CHAPTER 2
LITERATURE REVIEW

The world is presently confronted with twin crises viz. fossil fuels depletion and environmental degradation. Indiscriminate extraction and lavish consumption of fossil fuels have led to reduction in underground-based carbon resources. The search for alternative fuels, which promise a harmonious correlation with sustainable development, energy conservation, efficiency and environmental preservation, has become highly pronounced in the present context. The fuels of bio-origin along with zero emission hydrogen can provide a feasible solution to this global fossil fuel crisis. The theoretical aspects and its background along with literature related to present experimental investigation have been reviewed in this chapter

The inventor of the diesel engine Dr. Rudolph Diesel first developed his diesel engine run with 100% pure peanut oil (*Arachis hypogaea*) and demonstrated in world exhibition held at Paris in 1900. He used the renewable fuel for his experimentation due to its cheaper price and availability when compared to the fossil fuel at that time. In this way, use of alternative fuels in compression ignition engines were introduced. Since then the use of alternative fuels was reported in the literature. Further, various approaches of usage of vegetable oils in compression ignition engines are mentioned below:

1. Use of esters of vegetable oils and its blends with conventional diesel
2. Use of vegetable oil blends with mineral diesel
3. Use of straight vegetable oils
4. Use of hydrogen as a supplemented mode with conventional diesel, vegetable oils and their blends

2.1. USE OF ESTERS OF VEGETABLE OILS AND ITS BLENDS WITH CONVENTIONAL DIESEL

The main drawbacks of vegetable oils are the high viscosity and low volatility, which causes poor combustion in diesel engine. The trans-esterification is the process of removing glycosides and combining oil esters of vegetable oil with alcohol. Due to trans-esterification the viscosity of the vegetable oil would be reduced nearer to diesel leading to improvement in combustion. Available literature related to use of the bio-diesel have been grouped and presented in this section.

Scholl et al.[19] done an exhaustive study on combustion of soybean (*Glycine max*) oil methyl ester (SME) in a direct-injection diesel engine. They understood the combustion parameters like ignition delay, peak pressure and rate of pressure rise were close to diesel operation at same load and engine variables. Also investigated the combustion and emissions characteristics with SME and diesel for different injector orifice diameters. Ignition delay for the both fuels were comparable in magnitude, whereas SME was more sensitive to nozzle diameter than diesel. CO emissions of SME was lower compared to diesel. HC emissions drastically reduced. NO_x was similar for both. Smoke was less for SME than the diesel. Yusuf Ali et al. [20] have tested the use of blends of methyl ester of soybean (*Glycine max*) with diesel in a 6 cylinder, turbocharged diesel engine. Two blends of 70/30 and 80/20 were tested. At 70/30 blend, the performance remains same as of diesel. Whereas at 80/20 gave better performance and lower emissions when compared to diesel.

Laforgia et al. [21] had done investigations on methyl esters of rapeseed(*Brassica napus*) oil as a fuel in an indirect-injection diesel engine. It was observed that, brake thermal efficiency was increased by 10% and emissions CO was increased by 6% there was mild effect on smoke, whereas unburned HC were reduced by 45%. Schumacher et al. [22] have investigated the use of soybean (*Glycine max*) methyl ester in a heavy duty diesel engine. It has been reported that the bio-diesel blends reduced the particulate matter, whereas NO_x was increased. By retarding the injection timing by 3^o CA, NO_x emission was decreased without altering the other emissions.

Masjuki et al. [23] done experimentation with pre-heated palm oil (*Elaesis guineensis*) methyl esters (POME) in the diesel engine and it was observed that increase in performance and reduction in emissions with POME. Agarwal et al. [24, 25] done experimentation on trans-esterified Linseed (*Linum usitatissimum*) oil and Linseed (*Linum usitatissimum*) oil methyl ester (LOME) and preferred the engine experiments with different blends of bio-diesel (LOME) and diesel and compared the data with conventional diesel. It was found that, with all blends thermal efficiency was increased and higher the exhaust temperature was observed. However, B20 blend was identified to be the best blend with respect to higher thermal efficiency and lower emissions. Murayama et al. [26] used waste vegetable oil esters in a direct-injection and indirect-injection engines. It was observed that, in both engines the performance was similar to that of light oils.

Masjuki et al.[27] done experimentation with coconut (*Cocos nucifera*) oil esters-diesel blends in an indirect-injection diesel engines. It was shown that results from the blends are similar to that of diesel. With 30/70 coconut oil ester/diesel blend brake thermal efficiency was even higher than the diesel. Emissions and lubricating oil contamination was also less with this blend when comparing to diesel. Dorado et al.[28] had executed experimentation with olive (*Olea europaea*) oil methyl ester. It has been reported that CO, CO_2, NO_x and SO_x were reduced when comparing to diesel.

He Y et al.[29] studied the influence of different methods in order to reduce the viscosity like blending, transesterification and dual fuel operation of Jatropa (*Jatropa curcas*) and methanol. Brake thermal efficiency was 27.4% with neat Jatropa oil to a maximum of 29% with methyl ester and 28.7% with the dual fuel mode. Bhattacharya et al.[30], Babu A K et al.[31] and De Almeida[32] also concluded with similar results of He Y. Further, explained that, the diesel engine would run successfully without any modification of the engine on various blends of vegetable oil and diesel fuel.

Nazar et al.[33] tested their diesel engine with karanja (*Pongamia pinnata*) oil ester. Thermal efficiency was increased to 29.6% which is lesser than the diesel having 31.5%. HC, CO were lower for esters in all loads. Further peak pressure and maximum rate of pressure rise were also similar to diesel. Usta et al.[34] done an exhaustive study on turbocharged indirect-injection diesel engine with tobacco seed (*Nicotina tabacum*) oil methyl ester. Performance was slightly deteriorated compared to diesel. CO was reduced and NO_x was increased. SO_2 traces also identified. It has been concluded that the tobacco seed oil methyl ester could be the partially substituted for diesel fuel as blend without any modification.

Sukumarpuhan et al.[35] used mahua (*Madhuca longifolia*) bio-diesel and tested in single cylinder, 4 stroke, direct-injection, constant speed compression ignition engine. It was observed that brake thermal efficiency was slightly deteriorated. Emissions like CO, HC, NO_x were also slightly reduced. Gvidonas et al.[36] done experimentation with rapeseed (*Brassica napus*) oil methyl ester (RME) and its diesel blends in a high speed diesel engine. Brake thermal efficiency was decreased with both bio-diesel and its blends. Emissions like CO, HC Smoke were lower with RME blends. NO_x and CO_2 were increased.

Kayisoglu et al.[37] had conducted some experimentations on sunflower (*Helianthus annuus*) and soybean (*Glycine max*) methyl ester blends in a direct-injection diesel engine. Brake thermal efficiency was slightly deteriorated with blends when compared to diesel. Further sunflower oil methyl blends shown better performance than the soybean methyl ester in performance as well as emissions. Nurun Nabi et al.[38] developed the bio-diesel from neem (*Azadirachta indica or Antelace azadirachta*) oil by esterification process with methyl alcohol. And tested this bio-diesel from neem oil in different blends with diesel in a naturally aspirated direct injection diesel engine. Performance was improved and emissions like CO and smoke were reduced with diesel.

Deepak Agarwal et al.[39] have investigated the influence of linseed (*Linum usitatissiumum*) oil, rice bran (*Oryza sativa*) oil, mahua (*Madhuca longifolia*) oil and linseed methyl ester in a diesel engine. In all the vegetable oil operated modes the brake thermal efficiency was slightly lower than the diesel. Further, among the identified vegetable oils, 20% of linseed oil methyl ester blend was found to be optimum performance when comparing to others. Sudhir et al.[40] have tested the effect of waste cooking oil methyl ester in a diesel engine. Thermal efficiency was decreased. HC emissions were decreased by 35% and NO_x remains same when comparing to mineral diesel. Further, peak pressure and the ignition delay were increased with esters.

Raheman et al.[41] have investigated the influence of engine variables like compression ratio and injection timing with mahua (*Madhuca longifolia*) oil methyl ester on a diesel engine. Optimized performance with reduction in emissions was observed at 20% bio-diesel blend with a compression ratio of 20 and at 40^o injection advancement. Lakshminarayan Rao et al.[42] done experimentation on a diesel engine with rice bran (*Oryza sativa*) oil methyl ester and its diesel blends. Ignition delay and heat release rate were lower with minimum RBME blends. As blending is increasing, it was observed that ignition delay and heat release rate were also increased. Emissions like CO, HC and Smoke were increased but NO_x was similar to that of diesel.

Zafer Utlu et al.[43] had done experimentation with waste frying oil methyl ester in a turbocharged variable speed diesel engine. Brake specific fuel consumption was increased by 14.4%, CO, NO_x & Smoke emissions were decreased by 17.1%, 14.5% and 22.46% respectively. Exhaust gas temperature was increased by 6.5% for bio-diesel compared to diesel.

Edwin Geo et al. [44] had investigated a diesel engine with rubber seed (*Hevea brasiliensis*) oil blended with Diethyl ether (DEE). It was observed that the brake thermal efficiency was increased by 7% and the smoke emissions were significantly reduced. CO & HC were reduced, whereas NO_x was slightly increased. Cylinder pressure and ignition delay were increased by 4.2%, 1^o CA respectively with DEE at the optimum quantity of 200 gm/hr. at rated power. Pugazrvadivu et al.[45] studied the performance and emission characteristics of a single cylinder direct-injection diesel engine fuelled with blends of bio-diesel and diesel with fuel additives of Diethyl ether (DEE) at different proportions. The addition of DEE to blends reduced the NO_x at part as well as medium loads, whereas peak loads the same was increased when compared to diesel and lowered to the corresponding bio-diesel blend.

Kandasamy et al.[46] have investigated the reduction of NO_x and smoke of diesel engine with the vetia peruviana bio-diesel emulsified with water in the ratio of 5, 10, 15 and 20% in order to investigate the engine performance and emissions characteristics. They concluded that emulsified fuels showed improved brake thermal efficiency with reduction in NO_x. Optimum % of water was found to be 15% for best performance and fewer emissions.

Ganapthy T et al.[47] studied the effect of injection timing along with engine operating parameters in Jatropa *(Jatropa curcas)* bio-diesel based engine. Advancing of injection timing from manufacturer recommended value for Jatropa's bio-diesel reduces the bsfc, CO, HC and Smoke emissions and subsequently increase in thermal efficiency, Pmax, Heat release rate and NO_x emissions . At 15 N-m torque, 1800 rpm and 340 CAD injection timing the percentage of reduction of bsfc, CO, HC and Smoke levels were 5.1%, 2.5%, 1.2% and 1.5% respectively. Similarly percentage increase in brake thermal efficiency, Pmax, HRR and NO_x were 5.3%, 1.8%, 26% and 20% respectively.

Arun Balasubrahmanian[48] done an exhaustive study on a stationary, single cylinder, 4 stroke, direct-injection, diesel engine with various dual bio-diesel blends like mixture of Jatropa (*Jatropa curcas*) bio-diesel and neem (*Azadirachta indica or*

Antelaea azadirachta) bio-diesel with a combination of first set contains 90% diesel by volume and 5% Jatropa bio-diesel and 5% neem bio-diesel (BB10) and second set contains 80% diesel and 10% Jatropa bio-diesel and 10% neem bio-diesel named as BB20 and others like BB40, BB80 and BB100. Among these BB20 given good response when comparing to others.

Venkayta Subbiah et al.[49] done an exhaustive study on performance and combustion characteristics of direct injection diesel engine with the effect of Triacetin as an additive to with coconut (*Cocos nucifera*) oil methyl ester. It was identified with addition of additives to bio-diesel reduced the CO, HC and smoke emissions. Even NO_x was also considerably reduced when compared to diesel.

