperature at higher loads. Because of higher oxygen content the J20 had highest rate of NOx production. Proper oxygen content and appropriate temperature was necessary for complete combustion (Biju Cherian Abraham and Chindhu Prasad, 2015).

Hariram *et al*. (2015) found that, NOx emissions increased with increased load which was due to the high flame temperature in the combustion chamber. NOx emissions increased overall with B20 blend by 19% at standard setting due to the higher oxygen content of biodiesel blend and better combustion. NOx emissions increased by 35% and 27% when CR was increased from 16 to 18 and 16 to 17 respectivily at IT 23. At full load conditions, about 15.7% increase in NOx emission was found at all IT and blends. Advanced the IT increased the NOx emissions where the peak pressure and peak temperature were higher due to advanced start of combustion. The increase in NOx emission was 16%, 16.5% and 21% for diesel when IT was advanced at CR 18, 17 and 16 respectivily and for B20 blends the increase was 12.2%, 13.5% and 22.5%. The retarding of IT reduced the peak pressure and temperature enabling reduction in NOx emission where NOx emissions reduced by 12.5%, 13.8% and 23% at CR 18, 17 and 16 for diesel and the reduction for B20 was 12%, 16.5% and 21% respectivily (Hariram and Kumar, 2015).

O_2 Emissions

Pavanendra Kumar *et al*. (2013) found that, the oxygen emission decreased with increase in load. That may be due to better combustion at higher loads. Oxygen emission of biodiesel (B50) was more than diesel that may be due to the fact that the biodiesel contains nearly 10% inbuilt oxygen. Similar trend was obtained on increasing the compression ratio this is again due to nearly complete combustion of fuel (Pavanendra Kumar *et al*., 2013).

Smoke Opacity

Anand *et al*. (2009) found that, soot emitted by all biodiesel blends was lower than neat diesel at low loads and lower compression ratio. That was attributed to the combustion being mixed controlled for these blends, as also the case for neat diesel, which was assisted by the presence of the fuel bound oxygen. There was no definite trend found in smoke density with increase in blend percentage of biodiesel. At all compression ratios, B15 was found to emit maximum smoke at full load. The minimum and maximum smoke opacities produced for B5 and B20 at 15:1 were 10.3% and 65.8% with a maximum and minimum reduction of 84% and 33% respectively, as compared to diesel. that trend continues for 17:1 and 19:1 CR for all the loads but the decrease in smoke opacity was comparatively higher for both compression ratios and that may be due to increase in cylinder temperature. At maximum load reduction was low, whereas the most beneficial reductions appeared for the 75% load. The reason for that behavior was the different amount of sulphur between the diesel and biodiesel blends. Smoke emissions generally increase or decrease in relation to the sulphur concentration. Sulphur in the fuel, results in sulphates that were absorbed on soot particles and increase the smoke emitted from diesel engines. In addition, the increase of oxygen content in the fuel contributed to a complete fuel oxidation even in locally rich zones, leading to a significant decrease of smoke (Anand *et al*., 2009).

Pavanendra Kumar *et al*. (2013) found that, formation of smoke was basically a process of conversion of molecules of hydrocarbon fuels into particle of soot. The soot was an agglomeration of very large polybenzenoid free radicals. The soot formation takes place during early part of actual combustion but it was consumed during later part of combustion. Pyrolysis of fuel molecules themselves thought to be responsible for soot formation. The fuel heated with insufficient oxygen will give carbonaceous deposits. Among the particulate matter components, soot was recognized as the main substance which was responsible for the smoke opacity. Smoke opacity formation occurred at the extreme air deficiency. The air or oxygen deficiency was locally present inside diesel engines. The significant reduction in smoke emission for the biodiesel (B50) may be due to the oxygenated blends. The 10% inbuilt oxygen provided better and nearly complete combustion. Smoke was mainly produced in the diffusive combustion phase, the oxygenated fuel blends lead to an improvement in diffusive combustion for biodiesel. Further, increasing compression ratio, in general, reducedsmoke due to better combustion. For all loads, the smoke emission was lowest for B50 and at CR of 18 (Pavanendra Kumar *et al*., 2013). Shelke, (2013) found that,100% diesel oil had lower smoke density than the other blends and esterified soyabean oil. Smoke density in case of the methyl ester of soyabean oil was in between diesel oil and soyabean oil-diesel oil blends. Smoke density in case of CR-16 was less than that of in CR-18 (Shelke, 2013).