In addition to bio-diesels, researchers also concentrated on different straight vegetable oil blends with diesel in the transition time before shifting to dedicated straight vegetable oil operated diesel engines. Some such attempts towards usage of straight vegetable oil blends with mineral diesel were grouped together and showed in the next section.

2.2. USE OF STRAIGHT VEGETABLE OIL BLENDS WITH CONVENTIONAL DIESEL

Researchers made an attempt to have the straight vegetable oil blends in the transition time before they shifted to dedicated straight vegetable oil operated engines. Available literature related to use of straight vegetable oil blends with conventional diesel have been grouped and presented in this section.

Engelman et al.[50] discussed their data with range of blending from 10%-50% Soybean (*Glycine max*) oil fuel blends used in unmodified diesel engines. The brake specific fuel consumption and power were slightly effected with pure vegetable oil blends when compared to diesel. Minimum concentration blends helped in continuing the experimentations without much burden the engine performance. Whereas higher concentration leads to sputter the engine.

Sims et al.[51] tested the diesel engine with particularly rapeseed (*Brassica napus*) oil as 50% blend. There is no significant effects as fuel injection system. Whereas long run, fuel injector pump failure was taken place. Carbon deposits on combustion chamber remains same as of diesel operation. Bartholomew et al.[52] of Caterpillar, done an exhaustive study on blends of vegetable oils with diesel. They concluded that, with small blends, there is no effect on engine performance. They are able to succeed to make use of blend up to 50/50. However, 20% blend was considered to be a successful attempt.

Barsic et al.[53] of Deere company, studied the influence of mixing of peanut(*Arachis hypogaea*)oil and sunflower (*Helianthus annuus*) oil with diesel in a single cylinder diesel engine. Carbon deposits were more with vegetable oils when compared to diesel. Filter plugging was considered to be the one of the major problem with vegetable oil. Ziejewski et al.[54] have conducted experiments with 25/75 and 50/50 sunflower (*Helianthus annuus*) oil/diesel blends in a turbocharged diesel engine at various injection pressures. With both the blends the performance of the engine was slightly reduced. At long runs, emissions were increased. Carbon deposits in the combustion chamber and nozzle were also high.

Yu et al.[55] have studied the use of waste cooking oil/diesel blends preheated to 70^oC in direct-injection diesel engines. Engine performance was deteriorated in long run testing. The peak pressure was higher by 1.5 bar when comparing to diesel and occurred 1.1 – 3.8^o CA earlier. CO, NO and SO_2 were higher when compared to diesel.

Venkataramana et al.[56] had conducted experiments with rice bran (*Oryza sativa*) oil /diesel blends from 10-90% on a 3.5 kW diesel engine. They concluded that 50/50 rice bran oil and diesel blend shown good performance when comparing to other blends. Pramanik [57] had done an exhaustive studies on Jatropa (*Jatropa curcas*) oil/diesel blends in a direct-injection diesel engine. They tested up to 50% blends. It was concluded that with Jatropa/diesel blend the brake specific fuel consumption was increased when comparing to pure diesel due to its lower calorific value.

Forson et al.[58] concluded through their experimentation on performance of Jatropa (*Jatropa curcas*) oil blends in a single cylinder, air cooled, direct-injection, 4 stroke diesel engine showed that a blend of 2.6% of Jatropa oil and 97.4% of fossil diesel by volume shown increase in brake thermal efficiency in comparison with pure diesel. Further, NO_x was controlled through water emulsified diesel.

Suryawanshi[59] had done experimentation with neat palm (*Elaesis guineensis*) oil and its blends with mineral diesel. At part and as well as full loads the brake thermal efficiency was higher when compared to diesel. Further, maximum cylinder gas pressure was lower with vegetable oil blends. Significant reduction in emissions like smoke and HC, but NO_x compared to diesel.

It was seen from the literature that, wide variety of SVOs like from edible peanut (*Arachis hypogaea*) oil to non-edible Jatropa (*Jatropa curcas*) oil in the blending range of from 10% to 90% were tested. Results showed that, small deterioration in brake thermal efficiency and increase in emissions took place due to change in viscosity when compared to the conventional diesel. Further, researchers also concentrated on pure straight vegetable oils due to its simplicity of processing and for being more environmental friendly as compared to bio-diesel and straight vegetable oil blends. A detailed study of use of straight vegetable oils in diesel engines was discussed in the next section.

2.3. USE OF STRAIGHT VEGETABLE OILS (SVO)

Straight Vegetable oil (SVO) can be used rather than the bio-diesel because SVO can be produced in local co-operatives through processing the seeds like crushing, filtering and conditioning processes. This is very simple and proved process and is also helpful in developing the man power in rural areas. Further, Comparative Life Cycle Assessment (LCA) studies shown that, straight vegetable oil was given preference (as alternative) comparing to its bio-diesel, based on Energy Return on Investment Index (EROI). Further, LCA was carried out on non-toxilogical and toxilogical categories by CML and EDIP methods also revealed that, SVO is more environmental friendly than the bio-diesel [6].

Different experiments were conducted with various number of straight vegetable oils like coconut (*Cocos nucifera*) oil, Jatropa (*Jatropa curcas*) oil, karanja (*Pongamia Pinnata*) oil, neem (*Azadirachta indica or Antelaea azadirachta*) oil, and sunflower (*Helianthus annuus*) oil etc. as an alternative fuels in diesel engines. It was understood from the literature, acceptable performance was noticed. Under short and long term applications with these fuels showed heavy smoke and carbon deposits at various parts of the engine due to high viscosity and carbon residue. Further, use of these 100% alternative fuels in place of conventional diesel results almost same engine power with slightly lower thermal efficiency in comparison with diesel fuels.

Bruwer et al.[60] have studied the use of sunflower (*Helianthus annuus*) oil as fuel in farm tractors. It was observed that in long duration the power loss was to be 8% when comparing to pure diesel. Further, higher carbon deposits were observed with sunflower oil. Barsic et al.[61] done an exhaustive study on a single cylinder, naturally aspirated , direct-injection diesel engine for its performance with 100% sunflower (*Helianthus annuus*) oil and 100% peanut (*Arachis hypogaea*) oil and their 50% by volume blends and compared these results with diesel. Performance was slightly deteriorated and emissions were increased due to their high viscosity when comparing to diesel.

Bacon et al.[62] had done some experimentation with different Straight vegetable oils for their diesel engines. It was observed that vegetable oil based engines developed similar power with diesel but carbon deposit in the combustion chamber and at the injector tips at part and as well as medium loads were very high during their short term testing. It has been further concluded that, long-term engine testing has to be carried out to determine the overall effects of using vegetable oils in a diesel engine.

Yarbrough et al.[63] had conducted experiments on a diesel engine with six variants of sunflower (*Helianthus annuus*) oil and understood their performance. Refined sunflower oil had shown the good satisfactory results when comparing to raw sunflower oil. With these studies it was further concluded that, degumming and de-waxing the vegetable oil prevented engine failures. Goering et al.[64] studied the characteristics properties of eleven vegetable oils to determine which oil would be the best suited for use of as an alternative fuel source. Out of selected eleven oils corn (*Zea mays*), rapeseed (*Brassica napus*), sesame (*Sesamum indicum*), cotton seed(*Gossypium hirsutum and Gossypium herbacum*), and soybean (*Glycine max*) oils had the most favorable fuel properties.

Tahir et al.[65] tested for agricultural tractors with sunflower (*Helianthus annuus*) oil as a replacement of diesel fuel. The viscosity of the sunflower oil was 14% higher than the vegetable oil at 37^{o}C. It was understood that, the performance of sunflower oil operated engine was similar with diesel operation but there was a slight increase in fuel consumption was observed. This is due to difference in heating values of both fuels. But sunflower oil left with heavy gum and wax deposits on test equipment due to its oxidation which could lead to engine failures.

Schoedder[66] had worked with rapeseed (*Brassica napus*) oil as an alternative to diesel in a diesel engine. Short term tests shown good performance with using of rapeseed oil whereas in 100 hours long run, due to carbon deposits on piston rings, valves and injectors. Further, they concluded that, long term durability tests required in order to understand these difficulties. Auld et al.[67] had used rapeseed (*Brassica napus*) oil to study the effects of using an alternative fuel in diesel engines. An analysis of rapeseed oil showed a relationship between viscosity and fatty acid chain length. Short term duration of engine testing shown that engine power results with rapeseed oil was similar with diesel.

Bettis et al.[68] done an exhaustive study with different alternative fuels like sunflower (*Helianthus annuus*), saffllower (*Carthamus tinctorius*) and rapeseed (*Brassica napus*) oils as a possible alternatives for diesel. These vegetable oils contain 94% to 95% of energy content of diesel fuel and of 15 times more viscous than the diesel. During short term engine testing, the performance was very good but at long term due to carbonization in the combustion chamber the performance was deteriorated.

Engler et al.[69] done a comparative study of raw and refined sunflower (*Helianthus annuus*) and cotton seed (*Gossypium hirsutum and Gossypium herbacum*) vegetable oils as alternative to diesel raw fuel. Engine performance was slightly good with refined fuels when comparing to raw fuels. However, in both cases carbon deposits and lubricating oil contamination was observed in long duration run. Pyror et al.[70] had done some experimentation with soybean (*Glycine max*) oil in a small diesel engines in short as well as long duration engine testing. At short term operation, engine with soybean (*Glycine max*) oil shown similar performance when compared to diesel but due to carbon deposits and viscosity at long run the performance was deteriorated. Seppo et al.[71] had tested a turbocharged, 4 cylinder, direct-injection diesel engine using mustard (*Sinapis alba*) oil. Engine power is similar to diesel and smoke, NO_x were reported to be less than the diesel.

Narayana Reddy et al.[72] have tested a diesel engine with neat Jatropa (*Jatropa curcas*) oil with various parameters like injection timing, injection opening pressure injection rate and swirl level. It has been observed that, marginal improvement in brake thermal efficiency and reduction in smoke and HC emissions with an injection advance of 3^o CA and injection pressure also increased the thermal efficiency and reduction in emissions due to better spray atomization. Deshmukh et al.[73] had tested the single cylinder, direct injection diesel engine with hingan (*Balanites*) oil as alternative fuel for diesel. It was observed that, slightly power loss and higher specific fuel consumption with hingan oil. CO & HC were reduced, NO_x was similar to diesel.

Use of raw vegetable oil as a fuel for diesel engine has an important drawback due to its high viscosity. This high viscosity effects the spray formation leads to poor combustion. Pre-heating of these vegetable oils would reduce the viscosity. During heating the heavy fatty acids components of the vegetable oil are broken or cracked in to lighter components and there by viscosity reduces. Hence, literature related to use of pre-heated straight vegetable oil were grouped together and presented below.

Ryan et al.[74] have studied the effect of pre-heating the vegetable oil in direct injection diesel engine. It was understood that, pre-heating of vegetable oil to 140^oC reduced the viscosity of vegetable oil and brings to nearer to diesel. Further, reduced viscosity caused increase in performance and reduction in emissions. Pugazvadivu et al.[75] had done some experimentation with pre-heated waste frying oil to 70^o-135^o, in a direct-injection diesel engine. With pre-heated mode, performance was improved and emissions were reduced. They concluded that, use of waste frying oil with 135^oC pre-heated condition will acts like as a diesel fuel for short term duration.

Murayama et al.[76] have studied the effect of rapeseed (*Brassica napus*) oil pre-heated at 200^oC in naturally aspirated direct-injection diesel engine. An empirical relation was identified for different pre-heating temperatures with its viscosity. Further, due to pre-heating carbon deposit and ring sticking were reduced. It has been further reported that the brake specific fuel consumption of pre-heated rapeseed oil was reduced than that of raw rapeseed oil at ambient conditions. Rajasekaran et al.[77] had investigated the indirect-injection diesel engine with preheated diesel as fuel at different speeds. Temperature was varied in the range of 60^o-75^o C in steps of 5^oC. With pre-heated diesel soot emissions were reduced by 50% when comparing to normal diesel. There is no effect on brake thermal efficiency. They were concluded that pre-heated diesel can reduce the soot emissions in a diesel engine. Bari et al. [78] investigated the effect of pre-heated palm (*Elaesis guineensis*) oil in a diesel engine.

With pre-heating peak combustion pressure was increased by 6% and ignition delay was reduced by 2.6^o CA CO and NO_x were increased by 9.2% and 29.3% respectively compared to diesel.

It was seen that use of bio-diesel, straight vegetable oils as an alternative fuels for compression ignition engines showed a promising growth with mild penalty on brake thermal efficiency due to lower heating value and higher viscosity when compared to conventional diesel fuel. However, in parallel to use of vegetable oils even some researchers and scientists did an exhaustive study on use of another alternative fuel, hydrogen has high heating value, minimum ignition energy, high flame velocity, and free form carbon for internal combustion engines. The first attempt as per literature reported in 1807 by Francois Issac de Rivaz, of Switzerland, politician and inventor of first internal combustion engine run with hydrogen and oxygen mixtures. Manual ignited spark was used to ignite these mixtures. Later, 1808, he developed an automobile with his earlier developed engine but was not successfully commercialized [79].

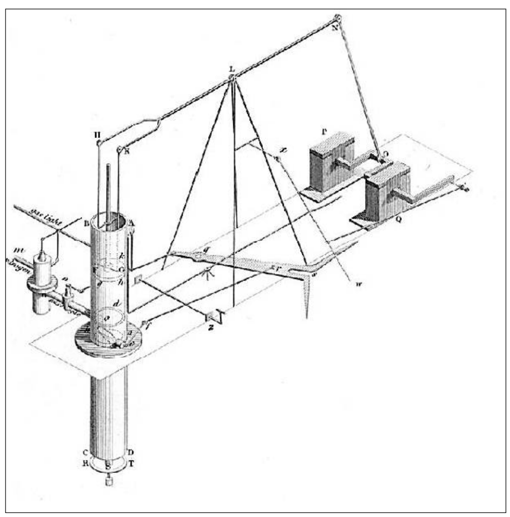

Figure 2.1 Cecil developed hydrogen engine

Later in 1820, Cecil, M.A., Fellow of Magdalen College, Cambridge University, England proposed and successfully tested the usage of hydrogen in internal combustion engines [80]. Later the same was patented by Jean Joseph Etienne Lenoir in the year 1860. The engine was powered by hydrogen generated via the electrolysis of water [81].

Ricardo and Burstall established the usage of hydrogen in internal combustion engine. But they are not able to succeed due to pre-ignition and back fire in the fuel supply system. As early as 1933, Norsk [82] developed , hydrogen operated internal combustion engine vehicle with onboard development of hydrogen by reforming the Ammonia. Erren R, around 1930 [83] carried out a detailed work on hydrogen application in internal combustion engines and developed knowledgeable literature but he lost the same in Berlin war. Later he continued his work in England and finally he was able to document it with Campbell at Institute of Fuel in London. In his exhaustive study, he claimed that very good brake thermal efficiency, as high as more than 60% could be achieved. His work also includes detailed study on controlling the backfire and pre ignition.

Oehmichen [84], worked on a single cylinder engine by mounting a hydrogen fuel injector in between the TDC and BDC and understood various parameters like thermal efficiency, effect of equivalence ratio, cylinder pressure versus CA, knock and its limitations and suggested to make use of hydrogen in stationary engines. King R O, 1948-1958, [85-87], and his collaborations were done very interesting work on carbureted engines by increasing the compression ratio up to 14 with chemically correct mixtures and able to control the pre-ignition, back firing and knocking. Compression ratio was increased up to 20 maintaining the equivalence ratio less than 0.5.

However, to overcome the transition phase of converting from dedicated diesel and alternative fuel operated engines to sole hydrogen operated engines; supplementation of hydrogen was concentrated by that, major changes in the engine design, material compatibility, abnormal combustion anomalies and huge rise in cost can be avoided. Further, use of gaseous hydrogen as a supplemented mode is discussed in the next section.

2.4. USE OF HYDROGEN AS A SUPPLEMENTED MODE WITH CONVENTIONAL DIESEL, VEGETABLE OILS AND THEIR BLENDS

As hydrogen has high self-ignition temperature of 568^{o}C, which is much higher than the most of CI engine fuels makes it not a successful attempt as a sole fuel. Researchers need an external ignition source to ignite this high self-ignition temperature hydrogen-air mixture. Wong [88] and Homan [89] suggested that, use of sole operated hydrogen compression ignition engines are very difficult to operate without any external ignition source. There are number of researchers [88-92], proposed and successfully tested the external source of ignition like spark plug, glow plug and pilot fuel injection of fossil fuels. Out of these, pilot fuel injection at the end of compression stroke, was considered being a reliable ignition source of hydrogen called dual fuel mode of operation.

Hence, dual fuel mode seems to be the most popular and stable combustion, when conventional diesel supplemented with gaseous hydrogen. Generally, hydrogen was supplemented through the inlet manifold into main stream of inlet air. Thus, inducted hydrogen gets properly mixed with air prior to getting injected as pilot fuel during compression stroke. This hydrogen-air mixture gets compressed during compression stroke and reaches the highest temperature to ignite the injected diesel fuel. But this temperature is not sufficient to burn the high self-ignition temperature of the inducted hydrogen. When pilot fuel is injected into the hot hydrogen –air mixture, the pilot fuel in turn gets atomized, vaporized and mixed with hot hydrogen –air mixture leads to self-ignition of mixture. The heat released by pilot fuel serves the ignition energy source to ignite the hydrogen. Further, dual fuel operated CI engines offers the potential of reduced exhaust emissions with increased engine performance [93].

When hydrogen is added in the intake manifold during suction stroke of a compression ignition engine, the burning of hydrogen will depend on the following as individual or combination of three approaches.

(1) Immediate or spontaneous combustion of inducted hydrogen with the burning of pilot fuel with in the pilot fuel spray plumes and got properly mixed with pilot fuel vapours.
(2) If sufficient rich mixture of hydrogen –air is available to support the initiation and propagation of multiple turbulent flames during the combustion makes utilization of inducted hydrogen.
(3) Due to increase of temperature of inducted charge during compression, combustion of hydrogen and pilot fuel and the heat absorbed by the combustion chamber leads to heat transfer from the unburned hydrogen –air diluent mixtures.

However, the influence of hydrogen supplementation/substitution in compression ignition engines with respect to performance, emissions and combustion were discussed below.

Haragopal Rao B. et al. [94], reported their experimental results on a single cylinder, water cooled, 5 HP, constant speed at 1500 rpm engine fuelled with diesel and supplemented with 10% hydrogen energy share of the total energy input of the engine. They claimed that maximum thermal efficiency was observed at peak loads rather than at part loads. Reduced quantity of hydrocarbon fuel due to supplemented hydrogen, the total hydrocarbon content in the engine exhaust was observed to be less at all loads with hydrogen introduction into air intake. Whereas, NO_x was increased due to increased heat release rate because of faster energy release. Maximum cycle pressure difference was observed at peak loads rather than part loads at 10% hydrogen supplementation with diesel operated compression ignition engine. Maximum cylinder Pressure (Pmax), is more in all loads with hydrogen supplementation when comparing to pure diesel operation. At full load, the ignition delay was decreased with hydrogen introduction. Increased Pmax is due to rapid combustion of inducted hydrogen burned rapidly. Rate of pressure rise not greatly affected.

Li Jing and Lu Ying-Qing and Wang Weing [95, 96] had conducted some experiments on a diesel engine at various speeds like 2600 and 3000 rpm with injection advancements of 21.5^o CA and 20^o CA respectively. They brought out some interesting results that smoke in dual fuel operation with introduction of hydrogen at peak loads was less when comparing to pure diesel operation.

Mathur H B, et al.[97], had conducted some experiments on a small utility , 4 kW, single cylinder, water cooled, diesel engine with neat hydrogen substitution up to 38% of total energy intake of the fuel. Smoke emissions were reduced considerably and NO_x emissions were increased. At 50% load as the hydrogen substitution increases in the range of 20, 30, 40 and 50 lpm smoke emissions were decreased to 15.6%, 22.2%, 26.7% and 33.3% respectively and at 100% load the same smoke emissions were reduced by 16.42%, 24%, 28.4% and 37.4%.

Naber J D et al.[98], their work highlights the significant pursuits and attainments of the combustion delay and hydrogen concentration under direct injection compression ignition engine. Specially designed disc shaped combustion chamber with transparent windows to access the internal mechanism during combustion of hydrogen was used. Ignition delay was less for such applications, where ambient gas temperature is more than 840 K and oxygen concentration is in between 5-21%. It was also understood that with increase in fuel temperature the effect of ignition delay with respect to ambient pressure was also small. They suggested that, late cycle direct injection of hydrogen helps to meet the CEV emissions standards without any type requirement for after treatment of gas.

Senthil Kumar M et al.[99], conducted an exhaustive experimentation on a 3.7 kW, single cylinder, 4 stroke, direct-injection fuelled with diesel and Jatropa's (*Jatropa curcas*) straight vegetable oil (SVO) and supplemented with gaseous hydrogen through the inlet manifold of the engine.

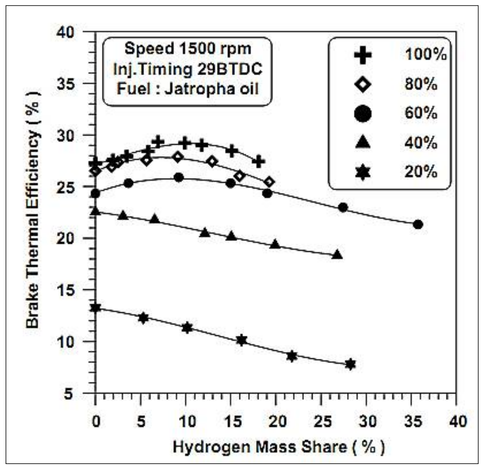

Figure 2.2 BTE vs. Hydrogen mass share of a Jatropa based SVO

They claimed that, the brake thermal efficiency of Jatropa based straight vegetable oil (SVO) as a pilot fuel operated engine was increased its efficiency from 27.3% to 29.4% at full load with 7% hydrogen mass share when comparing to pure Vegetable oil based engine. Whereas with diesel as a pilot fuel, the brake thermal efficiency (BTE) was increased from 30.3% to 32% at 5% hydrogen mass share. Smoke was reduced from 4.4 to 3.7 BSU at maximum efficiency point. HC and CO were reduced from 130 to 100 ppm and 0.26 to 0.17 % by Vol., respectively at maximum power output, NO_x was increased from 735 to 875 ppm at full load in dual fuel operated mode comparing to diesel as a pilot fuel supplemented with 5% hydrogen mass share: where smoke was reduced from 3.9 to 2.7 BSU and NO_x were increased considerably. Peak Pressure, maximum rate of pressure rise, Ignition delay were increased in dual fuel mode in both vegetable oil and diesel supplemented with hydrogen. Combustion duration was reduced due to high flame speed of hydrogen. Further, higher pre-mixed combustion rate was observed with hydrogen induction.