Santosh Kumar Kurre *et al*. (2013) found that, smoke reduced as compression ratio increased. When compared with diesel fuel smoke increased for all blends for lower CR while for higher CR smoke reduced drastically. The oxygenated fuel had the oxygen atom strongly connected to carbon atom since it was difficult to break the bond between them. That restrained the formation of aromatic hydrocarbon and black carbon. So ethanol may provide oxygen atom in the fuel rich region and inhibit the formation of smoke (Santosh Kumar Kurre *et al*., 2013).
Silambarasan *et al*. (2015) found that, the smoke gradually decreased with increased compression ratio. At maximum load with the compression ratio of 19.5, the smoke for A20 was 17.6HSU and 21.4% reduction of smoke was achieved when compared with the neat diesel fuel. That may be due to biodiesel consists of two oxygen atoms which lead to the oxidation of soot and thereby reducing the soot emission (Silambarasan *et al*., 2015). Venkateswara Rao P. (2015) found that, the smokedensity was decreased with increase in CR up to the brake powerof 1.8kW and increased thereafter. For blend fuel smoke was 9.2% less compared to diesel at maximum load for 20:1 CR. The blend fuel smoke emission was 20.11% and lower for maximum load by increasing CR from 14:1 to 20:1. Thatmay be due to better oxidation of fuel at higher temperature and pressure attained at higher CR (Venkateswara Rao, 2015). Hariram *et al*. (2015) found that, smoke density increased with increasing load conditions due to higher fuel consumption with more combustion at increased loading. The smoke emissions reduced by an overall 15.6% with B20 blend at standard setting. The smoke emissions reduced by 18% and 16.5% when CR was increased from 16 to 18 and then to 17 for diesel and for B20, it was 17% and 15.7% at standard IT due to more air temperature in the cylinder where complete combustion takes place. At full load condition, about an overall of 23.5% reduction was found at all IT and blends when CR was increased from 16 to 18. On advancing the IT, about 17.2%, 19.2% and 20.2% reduction in smoke was found at CR 18, 17 and 16 and for B20 blend it was 12.1%, 14% and 13.4%. Retarding the IT increased the smoke levels by 19.7%, 21% and 15.7% for diesel at CR 18, 17 and 16 and for B20 blend it was 21.3%, 23% and 22% respectivily (Hariram and Kumar, 2015).

Conclusion

From this review we conclude the following;

- Brake thermal efficiency increased with the increased compression ratio. That was due to the fact that at higher compression ratio the rate of combustion of increased which result in better thermal efficiency.
- Brake specific fuel consumption decreased with the increased compression ratio. That was due to fact that at higher compression ratio the pressure and temperature increased because of that the rate of combustion increases which resulted into decreased BSFC.
- Brake specific energy consumption decreased with the increased compression ratio. That may be due to the lesser percentage intake in fuel consumption with load compared to brake power and relatively lower heat loss at higher loads.
- Exhaust gas temperature decreased with increased compression ratio. That was due to fact that at higher compression ratio the pressure and temperature increased because of that the rate of combustion increased more complete combustion resulted in lower losses during exhaust.
- Brake power decreased with decreased in compression ratio. That may be due to the fact that when the compression ratio decreased, cylinder head slightly tilted. Also clearance volume increased in combustion chamber. Because of that peak cylinder pressure decreased and resulted into decreased brake power.
- Cylinder peak pressure was increased with the increased compression ratio. That may be due to the fact that increase in compression ratio enhances the pressure and temperature of air-fuel mixture in compression stroke resulted in increased peak pressure.
- Decrease in compression ratio increases heat release in premixed phase, however occurrence of maximum heat release moved away from TDC. That may be due to decrease in compression ratio increases the ignition delay period, which causes more fuel to burn late in the expansion stroke.
- CO emission decreased with increasing compression ratio. That may be due to the fact that at higher compression ratio combustion rate of fuel increases due to higher temperature and adequate turbulence was created in the combustion chamber to complete combustion and hence the emission of CO decreases.
- The HC emissions decreased with the increased compression ratio. That was due to the fact that increase the air temperature at the end of compression stroke, enhancement in combustion temperature and reduction in charge dilution leads to complete combustion and reduction in hydrocarbon emissions.
- CO_2 emission increased with the increased compression ratio. That may be due to better combustion and intermixing of fuel and air at higher compression ratio.
- The NOx emissions increased with the increased compression ratio. That may be due to increase in compression ratio increases the combustion pressure and temperature which accelerates the oxidation of nitrogen to form oxides of nitrogen.
- O_2 emissions decreased with the increased compression ratio. That may be due to nearly complete combustion of fuel at higher compression ratios.
- The smoke density was decreased with increase in compression ratio. That may be due to better oxidation of fuel at higher temperature and pressure attained at higher compression ratio.

REFERENCES

Anand, R., Kannan, G. R., Rajasekhar Reddy, K. and Velmathi, S. 2009. "The performance and emissions of a variable compression ratio diesel engine fuelled with biodiesel from cotton seed oil", *ARPN Journal of Engineering and Applied Sciences*, Volume-4, Number-9, Pages 72-87.

Babu, A. R., Amba Prasad Rao, G., Hari Prasad, T. 2015. “Performance and emission characteristics of CI engine with methyl esters of palm stearin under varying compression ratios and fuel injection pressures”, *International Journal of Engineering and Management Research*, Volume-5, Issue-6, Pages 61-69.