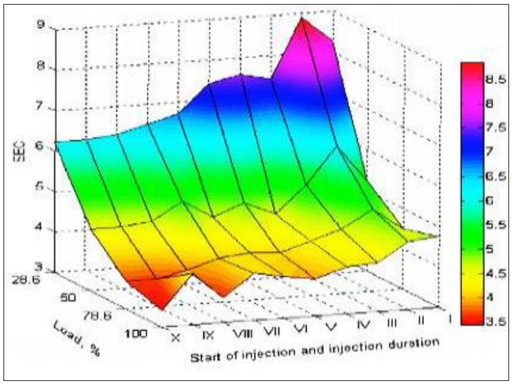

Figure 2.3 Variation of SEC with Load

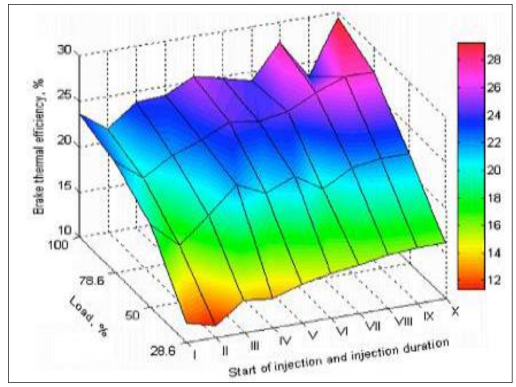

Figure 2.4 Variation of BTE with Load

Saravanan N et al.[100] had tried out some experiments to study the performance and emissions of a 3.7 kW, single cylinder, four stroke, water cooled, direct-injection diesel engine. Hydrogen injector was placed near by 13 mm from the intake valve seating portion. Hydrogen injection starts from 5^{o} bTDC and remains 15^{o} aTDC and duration was controlled from 30^{o} CA to 90^{o} CA.

From Figures 2.3 & 2.4 specific energy consumption (SEC) decreased from 4.7 to 3.4 at full load for the hydrogen injection timing of 5^{o} aTDC with injection duration of 30^{o} CA. This is because of better combustion in the combustion chamber due to uniform mixture of hydrogen with air. Brake thermal efficiency (BTE) increased from 23.6% to 31.67% at 15^{O} aTDC with 60^{o} CA duration, but where researchers experienced the knocking of the engine and optimized with 29.4% at injection timing of 5^{o} aTDC with injection duration of 90^{o} CA when compared to diesel. NO_x emissions were reduced from 1806 ppm with pure diesel operation to 705 ppm at full load with 60^{o} CA, because of leaner equivalence ratio.

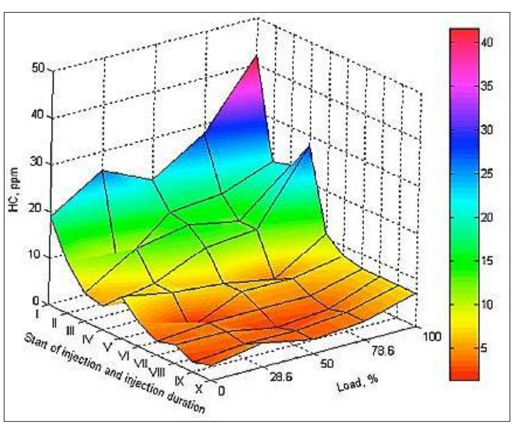

Figure 2.5 Variation of HC with Load

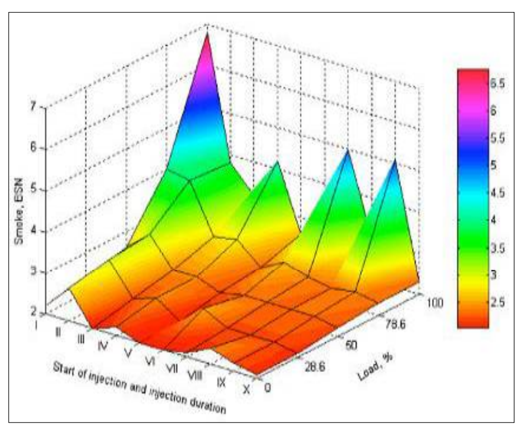

Figure 2.6 Variation of Smoke with Load

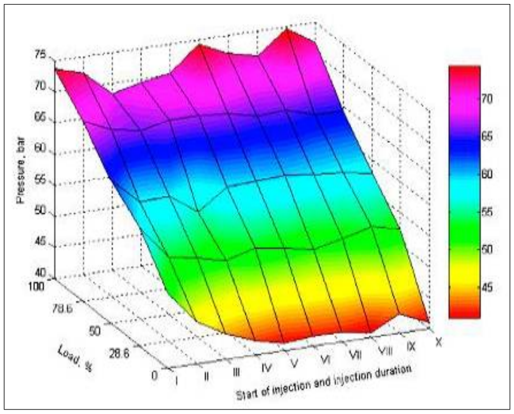

Figure 2.7 Variation of CO with Load

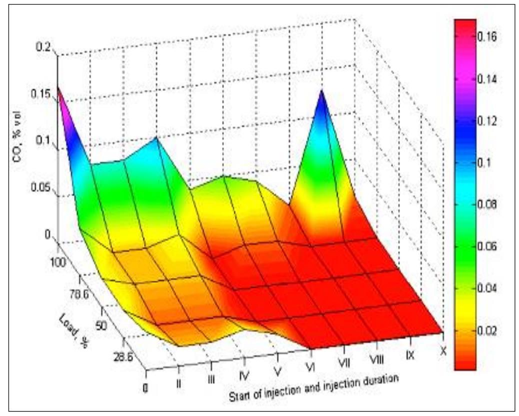

Figure 2.8 Variation of Peak Pressure with Load

From Figures 2.5, 2.6 and 2.7, HC were reduced from 42 ppm to 7 ppm at full load condition. Smoke was reduced from 2 BSN to 0.4 BSN at 75% load condition because of absence of carbon in hydrogen and homogeneous mixture of hydrogen with diesel rather than heterogeneous mixture like diesel. CO and CO_2 were reduced from 0.17 % by volume to 0.01 % by volume and 9.5% by volume to 2.1% by volume at 75% load at start of injection at 5^o aTDC with 90^o injection duration for supplementation of hydrogen and pure diesel operations. From Figure 2.8, Peak pressure was reduced from 73.7 to 71.7 bar at full load for hydrogen supplementation and pure diesel operation.

Edwin Geo V et al. [101], tried out their experiments on a diesel engine with dual fuel mode. Rubber seed (*Hevea brasiliensis*) oil (RSO), Rubber seed (*Hevea brasiliensis*) oil methyl ester (RSOME) and diesel as used as pilot fuels and hydrogen was used as a supplemented fuel. The optimum mass share of hydrogen for RSO, RSOME and diesel were 8.39%, 8.73% and 10.17% respectively. Maximum thermal efficiencies are of 28.12% for RSO, 29.26% for RSOME and 31.62% for diesel at their optimized hydrogen shares. The optimum hydrogen mass share for RSO and RSOME were 8.39% and 8.73% and the same were even tested up to 10.76% and 11.2% without knock. Later stages of hydrogen supplementation severe knock were observed. Peak pressure and rate of pressure rise were increased at peak loads comparing to part loads. Further, ignition delay and combustion duration were also increased in all supplemented modes.

Matthew G Shrik [102] carried out experiments on two different engines. The first set of experiments with 1.3 Liter, turbocharged, 53 kW, compression Ignition coupled with engine eddy current dynamometer under steady state conditions and another set of experiments were carried on 1.3 Liter, 66 kW compression ignition engine vehicle coupled with a chassis dynamometer and tested as per urban driving cycle. Both the engines were fuelled with B20 and supplemented with hydrogen in the range of 0-10% of total energy of B20 intake with an increment of 5%. The thermal efficiency was not so effected with engine dynamometer whereas with chassis dynamometer, the same was slightly reduced

Saravanan N et al.[103], carried out some experiments on single cylinder , 4 stroke, compression ignition, constant speed, vertical, water cooled , direct injection , 3.7 kW engine fuelled with diesel and substituted with hydrogen. Brake thermal efficiency was increased from 22.78% to 27.9% at 30% hydrogen substitution. It was understood that, specific energy consumption (SEC) was reduced with increasing the hydrogen share. At full load, with optimized 30% hydrogen share, SEC was reduced to 12.9 MJ/kW-hr, otherwise of 14.5MJ/kW-hr with pure diesel operation. Up to 30% of hydrogen share NO_x increased but after 30% share, NO_x decreased due to reduction of combustion temperature. At 75% load, the peak pressure was 64.3 bar at 5^o aTDC with 90^o CA duration. Which is 4.6 bar lesser than the pure diesel operation and the same was increased at full load to 71.1 bar, which is 6.8 bar more than the pure diesel. The increase in peak pressure is due to higher burning velocity of hydrogen, which makes combustion to be almost instantaneous resulting in increase of cylinder pressure.

Same researchers [104] extended their work on the same engine with different Exhaust Gas Recirculation (EGR) flow rates. Basically utilization of Exhaust Gas Recirculation was considered in two methods. First one is the replacement and the second one is the addition of exhaust gas recirculation. In replacement EGR air entering into intake manifold was replaced by the exhaust gas. This is mostly opted for compression Ignition engines whereas second one in which exhaust gases were added to intake air of an engine under aspiration process. This second one is mostly used in Gasoline engines. It was observed that application of EGR will reduce the NO_x emissions because of its dilution, thermal and chemical effects. Among these, dilution effect is the very effective one. It was observed that 20% EGR will be the best suitable option for this considered engine under hydrogen dual fuel mode. Performance and emission parameters were considered at this 20% EGR. The Brake

Thermal Efficiency (BTE) was increased from 15.6% to 23.11% and 24.2% without and with EGR. It was observed that, with EGR, BTE was slightly reduced. NO_x emissions were recorded that varies from 99 ppm to 526 ppm, whereas without EGR it was varying from 184 ppm to 2212 ppm. Smoke was decreased with increasing the percentage of hydrogen substitution. These smoke emissions were varied from 0.1 BSN to 4.9 BSN, whereas without EGR it was 0 to 2 BSN. It indicates smoke emissions were increased with EGR at full load conditions whereas GHG like CO was varied from 0.04% to 0.74% with 20% EGR and 0.02% to 0.09% without EGR and for HC emissions were increased from 12% to 19% with 20% EGR.

Saravanan N et al. [105] concluded in their research on AV1 make naturally aspirated, 4 stroke, vertical, compression ignition engine fuelled with diesel and supplemented hydrogen with Diethyl ether through carburetion as well as manifold injection. Optimum hydrogen injection timing for both methods were 5^o bTDC with injection duration of 30^o CA and thermal efficiencies were increased to 26.2% and 27.3% with both hydrogen inducting/injecting methods respectively. Manifold injection of hydrogen shown more brake thermal efficiency compared to carburation method of hydrogen injection. Smoke emissions were reduced under diesel-hydrogen dual fuel operation from 2 to 0.8 BSU and further reduced with DEE with hydrogen supplementation to 0.7 BSU. CO emissions remains same, not changed in both pure diesel and diesel- hydrogen supplemented modes of 0.316 by volume. But in DEE-hydrogen dual fuel mode the same CO reduced to 0.15% by volume CO_2 with pure diesel was 0.77 and reduced to 0.64 with diesel- hydrogen and further reduced to 0.33 with DEE - hydrogen at 78% load. At 78% load, peak pressure was observed. With pure diesel, the peak pressure was 78.5 bar and remains same even with dual fuel (diesel + hydrogen) mode. Whereas with diethyl ether + hydrogen dual fuel mode, peak pressure was reduced to 68 bar.

Gomes Antunes J M et al. [106], done their experimentation on modified single cylinder, air cooled diesel engine, naturally aspirated to accommodate both manifold and direct injection of hydrogen in dual fuel operation. In dual fuel mode, pre-heating of air is not required due to pilot fuel is sufficient to ignite the inducted hydrogen. Whereas in direct hydrogen injection mode, the inducting air has to be heated up to 120^o C to ensure the ignition of the hydrogen. In compression stroke air is further heated up to ignite the high self-ignition of hydrogen. Engine efficiency was achieved up to 42.8% with hydrogen under direct injection when comparing to 27.9% with conventional diesel. Whereas, with EGR, the BTE was further reduced. Further, with increase of flow rate of EGR, the BTE was reduced to that extent. Brake specific energy consumption (BSEC) without EGR is less, with EGR the same was increased. The reason for reduction in BSEC in hydrogen dual fuel operation is higher calorific value and operation of hydrogen fuelled engine under lean burn conditions. The increase in BSEC with EGR is due to negative effect of EGR on combustion. They observed that NO_x increased as load increases.

Szwaja Stanislaw et al. [107], highlighted through their significant pursuits and attainment of hydrogen combustion in HCCI mode under diesel-hydrogen dual fuel mode. Under this HCCI mode to ignite the hydrogen mixture compression ratio was optimized to 16. General hydrogen- air stoichiometric mixture combusted in internal combustion engines under HCCI mode makes severe knocking. Hence EGR and lean combustible mixtures were used to reduce the knock effect. Being a HCCI and supplemented with hydrogen makes the unstable combustion and start of combustion is varying in between 6^o to 15^o CA bTDC due to ignition of hydrogen at various hot places located on engine head and piston surface. Nearly 50% mass fraction burned (MFB), was taken place before piston reaches the TDC causes peak pressure was also located before TDC.

Probir Kumar Bose et al. [108], have reported the experimental results on diesel-hydrogen dual fuel mode converted 5.2 kW, single cylinder, Kirloskar diesel engine. They studied this diesel- hydrogen dual fuel with and without EGR. Timed Manifold Injection was considered to inject the hydrogen. At 0.15 kg/hr. hydrogen mass flow rate brake thermal efficiency was increased by 12.9% without any EGR. With EGR at same flow rate of hydrogen, the BTE was reduced due to dilution effect.

Saravanan N, et al. [109] had carried out extensive experiments on Kirloskar AV1 make, naturally aspirated 4 stroke vertical compression ignition with hydrogen diesel as a dual fuel mode. In their investigation, they studied the dual fuel mode hydrogen induction through the air manifold adopting carburetion, timed port and manifold injection techniques. Optimum hydrogen injection timing for hydrogen port injection and manifold injections were 5^o bTDC, with injection duration of 30 ^{o} CA respectively. Brake thermal efficiency with diesel is 23% and had been increased in timed port injections (TPI) with 26.2% and in Timed Manifold Injection (TMI) with 27.3%. NO_x emissions increased by 34% with TPI and 8% with carburetion when comparing to baseline diesel operation. With the use of SCR smoke emissions were near about nil even at full load CO emissions also considerably reduced. Timed Port Injection was decided to be the best among the TPI and carburetion for hydrogen induction as a dual fuel programme. But NO_x levels were increased with TPI but the same was reduced by 74% with Selective Catalytic Reduction (SCR) techniques with a flow rate of Ammonia to Nitrous Oxides (ANR) 1.1. Overall reduction of HC was reduced up to 73% with SCR at ANR of 0.9. Whereas CO was reduced by 33% and CO_2 was increased by 10% with SCR. CO_2 was increased due to oxidation of CO in SCR system. Hydrogen diesel dual fuel operation was a good combination to improve the performance and reduction in emissions except NO_x. Further, they concluded that, increased NO_x can be reduced by suitable in corporation with SCR system.

Murari Mohan Roy et al. [110], done their experimentation on 4 stroke, single cylinder, water cooled, constant speed, super charged diesel engine fuelled with conventional diesel and substituted with gaseous hydrogen up to 90% of total intake energy. With increased the thermal efficiency of 42% at Fuel to air equivalence ratio of 0.3. There were hardly any CO and HC emissions. With 50% exhaust gas recirculation (EGR) NO_x were reduced by 60% without effecting the remaining parameters. They are succeeded to substitute the hydrogen up to 90% of total energy. Smooth and knock free engine operation resulted from the usage of hydrogen in a supercharged dual fuel engine operation at leaner fuel-air equivalence ratios. The hydrogen operation produced the maximum indicated mean effective pressure (IMEP) of about 908 kPa. The two-stage combustion was found as a condition of higher engine power and a precursor of knocking was observed. The main combustion at the maximum IMEP conditions (with strong two stage combustion) was found much faster than that of normal combustion at other IMEP conditions. EGR was adopted to suppress the engine knock by reducing the gas temperatures.

Bari .S and Mohammed Esmail [111], had studied the impacts of small amount of H_2/O_2 mixtures as an additive to the performance of a 4 cylinder diesel engine. The required amount of the mixture was generated through onboard generation of hydrogen and oxygen from electrolysis of water. Introduction of 6.1% of total diesel equivalent H_2/O_2 mixture into diesel the brake thermal efficiency was increased by

2.6% at 19 kW, 2.9% at 22 kW, and 1.6 % at 28 kW. The bsfc of the engine was reduced by 7.3%, 8.1% and 4.8% at 19 kW, 22 kW, and 28 kW respectively. Emissions of HC, CO_2, and CO were reduced and NO_x was increased. higher peak pressure was experienced nearer to TDC due to high flame speed of hydrogen flame enhances the combustion up to 5% of total diesel energy of adding H_2/O_2 will enhance the combustion and later stages combustion was deteriorated.

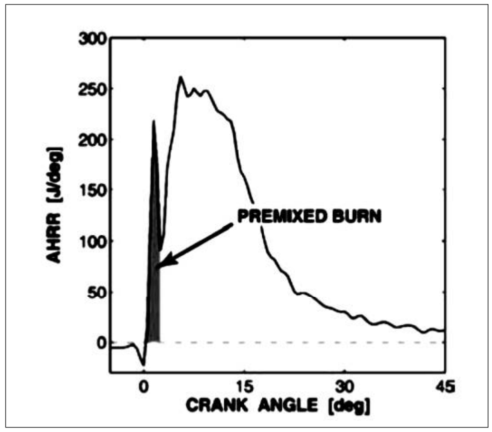

Figure 2.9 Apparent Heat release rate curve showing the fraction of heat release corresponding to pre-mixed burn

Jacob Wall [112] was concluded that, in dual fuel operation with diesel as pilot fuel and 8% by mass hydrogen as a supplemented fuel and with retarded injection timing the brake thermal efficiency was enhanced. At optimized 8% hydrogen mass share, HC, CO were reduced and NO_x were increased. Gatts T et al. [113], done an exhaustive experimentation work on a 2004 make Mack, MP7, 355E, 6 cylinder, 265 kW, variable geometry turbocharger with exhaust gas recirculation (EGR), 10.8 L capacity , direct- injection diesel engine fuelled with diesel and supplemented with hydrogen was shown at optimum hydrogen share of 6% by volume. Brake thermal efficiency was enhanced at peak loads rather than at part loads.

William F Northrop et al. [114] carried out extensive experiments on partially pre-mixed hydrogen with ultra-low sulphur diesel (ULSD) and soybean (*Glycine max*) based bio-diesel (B100) in 4 cylinder, direct-injection, 100 bhp, GM passenger car diesel engine. Baseline data was considered to examine the hydrogen emission under low temperature combustion (LTC) over a range of 0.5 to 1.1 equivalence ratio. These emissions are significant and needs to get treated through Lean NO_x Traps (LNT) and selective Catalysts Reduction (SCR) techniques. It was observed that, same emissions rose from ultra-low Sulphur diesel (ULSD) and B100 blends. They concluded that, methods and processes which were governed the hydrogen emissions are independent of the selected fuels but only dependent of controlling over the combustion. Optimum equivalence ratio found to be greater than 0.85 in representing the hydrogen emissions.

Gregory K Lilik et al. [115] conducted a series of exhaustive experiments on 103 kW, DDC direct-injection, turbocharged, light duty automotive diesel engine. Supplemented hydrogen was considered on energy basis of 0%, 2.5%, 5%, 7.5%, 10% and 15% in the intake manifold through aspiration process. They identified that, retarding the injection timing is making an important role in reduction of NO_x either with the hydrogen supplementation or with the baseline fossil fuels. With supplementation of hydrogen, NO/NO_2 ratio was shifted, in which level of NO emissions were reduced and NO_2 emissions were increased. Same experimental results were validated and tested with a transported probability density function (PDF method) and CFD model and they understood that, temperature alone will not make the NO to NO_2 with increasing hydrogen quantity, where H_2O also playing role in changing the NO to NO_2 in addition to temperature.

Chong J J et al. [116] were conducted experiments on Lister Petter model TR1, single cylinder, direct-injection, 8.6 kW diesel engine. They had introduced the CO and hydrogen to ULSD in addition to 2% by weight Silver dispersed on γ-aluminia selective catalytic reduction and reformed Exhaust Gas Recirculation System (REGR). Generally NO_x typically consists of 85-95% of Nitrogen Monoxide(NO) and 5-15% of Nitrogen dioxide (NO_2). It was studied the significant reduction of total NO_x emissions by introducing the hydrogen or reformate (H_2 CO and EGR gases) into the intake manifold of the engine. This paper brought out some interesting parameters like Air/fuel ratio, combustion efficiency temperature inside the cylinder and its effects on the production of NO_x in the exhaust after treatment methods like Selective Catalytic reduction technique (SCR) and diesel particulate Filter (DPF) will also improve in optimizing the NO to NO_2 oxidation. Reformate (H_2, CO and REGR) will reduce the NO_x and increases the NO_2 proportion in the exhaust. With REGR the ratio of NO_2/NO_x will be more when compared to the normal diesel operation. It was understood that using 15% hydrogen, 10% CO with REGR will be best influencing are to bring out to NO_2 from NO. Further use of Diesel Oxidation Catalyst (DOC) will reduce the CO and HC emissions also.

Liew C et al. [117], done an exhaustive experimentation and shown the significant pursuits and attainments on the combustion especially, heat release rate of a 1999 model Cummins ISM370, 10.8 Liter capacity, 6 cylinder, direct-injection, 276 kW diesel engine. When operated at pure diesel, the diesel engine featured two-stage combustion process was observed. The addition of hydrogen was shown that, compared to pre-mixed combustion, addition of hydrogen was found to have more significant on diffusion combustion on addition of hydrogen enhanced the HRR observed as shown in the Figure 2.10

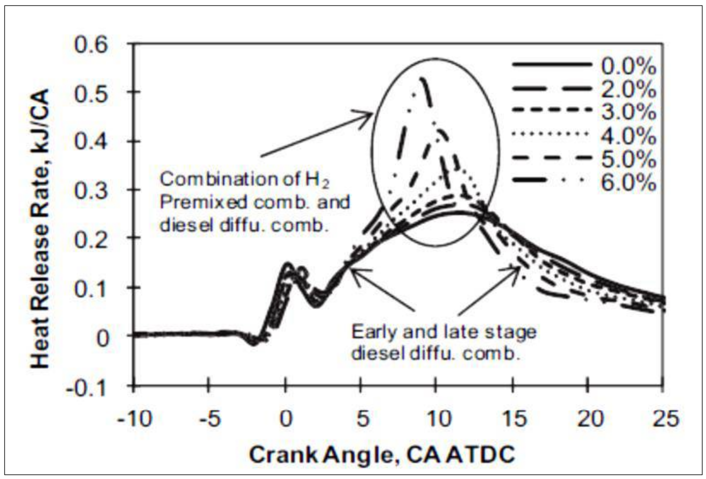

Figure 2.10 Effect of H_2 addition on heat release rate at 70% Load

Hohgsheng Guo, et al.[118], investigated the influence of hydrogen enrichment on the performance of diesel under homogeneous charge compression ignition (HCCI) engine and the experimental results shown that, brake thermal efficiency was increased due to enrichment of hydrogen with diesel under HCCI mode due to enrichment of hydrogen allows a diesel HCCI engine to operate at a higher compression ratio and leads to higher power output and fuel conversion efficiency. Peak pressure and HRR were increased in ultra -low sulphur diesel (ULSD), B50 blends. Enrichment of hydrogen not only reduces the overall indicated specific CO and HC emissions, but also decrease of CO emissions per unit mass. However, hydrogen enrichment does not significantly effect the HC emissions per unit burned diesel fuel. Further, there is no significant change in NOx emissions. Emission results indicates that, hydrogen enrichment retards combustion phasing and reduces the combustion duration of a diesel HCCI engine. Higher cetane number fuels require more hydrogen to retard the combustion phasing to the optimal values. Further, enriched hydrogen improves the combustion stability of a diesel HCCI engine under retarded combustion phasing. Knocking tendency may increase with increasing hydrogen enrichment.