Balajee, D., Sankaranarayanan, G., Harish, P. and Jeevarathinam, N. 2013. “Performance and combustion characteristics of CI engine with variable compression ratio fuelled with pongamia and jatropha and its blends with diesel”, *International Journal of Mechanical Engineering and Robotics Research*, Volume-2, Number-3, Pages 74-85.

Biju Cherian Abraham, Chindhu Prasad, 2015. “Performance characteristics of biodiesel mixtures in a variable compression ratio engine with CR 17”,*International Journal of Mechanical Engineering and Technology*, Volume-6, Issue-8, Pages 96-104.

Eknath, R. D. and Ramchandra, J. S. 2014. “Effect of compression ratio on energy and emission of VCR diesel engine fuelled with dual blends of biodiesel”, *Journal of Engineering Science and Technology*, Volume-9, Number-5, Pages 620-640.

Hariram, V., Kumar, G. M. 2015. “Effect of varying compression ratio and injection timing on performance and emission parameters in CI engine using mahua biodiesel”, *Journal of Scientific and Industrial Research*, Volume-74, Pages 695-701.

Jalpit, B. Prajapati, Parth R. Panchal, Tushar M. Patel, 2014. “Investigation on the effect of compression ratio for performance and emission characteristics of CI engine fuelled with diesel-biodiesel blends”, IOSR *Journal of Mechanical and Civil Engineering*, Volume-11, Issue-3, Pages 122-129.

Nageswara Reddy, V., Sreenivasa Rao, G. and Thirupathi Reddy, K., “Study of engine performance and exhaust emission characteristics of a 4-stroke CI engine operated with various blends of biodiesel and rice bran oil”, *International Journal of Engineering Research and Science & Technology*, Volume-2, Number-4, Pages 171-181, 2013.

Navaneetha Krishnan, P. and Vasudevan, D.2015. “Performance combustion and emission characteristics of variable compression ratio engine fuelled with biodiesel”, *International Journal of Chem Tech Research*, Volume-7, Number-1, Pages 234-245.

Naveen, K., Parameshwaran Pillai, T. Azhagiri Pon, 2014. “Experimental investigation of variablecompression ratio diesel engine usingziziphus jujuba oil”, *International Journal of Innovative Research in Science, Engineering and Technology*, Volume-3, Special Issue-3, Pages 1134-1139, 2014.

Pavan, G., Rajesh, M. and Srinivas, K. 2014. “Performance and exhaust emissions using blendson VCR diesel engine with varying compressionratio of hibiscus oil”, International *Journal of Engineering Trends and Technology*, Volume-17, Number-2, Pages 94-102.

Pavanendra Kumar, Niraj Kumar, Vineet Tirth, 2013. “Comparative analysis of performance and emission of diesel engine by varying compression ratio using different fuels”, MIT *International Journal of Mechanical Engineering*, Volume-3, Number-2, Pages 86-90.

Sanjay Patil, 2012. “Investigation on effect of variation in compression ratio on performance and combustion characteristics of CI engine fuelled with palm oil methyl ester (POME) and its blends by simulation”, *Global Journal of Researches in Engineering Automotive Engineering*, Volume-12, Issue-3.

Santosh Kumar Kurre, Shyam Pandey, Mukesh Saxena, 2013. “Effect of compression ratio on diesel engine performance and emission with diesel-ethanol blends”, International Journal of Scientific & Engineering Research, Volume-4, Issue-10, Pages 775-779.

Sejal Narendra Patel and Ravindra Kirar, 2012. “An experimental analysis of diesel engine using biofuel at varying compression ratio”, *International Journal of Emerging Technology and Advanced Engineering*, Volume-2, Issue-10, Pages 385-391.

Shelke, R.E. 2013. “Feasibility study of soyabean oil as an alternate fuel for CI engine at variable compression ratio”, International Journal of Chemical and Physical Sciences, Volume-2, Number-4, Pages 10-15.

Silambarasan, R., Senthil, R., Pranesh, G., Mebin Samuel, P. and Manimaran, M. 2015. “Effect of compression ratio on performance and emission characteristics of biodiesel blend operated with VCR engine”, National Conference on Recent Trends and Developments in Sustainable Green Technologies, *Journal of Chemical and Pharmaceutical Sciences, Special* Issue-5, Pages 23-25.

Venkateswara Rao P. 2015. “Compression ratio effect on diesel engine working with biodiesel (JOME)-diesel blend as fuel*”, Research Journal of Chemical Sciences,* Volume-5(7), Pages 48-51, 2015.

Venkatraman, M. and Devaradjane, G. 2010. “Effect of compression ratio, injection timing and injection pressure on a DI diesel engine for better performance and emission fueled with diesel diesel-biodiesel blends”, *International Journal of Applied Engineering Research, Dindigul*, Volume-1, Number-3, Pages 288-298.

****