Tira H S et al. [119], done an exhaustive study on liquefied petroleum gas (LPG) - diesel dual fuelled combustion experimental study to understand the impact of the properties of the direct-injection diesel fuels such as rapeseed (*Brassica napus*) methyl ester (RME) and gas to liquid (GTL) on combustion characteristics of a single cylinder, 4stroke, 773 cc, 8.6 kW, direct-injection engine.

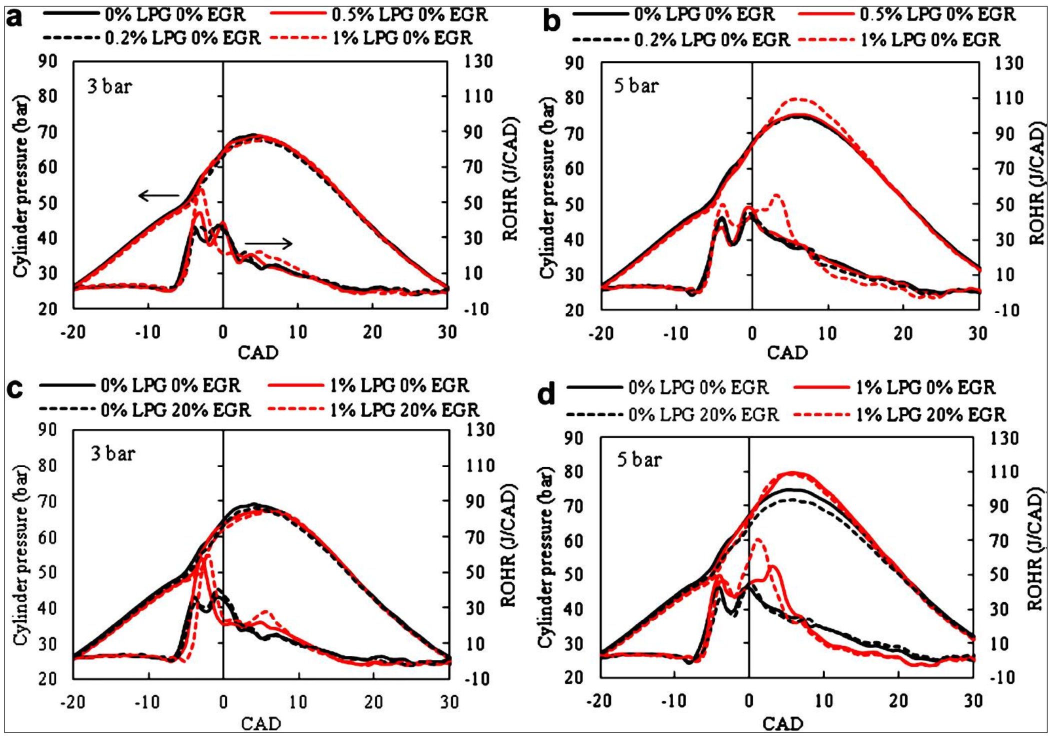

Figure 2.11 Cylinder pressure and rate of heat release for ULSD. a) 3 bar IMEP, b) 5 bar IMEP, c) effect of LPG and EGR at 3 bar IMEP, d) effect of LPG and EGR at 5 bar IMEP

From the Figure 2.11, cylinder pressure and rate of heat release rate of liquefied petroleum gas (LPG) – ultra low sulphur diesel (ULSD) for different proportions of LPG. As LPG, percentage increases ignition delay was increases due to lower cetane number and higher calorific value and lack of oxygen leads to higher ignition delay. At low engine load, in dual fuel mode, pre-mixed phase was dominated due to, as LPG was inducted along with air in the cylinder leads to formation of combustible mixture which will burn in the pre-mixed mode. At higher engine load, the ignition delay was shortened and liquid fuel injection was extorted compared to low engine load. The higher in cylinder temperature at high engine load promotes LPG combustion. In addition higher calorific value in turn leads to increase in peak pressure and extension of diffusion combustion phase as shown in Figure 2.11.

Zhou J H et al. [120], done an exhaustive study on 88 kW, ISUZU 4HFI model 4.3 Liter , direct-injection diesel engine fuelled with ultra- low Sulphur diesel (ULSD), Palm (*Elaesis guineensis*) methyl ester (PME) and blended fuel of 50% ULSD + 50% PME with naturally aspirated hydrogen at constant speed and varying loads. Maximum hydrogen share was up to 20% of the total fuel energy. The influence of fuel consumption and brake thermal efficiency is dependent on engine load is same scenario for all three selected fuels with hydrogen supplementation. At part load, for three fuels, the BTE was low whereas at peak load the BTE was improved for ULSD and ULSD + PME (B50) due to better participation of hydrogen in the combustion. Whereas with PME the change in BTE was negligible. But in Palm methyl ether (PME) at 10% and 20% hydrogen energy supplementation.

Hohgsheng Guo, et al. [121] extended their research on 612cc, variable compression ratio diesel engine fuelled with n -heptane and supplemented with hydrogen under HCCI mode. Results shown that at constant compression ratio the thermal efficiency was increased when an appropriate amount of hydrogen added to pilot fuel due to optimization of combustion phasing. When an appropriate amount of hydrogen added, it will reduce the indicated specific unburned HC, but slightly increases the unburned HC per unit burned n-heptane mass due to hydrogen addition slow down the low temperature kinetics of n-heptane and therefore relatively more HC are present during high temperature stage. The oxidation of the remaining HC during the high temperature stage is affected by hydrogen due to the competition for some key radicals and thus relatively more unburned HC are emitted.

Further, usage of hydrogen in the internal combustion engines have their own advantages, on other hands same hydrogen have some issues related to undesirable combustion phenomena is discussed in the following section.

2.5. UNDESIRABLE COMBUSTION PHENOMENA

The properties which make hydrogen as a desirable alternative fuel for internal combustion engines also cause abnormal combustion when the same is not properly handled. Especially, being low minimum ignition energy, high flammable and high flame speed limits to undesirable phenomena generally summarized as abnormal combustion anomalies like: Pre-ignition, backfire and knocking are discussed below[122].

2.5.1. Pre-Ignition

Wide flammability and low ignition energy of hydrogen prone to pre-ignition is often encountered in hydrogen operated internal combustion engine. Due to premature ignition caused by hot spots in the combustion chamber leads to increase the temperature of combustion chamber makes pre-ignition for next cycle. Further, it continues or prolongs up to intake stroke and causes back fire. As the minimum ignition energy is dependent on stoichiometric ratio, pre-ignition pronounced as shown below in Figure 2.12.

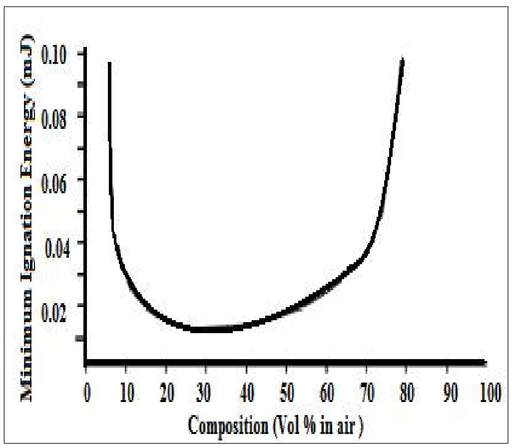

Figure 2.12 Minimum ignition energy of hydrogen in air as a function of air composition by % volume

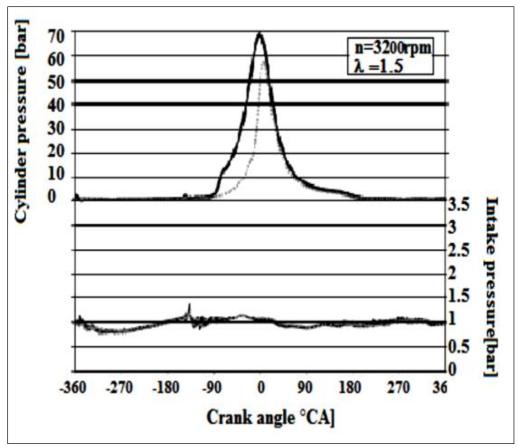

Figure 2.13 Typical cylinder and intake manifold pressure traces with pre-ignition (solid lines), compared to regular pressure traces (dotted lines) [30]

Further engine operating conditions at increased speed and load are more prone to pre-ignition in the combustion chamber. From the above Figure 2.13, data was taken on single cylinder, hydrogen research engine at engine speed of 3200 rpm and maximum indicated mean effective pressure (IMEP) of 7 bar. The pre-ignition data was shown with dark line and the same engine with normal operation (regular combustion) was shown with dotted line. In pre-ignition case, the peak pressure was higher than the regular combustion case and the same was advanced to 80^{o} CA bTDC, accordingly IMEP was 0 bar for this pre-ignition case. Further, as pre-ignition occurs after the intake valves there is no significance difference shown on the intake pressures [123].

2.5.2. Backfire

This backfire is a violent consequence of a pre-ignition phenomena. The enflamed charge that it travels part the valve and into inlet manifold resulting backfire due to occurrence of ignition at the inlet valve, when in open situation. The backfire is more pronounced in dual fuel engines, when the hydrogen is being inducted through the inlet manifold. The main difference between back firing and pre-ignition is the timing at which anomaly occurs. Generally, pre-ignition occurs during the compression stroke, where the intake valve is in closed position and backfire experiences during suction at that time the intake valve is in open position. The severity of backfire leads to not only remarkable audible sound and may damage the inlet systems of the engine. Generally, pre-ignition is a predecessor for the occurrence of backfiring. Optimization of injection strategy with combined effect of variable valve timing for intake as well as outlet valves [123] is the best solution to control the backfire.

2.5.3. Knocking

Temperature, Time and pressure of hydrogen –air mixture makes the spontaneous ignition of charge causing rapid release of heat leads to burning of available mixture, causing generating a high amplitude pressure wave known as knock. This developed pressure wave may increase the mechanical and as well as thermal stresses leads to damage of the engine. Further, knock widely depends on engine design and Fuel –air mixture properties [123].

Induction or injection of hydrogen into the engine plays a vital role in developing and as controlling these abnormal combustion anomalies. Hence, different hydrogen induction or injection techniques are described in the next section.

2.6. DIFFERENT HYDROGEN INDUCTION OR INJECTION TECHNIQUES

Effective participation of supplemented or substituted hydrogen in the combustion chamber and undesirable combustion phenomena like pre-ignition and backfire depend on the method or mode or technique adopted through which hydrogen is inducted or injected into the engine cylinder. Hence, the mode of hydrogen induction or injection played a vital role in either developing or controlling the anomalies of running hydrogen based internal combustion engine.

There are mainly 3 different techniques of hydrogen supplementation are available from the literature [124, 125].

1. Continuous Carburetion (CC)
2. Manifold Induction or Injection (MPI) and
3. Direct cylinder Injection (DCI).

2.6.1. Continuous Carburetion (CC)

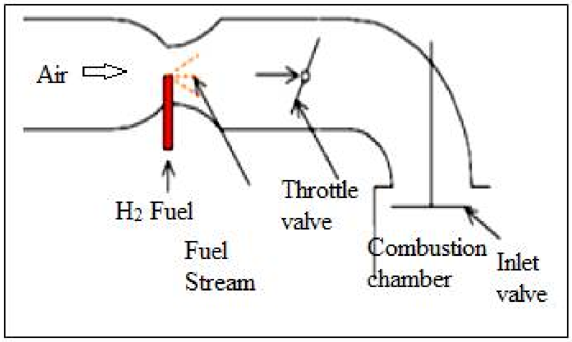

Figure 2.14 Continuous carburetion technique

Carburetion by the use of conventional gas carburetor is a simplest, oldest and proved technique. Generally, this method is preferable for gasoline engines because of minor modifications in the intake induction system. The advantage of this mode of hydrogen induction is by either up draught, or down draught or cross draught methods and does not require maximum pressure of the hydrogen supply.

Earlier research on hydrogen induction was totally based on carburetion. Ricardo [126] and King [85-87] had adopted the carburetion technique for injection of hydrogen. Ricardo experienced that, "pupping back into carburetor" throughout his experimentation even as he selected as low as possible 3.8 compression ratio engine. Whereas King also concentrated mostly on the same theme, but he always experienced severe undesirable combustion phenomena like pre-ignition and backfire. He identified that, free floating carbons and hot spots are main reasons for these undesirable anomalies. Due to these problems, later on several other researchers tried to have different alternative modes of fuel inductions. In such case, Swain & Adt [127, 128] supplied the hydrogen through a passage provided through the inlet valve. Further, they regulated the hydrogen pressure by varying the hydrogen flow rate. Bindon J et al. [129], had tried a good technique in providing the quality controlled mixture through a lean burn carburetor. The disadvantage of the carburetion method is the pre-ignition and backfires due to greater amount of hydrogen –air mixtures available in the inlet manifold.

2.6.2. Manifold Induction or Injection (MPI)

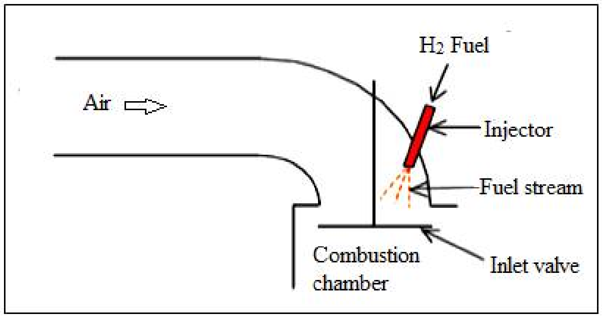

Figure 2.15 Manifold Induction or Injection technique

In inlet manifold or port injection technique, the gaseous hydrogen will be supplied either through mechanical or electronic injectors. Generally, in manifold induction/injection, air is inducted through either port or inlet manifold during suction [130]. Since the quantity of hydrogen is less when comparing to central carburetion injection, the severity of pre-ignition and backfire is less in this case. In this technique, the volume of air inducted is being constant and hydrogen can be varied to control the power output by controlling the amount of hydrogen inducted/injected in the air stream. Further, there are two approaches available, those are; continuous manifold injection (CMI) and timed manifold injection (TMI). In CMI, the hydrogen is being injected continuously throughout the intake duration of crank angle. But either due to valve overlap, or increase in speed or load, there is maximum prone to backfire in this case. Whereas in TMI, duration of injection and time of injection can be controlled by which the injection of hydrogen can be pursued at the end of suction stroke.

Lynch F E [131], done an experimentation on parallel induction of hydrogen through a separate copper tube in the air intake system. He was able to control the backfire to some extent. Varde et al. [132] had done successful investigations with electronically controlled fuel injection for given duration nearer to the intake valve, when the valve is in open condition. Mac Carley et al. [133], adopted port injection and being controlled not only with the air entry but also considered various engine parameters to avoid the backfire. Karim et al. [134] used delayed port injection at low pressure to avoid the backfire. Majorly, Das L M [124] done an exhaustive experimental investigations and studies on various hydrogen induction techniques and concluded that TMI is the best among other techniques available.

2.6.3. Direct Cylinder Injection (DCI)

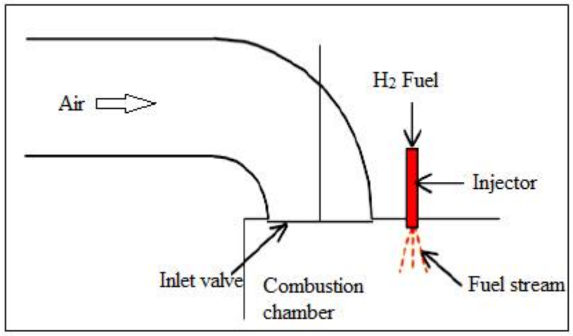

Figure 2.16 Direct Cylinder Injection Technique

Direct cylinder injection system is further categorized into low pressure direct injection (LPDI) and high pressure direct injection (HPDI). In both the cases gaseous hydrogen is directly supplied into the combustion chamber. With LPDI, hydrogen was supplied during intake stroke, whereas in HPDI the same was supplied at the end of the compression stroke [130]. In both LPDI and HPDI, as they both system injectors opened into the combustion chamber, leads to flushing of combustion hot and high pressure products to the injector makes questioning the reliability of the injector in course of time. Further, in direct cylinder injection , time available for hydrogen to get interact with air is very minimum causes poor participation of hydrogen-air mixture in the combustion [135]. However, with direct cylinder injection technique, abnormal combustion anomalies like pre-ignition and backfire can be avoided.

Erren [83], who used this concept of hydrogen injection directly into the combustion chamber in the year 1933 only. Homan H S [89] and [136], had developed late injection , rapid ignition and mixing (LIRIAM) technique to inject the hydrogen directly into the combustion chamber. Varde and Frame [132] and [137] had developed the hydraulically operated hydrogen injection and was used with direct cylinder configuration. Among these methods, to avoid the abnormal combustion anomalies like pre-ignition and backfire to an extent further to enhance the proper mixing of hydrogen in intake air for better combustion, timed manifold injection (TMI) is preferred. However, conventional diesel injectors are not successful and also not recommended for delivering the gaseous hydrogen due to difference of bulk modulus of gaseous and liquidous fuels.

Further details and comparison of these methods were shown in Table 2.1[124].

Table 2.1 Different Hydrogen injection techniques/methods [36]

Sl.No	Mixture formation	Classification	Hydrogen flow timing	Supply pressure
1	Continuous Carburetion	Pre-IVC	Continuous flow	A little above the atmosphere
2	Continuous Manifold induction/injection	Pre-IVC	Continuous flow	Slightly greater than the atmosphere
3	Timed manifold induction/Injection	Pre-IVC	Hydrogen flow commences after opening of the intake valve and closes before inlet valve closes	1.4 – 5.5 kg/cm^2
4	Low pressure direct injection	Post IVC	Hydrogen flow commences after the intake valve closure and is completed before significant compression pressure rise	2-8 kgf//cm^2

2.7. SUMMARY OF LITERATURE REVIEW

After exhaustive literature review it is concluded that, there is a possibility of fossil fuel replacement by vegetable oils for meeting the demands of irrigation pump diesel engines in agricultural sector especially in rural areas. Vegetable oil was one of the first fuel used in internal combustion engine especially in compression ignition engines in the year 1900. Since then, the use of vegetable oils as a fuel for diesel engines has been reported in the literature. But the viscosity of the vegetable oils makes the researchers work on different methods of reducing the viscosity. Some of such experiments based on use of esters to make bio-diesels derived from vegetable oils like: Jatropa's (*Jatropa curcas*) bio-diesel, sunflower's (*Helianthus annuus*) bio-diesel, soybean (*Glycine max*) based bio-diesel, etc. ; Pure vegetable oils and its blends with diesel like: sunflower (*Helianthus annuus*) oil blends with diesel, rice bran (*Oryza sativa*) oil blends with diesel, Jatropa (*Jatropa curcas*) blends with diesel and palm(*Elaesis guineensis*) oil blends with diesel etc. ; Methyl esters like: soybean methyl ester, methyl esters of safflower (*Carthamus tinctorius*) , methyl esters of karanja (*Pongamia Pinnata*) etc., and different dual fuel blends like: rubber seed

(*Hevea brasiliensis*) oil supplemented with diethyl ether (DEE), bio-diesel with diethyl ether etc. had carried out their experiments and showed positive results of using the same as alternative fuels for diesel engines with a minimum penalty on environmental burden. Out of these, as per literature it was seen that, as per comparative Life Cycle Assessment (LCA) and Energy Return on Investment (EROI) straight vegetable oils (SVO) are considered to be more potential and environmental friendly if they used in compression ignition engines. Further, India has diversified climatic conditions, Jatropa (*Jatropa curcas*) was considered to be the best suitable alternative as non-edible oil. These plants can grow in harsh climates, saline and alkaline soils and oil obtained from these plants have almost same power output with slight lesser thermal efficiency when used in diesel engines. But, at the same time, this Jatropa based straight vegetable oil (SVO) has high viscosity and high density when compared to diesel fuel. Though it was heated to 90^o to 100^oC [138], the kinematic viscosity is still double the diesel fuel makes poor atomization, further vaporization leads to poor combustion resulting in reduction in performance and increase in emissions with these straight use of Jatropa based vegetable oil. To avoid these issues, hydrogen supplementation was considered to be the best suitable option in order to augment the combustion thereby enhancing the performance and reducing the emissions. Further, as high self-ignition temperature of the hydrogen is, 568^oC, which is much higher than the most of CI engine fuels makes it not a successful attempt as a sole fuel. Some researchers [88, 89] suggested that, use of sole operated hydrogen compression ignition engines are very difficult to operate without any external ignition source and the same researchers [88-92], proposed and successfully tested the external source of ignition including pilot fuel injection of fossil fuels. Out of these, pilot fuel injection during the end of compression stroke seems to be considered and reliable ignition source.

Hence, dual fuel mode seems to be the most popular and stable combustion, when injected pilot fuel versus inducted hydrogen is used as a supplemented fuel. Generally, hydrogen was supplemented through the inlet manifold into main stream of inlet air. Thus, inducted hydrogen will properly mix with air prior to injected pilot fuel at the end of the compression stroke. This hydrogen-air mixture will be compressed during compression stroke and reaches the highest temperature to ignite the injected pilot fuel. But this temperature is not sufficient to burn the high self-ignition temperature of the inducted hydrogen. The injected pilot fuel in turn atomizes, vaporizes and mixes with high temperature –air mixtures. Then self-ignited and burned through all combustion phases of compression ignition engines. Then the heat released from the pilot fuel due to combustion will serve as ignition source to ignite the inducted hydrogen fuel. Effective participation of supplemented hydrogen in the combustion chamber and to control the undesirable combustion phenomena like pre-ignition and backfire all are dependent on the technique or mode of hydrogen inducted or injected into the engine cylinder. Hence, the technique or mode of fuel induction plays a vital role in hydrogen based internal combustion engine. There are various techniques of hydrogen induction/injection strategies out of which Timed manifold injection/induction (TMI) was identified to be the best suitable technique to inject or induct the gaseous hydrogen. Further, dual fuel operated CI engines offer the potential of reduced exhaust emissions with increased engine performance [93]. Further, these pilot vegetable oils need to be pre- heated to 90^o C [138] before injecting into the combustion chamber of the engine in order to reduce the viscosity.

A glimpse of literature review was shown in Table 2.2

Table 2.2 Glimpse of Literature review

Sl. No	Fuel used	Researcher	Mode of operation
1	Soybean methyl ester	Scholl et al., 1983 Yusuf Ali et al., 1997 Schumacher et al., 1996 Kayisoglu et al., 2006	Use of esters
2	Rapeseed oil methyl ester	Laforgia, 1995	
3	Pre-heated Palm methyl ester	Masjuki et al., 1996	
4	Linseed oil methyl ester	Agarwal et al., 1998, 2001 Deepak Agarwal, 2006	
5	Coconut oil esters	Masjuki et al., 2001	
6	Olive oil methyl ester	Dorado et al., 2003	
7	Jatropa oil methyl ester	He et al., 2003 Ganapathy et al., 2011 Arun Balasubrahmanian et al., 2012	
8	Karanja oil ester	Nazar et al., 2004	

9	Tobbacco seed oil methyl ester	Usta et al., 2005	
10	Mahua bio-diesel	Raheman et al., 2004 Sukumarpuhan et al., 2005	
11	Sunflower oil methyl ester	Kayisoglu et al., 2006	
12	Neem oil with methyl alcohol	Nurun Nabi et al., 2006 Arun Balasubrahmanian et al., 2012	
13	Waste cooking oil methyl ester	Sudhir et al., 2007 Zafer Utlu et al., 2008	
14	Rice bran oil methyl ester	Lakshminarayana Rao et al., 2008	
15	Rubber seed oil blended with Diethyl ether	Edwin Geo et al., 2009	
16	Soybean oil blends	Engleman et al., 1978	Use of Straight vegetable oil blends with mineral diesel
17	Rapeseed oil blend	Sims et al., 1981	
18	Peanut oil blends	Barsic et al., 1981	
19	Sunflower oil blend	Barsic et al., 1981 Ziejewski et al., 1982	
20	Cooking oil blend	Yu et al., 2002	
21	Rice bran oil blend	Venkataramana et al., 2003	
22	Jatropa oil	Pramanik et al., 2003 Forson et al., 2004	
23	Palm oil	Suryawanshi et al., 2009	
24	Sunflower oil	Bruwer et al., 1981 Yarbrough et al., 1981 Tahir et al., 1982 Bettis et al., 1982 Engler et al., 1983	Use of Straight vegetable oils (SVO)
25	Corn oil	Goering et al., 1981	
26	Rapeseed oil	Goering et al., 1981 Auld et al., 1982	
27	Sesame oil	Goering et al., 1981 Auld et al., 1982	
28	Cotton Seed oil	Goering et al., 1981 Auld et al., 1982 Engler et al., 1983	
29	Soybean oil	Goering et al., 1981	

		Auld et al., 1982 Pyror et al., 1983	
30	Safflower oil	Bettis et al., 1982	
31	Mustard oil	Seppo et al., 1997	
32	Waste cooking oil	Yu et al., 2002 Pugazvadivu et al., 2005	
33	Rice bran oil	Lakshminarayana Rao et al., 2008	
34	Neem oil	Lakshminarayana Rao et al., 2008	
35	Coconut oil	Lakshminarayana Rao et al., 2008	
36	Jatropa oil	Narayana Reddy et al., 2006	
37	Hingan oil	Deshmukh et al.,	
38	Coconut oil	Barsic et al., 1981	Use of Pre-heated Straight vegetable oils (SVO)
39	Rapeseed oil	Maurayama et al., 1984	
40	Palm oil	Bari et al., 2002	
41	Karanja oil	Nazar et al., 2004	
42	Waste frying oil	Pugazvadivu et al., 2005	
43	H_2 + O_2 mixtures	Francis Issac de Rivaz, 1807	H_2 operated engines earlier developments
44	Pure H_2	Cecil, M.A, 1820	
45	Pure H_2	Jean Joseph Etinne Lenoiur, 1860	
46	Pure H_2	Erren R et al., 1933	H_2 operated engines medieval developments
47	Pure H_2	Oehimichen, 1942	
48	Pure H_2	King R O. 1948-1958	
49	H_2 + Diesel dual fuel	Karim et al., 1975, 1976	
50	H_2 + Diesel dual fuel	Homan et al., 1979	
51	H_2 + Diesel dual fuel	Gopal et al., 1982	
52	H_2 + Diesel dual fuel	Hargopal Rao et al., 1983	
53	H_2 + Diesel dual fuel	Li Jing et al., 1984	
54	H_2 + Diesel dual fuel	Wang Weing et al., 1985	
55	H_2 + Diesel dual fuel	Mathur et al., 1992	
56	H_2 + Diesel dual fuel	Naber et al., 1998	
57	H_2 + Jatropa based straight vegetable oil dual fuel	Senthil Kumar et al., 2003	
58	H_2 + Diesel dual fuel,	Saravanan et al., 2007	
59	H_2 + Rubber seed oil and H_2 + Rubber seed oil methyl ester	Edwin Geo et al., 2008	
60	H_2 + B20 diesel dual fuel	Matthew et al., 2008	H_2 operated engines

61	H_2 + Diesel dual fuel with EGR and SCR	Saravanan et al., 2008, 2009	modern developments
62	H_2 + Diesel dual fuel under HCCI Mode	Szwaza Stanislaw et al., 2009	
63	H_2 + Diesel dual fuel	Probir Kumatr Bose et al., 2009	
64	H_2 + Diesel dual fuel	Muraimohan Roy et al., 2010	
65	H_2/O_2 mixtures as additives to diesel	Bari et al., 2010	
66	H_2 + Diesel dual fuel	Jacob Wall et al., 2010	
67	H_2 + Ultra Low Sulphur Diesel (ULSD) dual fuel	William F Northrop et al., 2010	
68	H_2 + Diesel dual fuel	Chong J J et al., 2010	
69	H_2 + Diesel dual fuel under HCCI mode	Hohgshung Guo et al., 2011	
70	H_2 + ULSD + Palm methyl ester	Zhou et al., 2013	

With this background, in this research, Jatropa based straight vegetable oil was identified as a pilot fuel with pre-heating to a pre-determined value i.e., 90^{o}C and gaseous hydrogen supplemented through the inlet manifold under timed manifold induction or injection technique (TMI).

1. It was seen that, being an equivalent energy content of vegetable oils with mineral diesel, vegetable oils can be used in the place of mineral diesel.
2. As per various attempts shown in the literature and observed through the comparative Life Cycle Assessment (LCA), Energy Return on Investment Index (EROI) and toxical and non-toxical assessments, straight vegetable oil will be the alternative fuels when compared to bio-diesels derived from vegetable oils, their blends and additives.
3. As Jatropa (*Jatropa curcas*) plants can grow in harsh climates, saline and alkaline soils and can resist the diversified climatic conditions of India, further, its straight vegetable oil was considered to be a good competitive alternative to the conventional diesel fuel.
4. In order to overcome the poor combustion of Jatropa straight vegetable oil due to its viscosity even after pre-heating to 90^{o} C, high flame speed, high burning

velocity, zero GHGs, high heating value gaseous hydrogen can be used as a supplemented mode to augment the combustion there by enhancement of performance and reduction in emissions can be achieved.

5. A wide variety of engines varying from 3.7 kW to 265 kW, single cylinder to multi cylinder, fueling from conventional diesel to different alternative fuels and additives like; DEE, RSO and RSOME, hydrogen etc. using different fuel induction techniques; carburetion system to High Pressure Direct Injection system have been studied. The results are promising to represent hydrogen as good alternative fuel as sole and supplemented mode.
6. Hydrogen as sole fuel utilization would require major modifications starting from engine material compatibility to utilization of hydrogen, its safety and economical aspects.
7. In supplemented mode of hydrogen gives better substitution in transition phase to switch over to the sole hydrogen engines. The supplemented hydrogen engines would require minor hardware modifications.
8. Hydrogen supplemented CI engine operating on the conventional or alternative fuels would require an approach to be developed considering optimization of heat release with optimal proportion of hydrogen to supplement the conventional and/or alternative fuelled CI engines to enhance their performance and reduction in emissions.
9. Further, no substantial work has taken place with in-direct Injection engines, needs to be concentrated.

2.8. MOTIVATION AND GAPS

The ever increasing fossil fuel consumption, cost and environmental concern has forced the researchers globally to look at different alternative fuels. Usage of straight vegetable oils in compression ignition engines will be a readily available solution. Though the SVO have some disadvantages, because of its easily availability, renewable and heating values nearer to diesel makes SVO be a suitable replacement for diesel. However, being in diversified climatic conditions Jatropa (Jatropa curcas) was considered to be the best plant species for selected SVO. Further, at some time in the not- too- distant future, hydrogen with its infinite fuel source potential has to be accepted as a very clean chemical fuel for engine applications. It has been clearly understood that hydrogen can burn in an engine with an increased efficiency and with much less pollution hazards compared to fossil fuels. Most of the abnormal combustion anomalies of using hydrogen in internal combustion engines were identified with possible solutions. Further, supplementation of hydrogen does not require much substantial design modifications on the existing engines. Hence adopting the hydrogen as an energy carrier rather than energy source as in dual fuel mode application in the existing engine can be looked into with all safety related issues in handling and storage. The unique properties of hydrogen compared to both conventional liquid fuels, gasoline and diesel make it a challenging yet a promising fuel for internal combustion engines. In particular, wide flammability range 4-75% by volume of hydrogen in air, low minimum ignition energy of 0.02 mJ and high flame speed nearly 2.65 to 3.25 m/sec at stoichiometric condition at 360 K, leads to operate even at lean mixtures, in other words, a fuel –air mixture in which amount of fuel is less than the stoichiometric or chemically ideal amount, by providing a better uniform mixture of hydrogen and air allows for greater fuel economy due to a more complete combustion of the fuel.

Further, supplementing of hydrogen in SVO operated engines shown a promising growth towards enhancement of performance and reduction in exhaust emissions. About 2/3rd of country's population is employed in agricultural sector, may give good response of using SVO in irrigation pump, diesel mode operated, agricultural, stationary engines with hydrogen supplementation. This will help to reduce the usage of conventional diesel fuels. Therefore present work is in step in the direction of using pre-heated Jatropa based straight vegetable oil as an alternative to conventional diesel for an agricultural stationary engine, vertical, lister, 7.35 kW, 4 stroke, indirect-injection, diesel engine supplemented with small doses of hydrogen on the mass basis to enhance the performance and reduction in emissions was considered. Further, there are certain identified gaps from the literature to extend this research towards this direction:

1. Use of vegetable oils in compression ignition engines showed a good promising results to replace the conventional diesel. As per Comparative Life

Cycle Assessment (CLA) and Energy Return on Investment Index (EROI), Straight Vegetable Oils are good and environmental friendly to be made use as alternative fuels in place of conventional diesel rather than bio-diesel derived from vegetable oils. However, a lot of work has already been done on bio-diesels, its blends and additives. But still pre-heated straight vegetable oils needs to be concentrated.

2. The presence of chemically bound O_2 will reduce its heating value of vegetable oil which can be compensated with supplementing with high heating value hydrogen with small doses that leads to enhancement the combustion of oxygenated fuels, thereby improving the performance and reducing the emissions. Hence, in this direction of hydrogen supplementation can be concentrated.
3. Taking the advantage of lower minimum ignition energy of, 0.02 mJ for hydrogen most of research was concentrated on the spark ignition (SI) engines. Hence, there is a good scope to work on compression ignition (CI) engines with usage of hydrogen as supplemented mode.
4. Very limited work done on the engine parameters like fuel injection strategy as: change of injection timing (preferably injection advancement) and injection pressure (preferably increase of injection pressure) due to high viscous of the SVO and high self-ignition temperature of the hydrogen. Hence, concentrating in this direction will enhances the combustion of SVO-hydrogen dual fuel operated engines thereby improving the performance and reducing the emissions.
5. Pre-chamber diesel engines are more suitable for straight vegetable oil based fuels [4]. However, no substantial work has been done on these pre-chamber indirect injection engines fuelled with straight vegetable oil supplemented with hydrogen.
6. Further, to start the usage of hydrogen in internal combustion engine for researchers, a comprehensive document is required towards safety aspects of handling and storage of gaseous hydrogen needs to be developed.

2.9. OBJECTIVES OF THE PRESENT INVESTIGATION

- Establishment of an experimental set up to obtain baseline data for conventional Diesel, Pre-heated SVO at 90^{o} C (PHSVO 90) and PHSVO 90 with gaseous hydrogen supplementation operations.
- To investigate the effects of gaseous hydrogen addition on performance and Combustion parameters
- To study the effects on exhaust emissions of PHSVO 90 operated IDI CI engine with gaseous hydrogen supplementation
- To optimize the engine parameters for improved performance and reduction in emissions